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Tread Brake Systems in Metro Vehicles

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Mechanical and Industrial Engineering

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Jede neue Herausforderung ist ein Tor zu neuen Erfahrungen.

Ernst Ferstl

Statutory declaration

I declare that I have authored this thesis independently, that I have not used other than the declared sources/resources, and that I have explicitly marked all material which has been quoted either literally or by content from the used sources.

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Date Signature

Notes of thanks

I want to take the opportunity to thank those people who supported and encouraged me during my thesis.

Special thanks go to Rüdiger Zenz, Siemens AG Österreich, who introduced me to the topic of railway brakes. He always took his time to answer my questions and provided me with the vital basics and initial contacts for the thesis on hand. Additionally I want to thank Thilo Hoffmann and Martin Teichmann, Siemens AG Österreich. Both were reliable sources for information, I appreciated. Especially Martin Teichmann always provided me with additional creative ideas. I am grateful for the time he spent with me, sharing his experience and knowledge. Furthermore, I thank Dominik Festl, Andreas Kitzmüller and Gerard Salzgeber for their advice.

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Abstract

The thesis at hand deals with railway vehicle braking systems; more precisely it is focused on the metro vehicle Siemens Syntegra and its tread brake units. These are mainly used for immobilization brake applications in this context. This is caused by the usage of the safe innovative brake (SIB). Actually SIB is a brake control system invented by Siemens AG. It enables the electro dynamic brake to be used as a service brake and additionally as an emergency brake. This reduces the demands on the friction based tread brake.
Moreover, this thesis clarifies the question whether the decrease in demands provides improvement potential to the currently used tread brake unit. The focus lies on the design of the necessary brake actuator. Another aim of the paper is to find improvement potential on the currently used brake as well as it provides basic information to support the strategic decision between the two actuation principles, the electro-pneumatic, and the electro-mechanic of the mentioned tread brake unit.

Initially a general understanding needs to be ensured. Therefore, the thesis starts with an introduction to the multiple braking applications which a railway vehicles needs to be capable of. Additionally the different braking systems are introduced and the brake force creation mechanism is described. Certainly the overall focus is related to metro vehicles.

In the second part the SIB is introduced. Furthermore an in-depth introduction to related standards and performance specifications is provided. Additionally the brake actuation is investigated in detail. A design catalogue is used to evaluate the possibilities for actuation from a holistic perspective. This covers the three major parts of the actuator used to perform immobilization brake applications, more precisely the power amplification, the energy storage, and the charging of the energy storage. Moreover, the current benchmark is introduced and some electro-mechanical actuation possibilities.

The third part provides calculations of the necessary parking force, deceleration or thermal stresses. Both these computations and an intensive simulation study of the brake unit setup enable the creation of the framework for the detailed considerations related to the design of the tread brake actuator. The ideas of power amplification, energy storage, and charging of energy storage are evaluated in detail. Finally the presentation of certain electro-pneumatic and electro- mechanic actuators pinpoints the necessity of a strategic decision.

The conclusion together with a future outlook point out perspectives for the different decision opportunities.

Kurzfassung

Die vorliegende Arbeit befasst sich mit Schienenfahrzeugbremssystemen. Im Speziellen mit den Klotzbremseinheiten eines bestimmten Metrofahrzeugs, des Siemens Syntegra, welche hauptsächlich als Feststellbremsen eingesetzt werden. Dies resultiert aus dem Einsatz der Sicheren Innovativen Bremse (SIB). Das Bremskontrollsystem ist eine Siemensentwicklung und trägt dazu bei, dass neben der Betriebsbremsung auch die Notbremsung mit einer elektrodynamischen Bremse durchführbar ist. Dadurch wird die reibungsgebundene Klotzbremse entlastet. Es wird im Rahmen der Diplomarbeit untersucht ob diese Verringerung der Beanspruchung das Potential zur Verbesserung der Klotzbremseinheit birgt. Dabei ist im vorliegenden Fall die Aktuatorik der Bremse im Fokus. Ziel dieser Diplomarbeit ist die Evaluierung von Verbesserungsmöglichkeiten der derzeit eingesetzten Einheit. Zusätzlich werden Informationen bereitgestellt, welche die strategische Entscheidung zwischen den Aktuatorprinzipien elektropneumatisch und elektromechanisch für die eingesetzten Klotzbremseinheiten unterstützen.

Im ersten Teil der Arbeit werden die Grundlagen dargelegt. Diese beinhalten die Vorstellung der verschiedenen Bremsanforderungen welche von Schienenfahrzeugen umsetzen werden müssen, sowie die unterschiedlichen Bremssysteme und die Bremskrafterzeugung im Allgemeinen. Selbstverständlich liegt der Schwerpunkt dabei auf Metrofahrzeugen.

Der zweite Teil beginnt mit der Vorstellung der Sicheren Innovativen Bremse, SIB. Zusätzlich werden Normen und das Lastenheft vorgestellt, die als Grundlage für die Dimensionierung des Bremsaktuators herangezogen werden. Anschließend wird der Aktuator im Detail betrachtet. Dazu werden seine drei Hauptbestandteile, Kraftverstärkung, Energiespeicherung und Energieeinbringung untersucht. Auf Basis eines Konstruktionskataloges werden unterschiedliche Realisierungsmöglichkeiten des Aktuators vorgestellt. Zusätzlich werden die derzeit verwendete Klotzbremseinheit sowie eine Auswahl elektromechanischer Aktuatoren vorgestellt.

Im dritten Teil wird die Berechnung der Parkbremskraft, der Verzögerung und der thermischen Beanspruchung dargestellt, sowie eine Simulation der Bremseinheitenverteilung. Damit ist es möglich eine weitere Einschränkung der Realisierungsmöglichkeiten aus Teil zwei vorzunehmen. Schlussendlich zeigt die Auswertung die Notwendigkeit einer strategischen Entscheidung zwischen elektropneumatischen und elektromechanischen Aktuatoren.

In der Zusammenfassung und dem Ausblick finden sich zusätzliche Informationen welche die unterschiedlichen Perspektiven für die Entscheidungsfindung darstellen.

Table of content

1 Introduction

When Wilhelm Jaxtheimer published his book on railway braking systems in 1933, he considered this issue as quite bottom-of-the-line until 1918. Well! It was 1933 and the topic got complicated, a book considering the basics became necessary. People related to railway vehicles, from the engine driver to the student had to be provided with the general idea on braking systems, because it had become the limiting factor to the top speed of railway vehicles. (Jaxtheimer, ca 1935)

Certainly, the issue did not get easier over the years. Speed increased further and every new braking system had to deal with the enhanced demands.

Today we are standing at an edge of development again. The trend is to find alternatives to pneumatic systems in railway brakes. The aim is to establish an all- electric vehicle.

Along with this trend, the aim of the thesis on hand is to investigate the opportunities provided by the SIB. The 'Safe Innovative Brake' invented by Siemens AG. Due to this all new brake control principle a decrease in braking force is expected.

1.1 Initial Condition

Electrical drive motors used in railway vehicles offer the possibility of braking. In case the drive motor is used as a generator, a deceleration of the vehicle can be achieved. Basically there are two types of this so-called electro dynamic braking. On the one hand it is possible to have a rheostatic brake system and on the other a regenerative brake system. Especially the regenerative brake system is popular at the moment because it serves the trend of sustainability. The brake application causes the generator to produce electric current which can be fed back into the grid. This possibility promotes the idea of using generators to a large extent. In course of this trend, a reduction of friction brakes that simply turn motion energy into heat is very popular. However, it is not possible to reduce all the friction brakes in railway vehicles at the moment. Still a lot of research work has been done over the years to find ways for using the dynamic brakes to a larger extend; especially considering the emergency brake. The goal is to use the dynamic brake besides the service brake application also as emergency brake. Along with this idea, the Siemens AG invented the SIB, the 'Safe Innovative Brake'. This braking control system is supposed to reduce the necessity of friction brakes in rail vehicle motor bogies. It can be considered, that with this system, the friction brake will mainly be used as immobilization brake and in some rare cases as last safety device to stop the vehicle if all other brake applications fail.

Due to this decrease in demands on the friction brake it is expected to need less braking force and this in turn can cause the braking actuator of the friction brake to be smaller and posses of reduced weight.

Moreover the SIB can facilitate the all-electric vehicle. While the service and emergency brake application are realized with the traction unit, it is only necessary to find a solution for a non-pneumatic immobilization brake. The idea of an all-electric vehicle is especially popular due to the reduction of weight and costs caused by the reduction of the compressor, brake piping and valves.
Certainly these considerations are valid for motor bogies only, because they possess an electrical drive motor. Yet in fact many metros sold by Siemens possess of an only motor bogies, which is called 100% motorization.

1.2 Objective

The objective of this thesis is to provide a decision- making support on which type of braking system is most appropriate for an immobilization brake actuator used together with the SIB in a metro.

Certainly the actuator has to be compliant with the EN 13452:2003, the leading guideline for mass transit vehicles. Especially the mentioned standards for metro vehicles with steel wheels are considered in the first place. Particularly for this thesis also the standards derived from the MMC (Metro Mid Cap) Lastenheft, (Siemens AG, 2010), are taken into account.

Object to compare for all found opportunities is the currently used electro-pneumatic Faiveley Transport BFC Tread Brake. Improvement potential concerning and alternatives to this presently used brake are considered with respect to defined characteristics and an outlook on future developments of braking systems.

1.3 Methodology

In the first place a literature study on regulations and basics of braking technologies is done to provide general understanding of the topic. Hereafter the Siemens SIB is introduced and several characteristics for the braking system are derived from the EN 13452-1:2003 and the MMC Lastenheft, (Siemens AG, 2010).

Then calculations considering the braking force, deceleration, braking distance and thermal conditions are conducted. Moreover a simulation study is presented which enables an objective choice of the brake configuration.

In this thesis the basic approach points out the different physical possibilities to design actuators and provide power amplification. Hence possible combinations are derived and the spring as lasting energy storage is defined.

In the next step a decision on the type of spring and a dimensioning calculation is conducted, which leads to different design approaches in detail.

For this purpose a second literature review is done to find current developments and possibilities. Finally a decision support is developed by means of the established criteria and additionally evaluation trends on the market are studied.

2 Siemens AG – Industrial partner

The host company for this thesis was Siemens Graz. As subsidiary of the Siemens AG Österreich with its specialization on bogie production, the branch office in Graz belongs to the Sector of Infrastructure and Cities, Division Mobility.

The production site, located in Graz has been a partner for the railway industry for about 150 years.

In the beginning it was a small blacksmith's shop, founded by Johann Weitzer in 1854. (Siemens STS, p. 5)

Nowadays, after a changeful company history, Graz is a subsidiary of Siemens AG Österreich and known as the world competence center for bogies (Siemens AG Österreich, 2011, p. 19). The product portfolio consists of bogies for mass transit like metro cars, trams, and urban vehicles as well as main line bogies for locomotives, passenger caches, and trains. Currently there are about 30 different bogies available.

These bogies are known for their high operation safety, particularly smooth running, reliability, low LifeCycleCosts and maintainability.

The proudly cherished specialty of the subsidiary in Graz is the automated bogie production. It is considered as the one with the highest automation degree worldwide. (Siemens STS, pp. 5-6)

Also sophisticated new developments are made by the engineers in Graz; e.g. is the Syntegra bogie.

Figure 1 Siemens Syntegra (www.mobility.siemens.com)

Special about the Siemens Syntegra bogie is the highly integrative design. A unified mechatronic system is created, combining traction, braking and bogie technology.

This type of motor bogie provides the advantage of high efficiency, reduced LifeCycleCosts and reduced weight. The present weight is to be 5t and the maximum axle load 14t.
This is made possible by using technologies like the completely encapsulated permanent magnet synchronous machine without a gear box as traction unit, a flexible and compact bogie design considering the lightweight design approach and an electric safety brake.
As a result the Syntegra bogie is more than 15% lighter than a conventional bogie with the same payload and traction power. Additionally the energy consumption is reduced due to the reduced weight and increased efficiency. It is expected that the decrease will be about 20%.

Due to the context of this thesis it is reasonable to take a closer look to the traction unit and the innovative brake design.

As mentioned previously the traction unit is an encapsulated permanent magnet synchronous machine. This is probably its most important advantage. The machine combines rotor and axle. Therefore two axles are powered separately. It is considered that no wearing parts are implemented. Additionally the necessary encapsulation provides maintenance free operation and a point of application for linkage to the car body. As a result the bogie frame does not have to endure any traction forces. However, the most important thing about the motor is its ability to provide an inherent braking torque in case a suitable external snubber circuit is installed. This physical characteristic provides a fail-safe braking ability. Nevertheless a special braking control has to be introduced to achieve compliance with the standards on braking systems. (Siemens AG, 2012)
This topic will be covered later on in Chapter 3.2.1.

In order to provide a holistic understanding of the Syntegra bogie the following figure shows the different components.

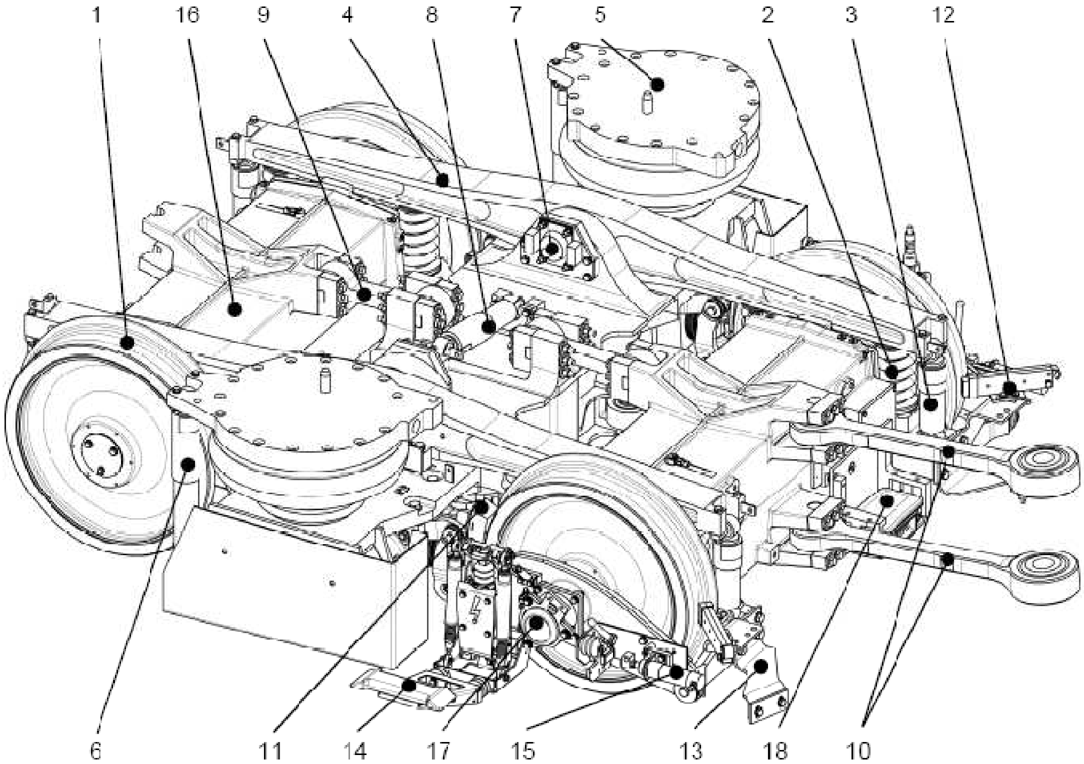


Figure 2 Parts of the Syntegra bogie (Siemens TS, 2008)

1 wheel set
2 primary suspension
3 primary damper
4 undercarriage frame
5 secondary suspension
6 secondary damper
7 secondary lateral suspension
8 secondary lateral damper
9 motor linkage - transmission of longitudinal force
10 bogie linkage - transmission of longitudinal force
11 tread brake
12 flange oilers
13 rail guard
14 current collector
15 short circuiting device
16 traction motor
17 earth-return brush
18 train stop

3 Literature outline

The literature outline is divided into two major parts. The first part considers common aspects of railway vehicles, with special focus on metro vehicles. The second part provides the theory necessary for the thesis on hand. The topics are related to the steps necessary to fulfill the tasks given by the thesis.

3.1 Railway vehicle related background

This part of the literature outline is supposed to provide a general overview on important keywords concerning braking and the different types of railway brake systems. Firstly it must be clarified what the duties of a braking system are and how braking can be characterized using different diagrams. Secondly the different braking systems are described. Furthermore advantages and disadvantages of the systems are studied. The last part is the introduction to the brake force creation and its limitations.

3.1.1 Duties of the braking system

Braking systems face limitations due to physical conditions. The aim of a braking system is to grant three main functions. Firstly, it has to transform kinetic energy of the moving vehicle into other forms of energy. As a result, the vehicle reduces speed if necessary until it stops. Secondly, the braking system has to ensure that the vehicle runs with constant speed, also on an incline. The deviation due to the weight force component in driving direction must be considered via braking. Thirdly, parking vehicles have to stay in position even if the track has a gradient or the vehicle is exposed to other forces, like wind. (Knorr-Bremse, 2002, p. 16)

Basically braking creates a force which counters the motion of a rail vehicle or prevents movement of a stationary train (EN-13452-1, 2003). Several braking situations occur during the work life of a railway vehicle.

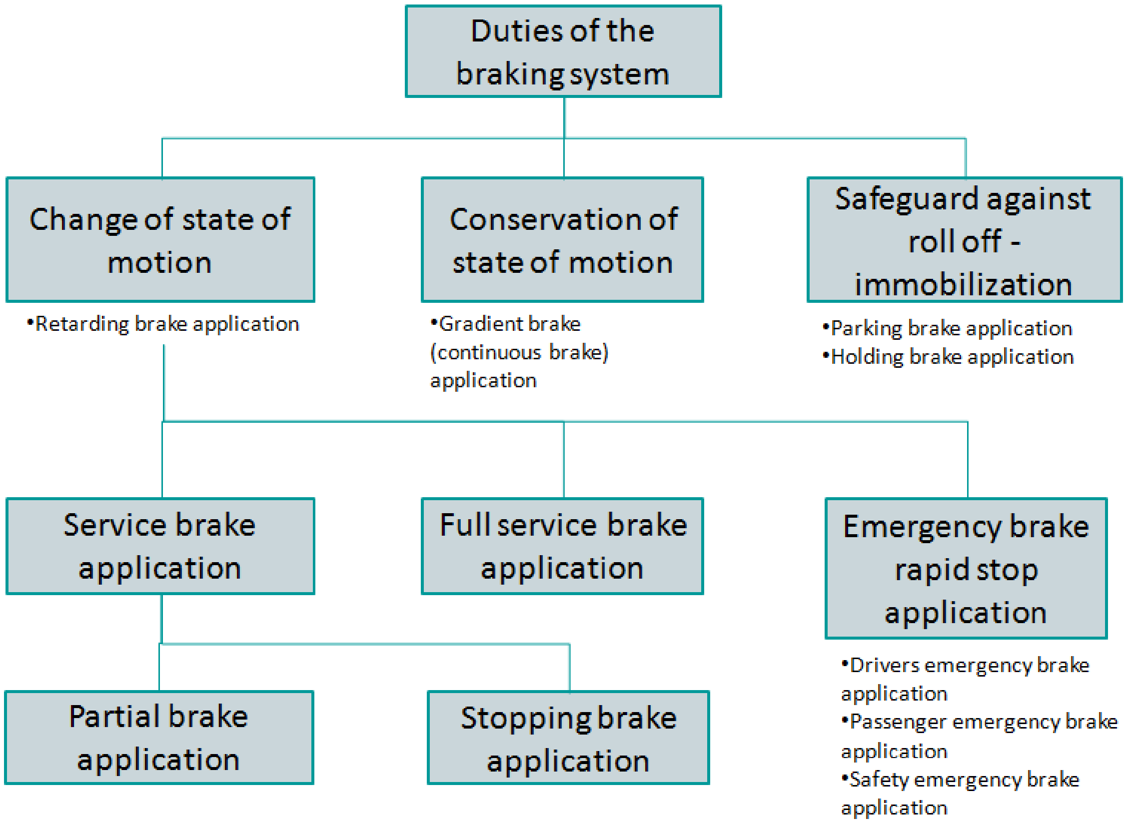


Figure 3 Duties of the braking system based on (Wende, 2003)

The main duties of the braking system are the change of state of motion, conservation of the state of motion and safeguard against roll off. Different brake applications enable the rail vehicle to perform the necessary braking operation. (Wende, 2003, p. 219)

DIN EN 13452-1:2003 especially distinguishes between the following braking applications:

- Service brake application
- Full service brake application
- Emergency brake application
- Safety brake application
- Stopping brake application
- Gradient brake application
- Immobilization brake application
  > Holding brake application
  > Parking brake application

Change of state of motion

The braking force has to reduce the rail vehicles speed to a certain limit if necessary until the vehicle stops. Basically, this is called retarding braking. Retarding braking consists of different braking types like the service brake application, the full service brake application and the emergency brake application which differ in the intensity of the braking and therewith in the resulting braking distance. (Wende, 2003, p. 220)

The service brake application uses the brake capacity partially. (Wende, 2003, p. 220) Its task is to influence the rail vehicles velocity. (EN-13452-1, 2003)
The service brake is used to adjust the velocity of the vehicle via the partial brake application or to stop the vehicle by the holding brake application. Both operations may either be velocity spot braking or velocity and displacement spot braking. (Wende, 2003, p. 220)

The full service brake application slows down the vehicle with the maximum force possible. If a pneumatic braking system is used, the brake pipe pressure will be reduced. (Knorr-Bremse, 2002, p. 32)

The emergency brake is considered to be a rapid stop brake application. The aim of the emergency brake application is to stop the railway vehicle in the shortest possible physical, but still for the passenger safe, way (Wende, 2003, p. 220). According to DIN EN 13452-1:2003 the intention of the emergency brake application is to provide maximum safety for passengers and personnel. An emergency brake has to stop the vehicle within a determined velocity/distance ratio considering a defined deceleration and jerk.
If a pneumatic brake is used, the brake pipe is vented rapidly and completely in order to ensure the maximum brake efficiency during emergency braking. Further the recharging of the brake pipe is barred. (Knorr-Bremse, 2002, p. 32)

Basically the emergency brake applications can be grouped as follows:

- Driver’s emergency brake application
- Passenger emergency brake application
- Safety emergency brake application

Conservation of state of motion

The gradient brake application is necessary to sustain a certain velocity at an incline. The braking force has to countervail the additional weight force component in driving direction. (Wende, 2003, p. 220)
A frequent brake adjustment (braking and releasing) is necessary which considers the length and the gradient of the track. This can exhaust direct release brakes. The

mechanical components of an air brake may be stressed due to the thermal demands; therefore dynamic brakes should be used for this purpose. (Knorr-Bremse, 2002, p. 32)

Safeguard against roll off

The braking force has to ensure that the vehicle can withstand forces due to exterior forces. It is essential to keep the train in position during a holding or a parking brake application. (Wende, 2003, p. 220)

According to DIN EN 13452-1:2003 parking, holding and immobilization brake application are defined as follows:

The Immobilization brake application
The stopped train vehicle is inhibited of further movement under certain conditions.

The Parking brake application
A train vehicle with defined mass is lastingly stopped for indefinite time duration on a sloped track.

The Holding brake application
The train vehicle is stopped for a defined duration with a defined load.

3.1.2 Braking characteristic

The braking characteristic describes the changes in motion during the braking process. Three types are available to illustrate the braking process.

Certainly the most intuitive characteristic is the velocity over distance diagram. This diagram has a clear statement. It points out the braking distance until stop from a certain velocity. It is easy to read and understand. (Heckemanns, 2010)
Figure 4 Velocity/distance characteristic provides an example:

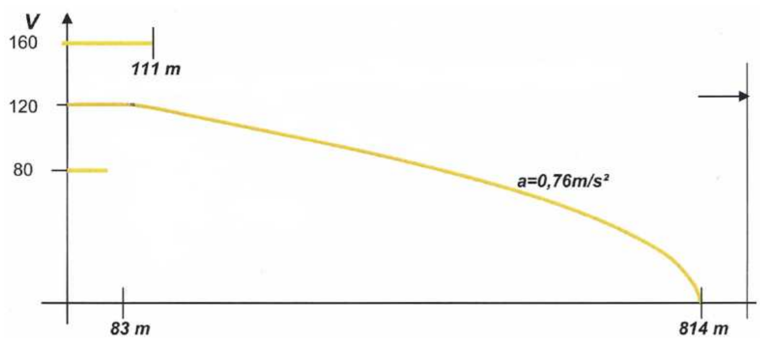


Figure 4 Velocity/distance characteristic (Heckemanns, 2010)

This particular characteristic starts at 120 km/h. After a certain reaction time the deceleration starts. More details on the reaction time and stopping distance will be provided later in this section.

Another way to illustrate the braking process is the characteristic velocity over time. Here a straight line representing the velocity implies constant deceleration; however any curve slope may be imaginable. Anyway with this type of diagram the braking distance is not visible. However, this actually is the most important thing in the first place. (Heckemanns, 2010)

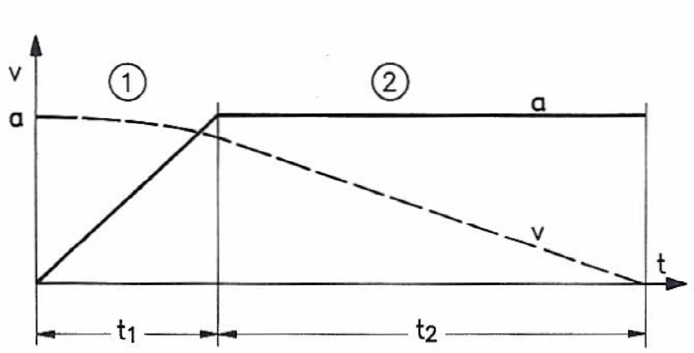


Figure 5 Velocity/time characteristic (Saumweber, Gerum, & Berndt, 1990)

Figure 5 shows an example of the velocity/time characteristic. However it also contains a deceleration characteristic. The figure illustrates that it takes some time to reach the constant maximum deceleration for a certain braking application. This figure is a simplification of the real characteristic. Real brakes do not contain constant deceleration during time t_2. (Saumweber, Gerum, & Berndt, 1990, p. 16)

Moreover a well known graphic representation of a braking process is the characteristic of braking force over velocity. Force and deceleration are proportional. Therefore this diagram characterizes the used principles very well. However it provides no information on the braking distance. (Heckemanns, 2010)

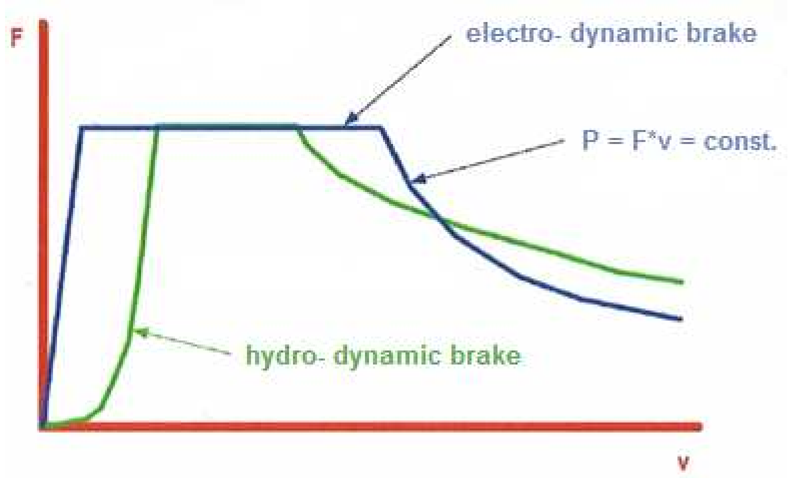


Figure 6 Force/velocity characteristic (Heckemanns, 2010)

According to (Heckemanns, 2010) the force/velocity characteristic is the most popular graphical representative. For all the different types of braking systems separate characteristics are necessary. (Heckemanns, 2010, pp. 23-24)
The braking force is used to perform the braking applications described in section 3.1.1. The sum of all braking forces of any braking system used in a vehicle as well as all rolling resistances together are known as the total braking force. Considering e.g. the characteristic of dynamic brakes, it gets obvious, that they need support by other brakes at low speeds. (Knorr-Bremse, 2002, p. 20), (Wende, 2003, p. 3) More details will be provided in section 3.1.3.

The characteristics mentioned above should help to clarify several expressions concerning distances, times and decelerations often mentioned in relation to the topic of braking applications.

Considering the distances, one has to differentiate between stopping distance and overall stopping distance. The stopping distance always goes with a certain velocity. The stopping distance is the distance covered between actuation of the operating controls and the achievement of the desired speed v_E. This distance can be divided into three sections. During the reaction section the braking force is zero. In most cases, there is still some deceleration due to resisting power of the vehicle and those parts of the weight force that result from the slope of the track. The dead time usually takes one to two seconds. The second section is the rise section. During this section the braking force changes from zero to the desired maximum. Normally the maximum is related to the type of the brake application. More or less the time needed can be considered as the force rise time. 5 to 25 seconds are considered as normal. (Gralla, 1999, p. 14) Dead time and force buildup time together are known as the reaction time. This is the time between the start of the brake application and the time when the brake forces reaches

90% of its maximum. (EN-13452-1, 2003) The full service section is the last section. In this section the braking force is available at the maximum allowed level for the specific brake application. The current deceleration depends on the prevailing velocity at each moment. (Gralla, 1999, p. 14)

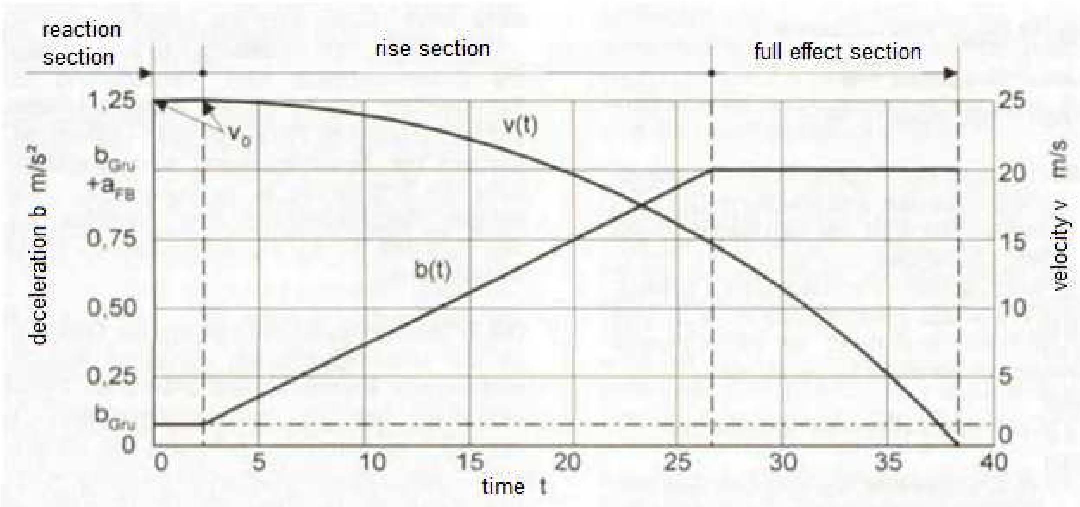


Figure 7 Three sections of the stopping distance (Gralla, 1999)

However, there is a second distance of interest: The overall stopping distance. This distance is derived from the moment the demand for braking occurs until the vehicle has stopped or reached a certain speed. Obviously this includes the time for detection, action and the preparation for braking as well as the stopping time itself. (Gralla, 1999, p. 15)

3.1.3 Realization of braking application

The rail vehicle has to be equipped with a braking system in order to perform the braking applications mentioned above. The braking system can be classified according to different perspectives:

(1) according to the tasks to be performed,
(2) according to the transmission of the braking force,
(3) according to the active principle and
(4) according to design.

Considering (1) the tasks to be performed, a braking system consists of a service brake, a parking brake and in some cases an additional brake.

The service brake controls the state of motion. It adjusts the intensity according to the brake application, whether a gradient brake application or retarding brake application is given. The service brake has to be designed to cope with a permanent load. Furthermore, it has to ensure as little wear as possible.
The parking brake provides safety against roll off for an unlimited time. The braking force has to maintain at the same level over time. Moreover the parking brake has to stop the vehicle in case of service brake failure.
Additional brakes, like the magnetic track brake, may be used to support the service brake in case of an emergency brake application. (Wende, 2003, p. 220)

(2) The transmission of the braking force can either be done using the wheel/rail force closure or without it. When the wheel is used, the braking system is called wheel dependent. In this case the limiting factor to the braking force is the friction coefficient between wheel and rail (see section 3.1.4). The wheel independent braking system on the other hand can achieve deceleration beyond the wheel-track adhesion limit. (Wende, 2003, p. 221)
This differentiation has been known for decades and is also used in (Jaxtheimer, ca 1935, p. 4). Interestingly three different possibilities for man-made deceleration are mentioned in this book: (i) Usage of wheel track force closure, (ii) track affecting and (iii) usage of air drag. However air drag cannot be used due to the size restrictions of possible braking surfaces.

Categorizing by the active principle (3) is another important way of differentiation. It applies only for pneumatic controlled brakes. These systems are either known as direct or indirect. Basically all braking systems have to provide braking force and some kind of signal transmission. Direct and indirect brakes refer to the type of braking force creation and signal transmission.
The direct pneumatic brake is very easy to realize and therefore was introduced very early in the history of railway vehicles. But it cannot fulfill some of the nowadays important standards. The biggest disadvantages are the non-uniform braking force creation and the slow output at the end of the vehicle. This problem can be reduced by introducing an electro-pneumatic system. Furthermore it cannot provide energy storage or self acting ability. The self acting ability is very important in case of breaking loose of a train. The following pictures give insight into the architecture of a direct brake.

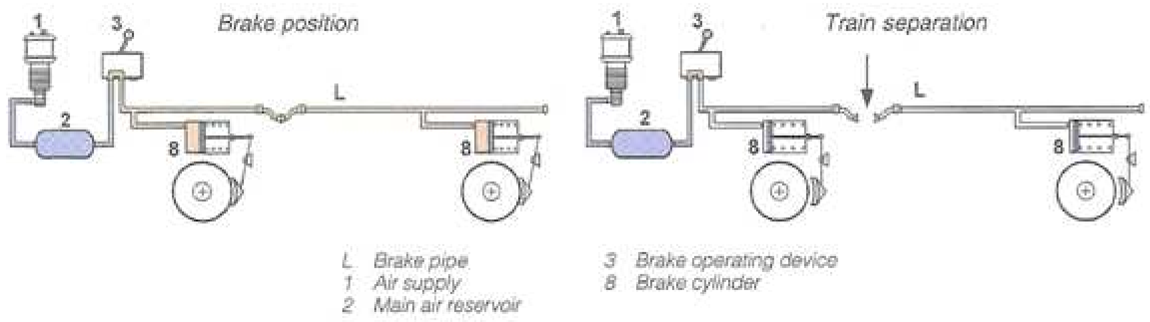


Figure 8 Direct working braking system in applied position and during train separation (Knorr-Bremse, 2002)

Figure 8 shows the system architecture of the direct braking system. The brake pipe is connected directly to the brake cylinder (8). Increased pressure in the brake pipe results in braking due to the movement of the piston in the brake cylinder.

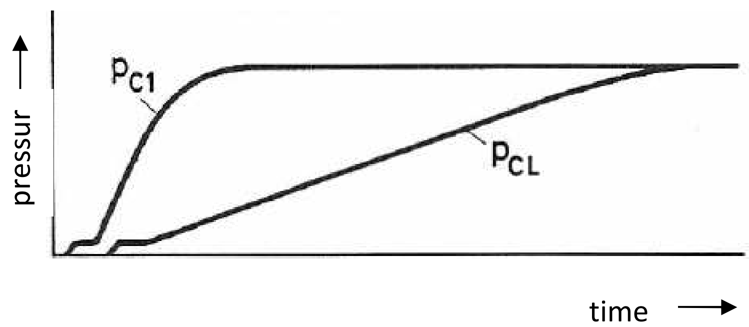


Figure 9 Pressure curve of brake cylinder using direct braking system at the first (p_{C1}) and the last (p_{CL}) coach (Saumweber, Gerum, & Berndt, 1990)

Figure 9 illustrates the difference in the pressure development over time. It points out that it takes the brake cylinder of the last coach longer to reach the necessary pressure than the first one. (Saumweber, Gerum, & Berndt, 1990, p. 85)
However, direct brakes are installed in some cases in the traction vehicle as additional brakes nowadays. They are used to increase the pressure in the brake cylinder and therefore allow utilizing the wheel track force closure in a better way. (Wende, 2003, p. 222)
However, the direct electro-pneumatic system (an electrically controlled pneumatic brake) is very popular for metro vehicles.

Indirect brakes are used in most cases for standard-gauge railway, but they are not common with metros. The most popular brake is the railway compressed air brake. Contrary to the direct brake, where pressure increase in the brake pipe is a result of the braking application, a decrease of brake pipe pressure is present when it comes to braking applications in indirect brake systems. (Saumweber, Gerum, & Berndt, 1990, p. 86) The following images show the principle of indirect braking systems:

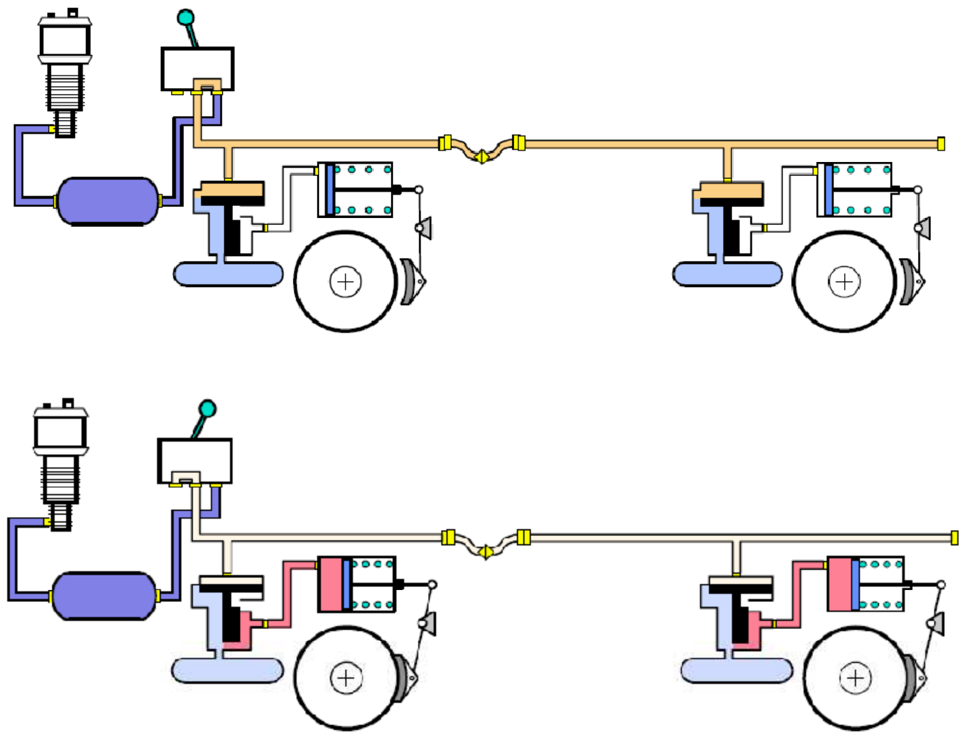


Figure 10 Indirect braking system in released and applied position (Kleemann, 2005)

As mentioned, a pressure decrease in the brake pipe enables the braking process.
Figure 10 shows the indirect braking system in the released position (upper) and the applied position (lower). In the released position the brake pipe is provided with pressure (orange) which is supplied by the compressor (dark blue). The compressed air furthermore charges the air reservoir (light blue), because the slide (black) is in its lowest position. The slide (black) furthermore inhibits the compressed air supply towards the brake cylinder. Therefore the spring is unloaded and the brake pad provides the necessary clearance. Additionally the slide (black) opens the inlet pipe to the surrounding, which guarantees that the brake piston is forced to the release position by the spring.
In the applied position on the other hand, the brake pipe is vented and the supply of compressed air from the compressor is prohibited by the main switch. In this case the slide black moves to its upper position, while closing the connection the brake pipe and the valve between the inlet pipe and the surroundings. In this situation the inlet pipe is

supplied with compressed air from the reservoir. This air compresses the spring in the braking cylinder and the brake shoe is applied to the wheel. (Knorr-Bremse, 2002, p. 25)

The major advantage of this architecture is the self-acting ability. In case the pressure decreases in the brake pipe due to braking loose of a train, the brakes are automatically activated. Furthermore, the braking force can be created faster because the necessary pressure is stored in the air reservoir, and the de-aeration of the brake pipe is done faster than a pressure increase can be realized. Certainly this is only valid for pneumatic and not for electrical controlled pneumatic brakes. In case of an electrical control, the direct brake is faster than the indirect according to Siemens experience.
Of course the air reservoir has to be filled with air again after it was used to activate the brake.
Certainly the indirect brake provides advantages concerning some of the standards that are not fulfilled by a direct system. However, the uniformity of the braking force creation is still not provided as seen in Figure 11:

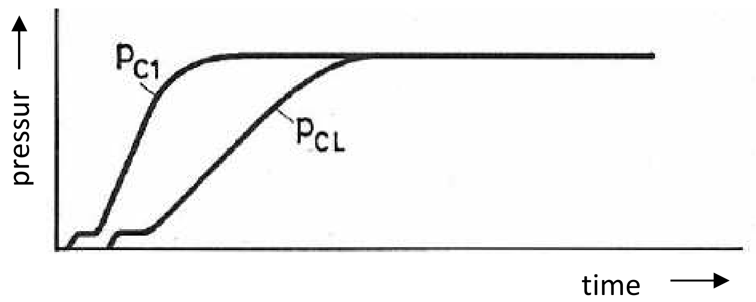


Figure 11 Pressure curve of brake cylinder using indirect braking system at the first (p_{C1}) and the last (p_{CL}) coach (Saumweber, Gerum, & Berndt, 1990)

Due to the faster change of the pressure in the front of the brake pipe also the braking force is created in front of the train faster than at its end. The brake cylinder pressure does not only differ due to the reaction time, it also has a different gradient. (Saumweber, Gerum, & Berndt, 1990, p. 86)
A development concerning the indirect braking is the electro pneumatic brake, where the brake application is transmitted to the coaches using electrical signals. This decreases the reaction time and the pressure build-up time. (Wende, 2003, p. 221)

However, the biggest advantage of an indirect pneumatic controlled brake is the automatic triggering of the brake application in case of breaking loose of a train car. This is especially important when it comes to standard-gauge railway vehicles.

(4) The design also provides a possibility to separate the different braking systems. Mechanical braking systems represented by tread or disc brake systems, sometimes also equipped with spring loading units are one design opportunity. Another type is typified by the electromagnetic track brake or the linear eddy current brake. Furthermore electrical braking systems using the electrical drive motor as electric generator are becoming very popular. Also hydrodynamic brakes in motor coaches with hydrodynamic power transmission are known. The rotating eddy current brake is also subject to planning for e.g. high-speed trains due to its ability to handle thermal stresses. In trams with DC series motor the short-circuit brake is used as well. (Wende, 2003, pp. 222- 224)

Braking systems

As mentioned above braking systems can be separated according to different perspectives. This chapter is supposed to provide a general idea on the present systems and their force creation principles.

A good overview on the present braking force creation opportunities is provided by (Heckemanns, 2010):

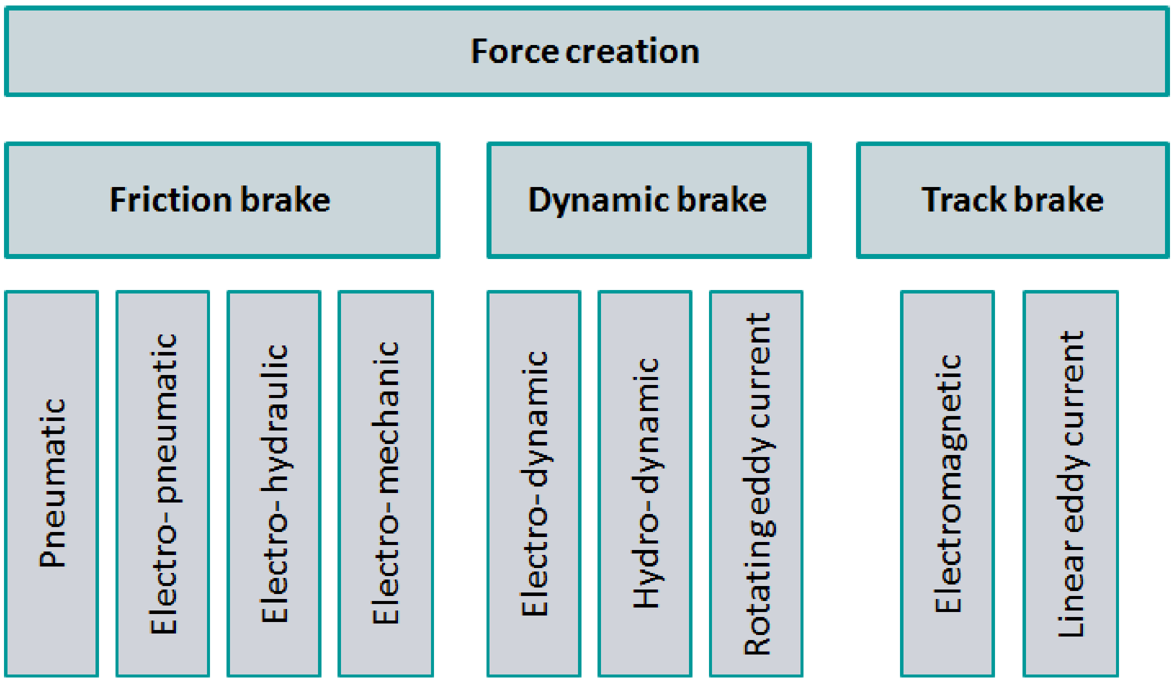


Figure 12 Force creation mechanisms (Heckemanns, 2010)

This view has a certain disadvantage. While dynamic brakes and track brakes apply their forces directly to slow down the vehicle, friction brakes possess of an intermediate step. The actuator force derived by one of the four principles of force creation is transformed into frictional force by mechanical means. The frictional force actually slows down the vehicle. Therefore a second possibility to describe the braking systems is based on the type of the brake itself.

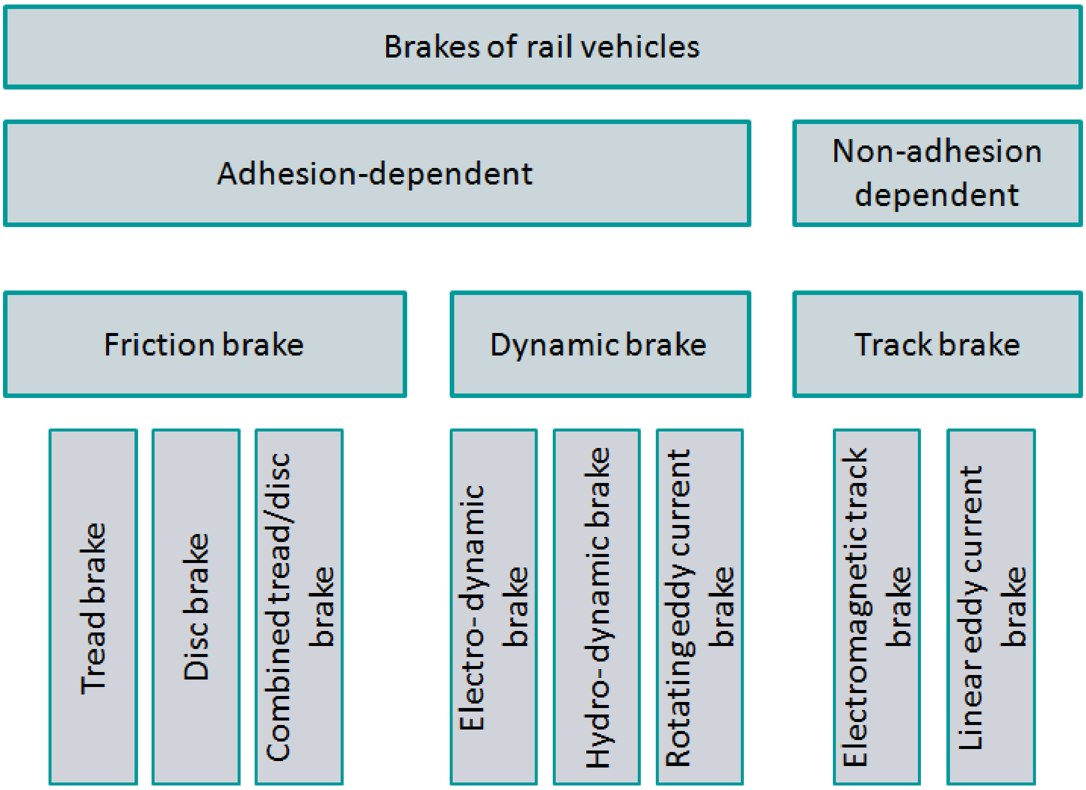


Figure 13 Categorization of rail vehicle brakes based on (Knorr-Bremse, 2002)

This figure classifies the different brake systems by means of the design of the brake.

Friction brake

As mentioned above brakes based on friction possess of an internal force creation principle and a friction creation concept. First there will be an explanation on the internal force creation mechanisms. Right after that the tread and the disc brake, representing the friction creation type, are described.

Simply put, a friction brake is a brake that uses brake pads that affect the wheel tread or a brake disc. (DIN EN 13452-1, 2003)

Pneumatic / Electro- pneumatic

As the name says, the pneumatic force creation is done using compressed air. (Knorr-Bremse, 2002, p. 23) defines the compressed air brake as follows:

A compressed air brake is a brake in which brake force is created by compressed air in a brake cylinder.

Such brakes have been used for decades in railway vehicles. The compressed air brake was the next evolutionary step in brake development after handbrakes such as lever brakes or such operated by cranks and screw spindles. The first compressed air brake used in 1868 was introduced by Georg Westinghouse. However, not until the beginning of the 20th century the compressed air brake got popular. (Knorr-Bremse, 2002, p. 183)

Nowadays, this long development time enables manufacturers to provide sophisticated solutions that are fail proof. Hence the pneumatic brake is still used in standard- gauge railways. The usage of such brakes is mandatory in international traffic. Binding regulations on technical and operative details are provided by UIC leaflets, yet these are not valid for metro vehicles. (Knorr-Bremse, 2002, p. 23) The UIC (Union Internationale des Chemis de Fer) consists amongst others of the European railway administrations. (Saumweber, Gerum, & Berndt, 1990, p. 91)

Compressed air brakes can be separated according to their working principles in direct-acting brakes and indirect- acting brakes as mentioned previously. Additionally in terms of characteristics and features a division is possible:

- *Automatic brakes / non-automatic brakes*

Automatic refers to the ability of the brake to apply force automatically in case of pressure drop in the brake pipe. On the one hand all indirect brakes can be considered as automatic. Direct brakes on the other hand are non-automatic, because they do not react to loss of brake pipe pressure.

- *Direct release brakes / graduated release brakes*

On the one hand direct release brakes are not able to interrupt or abort the venting of the brake cylinder. If the application is placed, the brake gets vented completely. Graduated release brakes on the other hand are able to interrupt the venting. Also a new brake application can be performed even if the brake is not released completely. Graduated release brakes are mandatory according to UIC.

- *Single chamber compressed air brakes/ dual chamber compressed air brakes*

The piston of the brake cylinder in single chamber compressed air brake is impinged only from one side. The piston return is provided by a spring in the opposite cylinder chamber. With the dual chamber, the piston is impinged by compressed air on both sides. Both sides of the piston are connected via grooves. One side is enlarged with an air reservoir. In the release position on both sides the same pressure is available. In case of a braking application the pressure in the brake pipe is reduced and the piston moves due to the pressure derived from the air reservoir.

- *Exhaustible brakes / inexhaustible brakes*

If there are multiple braking applications to perform within a short time, an exhaustible brake is in danger to run short of compressed air because the supply is too slow to refill. The air reservoir pressure can dramatically drop and therefore a reduction of the braking effect is to be expected. The inexhaustible does not have this problem. With indirect brakes the refilling has always priority before a new brake application is possible.

- *Single pipe brakes / dual pipe brakes*

With single pipe brakes the existing pipe is strictly reserved for signal and energy transmission for the brake. However, there may be other air consuming equipment in a railway vehicle. In order to provide air for this equipment as well a second pipe can be implemented. Then the air reservoirs of the brake can be filled using two pipes.

- *Slow-acting brakes / fast-acting brakes/ fast and strong acting brakes*

Here the brakes are classified according to the different time characteristics of the pressure change. It is used to control longitudinal forces in long trains with graduated release brakes. The aim is always to provide an as similar as possible pressure process in the front and the rear of the vehicle.

(Knorr-Bremse, 2002, pp. 23-30)

Considering the several ways of classification, not surprising the UIC leaflets set up a standard UIC brake. This brake is an automatic, indirect acting, graduated release brake. Signal and work transmission medium is certainly compressed air. Also components are defined for the UIC brake, like:

- A compressor with air drying to provide the working media
- A drivers brake valve, to create the signal for the braking application
- A brake pipe, to distribute the compressed air
- An air reservoir, to provide a pressure in case of the brake application
- A controlling valve, to transform the signals into cylinder pressure
- A brake cylinder, to transform pressure into braking force
- A brake rigging or brake caliper including a slack adjuster

This basic brake the UIC single pipe brake can be extended to other types like the UIC double pipe brake. Basically the idea is always the same. The compressed air is distributed and used to move the piston in the brake cylinder. This in turn causes the brake force. (Gralla, 1999, pp. 38-40)

Considering an electro-pneumatic brake the brake signal is not transmitted via the compressed air, electric signals are used. These signals have to be converted by the electro-pneumatic brake into pressure. This is done by using analog or digital converters.

The analog converter enables the signal to create a force using an electromagnet. This force controls the relay valve to provide the desired venting pressure. A second relay valve converts this pressure into the cylinder pressure. The problem of this system is the deviation of the hysteresis and nonlinearity due the electromagnet and internal mechanical friction. A feedback control loop has to be implemented in order to achieve better accuracy.
The control established by a digital converter provides various control steps. Seven are most common. (Saumweber, Gerum, & Berndt, 1990, pp. 95-96)

However, the most important thing about electro-pneumatic brakes is that even with very long trains the braking application is present on each brake at the same time. Thereby the propagation time and the pressure wave can be eliminated. Also the dynamic longitudinal forces in the vehicle are minimized. (Knorr-Bremse, 2002, p. 74)
Basically electro-pneumatic brakes can be designed as overlay systems to the UIC brake or as independent systems. Considering an overlay to the UIC brake more equipment has to be implemented. One of them is the signal generating unit right after the brake valve. Moreover the signal transmission via wiring has to be provided. Of course the signal interpreting unit has to be present.
Similar to compressed air brakes the electro-pneumatic brakes can either be direct or indirect. (Knorr-Bremse, 2002, p. 74)

Nevertheless, it is very important to know that no matter how useful the UIC brake might be its regulations only apply for standard gauge railways. For mass transit there is no UIC brake available. Furthermore no comparable regulation to the UIC is established. Therefore each manufacturer can come up with its own solution. Some basic requirements are known that need to be proven before a mass transit vehicle can be sold, though.

- Vehicles have to be coupleable from a mechanical point of view
- If the brake control is out of service automatic braking has to be provided
- Low-active
- Simple emergency control has to be provided to remove broken vehicles
- Brake pipe design in a way that other vehicle can be attached to provide pressure supply in case of tow is necessary

(Gralla, 1999, p. 48)

Hydraulic/ Electro- hydraulic

In prototypes sometimes hydraulic systems are used, though they are not very common. In contrary to the electro-hydraulic systems these systems use compressed air for signal transmission. Of course a pneumatic/hydraulic converter is needed. (Gralla, 1999, p. 107)

The term ‘electro- hydraulic force creation’ refers to an electrically controlled brake with hydraulic force creation. The braking force is the result of oil under pressure. This system is mainly used in trams. (Knorr-Bremse, 2002, p. 200)

The basic idea of the electrical control in an electro- hydraulic brake is equal to the one of electro- pneumatic brakes. However, some more monitoring functions are needed, such as (1) gas pressure monitoring in the reservoir, (2) leakage monitoring and (3) monitoring of oil volume in the reservoir. (Saumweber, Gerum, & Berndt, 1990, p. 96)

Furthermore, just as in pneumatic systems hydraulic systems can be designed two-tiered or continuously variable. The decision on the design is based on the question whether only loaded or unloaded conditions are possible or if also partly loaded settings are likely to occur.

The brake application is transmitted electrically via the train bus system. The application can either be converted by the vehicle electronics to an electronic control signal for the pump or the magnetic valve or with the usage of a compact control unit in a hydraulic brake cylinder pressure. The actuator is equipped with separate units that create force to complete the braking application. This separation makes the integration of service, rapid stop and emergency brakes as direct brakes and a spring loaded indirect brake possible.

The three phases, which together define the braking distance, certainly are valid for an electro- hydraulic brake. In comparison to the pneumatic brake the reaction phase is shorter due to the electronic signal transmission. The signal transmission time can be neglected. Furthermore reaction time is shorter using a control unit than with a pneumatic control valve. Due to this reduction higher speeds are possible from a braking perspective. Also the rising phase can be influenced by means of software. This is object to consider because the rising phase has an influence on the jerk. (Gralla, 1999, pp. 109-110)

Three types of system architectures are known when talking about electro- hydraulic brakes.

(1) Central hydraulic system

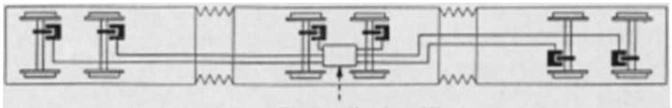


Figure 14 Central hydraulic system (Kipp, 1995)

The biggest advantage are low system costs. However some disadvantages are known, like missing redundancy and high effort for piping, especially in the area of joints. This might also be the reason why manufacturers try to avoid this design. (Kipp, 1995), (Gralla, 1999, p. 108)

(2) Bogie hydraulic system

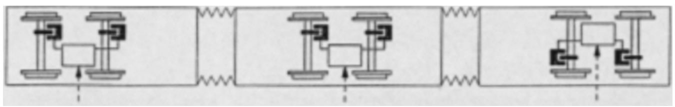


Figure 15 Bogie hydraulic system (Kipp, 1995)

The bogie hydraulic system avoids the disadvantages of central hydraulic systems. Each bogie can be seen as an independent hydraulic island. The result is less usage of piping. (Kipp, 1995), (Gralla, 1999, p. 109)

(3) Intelligent actuator

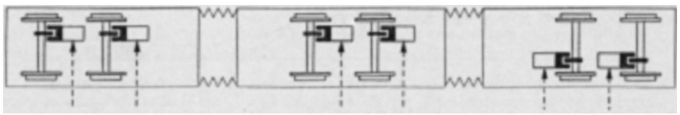


Figure 16 Intelligent actuator (Kipp, 1995)

In case an intelligent actuator is used, the hydraulic is only present within the system. No piping is necessary in the whole bogie. The system can be considered as compact. Still it needs large installation space. Furthermore costs increase due to the multiplication of components. According to Gralla (1999) this is one of the reasons why a bogie hydraulic system is given preference by the manufacturers. However the biggest advantage of an intelligent actuator is that there will never be a whole bogie out of service in case of hydraulic breakdown, in fact only one actuator will be idle.

Electro- mechanic

According to Knorr- Bremse (2002) the electromechanical brake can be of the following characteristic: *(1) Brake in which the brake force is created by means of an electrical drive motor. (2) Brake in which the brake force is created by the magnetic power of a solenoid.*

More detailed, the clamping force is generated by electrical means. As mentioned previously, electrical motors equipped with a reduction gear unit or solenoids can be used. (Saumweber, Gerum, & Berndt, 1990, p. 96) The force is generated directly in the actuator unit. (Gralla, 1999, p. 107)

The control is of equal structure as with electro- pneumatic brakes. (Saumweber, Gerum, & Berndt, 1990, p. 96)

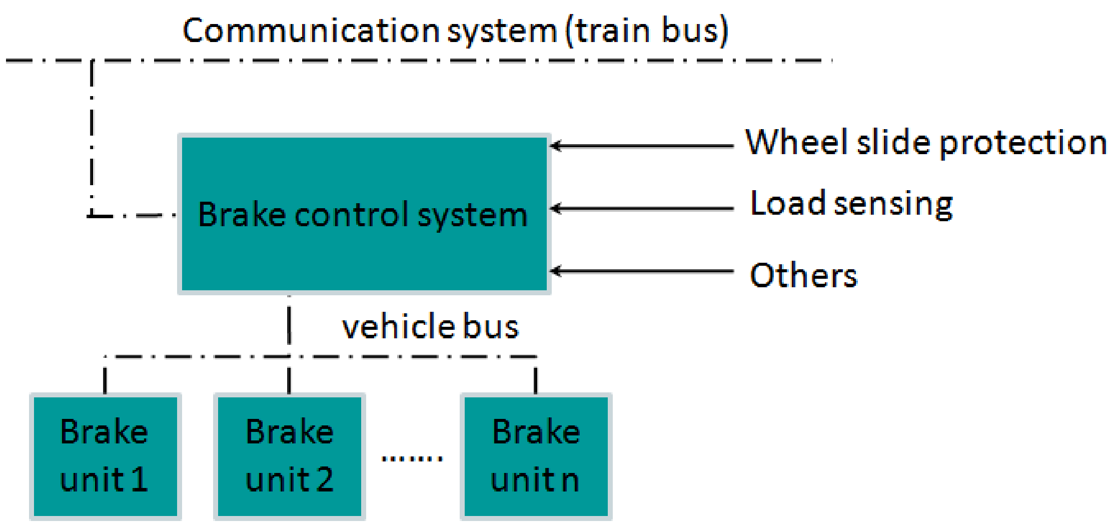


Figure 17 System of an electro- mechanic brake based on (Gralla, 1999)

Considering the advantages, most important about electro- mechanic brakes is their ability to represent a possibility to accomplish an all- electric vehicle. Furthermore, compact design, easy installation, low maintenance effort and low environmental impact are positive effects of this system. Certain disadvantages are known as well. At the moment this principle is mostly available for disc brakes. Moreover electrical energy is needed and it is not possible to provide the vehicle with a fallback level.

In view of braking distance, all the details mentioned in section Hydraulic/ Electro- hydraulic can be carried over to electro- mechanic brakes as well. (Gralla, 1999, pp. 110-111)

Tread brake

The most popular friction brake is the tread brake; especially when it comes to transportation of cargo. Via a brake rigging the braking force is transmitted as evenly as possible to the brake shoe inserts. The rigging also provides power amplification for the brake force. To provide a sufficient shoe clearance over the lifespan of the brake shoe lining some sort of adjusting mechanism is implemented as well.

The special characteristic of the tread brake is that the brake energy is converted to heat between the brake shoe inserts and the running surface of the wheel. Not surprisingly the braking is caused by friction. However the resulting heat has to be drawn off via the wheel and the brake shoe. It is very important that the wheel is not negatively affected during continuous braking. Otherwise the wheel cannot endure the demands that result due to bearing and guiding the vehicle. Especially with increased speeds and high axle load limits the demand on the wheel has been enlarged in the last few decades. This fact certainly limits the load a wheel can additionally bear due to thermal stresses and wear caused by braking. (Knorr-Bremse, 2002, pp. 86-87) (Saumweber, Gerum, & Berndt, 1990, pp. 39-40)

Basically there are two types of tread brakes concerning their actuation. On the one hand there is the type which uses a brake rigging and on the other hand there are entire systems called tread brake units.

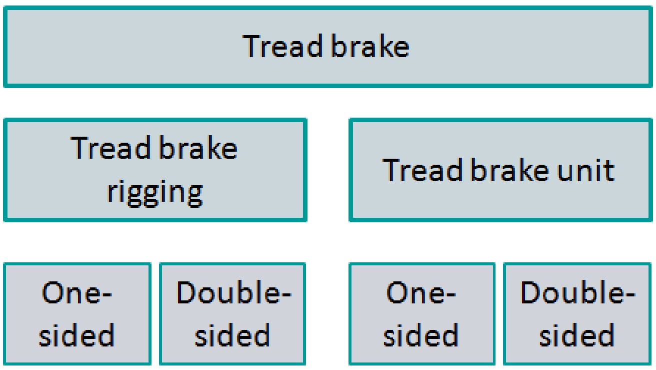


Figure 18 Overview of tread brake equipment components (Knorr-Bremse, 2002)

The tread brake which uses a tread rigging is commonly installed in vehicles used for cargo transportation. If there are two to four axles on the vehicle mostly one brake cylinder is sufficient. Only difficult assembling situations prevail an additional cylinder may be implemented. The rigging consists of several levers, shafts, and joints. Certainly the degree of efficiency suffers with a design like this. It is supposed to be between 80% and 90%. (Knorr-Bremse, 2002, pp. 86-87) (Saumweber, Gerum, & Berndt, 1990, pp. 39-40)

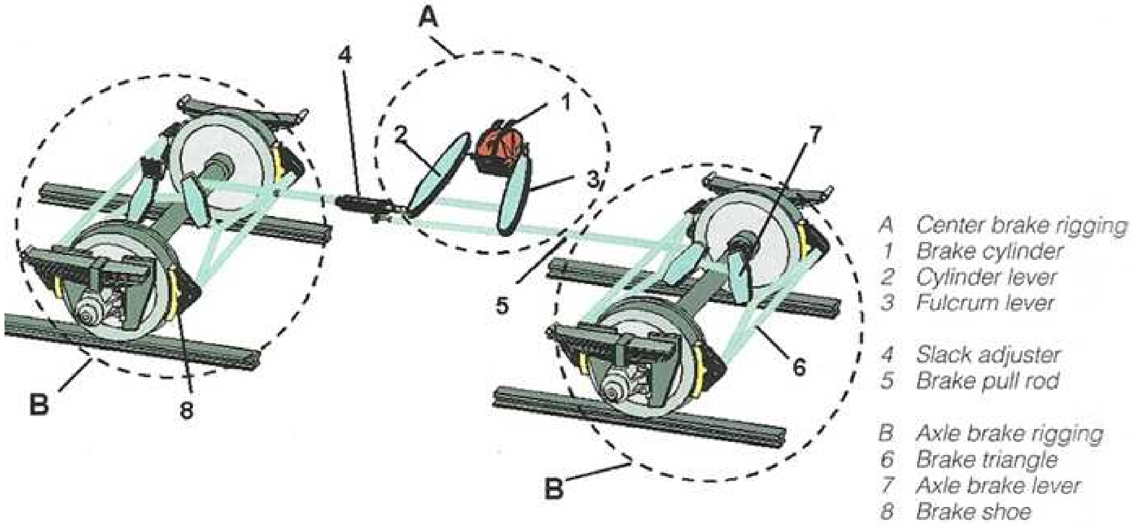


Figure 19 Example for brake rigging (Knorr-Bremse, 2002)

Tread brake units combine brake cylinder, lever transmission, adjusting mechanism, and brake shoe to one single unit. This is very space-saving compared to the brake rigging. However, the braking force can be adjusted via the cylinder pressure and/or the type of lever transmission. Another upside is that tread brake units can be equipped with a spring-loaded brake in order to enable the parking brake application. (Knorr-Bremse, 2002, p. 155)

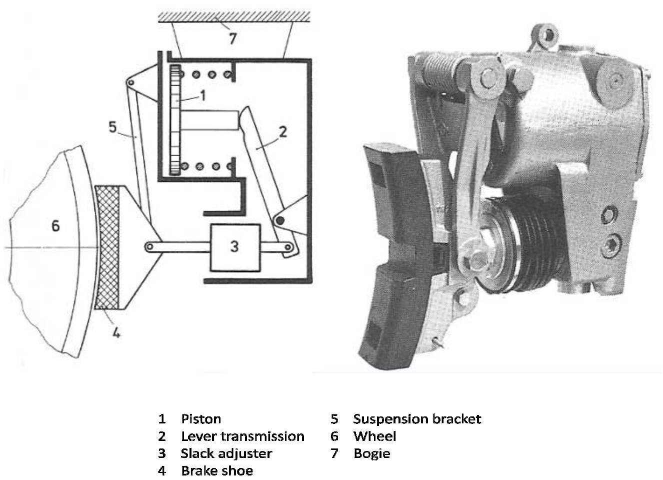


Figure 20 Tread brake unit (Saumweber, Gerum, & Berndt, 1990)

A special advantage of a tread brake is the simple design. It uses the wheel for heat transfer and as the friction partner. Moreover the positive cleaning effect on the wheel running surface has to be mentioned. This improves the adhesion between track and wheel. Brake tests can be carried out optically and the wear situation of the brake shoe insert is visible with very low effort. (Knorr-Bremse, 2002, p. 87)
According to experience at Siemens, the tread brake provides a weight advantage. Up to 130 kg can be reduced due to not installing brake discs.
Considered as negative is the high demand on the wheel surface as mentioned previously. Additionally tread brakes are quite noisy, especially when grey cast iron brake shoe inserts are used at low speed. Furthermore the brake rigging is responsible for noise emissions during the braking applications due to the high tolerances, too. (Gralla, 1999, p. 82)

Disc brake

A special characteristic of the disc brake is the transformation of braking energy to into heat between the brake pad and the brake disc. The heat is dissipated by cooling fins. Vented axle-mounted brake discs and wheel-mounted brake discs are most common. (Knorr-Bremse, 2002, p. 88)
Generally speaking the disc brake surpasses the tread brake due to several reasons: The performance limit of the tread brake, especially at high velocities, can easily be exceeded. Additionally the driving comfort is higher and the friction coefficient is less dependent on pressure and speed. (Saumweber, Gerum, & Berndt, 1990, p. 49)

The following paragraphs are based on (Knorr-Bremse, 2002, pp. 156-160), (Gralla, 1999, pp. 87-89) and (Saumweber, Gerum, & Berndt, 1990, pp. 49-51).
There are three different ways to fix the disc brake to the bogie. Hence it is distinguished between axle-mounted disc brakes, wheel-mounted disc brakes and gear brake discs.
The axle-mounted disc brakes are integrally molded to the axle. It is necessary to design this press fit in a way that it can sustain the transmission of the braking torque even at elevated temperatures. Axle-mounted disc brakes can either be made of one mono block or they are a combination of friction ring and hub. The friction rings can be either divided or undivided. Divided rings enable the removal without taking the wheel body from the wheel shaft. Undivided discs are used if low thermal stresses are expected. Their major advantage is their ability to transmit braking force and the transmission of accelerations when crossing track switches. However, high forces are created within the radial fixing due to the thermal expansion. Hence the thermal performance limit of undivided discs is lower than that of divided. Basically according to Siemens experience it is common the sell the first setup of the bogie with undivided discs, because they are

cheaper. In case the wheel is still serviceable but the brake disc is not at the inspection, divided rings are implemented.
As mentioned previously, most disc brakes are vented. These usually have two friction surfaces that are connected by ribs. In some cases the ribs are allocated in a way not to allow only radial but traveling direction through flow of air, too.
It is necessary to go through the two major problems concerning the venting of axle-mounted disc brakes. On the one hand that is power loss due to the turbine effect. On the other hand the heat gets transferred from the first bogie axle right to the next one. Higher surrounding temperature should be expected at the second axle.

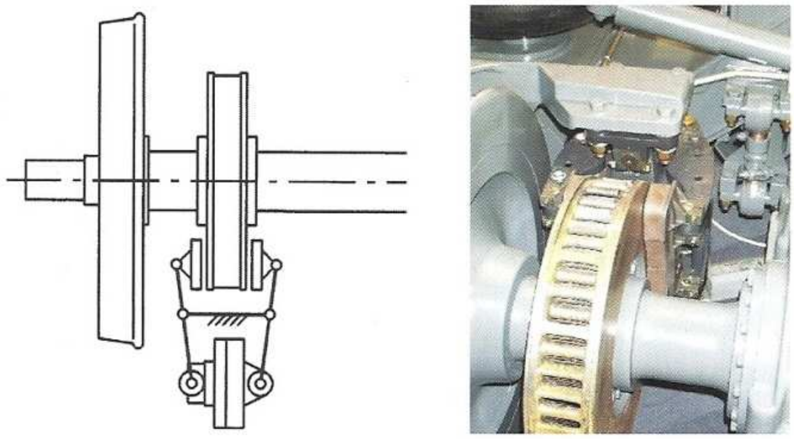


Figure 21 Axle-mounted disc brake (Knorr-Bremse, 2002)

As mentioned before another possibility is represented by the wheel-mounted disc brake. This type is used whenever there is no space to implement axle-mounted disc brake in the bogie. This can happen on motor bogies and wheels with small diameter. The brake disc is build up by two friction rings. These are positioned on both sides of the wheel using a key plug centering. Again, also these friction rings can be divided or not divided.

Wheel-mounted brakes have the advantage that their diameter is linked to the diameter of the wheel. It is not necessary to consider the clearance height. Theoretically the braking performance of a wheel-mounted disc brake can be increased by a larger friction radius, or in other words the same braking performance can be realized with lower stresses. In reality however, several problems occur, like problematic maintenance, cracks are tricky to detect, or difficult heat transfer in the gap between the brake disc and the wheel. In comparison to the axle-mounted disc brake the wheel-mounted disc brake has to sustain higher thermal stresses. Hence, the specific performance limit of wheel-mounted is about 20% less than that of an axle-mounted.

According to Siemens experience the decision on wheel-mounted or axle mounted also depends on the application of the railway vehicle. If majorly long term brake applications are demanded, the axle- mounted is given preference. For short time braking with high necessary forces, the wheel mounted is the better choice.

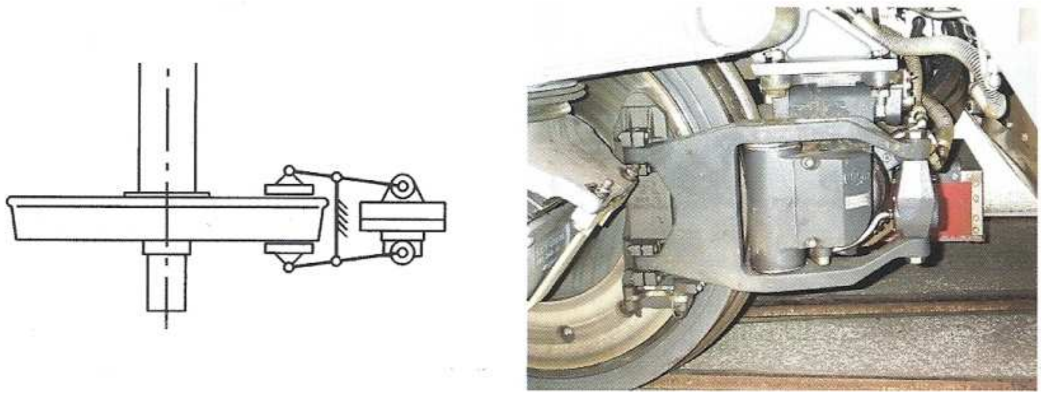


Figure 22 Wheel-mounted disc brake (Knorr-Bremse, 2002)

Gear brake discs are used when there is no space for proper installation neither for axle nor for wheel-mounted discs. Hence, gear discs are assembled to fast spinning gear shafts. A high brake performance is achievable with relatively light and small discs. However the radial stresses are very high in these discs.

Brake rigging is also possible with brake discs. The appearance depends on the number of discs which are pressurized by one cylinder. However, nowadays brake caliper units are most popular. The brake caliper unit consists of a slack adjuster, a brake cylinder, and a brake rigging, used as caliper. Moreover these units can be equipped with a spring loaded brake.

Certainly the disc brake itself implicates several advantages. In comparison to tread brakes the wheels are not affected by heat. Moreover the material combination between the friction partners can be solely defined with respect to the braking process. Therefore, higher brake performance is possible in comparison to tread brakes. Moreover, a disc brake has a better jerk behavior than a tread brake. Disadvantages are the necessary efforts for equipment when running braking tests and checks concerning the wear of the brake pads. (Knorr-Bremse, 2002, p. 88), (Gralla, 1999, p. 87)

Spring loaded brake

As mentioned previously the spring loaded brake is used to perform immobilization brake applications.

Its braking force is not allowed to decrease over time and certainly it has to be inexhaustible. (Wende, 2003, p. 220)

It can either be attached to a tread brake unit or to a disc brake actuator.

In order to visualize a possible design opportunity of a spring loaded parking brake attached to a tread brake unit Figure 23 is provided:

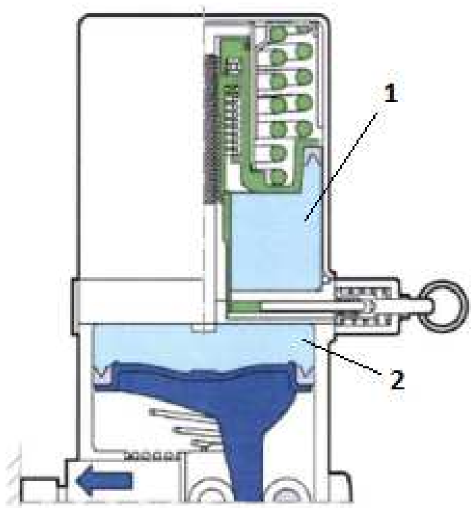


Figure 23 BFC Tread brake actuator with spring loaded parking brake (Faiveley Transport, 2007)

Basically the spring loaded brake is using spring load to accomplish the brake application. Figure 23 shows the spring actuator (1) and the compressed air service brake actuator (2). The present spring actuator is in charged condition. The springs (green) are compressed by the air (blue) in the chamber. If the chamber is vented and the air pressure decreases, the springs will expand and create the force needed for the immobilization brake application. (Faiveley Transport, 2007, pp. 12-13)

A detailed description on the operating mode of the Faiveley Transport BFC Tread Brake can be found in section 3.2.4.

Basically after initial startup of a railway vehicle equipped with spring loaded brake, the brake is loaded either pneumatically (hydraulically) or electrically and locked. In case of stop or service brake failure the spring loaded brake is activated. The spring load is now affecting the brake rigging. (Wende, 2003, p. 223)

Emergency release

It is reasonable to install an emergency release in case a spring loaded brake is used. In case the spring loaded brake is applied and cannot be released by the basic service mechanism, the emergency release can provide the necessary clearance of the brake pad or brake shoe. Using the emergency release it is possible to unfasten the spring loaded brake, basically the spring is relived from pressure. In most causes a mechanic mechanism is used for this purpose. The activation is usually direct at the actuator. This fact sometimes cases trouble, in case the brake actuator is hardly reachable. (Hobaum, 1998) Today this problem is solved using bowden controls.

Certainly to fulfill upcoming brake applications during the next assignment the spring needs to be loaded in advance again.

However the emergency release is useful to prevent wheel flats because unwanted blocking of the wheel is prevented.

Slack adjuster

Slack adjusters are necessary to compensate the wear of all parts related to braking in friction brakes - from the actuator to the friction partners, like tread brakes, brake shoes, brake linings, wheel tread, and joints and plugs of the brake rigging. The wheel and the friction lining have most important influence on the brake shoe clearance. (Knorr-Bremse, 2002, pp. 149-150)

A too large clearance is always negative. Especially on the spring loaded tread brake used in this thesis the clearance has an important influence. The spring has a characteristic of force over distance, as mentioned in section 3.2.4. If the spring deflection increases, the output force of the spring is reduced and the necessary braking force cannot be provided anymore.

Basically automatically-functioning slack adjusters are used nowadays. They can either be single-acting or double-acting. In case clearance as a result of wear should be adjusted, the single-acting adjuster is sufficient. The double-acting is used if the brake shoe clearance should be adjusted according to the load condition.

It is possible to differentiate between single- & double-acting clearance adjuster, single-acting stroke adjuster, and single-acting return joke limiter. (Knorr-Bremse, 2002, pp. 150-151)

According to a literature research, there are mechanical, electro-mechanical and hydraulic slack adjusters known. Considering tread brake units, the sack adjuster is located in the brake rigging. The current benchmark, the Faiveley BFC consists of a mechanic slack adjuster.

Dynamic brake

Basically dynamic brakes are systems that use the reverse function of parts of the power transmission in a vehicle. (Gralla, 1999, p. 102)

Electro- dynamic

Put simply, electro- dynamic force creation is used in electrical powered motor vehicles. The electro- dynamic brake works wearless. The basic idea is that in case of a braking application the electrical drive motor works as generator. It converts kinetic energy of the vehicle into electrical energy. (Knorr-Bremse, 2002, p. 175), (Wende, 2003, p. 224)

Electrical drive engines can be used very well for braking if a constant braking force is provided during the whole speed range. Certainly this is not possible when it comes to lower speeds or saturation. However during certain velocity ranges it is possible to achieve constant braking force.

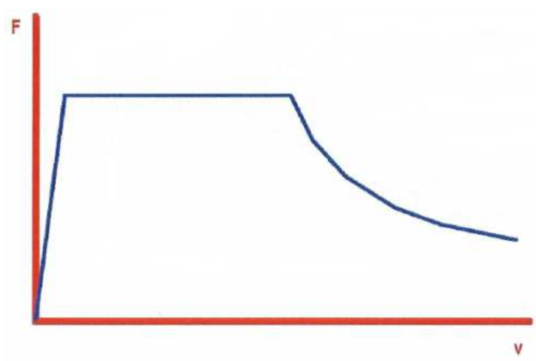


Figure 24 Theoretical braking force characteristic of an electro-dynamic brake based on (Heckemanns, 2010)

Considering the braking circuit two possibilities are available. (1) rheostatic brake circuit and (2) regenerative brake circuit. A rheostatic brake transforms the braking energy into heat, while regenerative brakes feed the braking energy in form of electrical energy back into the grid. Rheostatic brakes are favored when it comes to standard-gauge railways which are used on railway tracks with only a few sloped sections and large distances between the stations. In this case parts of the brake like resistors and ventilators can be designed for short-term operation. Moreover, regenerative circuits demand for more additional efforts than rheostatic does.

Interestingly in the early days of the regenerative brake energy recovery was not the main focus. The idea was to use the power grid as alternative braking resistor. Today regenerative braking is used in rapid transit and new developments in circuit technology may improve this idea.

Altogether the electro dynamic brake has several important characteristics:

- Wearless and low necessary maintenance
- Proper brake dynamic and thereby good controllability
- Adaption of braking force characteristic to certain demands using different circuits
- Possibility to feed the grid when using regenerative brakes.

(Saumweber, Gerum, & Berndt, 1990, pp. 56-57)

Aside these characteristics it has to be considered, that the electro- dynamic brake still depends on the wheel track force closure. Hence, the possibility of overbraking exists in case of combined brakes. (Knorr-Bremse, 2002, p. 175)
Another important issue becomes vital if the electric motor brakes down. In this case the braking application cannot be performed by the motor as well. However the vehicle can still be stopped if the electro dynamic brake is combined with e.g. a pneumatic brake.

Certainly it is vital for the electro- dynamic brake to have working resistors. If they miss because of e.g. a lowered pantograph, a catenary wire voltage breakdown, or a switched off main switch the transformation into electrical energy is not possible. Usually probability calculations are done in order to assess the risk. Currently there is a discussion on how the brake ability is credited to the whole system. (Gralla, 1999, p. 104) In 1999 it was credited to 50% but with new fallback policies this percentage is supposed to increase dramatically. See Chapter 3.2 for details.

Considering the passenger traffic, the electro-dynamic brake is able to provide the major part of the necessary service brake force. When it comes to freight transportation the electro-dynamic brake takes over the gradient brake and basically everything that is related to the conservation of state of motion. The other brake applications are done – in most cases – by the pneumatic braking system which is additionally installed in the vehicle. (Saumweber, Gerum, & Berndt, 1990, p. 56)

Hydro- dynamic

This system is used with motor vehicles that possess a hydraulic force transmission. Implementing a brake clutch the hydrodynamic transmission can be used as a wearless brake. The braking force is controlled by the liquid level. (Knorr-Bremse, 2002, p. 176)

There are two types of hydrodynamic brakes used in rail vehicles: firstly the (1) hydraulic retarder and secondly the (2) hydrodynamic converter.

(1) The hydraulic retarder

This fluid flow machine consists of a rotor and a stator; together they create a ring shaped working space. The rotor is connected to the power train. The balding on the rotor accelerates the fluid in the machine. Using turbulences the fluid is slowed down again in the stator. This process generates heat. The heat is transmitted via the housing to a cooling circuit. (Saumweber, Gerum, & Berndt, 1990, pp. 60-61), (Gralla, 1999, p. 105)

The braking torque can be varied using different liquid levels. Another possibility to vary the braking torque could be provided by adjusting the balding position.

Six different designs are known:

- Not adjustable
- Adjustable
- Switchable

Each assembled as single tier or double tier.

The effectiveness of a hydro-dynamic brake is to a large extent related to the heat conduction ability. Hence a temperature control has to be implemented in the retarder. As soon as a certain temperature is exceeded, the machine is taken out of service. (Gralla, 1999, p. 105) Cooling circuits have to be introduced in order to keep the temperature low. These circuits are easy to establish in diesel hydraulic motor coaches, because a cooling system is already installed. The retarder can be introduced to this system at relatively low effort. If trailer coaches are considered, the whole cooling system has to be implemented as well. This is the reason why retarders are rarely used in trailer coaches. (Saumweber, Gerum, & Berndt, 1990, p. 62)

Nevertheless retarders are capable of providing sufficient braking torque even with small installation size. If the liquid level is at its maximum the braking torque is quadratically increasing with the velocity. The constant torque needed for a retarding brake application is provided by the liquid level. At high velocity, only little liquid is needed. If the speed decreases, the level is increased until the whole unit reaches its maximum liquid level. In Figure 25 this situation is marked as A_i. After this point the torque decreases and a brake application needs to be supported by other brake systems, like a friction brake system.

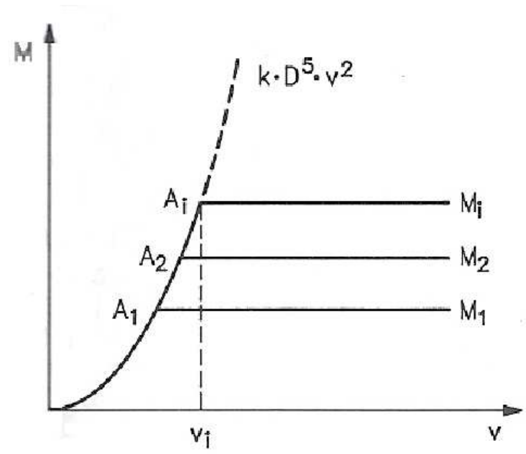


Figure 25 Torque characteristic of a hydrodynamic brake (Saumweber, Gerum, & Berndt, 1990)

Figure 25 clearly shows that the retarder is not able to perform a brake application until the vehicle stops. Hence this is perfectly suitable for gradient brake applications. (Gralla, 1999, p. 106)

Other advantages of the hydro dynamic brake are the good controllability, the wearless braking, and the good heat transfer possibility. A downside is the tractive resistance that is present even if the retarder is unfilled. (Gralla, 1999, p. 105)

(2) The hydrodynamic converter

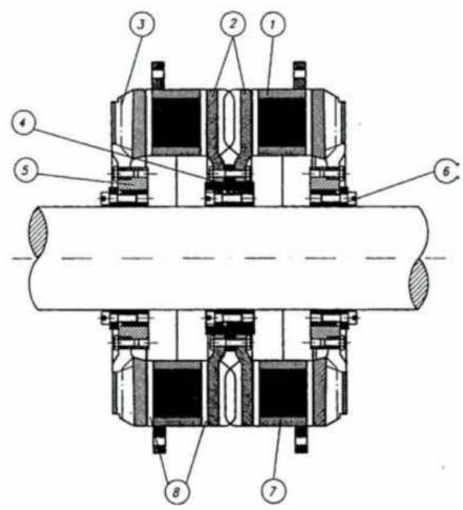
It represents a special design. When forward motion is present and a brake application is given, the fluid flow converter for backward motion is filled with fluid until stoppage. If the forward motion converter is defiled right after, the motion direction is reversed. (Gralla, 1999, p. 106)

Basically the hydrodynamic converter consists of a pump (P) which is powered by the motor, a turbine (T) that activates the wheelset, and a fixed guide vane apparatus (L). If the converter is supposed to power the rail vehicle, the mechanical power from the engine is converted into flow rate and by means of the turbine it is converted into mechanical power again. In case of a braking application the flow against the turbine blades is of the opposite direction to the drive. Therefore high losses and turbulences arise. Just as with the hydraulic retarder the fluid amount in the converter decreases with increasing speed. The converter's torque is controlled by the pressure within the system. It is very important, that the converter can only be used as brake if the pump power is provided by the drive motor. This power is around 10% of the nominal power. Furthermore in this system the heat creation due to braking influences the fluid the most and therefore a sufficient cooling system is vital. (Saumweber, Gerum, & Berndt, 1990, pp. 62-63)

Rotating eddy current brake (RWB)

The rotating eddy current brake is a member of the family of the dynamic brakes. It is further force closure dependent, because the braking torque is transferred between wheel and track. (Knorr-Bremse, 2002, p. 100)
The rotating eddy current brake was hardly of interest in Europe due to the force closure dependence. Still in Japan some test are made. (Gralla, 1999, p. 55)
Basically the usage of rotating eddy current brakes is not unconventional, especially with trucks. However, the idea to use such brakes within railways was dismissed many times. In the 80ies the reason lay in the material composition of the rotor and the maintenance of the air gap between rotor and stator. Nowadays these problems are overcome because of the development related to the usage of rotating eddy current brakes in trucks. (Sonntag & Jaensch, 2005)

The working principle is similar to the disc brake. (Gralla, 1999, p. 55) The braking is done by using a brake disk fixed to the axle, which penetrates a magnetic field continuously. (Wende, 2003, p. 224) More in detail, the brake consist of a rotor and a stator. The magnetic field is created by the stator. The rotor is represented by the brake disk. In case of a braking application the stator receives electric current, which creates a magnetic field. This field is intersected by the rotor. Using the Lenz's law, a force is created that opposes the rotating direction of the rotor. Certainly eddy currents in the disk are induced as well. These currents heat up the rotor and therefore sufficient cooling is necessary. This is often realized by self-ventilation of the rotor. (Sonntag & Jaensch, 2005)



1 = stator; 2 = inner rotor; 3 = outer rotor; 4 = inner sleeve; 5 = outer sleeve; 6 = fastening system; 7 = coil; 8 = pole plate

Figure 26 Cross- section of a RWB (Sonntag & Jaensch, 2005)

An important advantage of a rotating eddy current brake is the high thermal resistance. When it comes to high speed trains and trains that have to pass mountain routs with long sloped tracks that ask for extensive braking effort (e.g. Gotthart track) the rotating eddy current brake is perfectly suited. (Gralla, 1999, p. 55) The RWB allows full speed at the descents, certainly considering the route and train formation and if there are numerous speed changes or lots of stops the RWB is a good choice, to disburden the friction brake. Moreover the stops from high speeds can be supported. The friction brake usually engages at 40km/h, until this level the RWB is active.

It needs to be pointed out that a bogie decelerated by an RWB will never have wheel slide. The reason is simple; braking force is only applied as long as the rotor turns. The rotor turns only if the wheel turns because both are connected via the axle. Also if the rotation speed falls below a certain level, there is no braking force introduced to the rotor. This advantage pitifully has a disadvantage, too. A friction brake is needed to stop a vehicle completely. (Sonntag & Jaensch, 2005)

In comparison to the linear eddy current brake the RWB has several advantages. Most important is the fact that it does not influence the track. While linear eddy current brakes heat up the track and therefore introduce stresses into the continuously welded track, the rotating eddy current brake only creates heat in the rotor. Furthermore, the RWB can be implemented in vehicles that are already used, during defined speed ranges the RWB can be used solely, the number of friction brakes can be reduced in the vehicle configuration and the RWB can be used on any kind of track. (Sonntag & Jaensch, 2005)

Hence other advantages can be according to (Sonntag & Jaensch, 2005):

- *No wear*
- *Almost silent (< 30dB)*
- *No dust generated from braking*
- *Responsive to controls*
- *Almost maintenance free*
- *Durable equipment*
- *No effect on track mounted control equipment*

Certainly the RWB also has some disadvantages. It is not possible to implement a parking brake. It simply does not work with this principle. It is not possible to eliminate all the friction brakes, either. The RWB can only be considered as a supplementary brake. The rotors are heated to more than 400°C what implies high material demands. Compared to friction brakes, more space is needed, higher weight can be expected and increased capital costs. (Sonntag & Jaensch, 2005)

However, there is research done considering the implementation of freight vehicle RWB brake implementation in railway vehicles by e.g. the Deutsche Bahn AG. Research results, presented in (Sonntag & Jaensch, 2005), proof that the RWB is a good opportunity.

Track brake

As mentioned above the track effecting brakes apply the braking force directly to the track. This idea does not affect the force closure between track and wheel. These brakes can be used to increase the braking effect additionally to friction brakes. (Knorr-Bremse, 2002, p. 92)

This advantage is especially useful when it comes to surrounding conditions that have a bad influence on the adhesion coefficient. It can happen that rail vehicles, dealing with such conditions, cannot stop within a given distance. Situations like this demand force closure independent brakes like electromagnetic track brake and linear eddy current brake. (Saumweber, Gerum, & Berndt, 1990, p. 64)

Electromagnetic track brake

The electromagnetic track brake is mainly used in passenger cars that travel with more than 140 km/h. (Saumweber, Gerum, & Berndt, 1990, p. 64)

When talking about electromagnetic track brakes two types are known. (1) Electrically generated track brakes and (2) permanent magnetically generated track brakes. The track brake can either be triggered by pneumatic or by electrical means. (Gralla, 1999, p. 97)

Considering (1) electrically generated track brake, the basic principle is an electromagnet which consists of a drawn out coil and horseshoe shaped magnetic cores. The DC flows through the coil and creates a magnetic field that pervades the rail head. The braking force quantity depends on:

- Magnetic resistance of the circuit (geometry and permeability)
- Current linkage
- Friction coefficient between track and pole shoe (pressure, temperature, speed and material combination)
- Track condition (unevenness influences the air gap and therewith the possible brake force)

The geometry of the design is very important in order to achieve optimal braking forces, especially when it comes to the ratio of iron and copper. Furthermore the material for the magnetic core is of interest. None of the present materials is reasonable for all the possible applications. (Gralla, 1999, p. 64)

Considering the design of the brake magnet there are monoblock magnets and articulated magnets. The suspension comes in different types like high suspension combined suspension and low suspension. (Knorr-Bremse, 2002, p. 95)

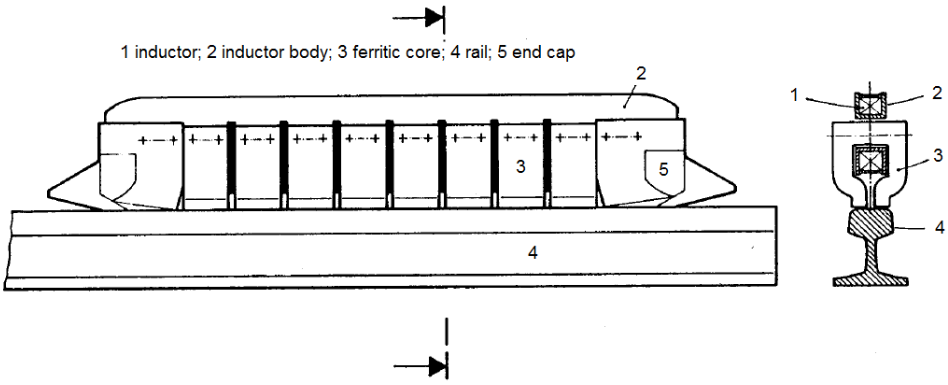


Figure 27 Electrically generated track brake based on (Gralla, 1999)

As opposed to (1) electrically generated track brakes the (2) permanent magnetically generated track brakes do not need electric current to be initiated. Hence, this brake can be used as parking brake as well; the spring loaded brake can be replaced. Other advantages are unlimited 'switch on time' and a constant attraction force, no matter how long the brake has been used earlier. Pitifully if the swelling motor is out of service the brake cannot be released anymore.

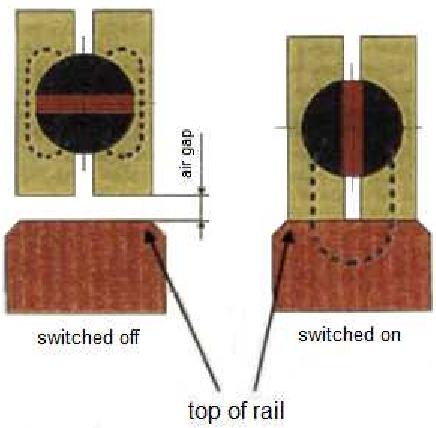


Figure 28 Rotatable magnetic poles – switched off and switched on (Gralla, 1999)

Figure 28 shows a cross-section of a permanent magnetically generated track brakes with the two positions of the rotatable magnetic pole (red). If the magnetic field is closed via the rail head, the brake is considered to be switched on. The necessary rotation to switch the brake on and off is triggered by a hydraulic swelling motor. It is important to adjust the magnetic line of force in a way to avoid deflection to its environment.

The braking force quantity depends on:

- Friction coefficient between track and pole shoe
- Size of the air gap
- Travel velocity, due to the speed depending friction coefficient characteristic

(Gralla, 1999, pp. 99-100)

Linear eddy current brake

In contrary to the rotating eddy current brake, the linear eddy current brake is force closure independent. (Knorr-Bremse, 2002, p. 100)
Compared to the electromagnetic track brake the linear eddy current brake has a crucial advantage: It is fixed about 7 mm above the track and therefore does not touch the track. Due to this reason the brake works wearless and can be used as service brake. The brake is fixed to the wheel set bearing to ensure a constant distance to the track. (Saumweber, Gerum, & Berndt, 1990, p. 67) Furthermore, it is necessary to check on the offset frequently, because it decreases due to wheel wear. (Gralla, 1999, p. 101)

Several electromagnetic coils assembled to a magnet yoke build the eddy current brake which is housed in a metallic guard casing in order to decrease environmental impacts and to shield the magnetic field from the environment. The coils are connected in a way that alternating magnetic north and south poles can arise. In case the brake is electrically actuated the resulting magnetic field pervades the rail head. If the speed of the vehicle is zero, a vertical attractive force is present. In case the vehicle has a certain speed, uneven zero, the magnetic field between brake and track is unsteady. Therefore, electrical voltage is created which further cause eddy currents due to the law of induction. These currents distort the magnetic field in the opposite driving direction. The previous vertical attraction force gets tilt; a horizontal force component is created. This force is the actual braking force of an eddy current brake. (Knorr-Bremse, 2002, p. 101), (Saumweber, Gerum, & Berndt, 1990, p. 68), (Gralla, 1999, p. 101)

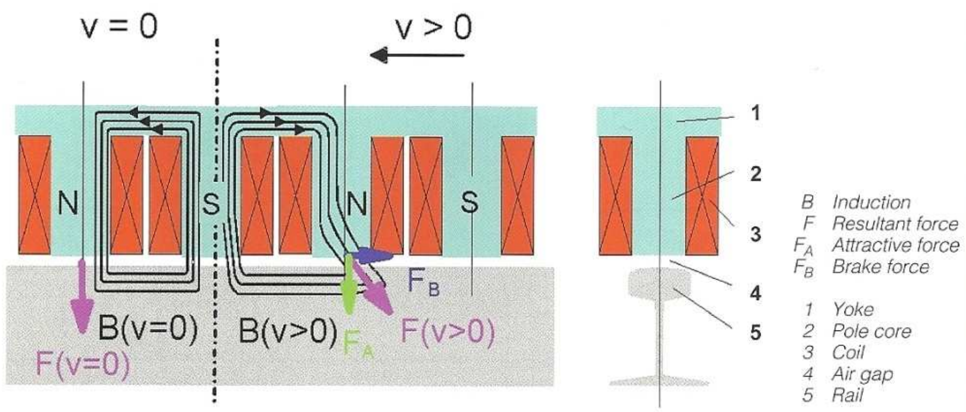


Figure 29 Functional principle of eddy current brake (Knorr-Bremse, 2002)

The characteristic of a linear eddy current brake is equal to that of an asynchronous motor. As mentioned above the braking force is zero in case the vehicle velocity is zero. In case of constant excitation the maximum force is present at about 75 km/h. If the velocity is higher the force decreases again. However, the vertical attraction force decreases exponentially with the increase of the speed. See Figure 30:

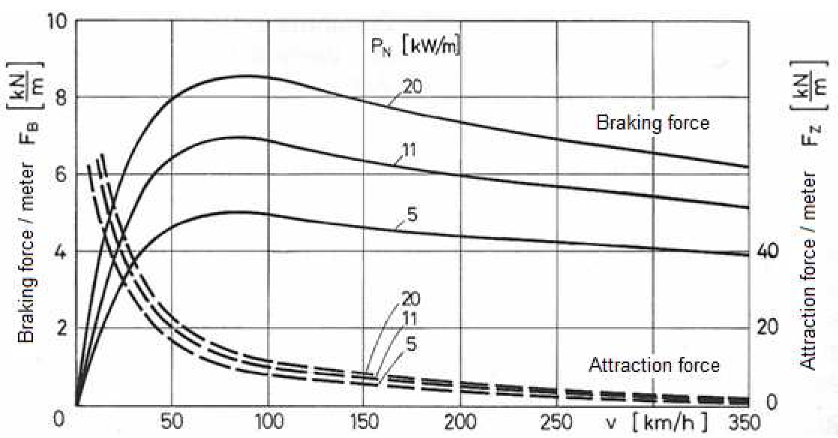


Figure 30 Braking force FB and attraction force FZ per length of the eddy current braking relation to velocity and exciter power PN, referring to 7mm air gap and 20°C. (Saumweber, Gerum, & Berndt, 1990)

A downside is that the braking force is limited due to the warming of the coil. Furthermore, sometimes the heat input in the track has negative effects on the stability.

3.1.4 Braking force – a theoretical outline

The braking behavior of a railway vehicle is determined by the total of the braking forces of the train. The total force results from the sum of the braking forces of all cars in the train as well as from the braking forces of all brakes involved.

Involved in braking might be:

- Wheel effective friction brakes
- Wheel effective dynamic brakes
- Track effective brakes

The actual braking force is calculated by adding the forces from involved brakes. It is mandatory to refer the braking forces to the same time *t* and velocity *v*; calculating the total of median forces to obtain a median braking force is not possible due to physical conditions.

The braking force of the rail vehicle is limited due to the (1) physical conditions concerning the wheel track force closure and the due to the (2) physical conditions of the brake force creation mechanism. (Gralla, 1999, p. 4)

Limitation due to the wheel- track force closure

Ever since the engineering of railway vehicles the wheel- track force closure was object to numerous experiments. As a result it was found that the closure depends on several influencing factors, like (i) weather conditions, (ii) contaminant, (iii) travel speed as well as on (iv) wheel and track material.

In most of the theoretical approaches the relation of force closure and slippage was examined. As condition to this evaluations wheel and track were defined as linear elastic and further the coulomb law was taken as given for the tangential force transmission in the wheel track contact area. (Saumweber, Gerum, & Berndt, 1990, p. 11)

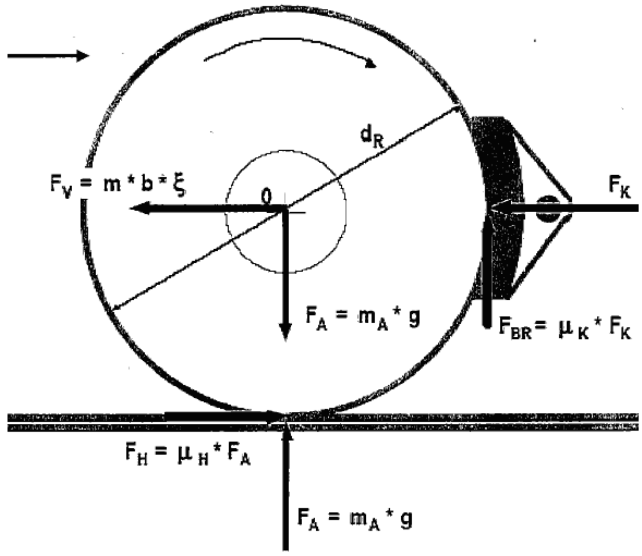


Figure 31 Forces at the wheel (Heckemanns, 2010)

The leading equation in this context according to (Gralla, 1999) is the Coulomb law. In this case expressed to describe the braking force limited due to wheel track force closure.

$$F_H = \mu_H \cdot F_A \qquad \textbf{3.1}$$

The braking force considering the wheel track force closure is known as F_H. It is a result of wheel set load F_A times the adhesion coefficient μ_H.

Basically during revolving the adhesion coefficient μ_{H}, is used. In case sliding occurs, the smaller coefficient, μ_{G}, is introduced to the equation. (Gralla, 1999, p. 5)

In a physical perspective a certain maximum adhesion coefficient is developable in the wheel track contact area. The existing adhesion coefficient is always smaller than this maximum.

Maximum tangential force = maximum adhesion coefficient * normal force

Existing adhesion coefficient = existing tangential force / normal force

(Wende, 2003, p. 157)

Wheel track force closure

The following paragraphs and equations are based on (Wende, 2003, pp. 157-158) .

Braking force in the wheel track force closure can be achieved via autonomous and non-autonomous operation.
Autonomous operation means, every axle creates independently its tangential force (braking force). If there are n axles involved, the total braking force F_H is calculated as follows:

$$F_H = \sum_{i=1}^{i=n} \mu_{Hi} \cdot F_{Ai} \quad \textbf{3.2}$$

μ_{Hi} in this case is related to the developable adhesion coefficient. F_{Ai} refers to the normal force at the ith axle.

When non- autonomous operation is considered, the control unit regulates on every axle the tangential (braking) force $F_{H\max}$, independent form the individual μ_{Hi} and F_{Ai} values. The limitation in this approach is the axle where the maximum adhesion coefficient is reached first. Therefore μ_H refers to the simple adhesion coefficient and F_A to the weight of the vehicle.

$$F_{H\max} = \mu_H \cdot F_A \quad \textbf{3.3}$$

Further a force closure utilization factor α is known.

$$\alpha = \frac{\mu_H \cdot F_A}{\sum(\mu_{Hi} \cdot F_{Ai})} \quad \textbf{3.4}$$

Force closure theory

There are two theories concerning the force closure:

a) physical force closure theory
The adhesion coefficient is described as discrete parameter according to the physical principles.

b) statistical force closure theory
The adhesion coefficient is defined as stochastic parameter and determined via the mathematical statistic.

(Wende, 2003, p. 158)

(a) Physical force closure theory

Basically if a wheel revolves, slippage is always related to the transmission of a tangential force. Slippage is known as the difference between translation velocity and wheel circumference velocity. (Saumweber, Gerum, & Berndt, 1990, p. 11)

The slippage velocity v_S is the difference between travel speed v_F and circumferential velocity v_U. The brake slip σ is defined as the quotient of slippage velocity over travel velocity. (Wende, 2003, p. 159)

$$v_S = v_U - v_F \quad \textbf{3.5}$$

Alternatively (Saumweber, Gerum, & Berndt, 1990) provide a second notation:

$$\Delta v = v - r\dot{\phi} \quad \textbf{3.6}$$

This equation is valid due to the basic dynamic relation auf velocity v is equal to radius r times angular velocity $\dot{\phi}$.

Concerning the brake slip following equations are known:

$$\sigma = \frac{v_S}{v_F} \quad \text{or} \quad \sigma = 1 - \frac{v_U}{v_F} \quad \textbf{3.7}$$

$$0 \le \sigma \le 1 \quad \textbf{3.8}$$

And alternatively:

$$s = \frac{v - r\dot{\phi}}{v} \quad \textbf{3.9}$$

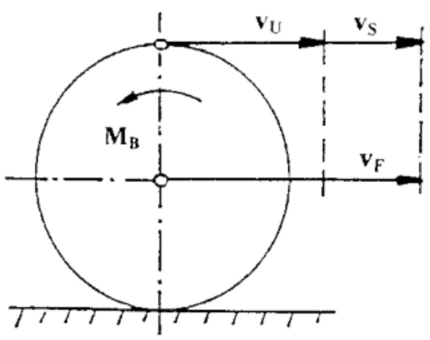


Figure 32 Velocities at the wheel (Wende, 2003)

To provide further information, the occurring effects in the contact area are described in the following paragraphs.

The wheel surface elements reach the contact area with tensile pre-loading, the track surface elements with compressive pre-stressing. The front contact section is the adhesion area. In this area the pre-stresses are retained. The second part of the contact area is the sliding area, where the surface elements slide relative to each other, while the pre-stresses get inverted. (Saumweber, Gerum, & Berndt, 1990, p. 11)

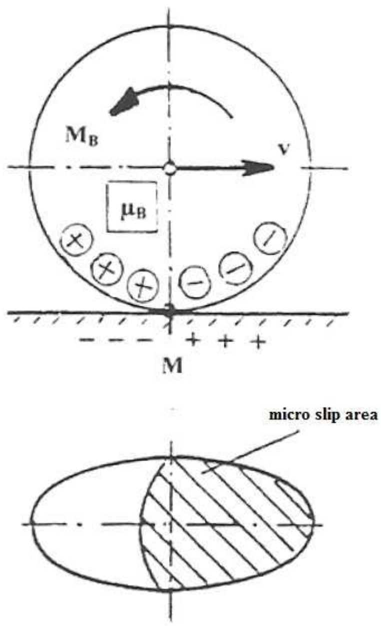


Figure 33 Wheel track contact area (Wende, 2003)

The mentioned circumstance causes that the circumferential velocity is always smaller than the travel velocity. This results in micro slippage. (Saumweber, Gerum, & Berndt, 1990, p. 11)
Basically the sliding movement in the wheel track contact area consists of micro and macro sliding movement.
Micro sliding movement is a reversible relative movement. It occurs due to plastic-elastic shear deformation of micro hills caused by the tangential force load in the contact area. When the micro hills have passed the contact area they move back to their initial position.
Macro sliding movement is an irreversible relative movement. It occurs due to the sliding displacement between the wheel micro hills and track micro hills in the contact

area. The micro hills have a different relative position to each other when they leave the contact area. (Wende, 2003, pp. 161-163)
If the tangential force is increased the proportion of the sliding area in the contact area is also increased until the whole adhesion area is vanished. In this case pure sliding occurs.

Furthermore in reality the bogie is never rolling straight the longitudinal direction of the track. Due to the concentricity of the wheels the bogie continues along the so-called sinusoidal run. This causes cross slippage and Bohr slippage besides longitudinal slippage in the contact area between wheel and track. Both further reduce the usable adhesion coefficient. (Saumweber, Gerum, & Berndt, 1990, pp. 11-12)

Adhesion coefficient – slippage law

Sliding is completely unwanted and consequently the force closure – slippage – characteristics need to be taken into consideration. (Gralla, 1999, p. 17)

The relation between adhesion coefficient μ_B and slippage velocity v_S can be illustrated. (Wende, 2003, p. 161)

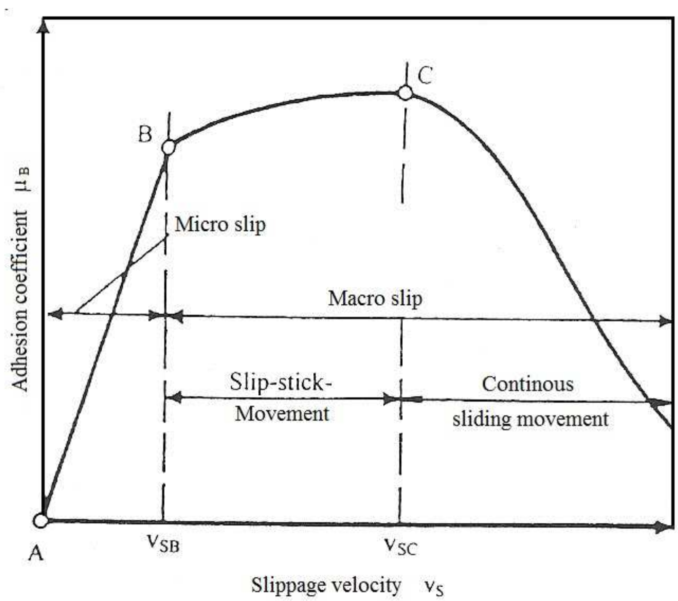


Figure 34 Adhesion coefficient – slippage law (Wende, 2003)

In the area between $0 \leq v_S \leq v_{SB}$ the force-closure creating slippage is predominant. It furthermore refers to the micro slip range.

The next area is known as $v_S \geq v_{SB}$ and belongs to the macro slip range. It is divided into two parts. The interval of $v_{SB} \leq v_S \leq v_{SC}$ and $v_S \geq v_{SC}$.
While $v_{SB} \leq v_S \leq v_{SC}$ refers to the so-called slip-stick effect, which has a force closure increasing effect, is the force closure rapidly decreasing in the interval $v_S \geq v_{SC}$. (Wende, 2003, p. 162)

However, the shape of the characteristic depends strongly on the condition of the track as Figure 35 shows.

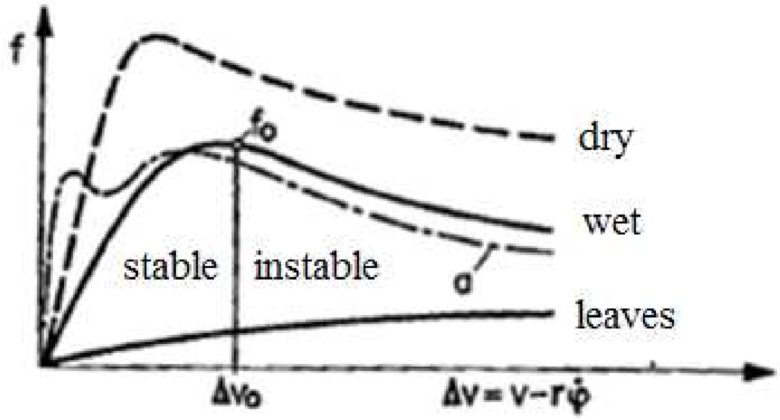


Figure 35 Adhesion coefficient depending on the velocity (Saumweber, Gerum, & Berndt, 1990)

The maximum adhesion coefficients are achievable with relative velocities between 5km/h and 20km/h, of course depending on the condition of the track.

(b) Statistical force closure theory

This theory uses a statistical approach to determine the adhesion coefficient. The adhesion coefficient is considered as a statistic variable U with the mean μ_M and the standard deviation σ. Both, μ_M and σ, are determined by measurement series. (Wende, 2003, p. 171)

Basically Siemens uses the following values for the calculations, based on Metzkow:

Dry track	0.17 to 0.26
Wet track, clean	0.17 to 0.22
Wet track, soiled	0.09 to 0.16
Wet track with leafs	about 0.03

Limitation due to the brake force creation mechanism related to wheel effective friction brakes

As wheel effective friction brakes tread brakes, disc brakes, drum brakes and rotating eddy current brakes are known. Nowadays tread and disc brakes are used in most cases.

When talking about the tread brake the brake radius is equal to the halved running wheel diameter. Moreover the sliding speed between the friction surfaces equals the travel speed.

The braking force F_H caused by a tread brake is the product of brake shoe friction coefficient μ_K and the brake shoe force F_K :

$$F_H = \mu_K \cdot F_K \qquad \textbf{3.10}$$

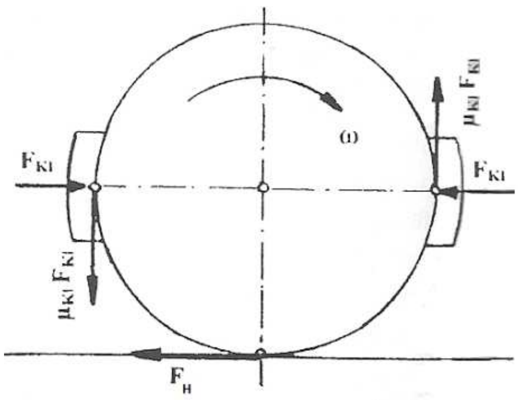


Figure 36 Braking force tread brake (Wende, 2003)

When considering a disc brake the brake radius is always smaller than the halved running wheel diameter and therefore also the sliding speed is less than the travel speed. (Gralla, 1999, p. 7)

Therefore the braking force has to be transformed from the brake disc radius to the running circle of the wheel. (Wende, 2003, p. 229)

The braking force of a disc brake is calculated as follows:

$$F_H = \mu_{Bel} \cdot F_{Bel} \cdot \frac{r_{Br}}{r_L} \qquad \textbf{3.11}$$

μ_{Bel}	brake pad friction coefficient	r_{Br}	brake radius of the disc
F_{Bel}	brake pad clamping force	r_L	running circle of the wheel

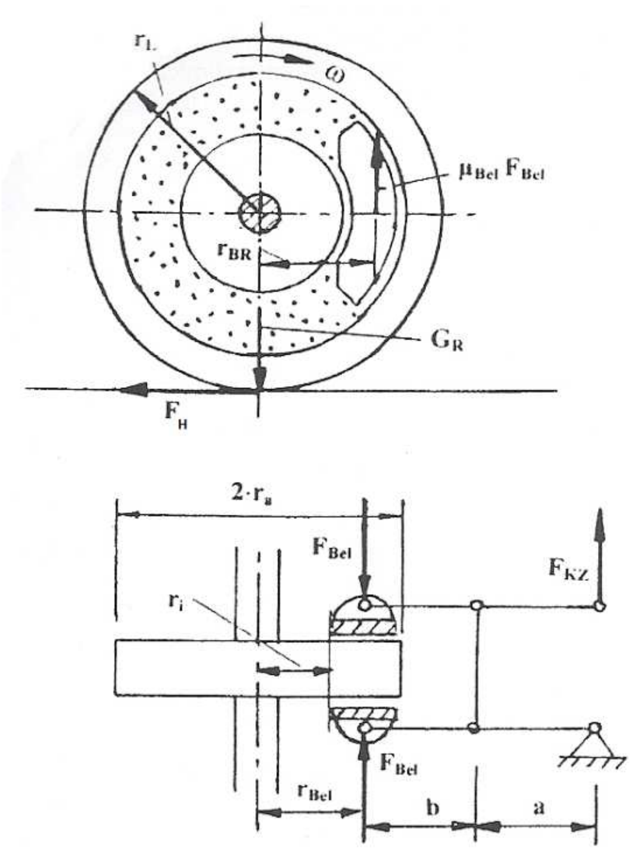


Figure 37 Braking force disc brake (Wende, 2003)

According to the mentioned equations the braking force of tread brake is limited due to the adhesion coefficient and the brake shoe force.
Concerning the disc brake the braking force is limited as well by the adhesion coefficient and the friction pad force, supplemented by the quotient of brake radius and running wheel radius.

(Gralla, 1999, p. 7)

Block- and brake lining coefficient

Brake shoe- and brake lining coefficient depend on several physical variables and are stochastic distributed. This circumstance also influences the braking force. Generally speaking the resulting variation of the braking force on each bogie is equalized when considering the whole vehicle. (Gralla, 1999, p. 7)

Following dependencies from physical variables are known:

- ***Actual travel speed***

With increasing travel speed the friction coefficient decreases.

- ***Initial braking speed***

With increasing initial braking speed the friction coefficient decreases.

- ***Block and brake lining pressure***

With increasing tread and coat pressure at the contact surface the friction coefficient increases.

- ***Material type***

Type and composition of the friction material determine the friction coefficient.

- ***Work by frictional force***

The more load is applied to the friction area due to the work caused by friction force; the smaller is the friction coefficient.

- ***Friction geometry and abrasion***

In case of a defective geometry due to faulty machining, temperature caused expansion and abrasion inhomogeneous surface pressure occurs what causes the decrease of the friction coefficient.

- ***Improvement in the beginning***

Right after the brake starts to operate, an improvement of approximately 20% is recognized. Still this improvement vanishes after a short time period.

- ***Environmental conditions***

Surrounding conditions like snow, ice and humidity influence the friction coefficient in a bad way.

(Wende, 2003, pp. 231-232)

The following paragraph and equation are based on (Wende, 2003, p. 234).

Different equations were established considering some of the mentioned variables. Especially for train brake simulations the equation developed by Gralla is useful. The tread friction coefficient μ_K is a product of term A the initial braking speed, tread pressure term B, current travel speed term D and work from frictional force term E.

$$\mu_K = A \cdot B \cdot D \cdot E \qquad \textbf{3.12}$$

Brake shoe force and friction pad force

Brake shoe force and friction pad force are initially caused by the ram force of the brake actuator. The linkage between the actuator ram and tread or friction pad transmits the actuator force and further provides a certain force ratio.

Additionally brake shoe force and friction pad force depend on development time and brake stage. (Wende, 2003, pp. 225-229)

3.2 Thesis related background

This part of the theoretical outline provides the background information necessary for the thesis at hand. This includes not only the description of the Siemens AG emergency braking control system, the SIB, but also the constraints for the brake actuator derived from the European standard on mass transit brake systems EN-13452-1:2003 and the Siemens Performance Specification for MetroMidCap.

Moreover in this part the practical Systems Engineering Approach, from general to detail, is used to pinpoint different principles to configure an actuator. This elaboration uses a design catalogue as basis and allows pinpointing physical effects and their real-world applications adaptive for the actuator.
The necessary theory to evaluate the feasibility of the found applications is provided as well.

Furthermore the presently used tread brake actuator by Faiveley Transport is introduced as well some of the electro mechanic actuation principles.

3.2.1 SIB – Safe Innovative Brake

Conventional underground motor coaches have at least two different possibilities to create braking force. This is a result of the technical standards of the EN 13452-1:2003, where it is said that every train has to have two separate braking systems, which might be of the same type. Still it has to be assured that the second system is fully operating while the other is out of order. (EN-13452-1, 2003, p. 23) Especially if an electro dynamic brake is used, there has to be another braking system available which can automatically perform at least a single emergency brake application in case of ED brake failure. (EN-13452-1, 2003, p. 18)

A conventional motor coach would possibly have an electro dynamic brake and an electro pneumatic brake to be compliant with the regulation. (Stützle & Schraud, 2010, p. 10) In this case one system usually uses pneumatic pressure and the second is realized via the electrical drive. So far the electrical drive is only used as service brake because the drive is considered not to be save enough to perform emergency brake applications with the current state of the art architecture. (Stützle, 2011, p. 1)

In case the SIB is used in a vehicle the motor bogie possesses of the following two brakes:

- Standard electro-dynamic brake
- Safe Innovative Brake

Then the service brake is performed by the standard electro-dynamic brake. The SIB is used to perform the emergency brake application. Therewith the electrical drive motor is used as generator in case of an emergency or safety emergency brake application as well. (Stützle & Schraud, 2010, p. 10) Essential to the technique is the usage of a permanent magnet synchronous motor. (Stützle & Schraud, 2010, p. 5)

Because the trailer bogies have no motor that can be used as generator, the SIB does not affect the equipment of these. Any considerations concerning the SIB are related to motor bogies. In case of an emergency brake application the trailer bogie will always use the pneumatic brake equipment. (Stützle & Schraud, 2010, pp. 10-15)

The principle of dynamic braking using the drive engine as generator is explained in section 3.1.3.

Nevertheless the motor coach has to be equipped with a spring loaded friction brake. (Stützle & Schraud, 2010, p. 10)

The SIB would be used in the Siemens Syntegra bogie. The idea of the SIB is popular with the Syntegra because it enables to get rid of some pneumatic components of a conventional braking system, which causes a reduction of LifeCycleCosts as mentioned previously.

Figure 38 shows the possible savings on pneumatic brake cylinders when using the SIB.

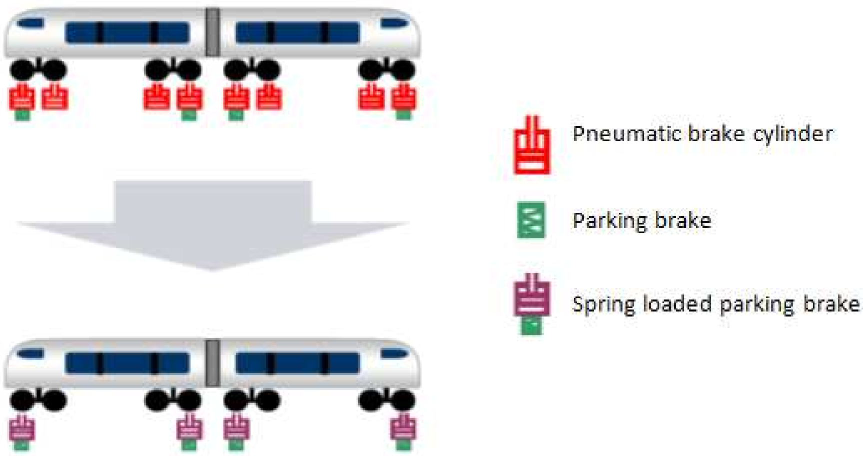


Figure 38 Reduction of brake cylinders (Stützle, 2011)

Every brake cylinder without spring loaded brake can be reduced due to the SIB. The parking brake cylinder for the SI- brake however has to provide force for parking brake and holding brake applications as well as it should be able to work as last safety device in case every emergency brake backup fails, to assure that the train will stop. (Stützle, 2011, p. 1)

Siemens expects that the spring loaded parking brake in case of emergency brake application will only provide approximately 50% of the required average deceleration of 1.3 m/s^2 (80km/h) and 1.4 m/s^2 (90km/h). Therefore it has to be ensured, that the spring parking brake will most likely not be necessary to provide emergency brake force. (Stützle & Schraud, 2010, p. 10)

The SI- brake in detail

If a motor bogie is equipped with the SI- brake, it possesses several controlled sub-brakes. These sub-brakes are applied to different fallback levels. These will be described later on. (Stützle, 2011)

The following paragraphs are based on (Stützle & Schraud, Inspiro - Konzeptionelle Machbarkeit, Sichere Innovative Bremse, 2010, pp. 10-15).

Basically the SI brake consists of several monitored sub-brake systems, like

- A software controlled electro dynamic (ED) brake including monitoring system with wheel slide protection
- An electric brake (FPGA) with wheel slide protection (FPGA brake, hardware controlled chopping) and monitoring system; it is using the same traction components as the ED brake
- Pneumatic spring loaded brake

Basically ED monitoring systems checks on the braking effect of the ED brake and the FPGA monitoring system on the effect of the FPGA brake. However the monitoring systems of ED and FPGA brake might be identical in case both systems are installed together. ED and FPGA use the electrical drive motor as generator in case of a braking application. The monitoring systems will detect failures and low brake force and ensure a safe changing to a backup brake system as shown in Figure 39 column 1. However, hardware failure concerning the traction unit cannot be avoided.

While the service brake application is performed by the standard ED brake, the emergency brake application is performed using the SIB brake. As mentioned before, the SIB uses the same hardware components, e.g. the traction unit to create braking force. However, the control depends on the fallback level.

There are three different strategies to design the fallback levels of the SIB:

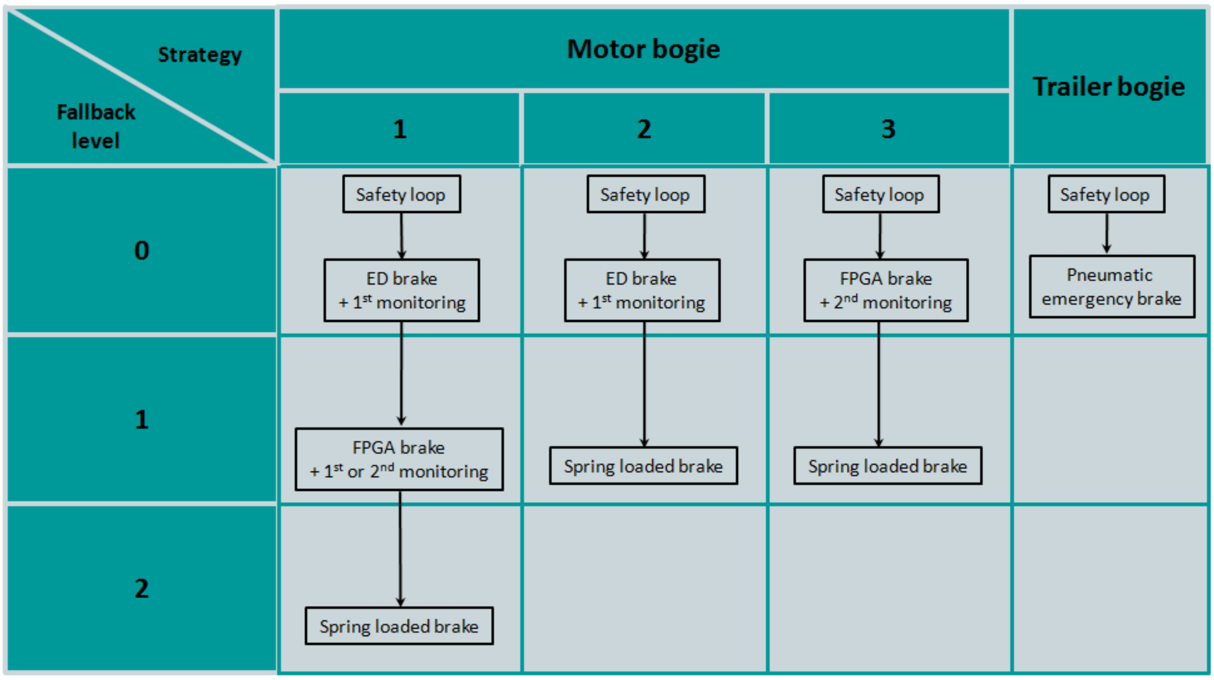


Figure 39 SIB strategies based on (Stützle & Schraud, 2010)

Strategy FPGA (3): In this case only the FPGA brake is used as emergency brake. As dynamic brake the FPGA uses a hardware pulsing procedure that causes an electrical braking torque within the permanent magnet synchronous motor. During an emergency brake application the line contactor is opened to inhibit powering the motor. Basically the FPGA brake is a rheostatic brake.

The resulting average deceleration of the vehicle is sufficient for a certain rpm- range. But if the vehicle reaches a certain low velocity, short before stoppage, the braking effect vanishes due to the pulsing procedure. In this case the FPGA brake activates the spring loaded brake to ensure stoppage. One possible problem has to be taken in consideration while the FPGA is replaced with the spring parking brake: it is most critical that wheel lock- up occurs because the spring loaded brake has no wheel slide protection. Wheel flats might be the worst case result.

In case the power supply for the FPGA brake is out of order or the FGPA monitoring system detects a too low braking effect, the spring loaded parking brake is used as first (and only one) fallback level. Still the possibility that a failure concerning the FPGA brake occurs is considered as low.

Strategy ED (2): This strategy uses the ED brake as emergency brake. As mentioned above the ED brake is considered as unsafe and therefore it needs a complex system

architecture that continuously detects failure during the emergency brake application. As a consequence the monitoring system ED is introduced. The ED brake performs the emergency braking till stoppage. The spring loaded brake is – again – the first fallback level if monitoring system detects an ED brake failure. When using strategy ED the emergency brake can be done regenerative. This is an advantage due to energy efficiency.
Yet it is assumed that ED brake failure happens rarely.
When comparing strategy FPGA and ED, strategy ED offers a major advantage. For the ED brake the approved SITRAC drive control is used. It has a smoother control mode than the FPGA pulsing procedure, due to the usage of field- oriented control.

Strategy Combination (1): This is basically a combination of strategy ED and strategy FPGA. The ED brake is used in the first place if an emergency brake application is triggered. If a too low braking effect is detected or the motor vehicle driver activates the emergency push-button, the system switches to strategy FPGA. When talking about strategy Combination the monitoring system for ED and FPGA brake is identical.
It is also necessary to take into consideration that the FPGA brake needs a certain amount of time to compensate the ED braking force loss. This affects the braking distance. However it can be regulated due to local circumstances, like legal requirements, safety distance or track signals, etc.
Basically the safety level with the FPGA brake can only restore the brake ability if the loss of braking power was not caused by the failure in electrical components that are used in the ED brake as well as in the FPGA brake. Therefore the FPGA braking strategy Combination is used to compensate failure in the signal processor of the converter data processor and the depending complex software.

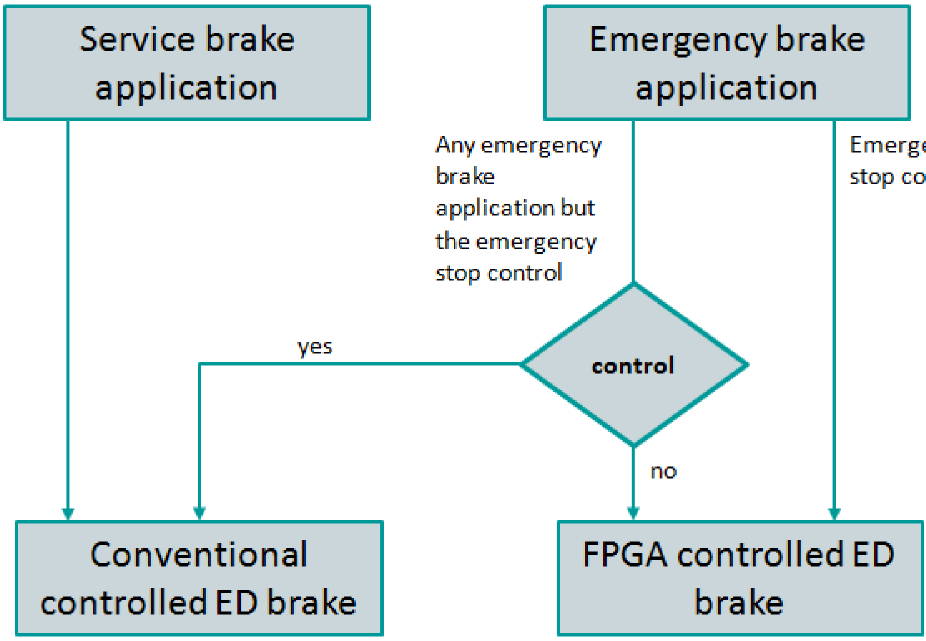


Figure 40 Brake applications with strategy combination

In order to decide on a strategy several requirements defined by regulations have to be met.

a) According to EN 13452-1:2003 every train vehicle and every car has to be equipped with at least two separate braking systems, which might be of the similar or same architecture, as long as one braking system is fully operable, while the other is inoperable.
b) EN 13452-1:2003 further defines if a railway vehicle is equipped with an electro dynamic brake a second brake assembly needs to be available, which in case of ED brake failure can at least perform a single emergency brake application. This application has to be fulfilled automatically.
c) BOStrab rail vehicles have to consist of two separate brakes. These have to be independent from each other in a way that if one fails the other can take over properly.

The spring loaded brake by itself cannot provide the requested average emergency brake application deceleration. Therefore it cannot be seen as a full- fledged second brake assembly. As a consequence strategy ED has only one full-fledged braking system.

As already touched previously the spring loaded brake has the purpose to be the last safety device in the most unlikely case of a fatal failure scenario. In this case, the spring loaded brake is applied automatically. In a thermal perspective the spring loaded brake is designed to endure a single stoppage brake application initiated at maximum speed.

Yet it is rather possible to consider the pneumatic spring loaded brake in combination with the electrical FPGA brake as second brake assembly to complete in relation with the ED brake the necessary braking systems in the motor bogie. Yet this point of view is uncertain because, in point a), mentioned separation, is only present in the control level while the FPGA brake is in operation. A complete separation is widely achieved only if the spring loading brake is in operation, though the spring loading brake is, as mentioned earlier, not considered as a full- fledged braking system.

So far the strategy Combination is the favored strategy. It contains strategy ED as well as strategy FPGA. With this combination a system is achieved which isn't more vulnerable than today's common systems. This is proven with accredited analytical methods in (Stützle & Schraud, 2010). Therefore it should be a matter of negotiation in order to adjust regulations to consider this combination as full- fledged alternative to nowadays common systems.

3.2.2 Performance specification of the braking system

Whenever it comes to security relevant components, there are lots of attributes defined by the legislature and some additionally defined in the performance specification between vendor and customer. In the present thesis the attributes of the braking systems are defined by (EN-13452-1, 2003), (EN-50126, 1999) and (MMC- Lastenheft, 2010) by Siemens AG.

While the design of the braking system has to be compliant to (EN-50126, 1999), the rough static and dynamic safety values are defined in (EN-13452-1, 2003). The detailed specification is negotiated between customer and vendor and documented in this case in (MMC- Lastenheft, 2010).

The EN 50126, (1999) defines a process to manage reliability, availability, maintainability and safety (RAMS) during the whole life cycle of a railway vehicle. Certainly this process can be applied to any kind of railway system, not just brakes. Aim of this EN regulation is to provide railway companies and their suppliers to find the optimal relation between RAMS and costs.

A big problem concerning rail-bound mass transit is that there are no valid, uniform regulations and rules for the braking application. To improve this situation, the EN 13452-1:2003 was published in 2003, providing minimum requirements for these vehicles. The details however are still defined by the transportation companies. The previous mentioned regulation is particularly of interest for this thesis because it consists of the regulations for subways with steel wheels.

Characteristics according to EN 13452-1:2003

Basically a mass transit vehicle has to be equipped with a braking system which is able to decelerate or stop it. Also the ability to perform gradient and holding brake applications has to be provided. These functions together need to be realized in a way that passengers and other road users are not exposed to hazards. Acceptable jerk and realistic adhesion coefficient need to be considered as well as a sufficient efficiency in respect to the operating conditions.

As explained in Chapter 3.2, the actuator this thesis deals with is supposed to provide the parking and holding brake application and to act as last safety device to stop the vehicle. The parking brake application has to provide a given brake ability and operational reliability. The application should be performed automatically. Hand operation is considered as optional. The holding brake application is optional and only necessary if the transportation company asks for it. Therefore also the slop of the track and the time the application has to be performed is defined by the transportation company.

Besides these basic standards there are special ones for the different applications of light rail vehicles. Tram lines have to achieve other stopping power than subway vehicles with steel wheels or pneumatic tires or suburban railway systems.

For this thesis the subway systems with steel wheels are of interest. These vehicles are defined by several characteristics:

- Station spacing of 0,5km to 2km
- Largest load capacity between 60% -100% of EL E
- Braking applications every 1 to 2 minutes
- Largest slope about 5%
- Below grade track section 15% to 100%
- The vehicles use tracks that are separated from other road users and pedestrians

Dynamic limits

	Service brake	Emergency brake 1	Emergency brake 2	Emergen cy brake 3	Emergenc y brake 4	Safety brake
Minimum deceleration [m/s^2]	variable 0 to 1.0	1.0	1.0	1.0	1.0	0.7
Maximum equivalent reaction time [s]	1.5	1.5	1.0	1.0	1.0	2

Table 1 Dynamic limits for metro vehicles with steel wheels (EN-13452-1, 2003)

Brake applications

	Holding brake	Parking brake
Load capacity	EL 6	EL E
Slope of the track	4%	4%
Duration	1 h	Unlimited

Table 2 Holding and parking brake application requirement (EN-13452-1, 2003)

Concerning the design possibilities for the parking and holding brake the standard provides the possibility to use a friction brake or a permanent magnet track brake.

As mentioned above there are also some special things to consider when talking about those mass transit vehicles that do not belong to the subway vehicles with steel wheels. If the actuator might also be used in a tram line, it is reasonable to consider some specifics for these vehicles in addition. The standard asks to use a spring loaded brake or a similar reliable working principle to perform the parking brake application. Also one of the two braking systems in the vehicle has to be independent form the wheel track force closure and independent from the drive power supply.

Characteristics according to the Siemens performance specification for MMC

The Siemens performance specification used is based on the contract for Metro Mid Cap.

- 14t of axle load need to be decelerated
- Usage of a tread brake
- Manual release of the spring loaded brake
- Two successive emergency braking applications to be performed
- Maximum slope of track 5%

1. Service brake:
   Present deceleration ≥ 1.2 m/s^2 (load capacity EL E to EL 6 (6 Pers./m^2), plane, straight, dry and clean track, Jerk ≤ 0.8 m/s^3

2. Emergency brake:
   Medial deceleration 1.3 m/s^2, optional 1.4 m/s^2 (with 80 km/h, optional 90 km/h), load capacity EL E to EL 6 (6 Pers./m^2), plane, straight, dry and clean track

Additional characteristics

Additionally to the requirements mentioned above, there are some additional characteristics supposed to be considered: (1) the mass of the actuator should be lower and the space requirements equal at max, compared to the one of the current Faiveley BFC, (2) the costs should not exceed the current ones, (3) controllability has to be provided, (4) slack adjuster and (5) emergency release should be installed.

3.2.3 The actuator – basic considerations

An actuator supplies and transmits a certain amount of energy for the operation of another mechanism. (Dictionary.com LLC's online)

As defined previously, the brake actuator this thesis deals with has to provide a parking brake application. Therefore it is essential to possess of some kind of inexhaustible energy storage in order to provide constant braking force over infinite time. Certainly this energy storage has to be charged somehow. The size of the storage depends on the amount of energy to be stored, the lower the energy, the smaller the storage. Therefore power amplifications are used. This amplification can be one-tier or multi-tier.

The tread brake, as described in section 3.1.3, has to contain some sort of rigging to activate the brake shoe. This rigging provides the power amplification, too. Furthermore some sort of slack adjuster completes the actuator in most cases. The slack adjuster is necessary due to the wear of the brake shoe lining. Otherwise the braking force would decrease due to the longer distance to the running surface of the wheel. More details are explained in section 3.1.3.

In order to visualize the functions of an actuator the following figure is provided:

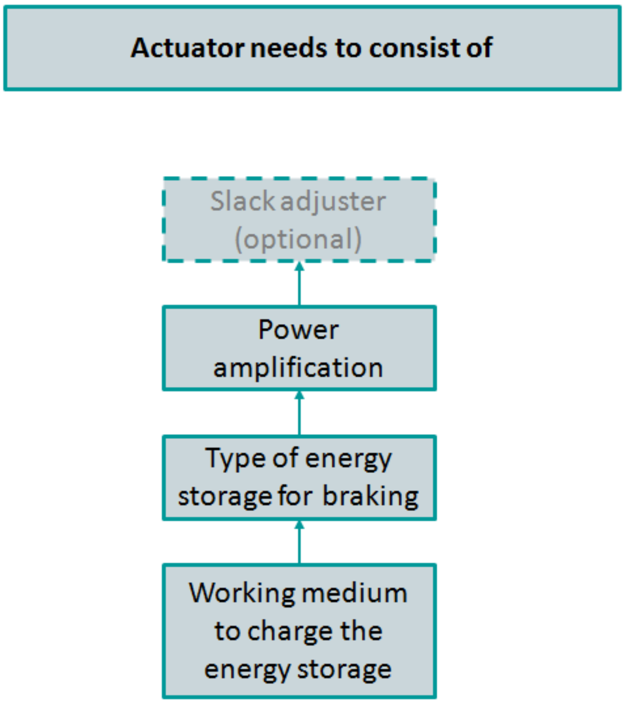


Figure 41 Components of an actuator

For the thesis on hand it is necessary to understand the different functions of the actuator and how they can be put into practice. Without this knowledge it is not possible to find improvement potential concerning the actuator used so far. Therefore structured basic considerations are necessary.

In fact the degree of elaboration and complexity of products on the market is nowadays quite high. Improvement potential is hardly to find if the redesign approach is not systematic. (Koller, 1985)
The first step is related to evaluation of the actuator principle.
It covers the search for effects that can be used to provide the must-have functions. Additionally the working principle and the working structure are defined. (Roth, 1994, p. 428)
For the thesis on hand a design catalogue is used to be consistent with a structured approach.
This catalogue can be considered as knowledge storage. In most cases it is a table that consists of systematic structured and complete knowledge about a certain topic. (Roth, 1994, p. 421)

The following sections provide the results of the design catalogue application for the three different parts of the actuator, the power amplification, the type of energy storage and the working media to charge the energy storage.

Power amplification

Basically, forces are available due to different energy sources. However in most cases the forces are not of sufficient size. In order to solve this problem, power amplification mechanisms are used.

The existing mechanisms can be categorized in energy transmitting systems, energy linking systems and energy storing systems. (Roth, 1994, pp. 92-93)

Figure 42 shows those mechanisms that fulfill the required properties.

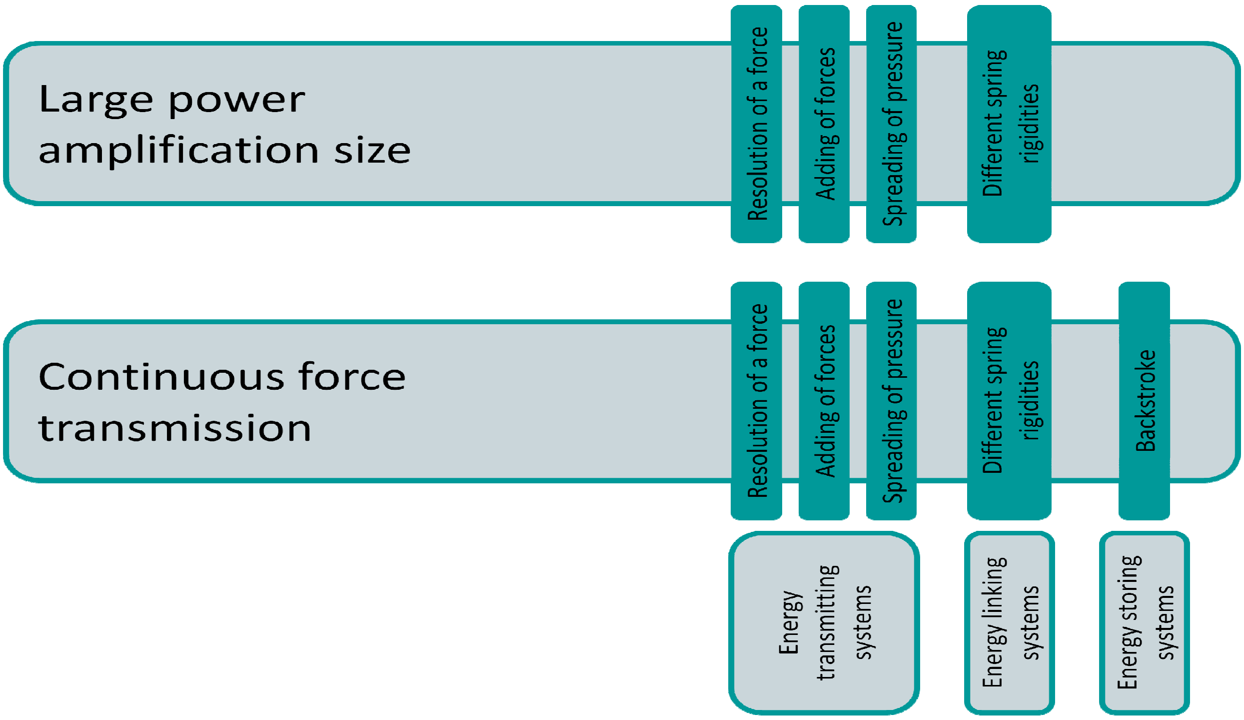


Figure 42 One-tier power amplification based on (Roth, 1994)

The most popular representatives of a real occurrence for 'Resolution of force' are wedges, screws, toggle joints and levers. The following table shows the simplified setups.

1 input 2 output

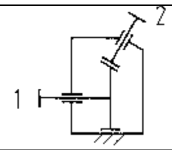
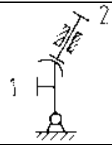
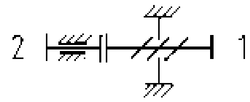
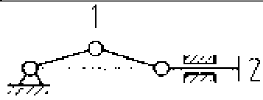
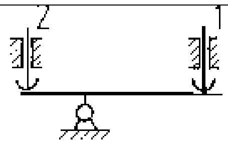
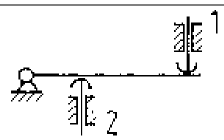
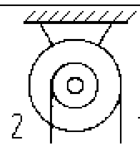
Wedge	
Rotating wedge	
Screw	
Toggle joint	
Two-armed lever	
Single-armed lever	
Double wheel	

Table 3 Types of force resolutions presented by (Roth, 1994)

The most promising ‘Resolution of force’ principles are object to consideration in terms of basic equations for power amplification and moving distance.

Wedge

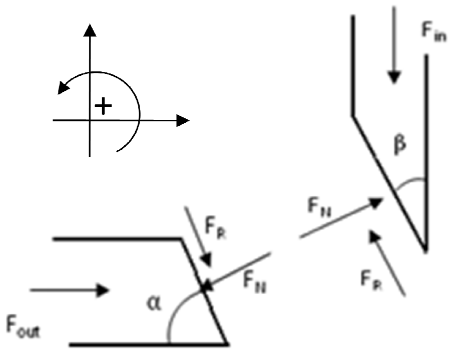


Figure 43 Forces at the wedge

$$\sum F_x = 0 = F_{out} + F_R \cdot \cos\alpha - F_N \cdot \sin\alpha \quad (3.13)$$

$$F_R = F_N \cdot \mu \quad (3.14)$$

$$\sum F_y = 0 = F_N \cdot \cos\alpha + F_R \cdot \sin\alpha - F_{in} \quad (3.15)$$

$$F_{in} = F_N \cdot (\cos\alpha + \mu \sin\alpha) \quad (3.16)$$

$$F_{in} = F_N \cdot (\mu \cos\beta + \sin\beta) \quad (3.17)$$

Distance

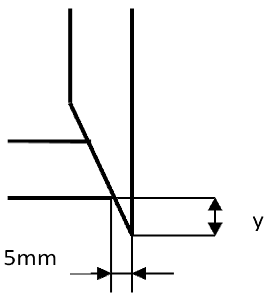


Figure 44 Distance related to the wedge

$$\tan\beta = \frac{5mm}{y} \quad (3.18)$$

Toggle Joint

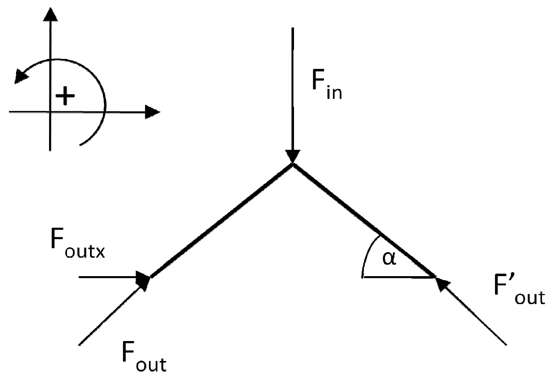


Figure 45 Froces at the toggle joint

$$\sum F_x = 0 = \cos\alpha(F_{out} - F'_{out}) \tag{3.19}$$

$$\cos\alpha = \frac{F_{outx}}{F_{out}} \tag{3.20}$$

$$\sum F_y = 0 = 2 \cdot F_{out} \cdot \sin\alpha - F_{in} \tag{3.21}$$

$$F_{in} = 2 \cdot \sin\alpha \cdot \frac{F_{outx}}{\cos\alpha} \tag{3.22}$$

Distance

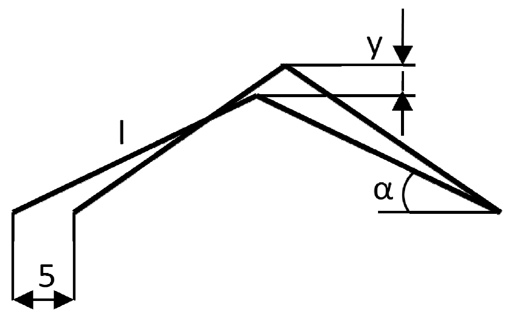


Figure 46 Distance related to the toggle joint

$$y = \sin\left(\arccos\left(\frac{\cos(\alpha) \cdot l - 5}{l}\right)\right) \cdot l - \sin(\alpha) \cdot l \tag{3.23}$$

Lever

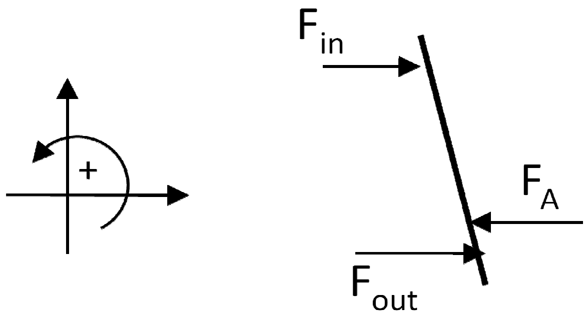


Figure 47 Forces at the lever

$$F_{in} = \frac{F_{out} \cdot l_{out}}{l_{in}} \qquad \textbf{3.24}$$

Distance

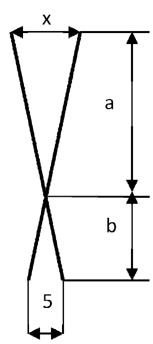


Figure 48 Distance related to the lever

$$\frac{5}{2 \cdot b} = \frac{x}{a \cdot 2} \qquad \textbf{3.25}$$

‘Adding of forces’ can be done in the following ways:

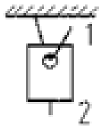
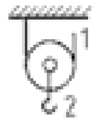
Draw-string	
Pulley-block	

Table 4 Types of adding possibilities of forces based on (Roth, 1994)

Even though these power amplification methods are very effective, still they seem not appropriate for the current application. A more detailed consideration is therefore not conducted.

‘Spreading of pressure’ is the third possibility representing energy transmission.

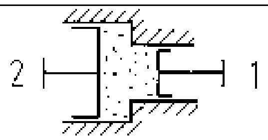
Hydraulic cylinder	

Table 5 Type of spreading of pressure based on (Roth, 1994)

Hydraulic

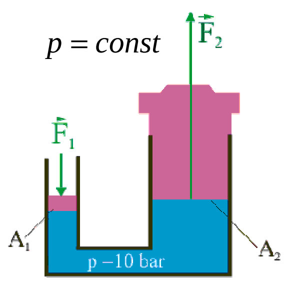


Figure 49 Forces at the fluid transmission

$$\frac{F_{out} \cdot A_{in}}{A_{out}} = F_{in} \qquad \textbf{3.26}$$

‘Different spring rigidities’ refers to clamping devices like e.g. bench vise. These are not part of the considerations for the present problem.

Basically the following power amplification possibilities are considered in detail in this thesis:

Application	Mechanism	Component
Power amplification	Resolution of force	Wedge Toggle Joint Lever
	Spreading of pressure	Fluid transmission

The energy storage

Next, the type of energy storage is considered. The basic physical principles to create force with other variables as documented in (Roth, 1994, p. 88) are used as starting point. Several characteristics have to be fulfilled by the principle. These characteristics are derived from performance specification of the actuator.

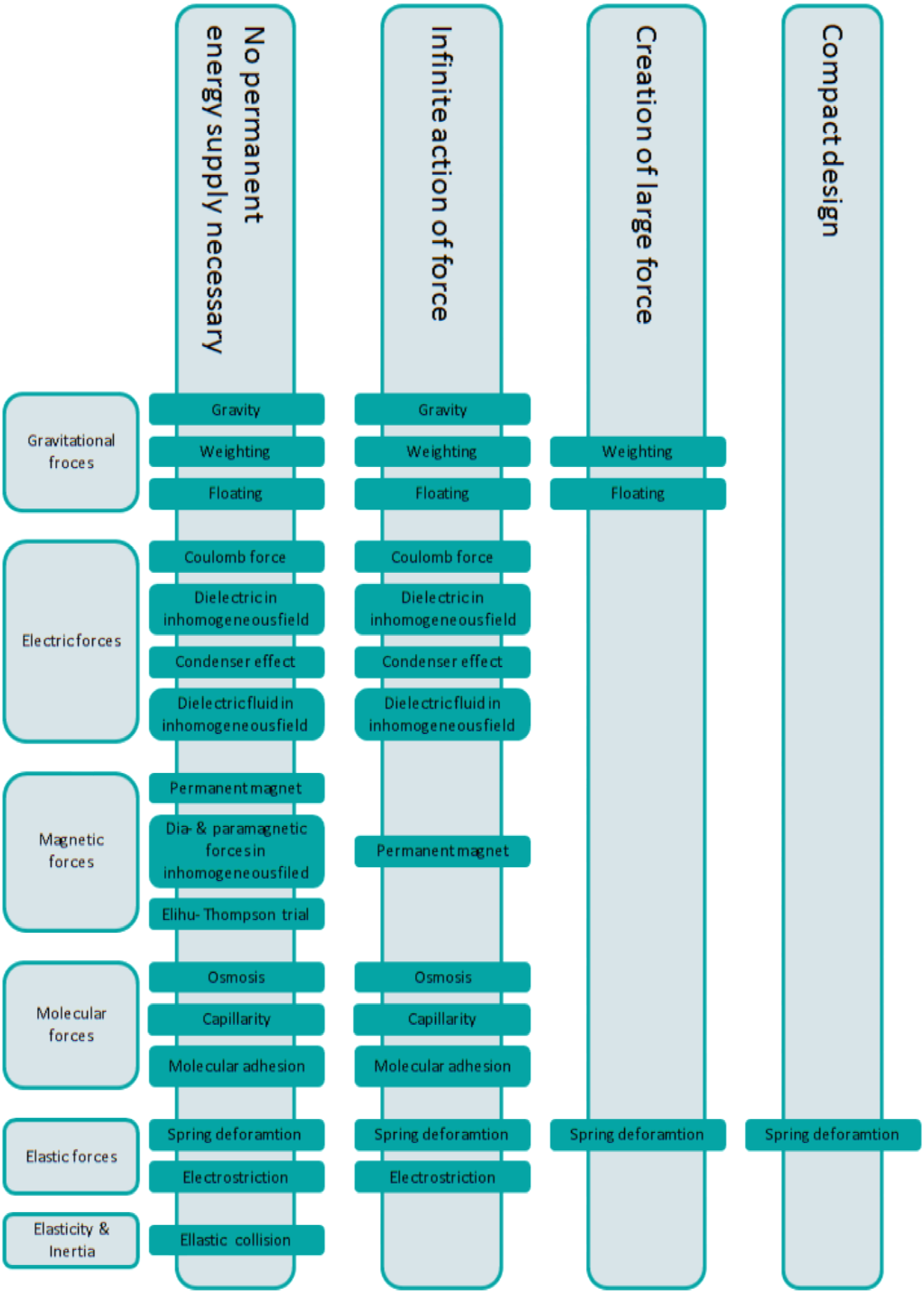


Figure 50 Possibility for energy storage based on (Roth, 1994)

Figure 50 points out the possibility to store the energy for the actuator. Not surprisingly the spring actuator is the best solution.

The following sections provide an overview on the different types of springs that might be used in an actuator.

The spring

As oldest design element, the spring has of a long history. Several different types were invented in order to achieve certain characteristics. However, all springs have the ability to perform reversible deformation under load. In most cases energy absorption and transmission of forces is related to massive deformation. The capability to do so is based on the usage of the right shape and material.
Springs may be assembled either as single or as spring system.
Springs are used as accumulator, measuring element, absorber, positioning element or as bearing support. (Meissner & Wanke, 1993, pp. 11-13)

For the present application, the ideal spring should have a characteristic that has nearly constant force over the deflection which is necessary for the power amplification movement. In this case less work is needed to release the spring loaded brake. In the best case, after having reached the necessary parking force, the unloading should be done with not even the same, but with lower force. The following figure shows the idealized characteristic.

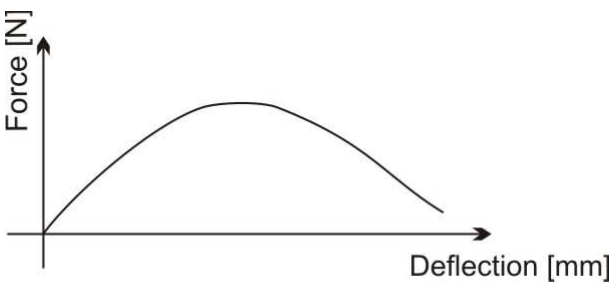


Figure 51 'Perfect' characteristic for a spring used in an actuator

However, a spring with a characteristic alike is not possible to achieve. Realistic spring characteristics are known as follows.

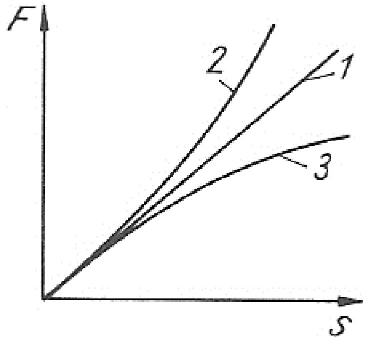


Figure 52 Different spring characteristics (Meissner & Wanke, 1993)

Basically there is a difference between the linear shape (1), the progressive shape (2) and the decreasing shape (3). The coil spring possess of a linear characteristic, but there are other popular springs which provide a different characteristic.

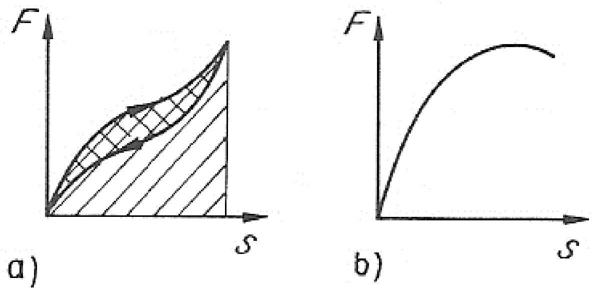


Figure 53 Spring characteristic a) spiral spring, b) disc spring based on (Meissner & Wanke, 1993)

For the thesis on hand the following spring types are considered to be most promising:

Application	Mechanism	Component
Type of energy storage for braking	Spring deformation	Disc spring Coil spring Spiral spring

Table 6 Components possible to use as energy storage

Disc spring

The literature research found that there are actuators which use the disc spring.

The following paragraphs are based on (Adolf Schnorr GmbH + Co. KG, 2004).

The first patent related to the principle of disc springs was issued on 26.12.1961 on behalf of Julien Francois Belleville. In comparison to other springs the disc spring is therefore quite new. Not until 1936 it was possible to calculate the disc spring with a practice –oriented calculation method.

Be it as it may, the disc spring possesses of several important advantages.

(1) It can accommodate very large forces at very small installation space.
(2) The spring characteristic can be linear, declining or progressive.
(3) Characteristic and block length can be adjusted within wide ranges due to the different possible combinations.
(4) In case of proper dimensioning the long operating life is achievable.
(5) No incorrect devolatilisation in case the tension limit is not exceeded
(6) Good damping ability in case of correct arrangement
(7) Concentrical load transmission

Besides these advantages the disc spring also has some characteristics that need to be considered. When it comes to static load the spring is not supposed to be applied with an equivalent stress σ_{om} larger than the tensile strength Rm of the material. This helps to avoid plastic deformation. Concerning the dynamic load it is necessary to reflect on load range and load level to obtain evidence of the operating life.

The disc spring can also be stacked. Several different opportunities to do so are known:

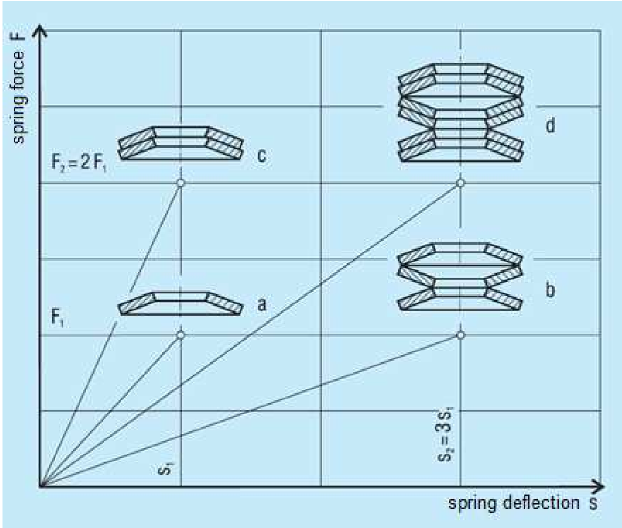


Figure 54 Different spring characteristics of disc springs related to the kind of stacking (Adolf Schnorr GmbH + Co. KG, 2004)

The characteristics of the different stacking possibilities are derived from the characteristic of a single disc spring (a). The characteristic (b) represents a mirror-inverted stacking (connection in series), (c) refers to a stacking in the same direction (parallel arrangement), and (d) is related to mirror-inverted stacking of spring packs.

Certainly spring force, spring deflection and block length are different for each blocking opportunity. The following variables to be used are listed and explained right after.

F	force of a single disc spring
F_{total}	force of a disc spring pack
s	deflection of a single disc spring
s_{total}	deflection of a disc spring pack
L_0	length of the unloaded disc spring or spring pack
l_0	installation height of a single disc spring
i	number of mirror-inverted disc springs or packs in a spring stack
n	number of parallel stacked disc springs
t	thickness of a single disc spring

Connection in series:

$$F_{total} = F \tag{3.27}$$

$$s_{total} = i \cdot s \tag{3.28}$$

$$L_0 = i \cdot l_0 \tag{3.29}$$

With this setting only the spring deflection increases but not the spring force. The spring force is still equivalent to the force of a single spring.

Parallel arrangement:

$$F_{total} = n \cdot F \tag{3.30}$$

$$s_{total} = s \tag{3.31}$$

$$L_0 = l_0 + (n-1) \cdot t \tag{3.32}$$

The spring force is to be multiplied by the number of parallel stacked springs. The spring deflection is equivalent to the deflection of a single spring.

Springs of spring pack:

$$F_{total} = n \cdot F \tag{3.33}$$

$$s_{total} = i \cdot s \tag{3.34}$$

$$L_0 = i \cdot [l_0 + (n-1) \cdot t] \tag{3.35}$$

While the spring force is proportional to the number of springs stacked in the same direction, the spring deflection is proportional to the number of spring packs.

Coil spring

Coil springs are very popular compression springs. In most cases they are made of round wire, but there are also other types available like, square or rectangle shaped ones. Many different designs are offered. They are different in respect to the pitch of their winding, the diameter of their winding and their shape of the end of the spring.

The spring characteristic of a coil spring with constant pitch is linear. A declining shape is hardly to achieve. Progressive shape however can be easily obtained. A changing pitch or diameter of winding is required in that case.

Basically the spring is deformed by a force applied in axial direction. While load is applied, the spring has to endure torsion strain. (Meissner & Wanke, 1993, pp. 196-197)

Concerning the production process it is necessary to differentiate between cold and hot formed springs. Cold formed springs are defined according to DIN2095 to have a wire diameter d of 17mm at maximum. Hot formed springs have a diameter between 8mm to 60mm, as required in DIN2096. However it is reasonable to know the difference in order to choose the right calculation equations. (Matek, Muhs, Wittel, Becker, & Jannasch, 2000, p. 299)

In order to design a spring several things have to be considered. The most important things are:

- The required force and stroke
- The type of load; static or dynamic
- The working temperature
- Buckling

(EN-13906-1, 2002, p. 8)

The required force for the calculation in this case is defined by the type of power amplification chosen, as well as the required stroke. Concerning the load type, the following consideration is done:

Springs are applied with dynamic load if the load changes over time and more than 10^4 load cycles with more than 0.1x fatigue stroke limit are performed. (EN-13906-1, 2002, p. 9)

This is valid for springs used in an actuator. It can be assumed that the spring will be in place for about 20 years. It can't be expected that the spring will only be used about 1.4 times a day. A better accuracy is provided considering the fatigue limit with more than $2x10^6$ load cycles for hot formed springs. Performing more than 274 applications per day is quite normal for the spring, if used as holding brake at each station.
Additionally it has to be motioned, that the spring to be used in the actuator will be under pulsating stress.

Basically the following equations are used to calculate the dimensions of a coil spring. The following ones are based on (Meissner & Wanke, 1993, pp. 192-195) and (EN-13906-1, 2002).

First a factor for stress correction k needs to be considered. It depends on the winding ratio w and is calculated using the Bergsträsser approximation:

$$k = \frac{w + 0.5}{w - 0.75} \quad 3.36$$

Considering the different shear stresses of dynamic application, a lower shear stress τ_{ku} an upper shear stress τ_{ko} and the stroke shear stress τ_{kh} have to be considered. Additionally the terms k, D_m, F_1 and F_2 as well as d are used. D_m refers to the mean diameter of the spring, F_1 and F_2 are the forces related to the upper and lower spring load. d defines the diameter of the wire. Additionally a safety coefficient S is necessary to be defined. This value is the ratio between the shear stress a certain spring is able to endure according to the Goodman characteristic and the calculated shear stresses. For the first approximation a value of 1.5 to 2.5 is to be reached (Meissner & Wanke, 1993, p. 40).

$$\tau_{ku} = \frac{8 \cdot k \cdot D_m \cdot F_1}{\pi \cdot d^3} \quad 3.37$$

$$\tau_{ko} = \tau_{ku} \cdot \frac{F_2}{F_1} \quad 3.38$$

$$\tau_{kh} = \tau_{ko} - \tau_{ku} \quad 3.39$$

$$S = \frac{\tau_{kO} - \tau_{kU}}{\tau_{ko} - \tau_{ku}} \quad 3.40$$

As spring rigidity c is defined as the ratio between the changing of force ΔF over the distance Δs.

$$c = \frac{F_2 - F_1}{\Delta s} \quad 3.41$$

The number of windings n is calculated using equation 3.42. It depends amongst others on the shear modulus G.

$$n = \frac{G \cdot d^4}{8 \cdot D_m^3 \cdot c} \quad 3.42$$

However, the total number of windings n_t is larger, depending on the manufacturing process. For hot worked springs the 1.5 windings need to be added and for cold worked 2.

$$n_t = n + 1.5 \qquad (3.43)$$

The dimension of a spring is defined using several equations to obtain the different lengths and also the different diameters.

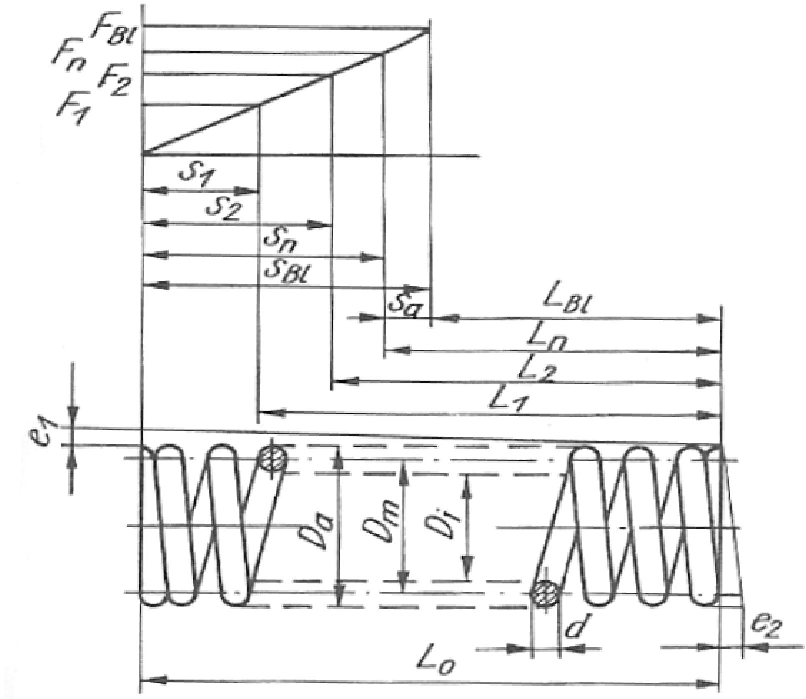


Figure 55 Different diameters, lengths and forces of a coil spring (Meissner & Wanke, 1993)

$$L_{Bl} = (n_t - 0.3) \cdot d \qquad (3.44)$$

Equation 3.44 is valid for hot worked springs with machined endings.

$$S_a = x \cdot d \cdot n \qquad (3.45)$$

$$L_n = L_{Bl} + S_a \qquad (3.46)$$

$$s_n = \frac{F_2}{c} \qquad (3.47)$$

$$L_0 = L_n + s_n \qquad (3.48)$$

$$d = \sqrt[3]{\frac{8 \cdot k \cdot (F_2 - F_1)}{\pi \cdot \tau_{kH}}} \qquad (3.49)$$

$$D_a = D_m + d \qquad (3.50)$$

$$D_i = D_m - d \qquad (3.51)$$

L_{Bl} block length
S_a sum of the minimum clearances between the windings
x factor depending on w
L_n length of the loaded spring
s_n serviceable spring deflection
L_0 length of the unloaded spring
D_a outer coil diameter
D_i inner coil diameter

Concerning the safety against buckling the following equations are recommended by the (EN-13906-1, 2002)

$$s = \frac{8 \cdot D_m^3 \cdot n \cdot F_2}{G \cdot d^4} \quad \textbf{3.52}$$

$$s_k = L_0 \cdot \frac{0.5}{1 - \frac{G}{E}} \left[1 - \left[\frac{1 - \frac{G}{E}}{0.5 + \frac{G}{E}} \cdot \left(\frac{\pi \cdot d}{\upsilon \cdot L_0} \right)^2 \right]^{\frac{1}{2}} \right] \quad \textbf{3.53}$$

In these equations the s_k is used to calculate the spring deflection for buckling and s the actual spring deflection. Additionally E refers to the Young's Modulus and υ is the bearing friction coefficient.

Additionally it is common to combine several coil springs to a spring set. Especially in case a single spring cannot endure the assignment on hand. Two to three springs are usually combined. It is not reasonable to use more, because in such a case, only the material demand increases but not the performance.
Concerning the calculation it is recommended to use the same spring deflection for all the springs in the set. Furthermore to avoid seizing, the springs should be of different winding direction. The distance between two springs is defined using diameter deviation and safety against buckling. (Meissner & Wanke, 1993, pp. 196-197)

Basically the equations mentioned above to calculate a single coil spring are valid as well. However, there are some additional equations to be used:

$$\frac{D_{m1}^2 \cdot n_1}{d_1} = \frac{D_{m2}^2 \cdot n_2}{d_2} = \frac{D_{m3}^2 \cdot n_3}{d_3} \quad \textbf{3.54}$$

This equation is valid in case shear stress and spring deflection are the same for all three springs.

Additionally the following equations are used:

$$n_1 \cdot d_1 = n_2 \cdot d_2 = n_3 \cdot d_3 \quad \textbf{3.55}$$

The overall force F for the spring set is calculated as follows:

$$F = F_1 + F_2 + F_3 = \frac{\pi \cdot \tau}{8} \cdot \left(\frac{d_1^3}{D_{m1}} + \frac{d_2^3}{D_{m2}} + \frac{d_3^3}{D_{m3}} \right) \qquad \textbf{3.56}$$

Additionally it has to be considered that the outer diameter of the spring enlarges if load is applied. ΔD_e describes the enlargement.

$$\Delta D_e = 0.1 \cdot \frac{m^2 - 0.8 \cdot m \cdot d - 0.2 \cdot d}{D_m} \qquad \textbf{3.57}$$

While m, the average center distance of the windings when no load is applied, is defined as:

$$m = \frac{L_0 - d}{n} \qquad \textbf{3.58}$$

Spiral Spring

Another alternative is the spiral spring. Concerning its spring characteristic it would fit well for the application in a brake actuator due to the flattened middle section. However it was not found to be used in an actuator in the literature research on hand.

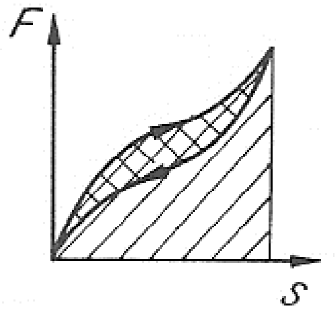


Figure 56 Spring characteristic of a spiral spring, based on (Meissner & Wanke, 1993)

These springs are able to create large righting moment. Together with a system to transform rotation in translation it seems to be a promising idea. However, a closer look into the design possibilities for spiral springs comes up with some problems:
As a spiral spring with no gap between the turns is favored. Very important for these kinds of springs is an appropriate lubrication; high maintenance effort in comparison to a coil or disc spring can be expected. (Meissner & Wanke, 1993, pp. 137-143)

The output of a spiral spring is a torque causing a rotation; therefore it is necessary to attach some sort of rotation – translation converter to this energy storage type. In this case the choice is a wheel-rod gear.

The spring has to provide the input force F_{in} required by the power amplifier. $F_{in} \cdot r_{wheel}$ causes a certain torque M . r_{wheel} is the radius of wheel used in the wheel-rod gear.

Knowing M it is possible to calculate an estimated width b . Needed for this purpose is an acceptable bending stress σ_{b_acc} derived from the tensile strength R_m of the used material.

$$M = F_{in} \cdot r_{wheel} \qquad 3.59$$

$$b = \frac{6 \cdot M}{\sigma_b \cdot t^2} \qquad 3.60$$

$$R_m = \sigma_{b\_acc} \qquad 3.61$$

Equation 3.61 is valid if certain process steps during production were conducted. Otherwise $R_m \cdot 0.75 = \sigma_{b\_acc}$ needs to be used. (Meissner & Wanke, 1993, p. 137)

For a long life expectancy non-rusting spring band steel is recommended. (Meissner & Wanke, 1993, p. 141)

Working media to charge energy source

Finally the working media to charge the energy source is analyzed.

The possible force creation mechanisms to charge the energy source are based on the same design catalog as those of the energy storage. Certainly different requirements have to be considered.

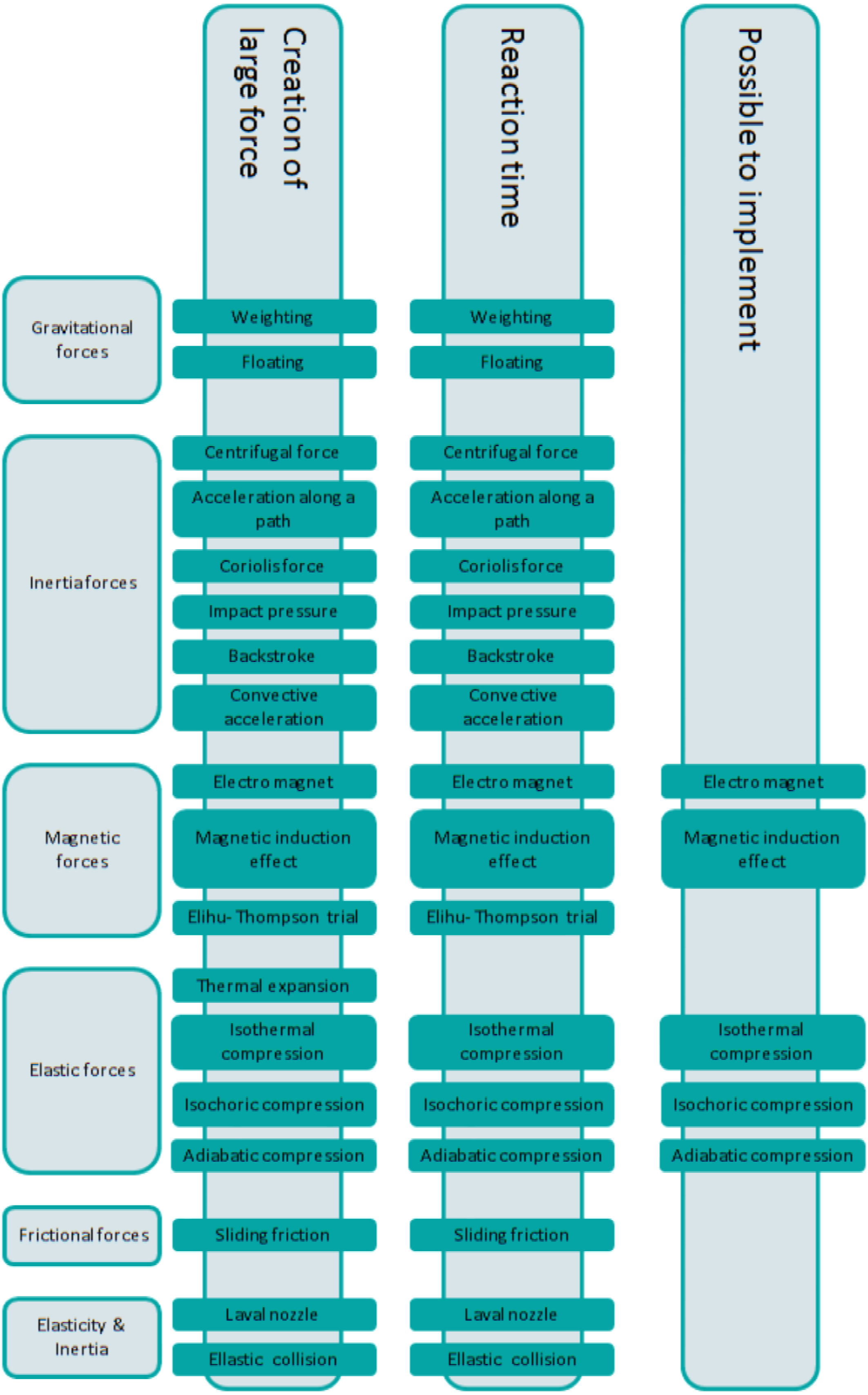


Figure 57 Working media to charge energy source based on (Roth, 1994)

As visible in Figure 57 as main mechanisms to charge the energy storage electro magnet, magnetic induction effect and different kinds of compression can be considered.

Considering the different mechanisms to charge the energy storage it is possible to derive several components.

Application	Mechanism	Component
Working media to charge energy storage	Electro magnet	Electro magnet
	Magnetic induction effect	Rot. electric motor Linear electric motor
	Compression	Pneum. / hydr. motor Pneum./ hydr. cylinder
Non-competitive		Shape memory alloy Piezo element

Table 7 Components useable to charge the energy storage

Electro magnet

The most appropriate type of electro magnet is supposed to be the stroke magnet. These magnets are characterized by having two discrete positions. Between these positions the magnet can vanquish counter forces. Usually there is only one active direction of movement. If the magnet is disconnected from the mains it usually returns to the initial position due to spring force. Most stroke magnets possess of an adjustment range of 10mm. However, a range up to 100mm is possible. (Kallenbach, Eick, Quent, Ströhla, Feindt, & Kallenbach, 2003, pp. 291-293)

For the thesis on hand the DC electro magnet is considered as most suitable. It offers a high operating life, many switching operations can be performed and an adjustment of the magnetic force characteristic using geometrical means to influence the characteristic is provided. (Kallenbach, Eick, Quent, Ströhla, Feindt, & Kallenbach, 2003, p. 363)

Considering the magnetic force – stroke characteristic, it is possible to adjust it using geometrical means. The different shapes of the matching piece and their effect on the characteristic are visible in the picture below. (Kallenbach, Eick, Quent, Ströhla, Feindt, & Kallenbach, 2003, pp. 74 -75)

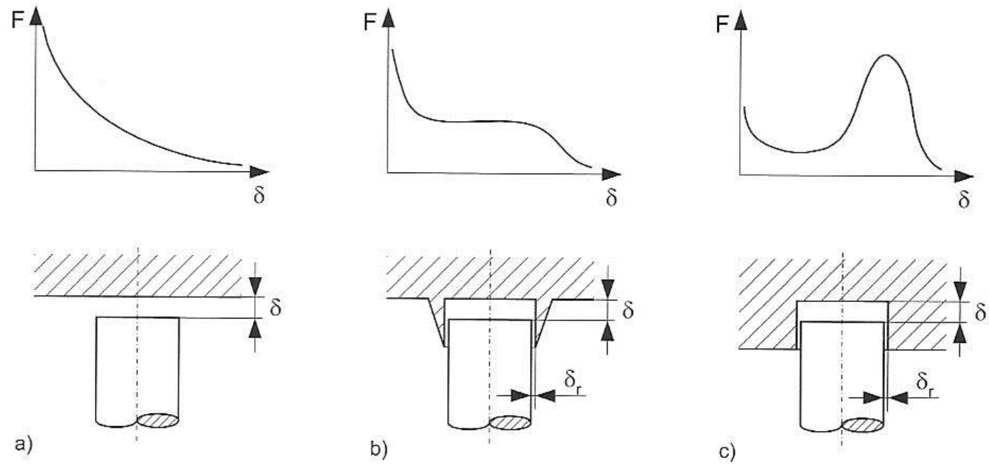


Figure 58 Force –stroke characteristic using different matching pieces (Kallenbach, Eick, Quent, Ströhla, Feindt, & Kallenbach, 2003, p. 75)

A combination of type b) and c) would be best for the thesis on hand considering a spring characteristic.

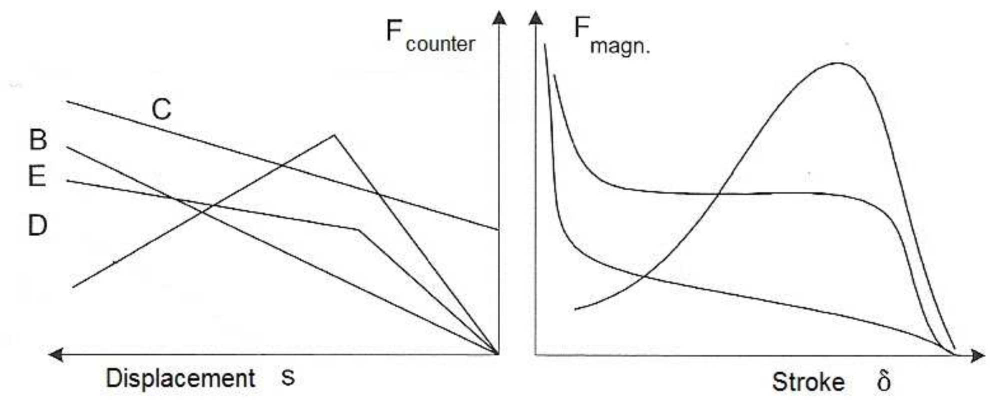


Figure 59 Comparison of spring characteristics and force-stroke characteristic of an electro magnet based on (Kallenbach, Eick, Quent, Ströhla, Feindt, & Kallenbach, 2003)

The figure above provides the comparison of spring and magnet characteristic. B refers to a conventional spring, C to a preloaded and D as well as E to combined springs. (Kallenbach, Eick, Quent, Ströhla, Feindt, & Kallenbach, 2003, p. 290)

In order to find out whether the electro magnet is a fitting component usable to charge the actuator source, a calculation is conducted, using the following equations:

$$\Theta = I \cdot N \tag{3.62}$$

$$\Theta = \delta \cdot \frac{B}{\mu_0} \tag{3.63}$$

$$J = \frac{\Theta}{A_{Cu}} \tag{3.64}$$

$$P_v = V_{Cu} \cdot J^2 \cdot \rho_{Cu} \tag{3.65}$$

Θ	magnetic flux [Wb=Vs]
I	amperage [A]
N	number of turns
δ	magnetic field line in the air gap [m]
μ_0	induction constant $1.257 \cdot 10^{-6}$ [WB/Am]
B	magnetic flux density [Wb/m^2]
A_{Cu}	cross sectional area [m^2]
J	current density [A/ m^2]
V_{Cu}	volume [m^3]
ρ_{Cu}	specific electrical resistivity of the element [Ωmm^2/m]
P_v	power loss [W]

Rotational electric motor

Basically all electric motors convert electric energy to mechanic energy. Every type consists at least of one stationary and one rotating element, called stator and rotor. The creation of torque is a result of force established due to the current passing in a conductor placed in a magnetic field. It is possible to divide between synchronous, asynchronous and direct current motor when talking about rotational electric motors. (Uhlmann, 2008, p. 6)

A major advantage concerning electric motors is that the same type of energy can be used for sensor system, actuation and control.

Most important when talking about the usage of electric motors for the actuation of a spring loaded brake, an adequate gear needs to be provided. (Hermanns & Müther, 2005, p. 13)

Linear electric motor

As the linear electric motor belongs to the electric motors, the same basics can be applied as with the rotational electric motor. However, the linear electric motor has only two popular designs, synchronous and asynchronous. (Uhlmann, 2008, p. 6) Nevertheless, it is possible to consider the linear electric motor as uncoiled rotational motor. Basically it consists of a primary part through which a current is passed (comparable to the stator) and a secondary part which is the moving translator (analogous to the rotor). (Grote & Feldhusen, 2005, p. T9)

The major advantage of the linear electric motor compared to the rotational is, that for the present purpose, no gear is needed to charge the energy source. (Hermanns & Müther, 2005, p. 13)

Pneumatic/ hydraulic Motor

Basically fluid motors are pumps used in reverse direction (Watter, 2008, p. 126). Both work using the suppression principle. With the right control of the fluid flow, switching between motor and pump is possible. Basically the motor converts hydraulic power into mechanic. According to the design it is possible to distinguish between motors working in rotational or translational direction. (Will & Gebhardt, 2011, pp. 121-122)

Concerning fluid motors, it is reasonable to divide between hydraulic and pneumatic types. The major difference is that pneumatic motors are equipped with an open system while hydraulic have to consist of a closed one. Certainly the hydraulic motors benefit from using an incompressible media. An increased force density can be achieved with the hydraulic motor compared to the pneumatic. However, fluid motors need a sufficient pressure supply. (Hermanns & Müther, 2005, pp. 11-13)

Pneumatic/ hydraulic Cylinder

The basic idea is a cylinder closed by a movable piston. In case some sort of fluid is feed into the chamber, created by cylinder and piston, a force F is created. The most important equation in this context is:

$$F = p \cdot A \tag{3.66}$$

In this case p refers to the pressure in the chamber and A to the piston surface. Basically the energy turnover is proportional to cylinder pressure and volume. Therefore the length of the cylinder should only be a little bit larger than the stroke of the piston.

Basically two types of fluids are used: compressed air and hydraulic oil.

Compressed air possesses of the advantage to be completely eco-sensitive. Ambient air can be used. The current pressure limit for air is defined for 10bar. This is a result of the fact that the human skin gets permeable for air with a higher pressure and the

compressed air system is an open system. Just as with the pneumatic motor. Pitifully it is compressible and a quite slow system. The hydraulic oil on the other hand in incompressible.
Prevailing pressure vanishes immediately in case leakage is present and therefore hydraulic systems can be operated with pressures up to 600bar. This makes it possible to achieve a compact design. However, the hydraulic oil is hazardous to the environment and burnable what causes big problems for railway vehicles. (Hermanns & Müther, 2005, pp. 9-10)

Shape memory alloys

A Shape Memory Alloy (SMA) is an alloy which, when deformed (in the martensitic phase) with the external stresses removed and then heated, will regain its original "memory" shape (in the austenitic phase). Shape memory alloys can thus transform thermal energy directly to mechanical work. (Otsuka, K. & Ren, X., 1999) and (Ju, D.-Y. & Shimamoto, A., 1999) as referred in (Vessonen, 2003, p. 12)

Considering the thermal memory effect, one-way and two-way effect can be differentiated. The one-way effect refers to material that changes shape when heated, but does not perform another shape change after re-cooling. The two way effect on the other hand is present with material that has two memorized shapes. One is achieved in case a certain temperature is reached, the other after cooling. This ability makes the two-way effect shape memory material perfect for actuation. This regaining of shape can be forced from the outside, by e.g. force, in this case it is called extrinsic two-way effect. The intrinsic two-way effect is assured if the material is trained due to thermo-mechanical treatment to regain the shape on its own. In general, the extrinsic two-way effect is favored, though actuators have to be built more complex, due to the following reasons:

- More work cycles can be performed
- Lower memory loss
- Higher actuation forces and larger actuation distances
- No training is necessary

(Memory Metalle GmbH, 2011)

Piezo element

Basis for operability of piezo-actuators is a piezo-electric material. This material deforms if an electric field is applied to it. The resulting deformation is able to create force. (Uni Saarland, 2012)

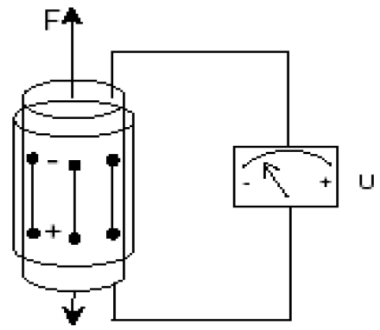


Figure 60 Visualization of the piezo-electric effect (Uni Saarland, 2012)

Piezo-electric material converts electric energy directly to mechanic energy. This is done very fast; the reaction time basically is within the microsecond range. Also the power density of piezo-electric material is tremendous compared to other actuation possibilities. Additionally due to the fact, that the piezo effect is based on an electrical field, no magnetic field is created. (Physik Instrumente (PI) GmbH & Co.KG, 2012, p. 4) This would be a special advantage considering the electromagnetic compatibility, which is regulated for railway application in DIN EN 50121-3:2006.
Moreover the piezo-actuators hardly need energy in the static load condition, even though heavy loads might be applied and finally piezo-actuators are not subject to wear. The movement is caused by effects in the crystalline rigid body. (Physik Instrumente (PI) GmbH & Co.KG, 2012, p. 4)

3.2.4 Basics for linear enhancement

In this thesis the term linear enhancement is used to describe a linear development of an existing system. In this case the presently used pneumatic tread brake by Faiveley, described below, is the origin. Basically there are two possibilities for this enhancement. On the one hand, the working media for the compression cylinder – compressed air – stays the same on the other hand the working media can be changed to hydraulic. Furthermore components which are part of the energy source or power amplifier can be modified in the course of linear enhancement.

Certainly it is necessary to understand the present solution before improvement potential can be spotted.

Current benchmark

Currently the Faiveley BFC Tread Brake is supposed to be installed at the Syntegra bogie for MMC.

The following paragraphs are derived from (Faiveley Transport, 2007).

The BFC (Brake Friction Concept) brake unit is a compact pneumatic brake unit with an internal mechanical force amplification and slack adjuster. The compact and low weight design makes the BFC especially suitable in installations with limited space availability and the internal amplification makes interface standardization possible.

The BFC actuator can be used not only to actuate tread brakes; disc brakes can also be operated using this actuator design. Additionally it has to be mentioned, that the BFC is not only a spring loaded actuator. It consists of an active pneumatic brake design and can additionally be equipped with a spring loaded parking brake.

For this thesis the spring loaded parking brake is of special interest. Certainly the overall size and weight used for the benchmark are based on the conventional design combining active and parking brake actuator. However, what is really needed about this actor is the parking brake.

Technical Data for the spring loaded parking brake

Cylinder diameter	178mm
Max. working pressure	6 bar
Dimensions width Height	200 x 200 mm 164,190,224,244 mm
Maximum parking brake force	35kN
Weight	63kg

Table 8 Technical data of Faiveley Transport BFC spring loaded parking brake

As one can see from the provided data, the parking brake force of 35kN is fitting perfectly for the demands calculated in section 4.1.

Principle of operation

As mentioned previous the BFC actuator is designed to use compressed air as activation media. In order to achieve sufficient braking force it utilizes the wedge principle for power amplification.

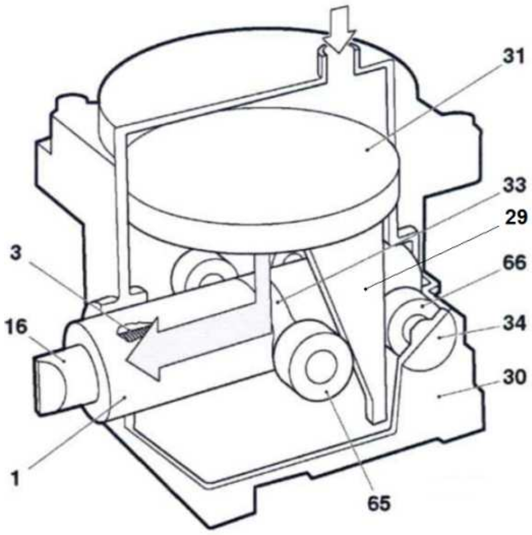


Figure 61 Principle of operation of the Faiveley BFC based on (Faiveley Transport, 2007)

Two wedges (29) are connected to the piston (31) and located at their lower end between two roller bearings (65, 66) each. As the piston (31) moves downwards due to increased pressure in the active brake or due to decreased pressure in the spring parking brake, the wedges (29) move down as well. This downward movement is transmitted via the movable left roller bearing (65) into a horizontal movement of the push rod (16). The actuator force is transmitted via the pushrod (16) and the slack adjuster (1, 3) to the brake shoes.

The slack adjuster works at the return stroke. Its purpose is to guarantee a constant brake shoe clearance no matter how much wear the brake shoe linings or the wheel already possess.

The spring parking brake is released by air pressure as mentioned previous. The parking brake of the BFC unit contains of an emergency release mechanism. This enables the user to release the brake in case no air pressure can be provided for some reason. Furthermore optional additional electronic features like blocking device or piston indicator can be applied.

Spring parking brake

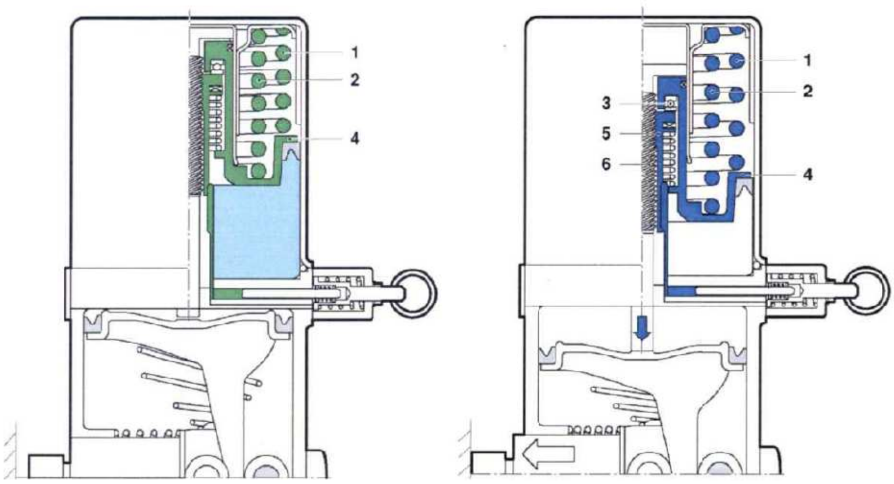


Figure 62 Spring loaded parking brake released (a) and applied (b) (Faiveley Transport, 2007)

Figure 62 shows the spring parking brake in its release and applied position. In the released position the springs (1, 2) are compressed by the air pressure in the blue chamber. In case the parking brake is activated the pressure chamber is vented. The springs expand. As a result the piston (4) moves downward. This in turn causes the spindle (6) to move down too. The downward force caused by the springs is transmitted into the piston, passes on the force to a bearing (3) and a nut (5). The nut is connected to the spindle.
The spindle pushes down the piston of the service brake and introduces the necessary movement of the wedges.

3.2.5 New actuation possibilities

Presently available are, electro- mechanic actuators besides the common electro-pneumatic and electro-hydraulic actuators. These actuators are considered to belong to the new actuation possibilities. According to the literature research there are two basic concepts are present. On the one hand the simple electro- mechanic and on the other hand the self amplifying electro mechanic actuators.

The following paragraphs are used to provide an overview on the different actuators found during patent and literature research.

Electro-mechanic actuators

As explained in Chapter 3.1.3 Realization of braking application electro-mechanic actuators use electric motors or solenoids as a power source.

These actuators aim to replace the high-maintenance electro-pneumatic and electro-hydraulic brake systems, especially in the public transport sector (Barner & Martensson, 2000).

Faiveley Transport

Faiveley Transport offers an electro-mechanic brake system EMB which can perform all brake applications required by railway vehicles. While service brake and emergency brake applications are performed using active electro- mechanic actuation the parking brake application is done using a spring loaded actuation.

Basically the system consists of three major components. The electronic brake control (1), a distribution unit (2) and the active electro- mechanic actuator combined with the passive spring loaded brake (3). The actuator is originally designed for a disc brake. However, with the right setting, it is possible to use it for triggering a tread brake as well.

The electronic brake control is the interface between vehicle and braking system. The brake offered by Faiveley Transport uses a regular unit, which is also used for electro-pneumatic or electro- hydraulic systems.

The distribution unit is used as physical connection between control unit and actuation system. It provides the power supply for the brake force creation mechanism, safety loops, network and so forth.

Finally the electro- mechanic brake actuator is designed to actuate friction brakes and is powered solely electrical. The active brake is powered by an electro motor. The force of the active brake can be maintained using a non-returnable clutch for an infinite time. Additionally the spring brake can be used to serve the same function. This causes redundancy of the braking system.

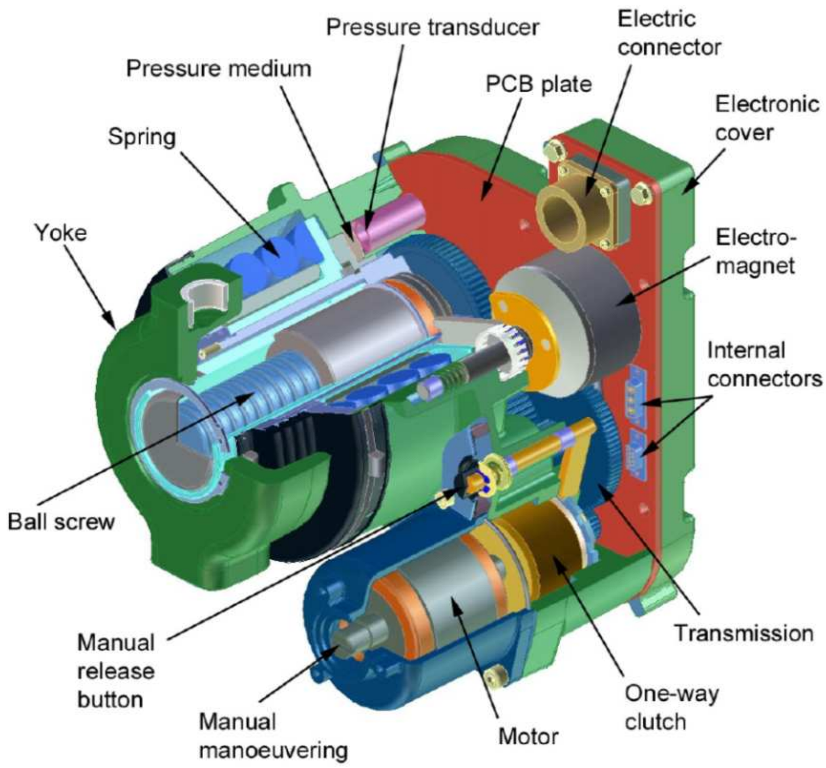


Figure 63 Electro- mechanic brake actuator by Faiveley Transport (Faiveley- Transport, 2008)

Considering the working principle of the actuator, it has to be divided between the active brake and the passive brake. The active brake application is executed by an electrical motor. If it turns in direction 'brake' the braking force is transmitted via one-way clutch, transmission and ball screw to the friction lining. The further the motor turns in this direction the higher the force. On the other hand if the motor turns in direction 'release' the transmitted brake force is decreased. Is the motor turned off, the one-way clutch secures the existence of the brake force.

The passive brake application is executed by the spring loaded actuator. The spring is loaded when the vehicle is first equipped with the brake. The spring is frozen in the loaded condition using an electro magnet. The magnet is powered by an emergency shutdown. In case this power supply disconnected, the spring is released.

In order to release active or passive brake, a manually operated release and an electrical release is provided.

(Barner & Martensson, 2000)

Raco Schwelm

The suitable brake actuator provided by Raco Schwelm is the GBM III. Just as the Faiveley Transport EMB, this actuator is able to perform all brake applications necessary for railway vehicles. Certainly also this system needs a brake control system.

Basically the GBM III is a spring loaded actuator and can be considered as indirect brake. It consists of three major parts. The first part (1) contains a disc spring set, a connecting rod and an eccentric shaft. The second part (2) is a planetary gear set and the last part (3) is represented by the direct current motor and the solenoid brake.

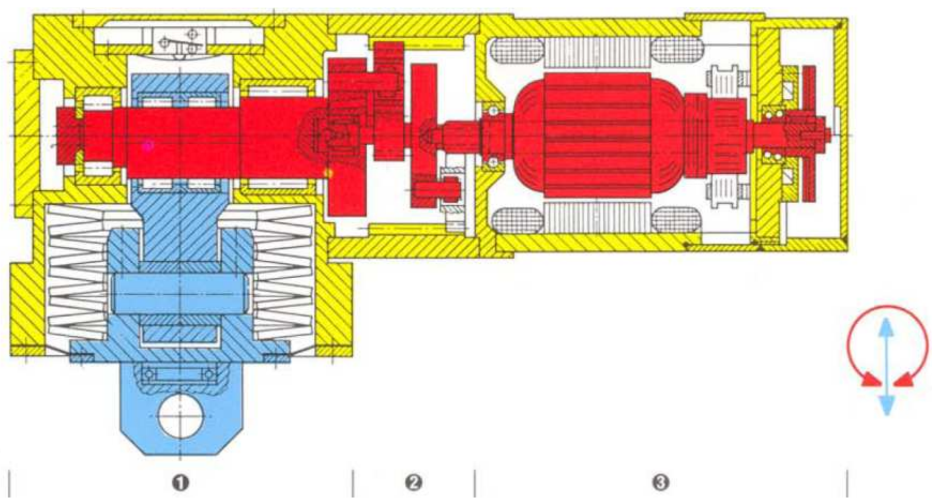


Figure 64 Raco Schwelm GBM III (Raco Schwelm, 1994)

In order to release the brake, the electric motor is switched on. The eccentric shaft turns the rotational movement of the electric motor into a linear movement of the connecting rod. Using this principle the spring is loaded. As soon as the spring set is fully loaded, the solenoid brake conserves the energy state of the spring. In case of a brake application the solenoid brake is released and the spring expands, causing the braking force. Additionally in case the energy breaks down the brake is applied automatically because the solenoid brake is not working.
(Raco Schwelm, 1994)

Knorr- Bremse

Knorr- Bremse owns several patents on electro-mechanic brakes. However, these patents are not known as sellable products. Knorr- Bremse keeps a low profile on the whole issue of electro-mechanic brakes. It is supposed by the author that Knorr- Bremse has no intention to provide electro-mechanic solutions as long as the electro-pneumatic are sufficient for customers and competition.

However, the two interesting actuators are EP 1218647B1 and DE 10058925A1 are presented here.

The following paragraphs are based on EP 1218647B1.

The actuator consists of the major parts spindle (60), electromagnetic immobilization brake (26), electric motor (24), gear (22), and spring (14). The idea of this actuator is the compact design combining two different clamping mechanisms. The compact design is achieved via coaxial arrangement of spring (14) and spindle (34). Basically the brake can be actuated via the electric motor (24) or the spring (14). The spring (14) represents the safety level in case the electro motor (24) is out of service and it is locked using a lock bar (76) which is actuated electromagnetically. In order to release immobilization brake application the motor (24) activates the spindle (34) which causes the spring (14) to release. Additionally an emergency release (88) is provided. It enables the release of the spring in case the motor (24) is out of service. (Wolfsteiner, Fuderer, & Staltmeir, 2000)

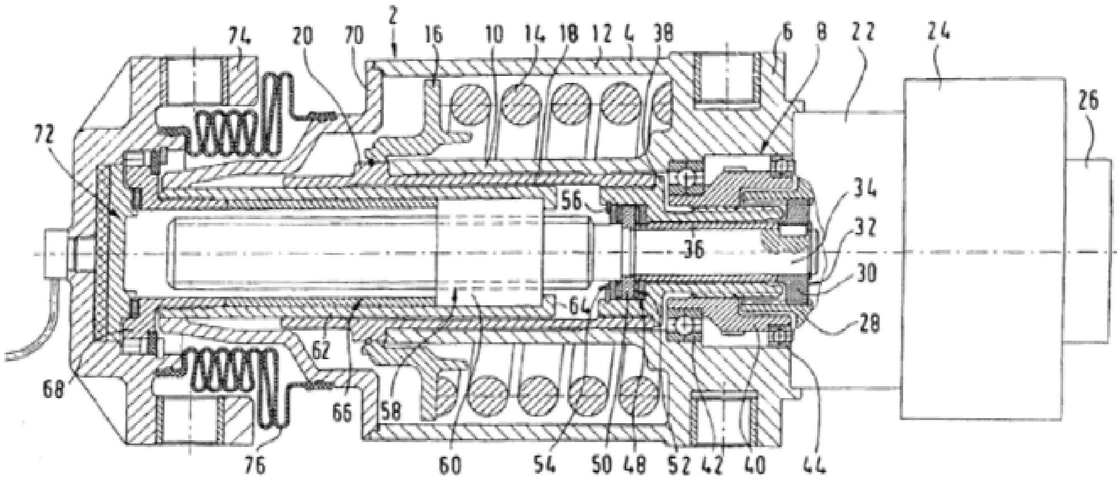


Figure 65 Knorr- Bremse actuator (Wolfsteiner, Fuderer, & Staltmeir, 2000)

The second actuator is based on the patent DE 10058925A1.

This patent also refers to the electro-mechanical actuators. Special about it is the parallel connection of spring (10) as well as electric motor (24) and spring (12).

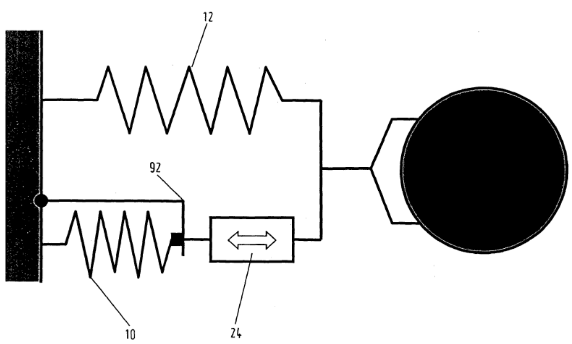


Figure 66 Parallel connection of DE 10058925A1 (Grundwürmer, Staltmeir, & Vohla, 2002)

Due to this special connection the electric motor (24) does not have to create braking force in a defined operating point. This point is larger than zero but smaller than the maximum braking force. In this case the spring storage provides the braking force. In case a larger braking force is desired the motor supports the spring, in case a lower force is needed, the motor works against the spring. Due to this fact the spring can be smaller than with a pure passive system. The maximum braking force is achieved in combination with the motor and not solely by the spring. On the other hand, the motor can also be smaller because it is supported by the spring as well. (Grundwürmer, Staltmeir, & Vohla, 2002)

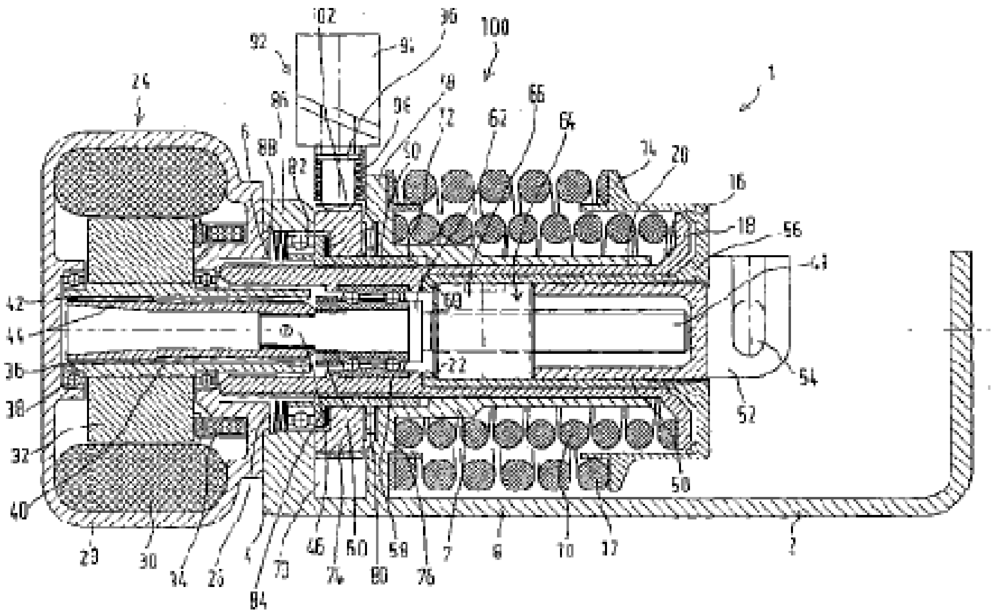


Figure 67 Cross section of actuator DE 10058925A1 (Grundwürmer, Staltmeir, & Vohla, 2002)

As visible Knorr- Bremse holds some interesting patents, however, the introduction of a real product to the market is uncertain.

Self amplifying actuator concepts

Self amplifying brake possess' the ability to create the force necessary to fasten the brake from the kinetic energy of the vehicle. It is not possible to achieve regenerative braking with this idea. However, the activation force necessary is very low. (Hermanns & Müther, 2005, p. 39)

Wedge brake

This brake uses the power amplification of a wedge. By the revolving of the wheel the wedge is forced deeper into an adapter, hence the braking force is increased the further the wedge moves. (Hermanns & Müther, 2005, pp. 41-42)

Basically a hyperbolic characteristic describes the relation between braking force and actuation force. The following figure illustrates the amplification factor C in relation to the friction coefficient.

$$C = \frac{2}{\tan(\alpha) - \mu} \tag{3.67}$$

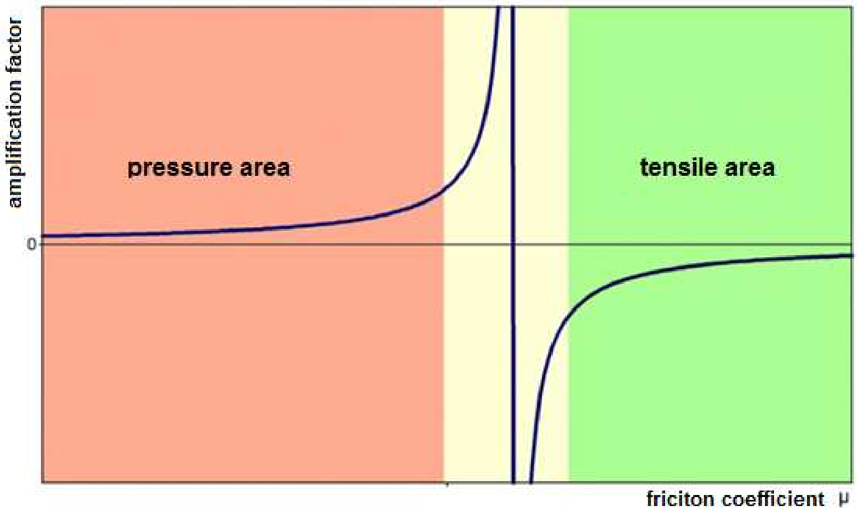


Figure 68 Amplification factor – friction coefficient (Hermanns & Müther, 2005)

$\tan(\alpha) > \mu$	Pressure area
$\tan(\alpha) \approx \mu$	Optimum
$\tan(\alpha) < \mu$	Tensile area

Figure 68 points out that there are different areas and amplification factors. By the time μ is small the actuator has to provide pressure force in case the wedge should create a constant normal force. If μ is large the wedge would be pulled into the adaptor without the application of tensile force. This force has to be provided by the actuator. In case the wedge is not applied with tensile force, it would lock. If μ is in the area of $\tan(\alpha)$ a nearly infinite amplification is possible, certainly with changing algebraic signs. Therefore a sufficient control of the actuator has to be provided to guarantee the appropriate loading of the wedge. (Hermanns & Müther, 2005, pp. 44-46)

In order to control the wedge brake a position control has to be applied rather than a force control. A force controlled system would reach its limit and fail. In order to apply the position control the disturbance variable is the friction coefficient. Basically it is possible to find a linear relation between wedge position and clamping force. Yet the system parameters change constantly. An elaborate control element is needed, because in order to obtain a stable system the control has to react faster than the system can change. (Gombert & Hartmann, 2006)

It gets obvious that the control of the wedge brake is very complex.

Additionally, if it should be possible to perform parking brake applications the wedge brake has a certain disadvantage. One brake can only perform the immobilization in one direction of movement. Therefore always a couple needs to be provided, with both wedges operating in the tensile area. (Hermanns & Müther, 2005, p. 48)

Vienna Engineering VE- Brake

One of the latest ideas on self amplifying brakes is the VE- brake.

This brake is an electro-mechanic brake with low actuation energy, non-linear actuation mechanism with low losses, high transmission ratio, and a designable part of self amplification at high braking forces. (Putz, 2010, p. 2)

It certainly consists of an electro motor with gear. As power amplification a crankshaft-like principle is used. (Putz, 2010)

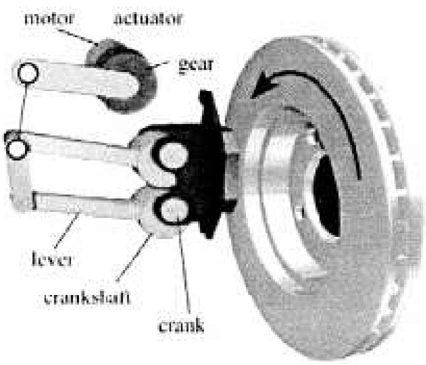


Figure 69 VE- brake principle (Putz, 2010)

Even though the presentation of results seems promising from an amplification perspective, the VE brake possess of a critical disadvantage for this thesis. The VE- brake is equipped with an auto release function in case motor or power failures occur. Therefore an immobilization brake application cannot be performed. (Putz, 2010)
However, at the Vienna engineering homepage it is mentioned that such an application should be provided. Yet, no advisory opinion or related press release on the way to achieve this function was available. Additionally the whole working mechanism is kept secret. It was not possible to find detailed press releases concerning this topic while this thesis was documented. Therefore the VE brake is simply mentioned to represent the latest developments.

4 Calculation results and decision on components

This chapter focuses on the different calculation results that provide the basis for the thesis on hand.

First it is required to clarify the necessary parking force. This force is most important for the dimensioning of the spring loaded actuator.
Furthermore, knowing this force it is possible to calculate the deceleration possible to achieve with the actuator. With this deceleration, the braking distance can be derived.
Another important thing about the design of a braking system is the thermal stability. This topic is covered in the present chapter as well. Purpose is to figure out which temperatures occur during the different brake applications given by the performance specification.
Additionally the brake setup is verified using a computerized simulation.

Having defined these general conditions, it is possible to evaluate the real-world applications derived from the physical principles for their applicability.

The next logical step is to search for improvement potential of the current benchmark, and also to evaluate the different alternatives provided on the market.

4.1 Necessary parking force

As clarified in chapter 3.2.2 Performance specification of the braking system, it is necessary for the braking system to provide parking brake application fulfillment at certain sloped tracks with a defined load capacity.

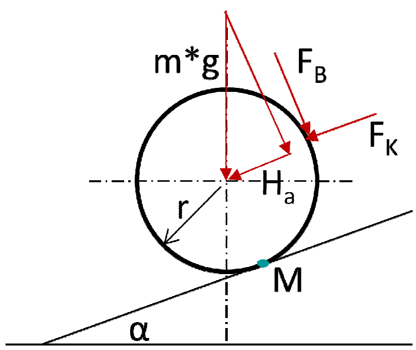


Figure 70 Forces during parking brake application

The equations to be used read as follows:

$$H_a = m_{axle} \cdot g \cdot \sin(\alpha) \tag{4.1}$$

$$F_B = F_K \cdot \mu_K \tag{4.2}$$

$$\sum M_M = 0 \tag{4.3}$$

In order to obtain a stationary condition the torque caused by the downhill slope force H_a needs to be compensated by the torque caused by the sliding friction force F_B. In this case the sliding friction force is the product of brake shoe force F_K times the friction coefficient μ_K.

The resulting overall equation to calculate the brake shoe force is defined as:

$$F_K = \frac{H_a}{\mu_K} \tag{4.4}$$

When considering the different requirements from the characteristics mentioned above, following results for the brake shoe force are found:

	Requirement specification MMC	EN 13452-1:2003
Gravity acceleration [m/s2]	9.81	9.81
Gradient [%]	5	4
Maximum axle load[kg]	14000	14000
Friction coefficient	0.2	0.2
Brake shoe force [kN]	**34.28**	**27.44**

Table 9 Brake shoe force and related values necessary for calculation

Certainly for the following considerations the parking brake force derived from the requirement specification MMC will be used.
The must- have for the braking system are 34.28kN per axle in order to provide enough force for parking brake application on a gradient of 5%.

However, a not-spinning wheel does not necessarily mean that the vehicle will not move. For this perspective, it is possible to defined the condition $H_a < F_{stick}$.

$$F_{stick} = m_{axle} \cdot g \cdot \cos(\alpha) \cdot \mu_H \tag{4.5}$$

The necessary minimum friction to stick on the gradient can be defined using Equation 4.1 equal to Equation 4.5.

$$\mu_H := \frac{\sin(\alpha)}{\cos(\alpha)} = 0.05$$

A minimum adhesion coefficient of 0.05 has to be prevailing to prevent sliding on the sloped track. This result is consistent with the adhesion coefficients measured by Metzkow. (Wende, 2003, p. 177)

4.1.1 Taw condition

Wheel slide is totally unwanted when it comes to railway vehicles.
Special equipment, such as the wheel slide protection, tries to prevent the blocking of the wheel set during braking. Blocking is a consequence of over braking. However the worst consequence of blocking is a wheel flat. It is caused by sliding of a blocked wheel on the tack. (Knorr-Bremse, 2002, p. 220)

During the life of a railway vehicle it can occur, that taw of the vehicle is necessary. In order to prevent wheel flats in case the brake can't be released any more it is required that the friction lock between brake shoe and wheel collapses before the one between wheel and track does $F_B < F_H$. Considering the minimum axle load, the adhesion coefficient necessary to achieve the condition mentioned above can be calculated.

This means Equation 3.3 has to be larger than Equation 3.10.

$$\mu_H := \frac{\mu_K \cdot F_K}{m_{axle} \cdot g} = 0.05$$

Again the minimum value of μ_H has to be 0.05.

4.2 Deceleration derived from actual actuator

In case an emergency brake application has to be performed, for whatever reason by the last safety device – the tread brake – the resulting deceleration is certainly lower than expected by the EN 13452-1:2003. Therefore it is reasonable to calculate the actual deceleration.

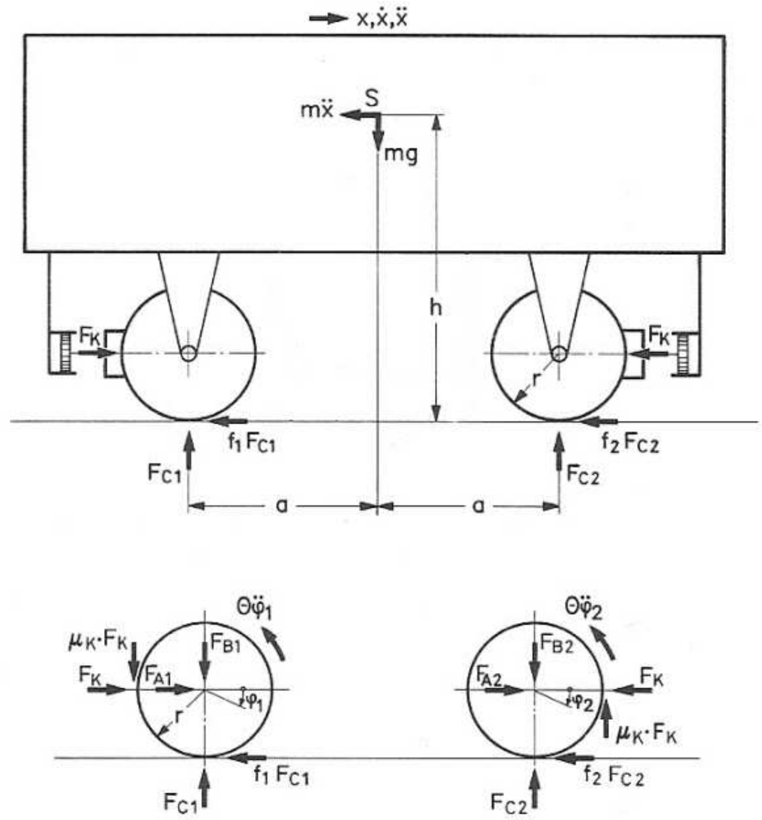


Figure 71 Forces on the wheel (Saumweber, Gerum, & Berndt, 1990)

According to literature, the calculation is done using the following equations:

$$\Theta \cdot \ddot{\phi}_{1,2} = -\mu_k \cdot F_K \cdot r + f_{1,2} \cdot F_{C1,2} \cdot r \quad (4.6)$$

$$F_{C1,2} = \frac{1}{2} \cdot m \cdot g \pm m \cdot \ddot{x} \cdot \frac{h}{2 \cdot a} \quad (4.7)$$

$$a = \ddot{x} \approx r \cdot \ddot{\phi} \quad (4.8)$$

These equations are quite elaborate. Many values need to be considered and therefore companies often simplify the deceleration calculation using a mass coefficient. So does Siemens. The calculation considers the basic equation of mechanics, based on Newton's second law. Whereby the sum of the forces F is known as the mass m times the acceleration a.

$$\sum F = m \cdot a \quad (4.9)$$

Certainly this equation does not consider the rotary mass therefore a coefficient λ is introduced.

$$\sum F_b = m \cdot \lambda \cdot a \qquad \textbf{4.10}$$

This equation is valid for a wheel translation without slippage. The coefficient λ is according to experience between 1.06 for trailer bogies and 1.1 for motor bogies. (Gralla, 1999, p. 2)

The present deceleration is considered to be as follows:

$$F_B := 7000N$$

$$\lambda := 1.1$$

$$m_{axle} := 14000kg$$

$$a := \frac{F_B}{m_{axle} \cdot \lambda} = 0.455 m \cdot s^{-2}$$

With the given values a deceleration of 0.455 m/s^2 is achievable.

4.2.1 Braking distance

With the deceleration defined, it is possible to calculate the braking distance in case the tread brake has to stop the vehicle. In this consideration it is assumed that the braking system is able to provide a maximum equivalent reaction time like defined in (EN-13452-1, 2003).

The basic equation for the braking distance is:

$$s_b = v_0 \cdot t_e + \frac{{v_0}^2}{2 \cdot a_e} \qquad \textbf{4.11}$$

According to the values given in (EN-13452-1, 2003) and a velocity of 90km/h, the braking distance would be as follows:

	Emergancy brake 1	Emergancy brake 2	Emergancy brake 3	Emergancy brake 4	Safety brake
ae [m/s^2]	1	1	1	1	0,7
te [s]	1,5	1	1	1	2
v0 [m/s]	25	25	25	25	25
Braking distance [m]	350,0	337,5	337,5	337,5	496,43

Table 10 Braking distance calculated with the values required by (EN-13452-1, 2003)

However, the present deceleration provided by the tread brake is higher. Therefore the braking distance decreases.

	Emergancy brake 1	Emergancy brake 2	Emergancy brake 3	Emergancy brake 4	Safety brake
ae [m/s^2]	0,455	0,455	0,455	0,455	0,455
te [s]	1,5	1	1	1	2
v0 [m/s]	25	25	25	25	25
Braking distance [m]	724,3	711,8	711,8	711,8	736,81

Table 11 Braking distance calculated with the deceleration provided by the tread brake

As to be seen from the calculation above, the longest braking distance would occur in case if a maximum equivalent reaction time of two seconds.

4.3 Thermal conditions

Also the thermal effects during braking are of interest. Thermal considerations are important because they can have a negative influence on the wheel, as Figure 72 proofs.

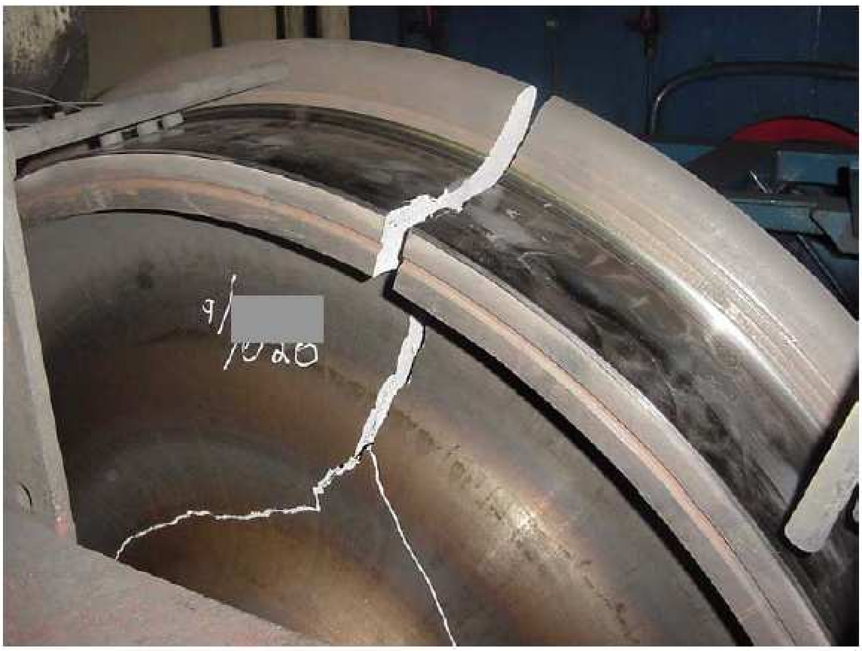


Figure 72 Thermal stress on wheel (Dr Merkel, 2005)

The wheel gets heated due to the friction between brake shoe and the running surface. The temperatures can be very high and due to spinning, the wheel gets in contact with the cold track. This causes rapid cooling. The material changes hardness due to the thermal condition change. Damages such as running surface cracks, striations and spall of brittle running surface are common results of the thermal stresses. Also thermal stresses are involved in case of wheel flats, however, in this case the temperature increases between the wheel and the track, as the wheel slides without rotation. (Gutschi, 2006, p. 1)

In the present case the thermal conditions for three braking applications were simulated with the Siemens AG TBS tool.

This tool is used to calculate the temperature characteristic in a wheel that is used in a tread brake setup. However, no thermal stresses are calculated. Still it is possible to derive important information on the braking system from the calculation. Defining the brake shoe lining is made possible as well as estimations on the operational demands on the wheel. (Gutschi, 2006, p. 1)

In order to perform a thermal calculation using the Siemens TBS tool, it is necessary to first define the car configuration. For the Metro Mid Cap there are seven configurations known. The configurations are known as follows:

Four cars	Low cost High cost	MC + M1 + T1 + MC MC + M1 + M1 + MC
Five cars	Low cost Medium cost High cost	MC + T1 + M + T1 + MC MC + M1 + T + M1 + MC MC + M1 + M + M1 + MC
Six cars	Low cost High cost	MC + T1 + M + M + T1 + MC MC + M1 + M + M + M1 + MC

Table 12 Different car configurations of the MMC (Stützle & Schraud, 2010)

MC = motor coach with driver’s cab,
M = motor coach (center),
M1 = motor coach (center) with vehicle power supply (HBU, BLG, battery),
T = trailer coach (center),
T1 = trailer coach (center) with vehicle power supply (HBU, BLG, battery)

(Stützle & Schraud, 2010, pp. 6-7)

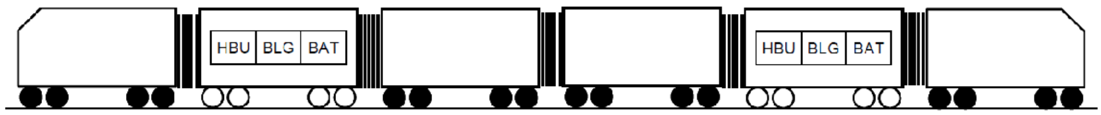


Figure 73 Six cars low cost (Stützle & Schraud, 2010)

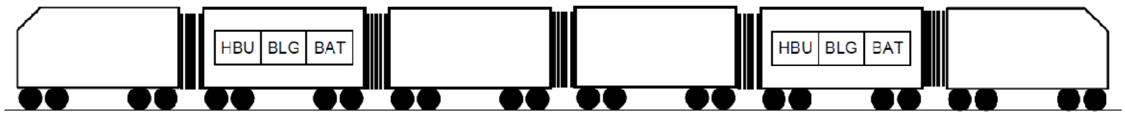


Figure 74 Six cars high cost (Stützle & Schraud, 2010)

The calculation made is focusing on the six car configurations due to the largest mass to stop and decelerate. Additionally the fife car low cost configuration is considered due to its unique combination ratio of two trailer and three motor coaches.
The simulation program additionally needs several input parameters presented hereafter using the example for six cars high cost configuration and an emergency braking application.

Simulation:	**Simulation 1**	**Explanation**
Load case:	Emergency 6 cars h.c, 2 times	Describes the calculation content; basically a name
Train configuration [-]:	6 cars h.c.	Defines the configuration of the vehicle, how many motor and trailer axles are present
Mass [-]:	high cost 6	Defines the mass and additionally the mass derived from the rotation coefficient.
Brake course scenario [-]:	emergency	Defines the scenario to be calculated. Possible are emergency brake applications, route profiles etc.
Brake failure scenario [-]:	emergency 6 cars h.c.	Defines how which and how many of the present brakes do work.
Axle configuration [-]:	“Tread only”	Defines the wheels, brakes and motor performances for a specific axle
Blending [-]:	ED out of service	Blending basically refers to the combination of different braking systems to provide an overall deceleration. In many cases the electro dynamic brakes are combined with friction brakes.
ED Brake [-]:	No	Presence of electro dynamic brakes
Additional brake 1 [-]:	No	Presence of an additional brake system
Additional brake 2 [-]:	No	Presence of an additional brake system

Train resistance [-]:	No	
Maximum speed [km/h]:	90	Defines the maximum speed of the vehicle
Deceleration [m/s²]:	0.455m/s²	Defines the deceleration. Can be a constant value or a characteristic curve. In the present case it was considered as constant value
Acceleration [m/s²]:	0.85m/s²	Defines the acceleration of the vehicle. This is necessary when it comes to route profile simulation, where acceleration and deceleration are needed.
Wheel-rail adhesion [-]:	0.15	Defines the adhesion between track and wheel.
Initial temperature [℃]:	40	Defines initial temperature of the brake.
Ambient temperature [℃]:	40	Defines the temperature of the environment.
Station time [s]:	120	Defines the time spent at a station.
Turn time [s]:	120	Defines the time to turn in case a certain route is simulated several times.
Simulated axle [-]:	MA1	Defines the axle to be simulated

Table 13 Input data outline for Siemens TBC

Special attendance is given to the axle configuration: for the present thesis "tread only" was developed. It defines the wheel, the brake and the motor performance. Concerning the wheel several parameters have to be considered:

Wheels:	Rd-690
Ø outer wheel rim, new [mm]:	690
Ø outer wheel rim, old [mm]: Ø inner wheel rim [mm]: Width of the wheel rim [mm]: Ø outer hub [mm]: Ø inner hub [mm]: Width of the hub [mm]: Width of the web [mm]:	630 585 135 210 138 130 20
Material [R8, R7]:	R7

Table 14 Necessary input concerning the wheels for Siemens TBC

Concerning the tread brake the following input is essential:

Tread brakes:	**TB-**
Tread configuration:	1Bg
Block width [mm]:	80
Block length [mm]:	320
Material:	Org.

Table 15 Necessary input concerning the braking system for Siemens TBC

The tread configuration refers to the different possibilities to adjust tread brake shoes

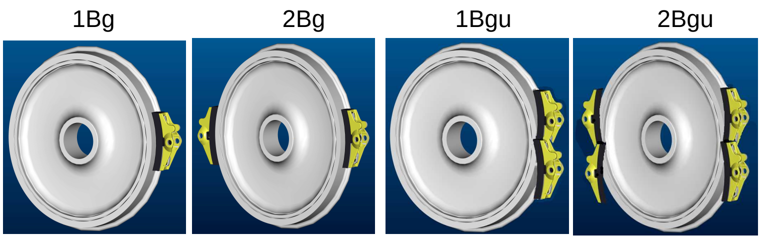


Table 16 Different tread brake setups for Siemens TBC

In the present thesis the configuration 1 Bg is chosen.

Also the material is of interest. There are basically organic and sinter brake shoe linings. Certainly several suppliers offer a large diversity of brake shoe linings. Companies of reputation are e.g. Becorit, Bremskerl or Jurid. For the thesis on hand an organic Bremskerl lining seems to be most promising. It is known as Bremskerl 302 and can endure temperatures up to 620°C if applied for a short time. The maximum temperature for a permanent application is supposed to be 450°C.
The upcoming calculation will proof that this high temperatures need to be endured by the brake shoe lining.

Additionally the motor performance needs to be considered in the simulation. It contains information on the traction force per axle and the electro dynamic braking force per axle.

Further to be mentioned is the fact that configurations consisting of motor and trailer bogies have a different deceleration compared to a configuration of solely motor bogies. As calculated in Chapter 4.2 the deceleration of a motor bogie is 0.455 m/s^2. It is considered due to experience that the trailer bogie has a deceleration of 1.2 m/s^2.

Used as basic equation is $\sum F_b = m \cdot \lambda \cdot a$

Braking force motor axle: 7kN

Braking force trailer axle: 18.489kN

Considering the two other configurations including trailer bogies, the overall deceleration is to be defined as follows:

Car configuration	Number of trailer axles	Number if motor axles	Maximum load [kg]	Coefficient	Deceleration [m/ s^2]
6 cars l.c.	8	16	14000	1.1	0.703
5 cars l.c.	8	12	14000	1.1	0.753

Table 17 Deceleration related to the car configuration

Certainly it needs to be evaluated if these different decelerations create coupling forces which overstress the couplings between the motor and trailer coaches. Yet this calculation is not provided in this thesis. Nevertheless, the following information can be derived from interviews conducted with Siemens representatives: a conventional coupling can endure 80kN to 100kN in case of compressive stress and 600kN to 800kN tensile stress; using special adapters the compressive stress can be increased to 200kN to 250kN.

After clarifying the necessary input, the results of the simulations conducted for the thesis on hand are presented. The details can be found below.

For each configuration four different situations were considered.

The first simulation considers the number of stops to be performed form a velocity of 10km/h until the brake reaches an equilibrate temperature. In this perspective it is expected that the braking system of the trailer bogie, which is considered to be electro-pneumatic as mentioned in Chapter 3.2, is working properly. However, the conventional ED- service brake of the motor bogie is out of order and the FPGA controlled ED brake pitches in, using the tread brake to perform the braking until stoppage.

	Fife cars low cost	Six cars low cost	Six cars high cost
Max. Temperature friction area [°C] at the motor axle tread brake	82	84	95

Table 18 Temperatures achieved when performing 200 stops

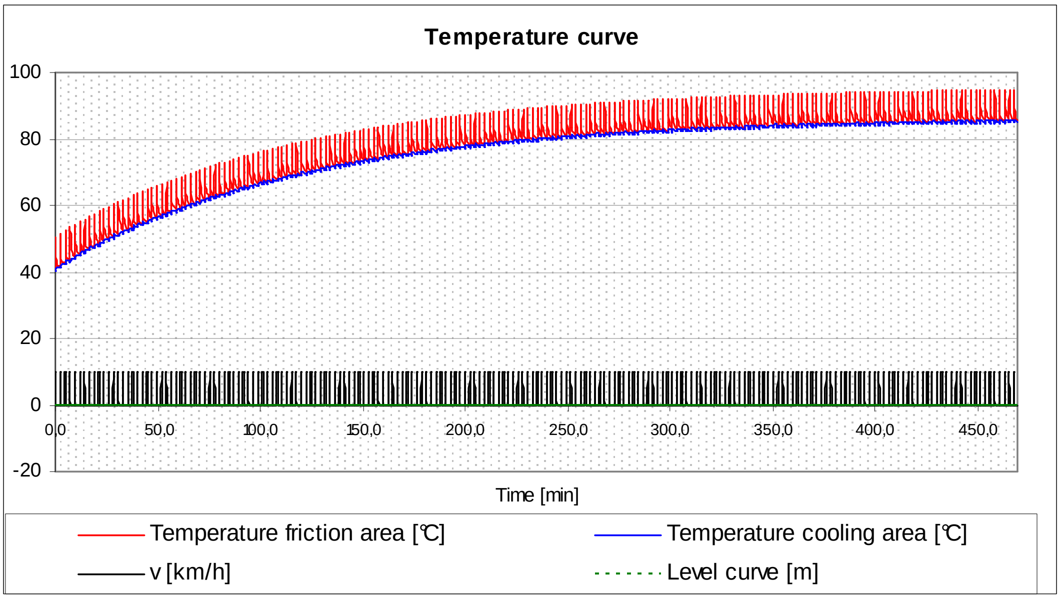


Figure 75 Temperature curve six cars high cost, 200 stops

Equilibrate temperature is reached around 200 stops. The resulting temperature is considered as low and nonhazardous.

The second calculation considers a track profile, in this case Metro Kolkata. Again, the conventional service ED brake is out of order at the motor bogie and the trailer bogie is working properly. Additional braking of the tread brake below 10 km/h is expected.

	Fife cars low cost	Six cars low cost	Six cars high cost
Max. Temperature [°C] at the motor axle tread brake	95	98	117

Table 19 Temperatures achieved performing track profile Metro Kolkata

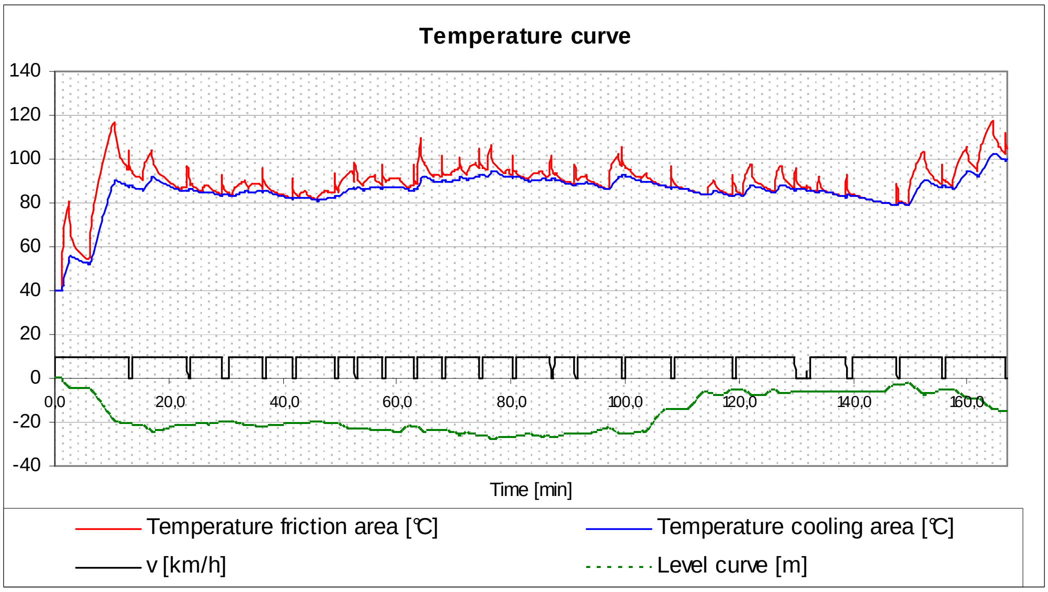


Figure 76 Temperature curve six cars high cost, track profile Metro Kolkata

As can be seen from results the maximum temperature at the friction area is low and nonhazardous.

The third and fourth simulation considers two successive emergency brake applications from 90 km/h. Once it is defined that all motor bogie tread brakes – and if available – trailer brakes are working properly, while the electro dynamic brake at the motor bogie is completely out of service. The second emergency simulation has additionally two motor bogie tread brakes out of service.

	Fife cars low cost	Six cars low cost	Six cars high cost
All working Max. Temperature [°C] at the motor axle tread brake	360	367	408
Two out of service Max. Temperature [°C] at the motor axle tread brake	383	387	440

Table 20 Temperatures achieved performing two successive emergency brake applications without and with two tread brakes out of service

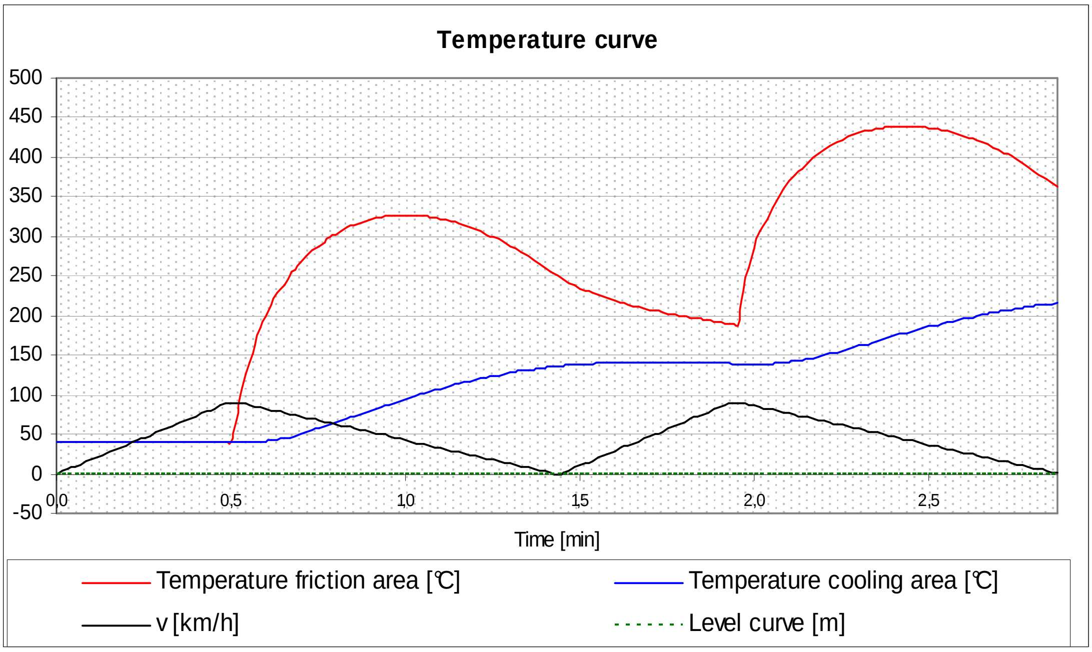


Figure 77 Six cars low cost, out of service scenario

As can be seen from the results, the temperature of the six cars high cost is quite high. However it is possible to use the organic tread brake lining due to experience up to 450°C.

Knowing these results, it is visible, that the tread brake can endure the demands due to different braking applications and failure scenarios.

4.4 Simulation of kinematics due to braking force application

As presented in section 4.1 Necessary parking force the current force that has to be provided by the actuator is supposed to be 34.28kN per axle. An axle of a bogie is equipped with two wheels. Theoretically there can either be one actuator providing the total force, affecting one wheel, or two actuators providing half the force each, affecting both wheels. Additionally, there can be either used one-sided or double-sided systems.

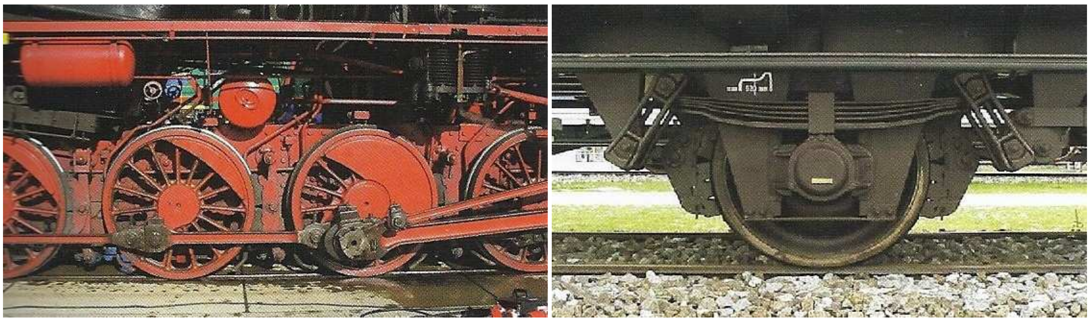


Figure 78 One-sided braking (left), double-sided braking (right) (Knorr-Bremse, 2002)

It can be assumed that a one-sided working tread brake unit is the lightest possibility to design the tread brake. However, this setup is the most critical from the force application perspective. The following analysis is used to proof the assumption '35kN applied to one wheel have no negative effect, in terms of unwanted deformations, on the bogie'. Used for simulation is the multi-body simulation program Simpack.

It is worth to figure out how much the motor axles turn in case force is applied. Basically there are two possibilities to apply the force.

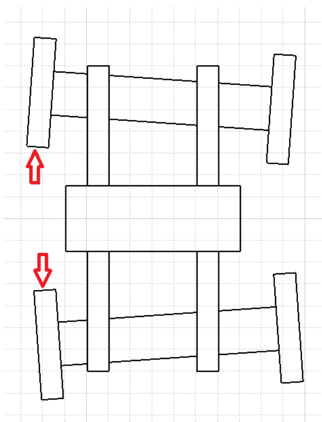


Figure 79 Opposed force application

The possibility in Figure 79 provides the first example. As visible the result of force application is tuning of the axles in a way that the bogie performs a curve, in case it rolls. Certainly this is not wanted therefore another possibility is used, presented in Figure 80.

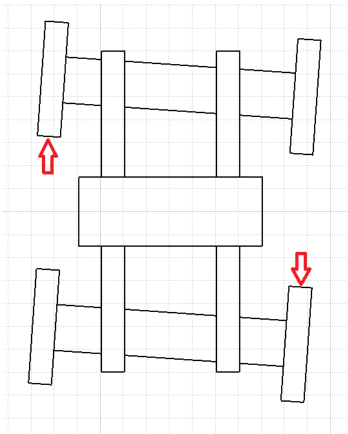


Figure 80 Diagonal force application

This setup is considered as less critical. However, the following simulation is supposed to prove it.
A model of the Siemens bogie Syntegra is used and applied with 34.28kN on one wheel of each axle. The brakes are positioned in a diagonal setup.

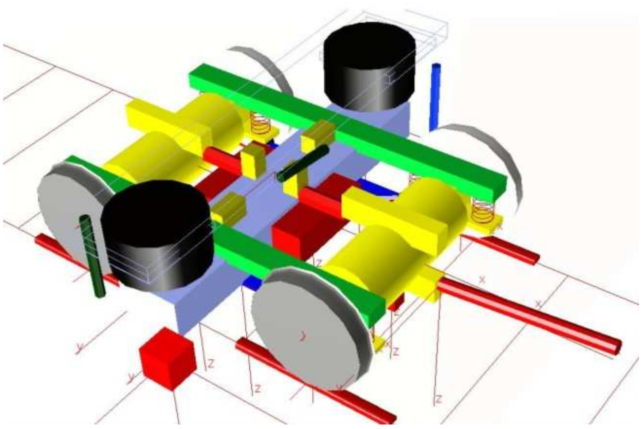


Figure 81 Simpack model of Syntegra bogie

The result of the simulations is summarized below.

The jaw angle was considered as the most important. Its effect is represented in Figure 80.

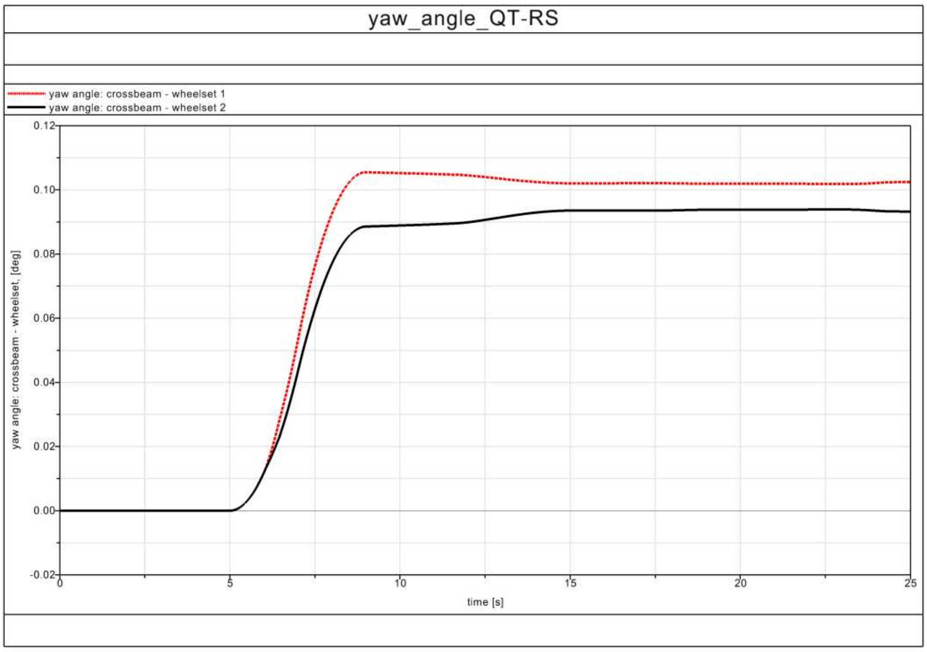


Figure 82 Yaw angle QT-RS

As visible in Figure 82 Yaw angle QT-RS, the yaw angle is not critical. The force is applied five seconds after start of the simulation. The yaw angle increases and reaches its maximum at the frontal wheel set at about 0.105deg. The rear wheel set always has a yaw angle lower than the frontal. However, considering a longer time interval (175s) the two yaw angles converge. The difference between frontal and rear behavior of the bogie axles lies in their asymmetry, e.g. the weight is slightly different.

Figure 83 Displacement Pitch Stop

A displacement pitch stop of 0.35mm is also considered as not relevant due to experience.

Also the displacement of the primary springs is very low. Related diagrams can be found in the appendix.

Overall the analysis proof that there is no negative influence if the Syntegra bogie axle is applied with a one-sided force of 34.28kN. Due to this result it is decided, that the present actuator will be designed as a unit applying force only to one wheel of the axle.

4.5 Components of the actuator

4.5.1 Effect of the power amplification types

Using the equations for the different power amplification types known from Chapter 3.2.3 it is possible to define the necessary input forces F_{in}. As shown in 4.4, it is possible to use one brake actuator per axle. Therefore the necessary output force F_{out} of the power amplifier is supposed to be about 35kN. Additionally the necessary spring deflection is also subject to consideration. According to experience the brake shoe clearance is supposed to be approximately 5mm. This will be considered in the calculations as well.

Single-tier power amplification

As visible by the name, the following considerations are made for a single amplification tier. Each amplification mechanism recommended by Chapter 3.2.3 is introduced.

Fluid transmission

For this calculation the following setup is defined:

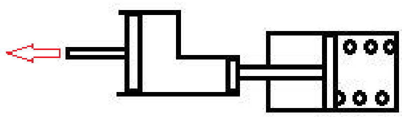


Figure 84 Concept draft of fluid transmission power amplification

Logically, it is reasonable to define the diameter d_{out} at maximum as large as the diameter of the spring loaded parking brake. It is expected that the current diameter from the Faiveley BFC, 178mm is a good choice to start a calculation. However, wall thickness and the seal have to be considered as well. It is estimated here, that a diameter of 150mm is acceptable.

The best combination of low force and not too elevated spring deflection is considered to in the diameter relation factor between 3 and 4.

dout	Aout	Ain	factor	din	Fin necessary to provide Fout=35kN	bar	distance
0,15	0,02	0,005	3,50	0,08	10,00	19,81	17,5

Table 21 Necessary input force into the power amplifier 'fluid transmission' and required moving distance

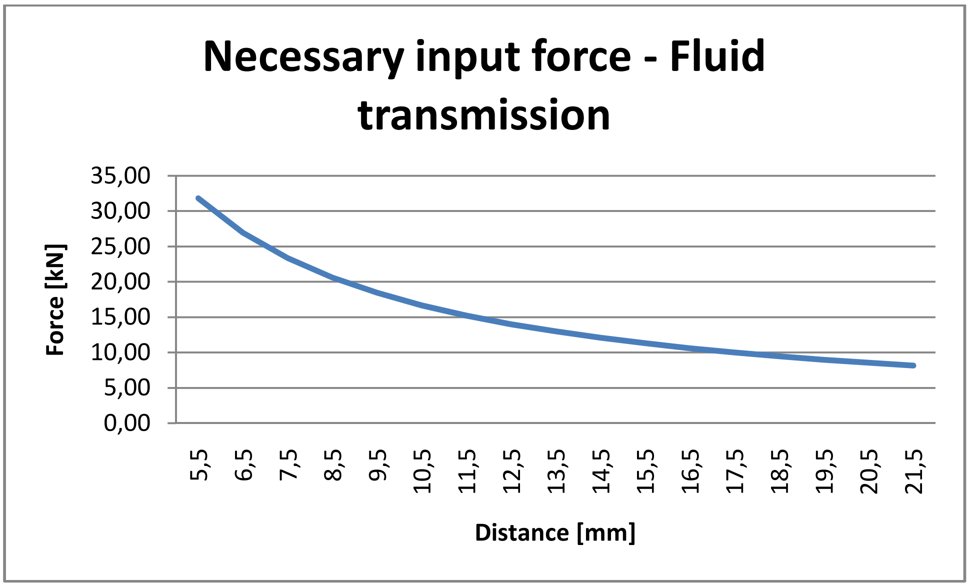


Figure 85 Necessary input force – Fluid transmission

Lever

In case the lever is used, certainly a different setup is needed in comparison to the fluid transmission:

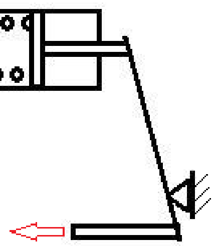


Figure 86 Concept draft of lever power amplification

In this case the radius from the Faiveley Transport BFC plus some additional distance due to housing and slack adjuster. It is estimated that the total length of the lever is 150mm.

ltotal	factor	lin	lout	Fin necessary to provide Fout=35kN	distance
150	0,8	120	30	8,75	20

Table 22 Necessary input force into the power amplifier 'lever' and required moving distance

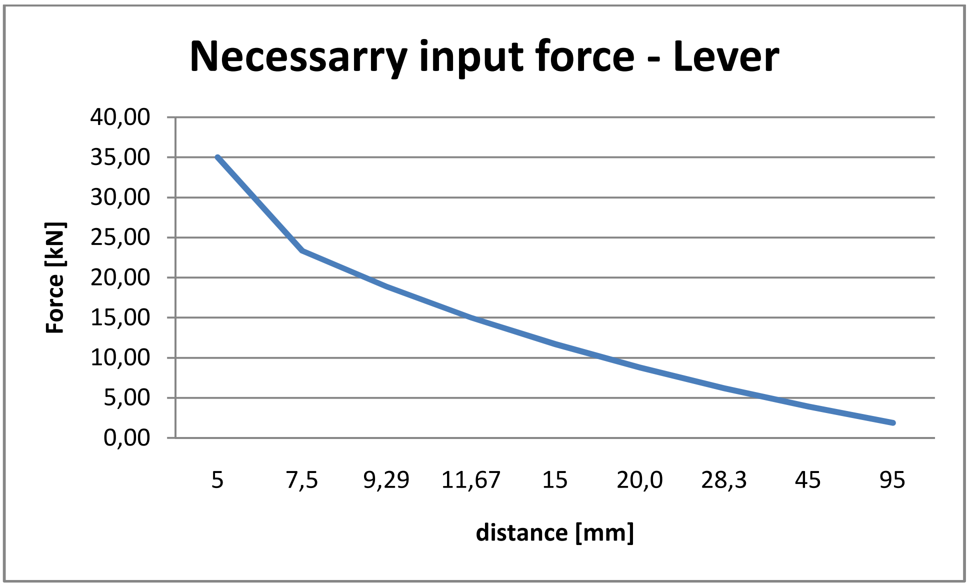


Figure 87 Necessary input force - Lever

Wedge

The wedge provides a major advantage compared to the principles named before. Its amplification depends on the angle and the friction in the connection area of the rods. It is possible to adjust the angle of the friction surface without any influence on the overall size of the power amplifier, certainly self-locking needs to be considered. The friction between two metal surfaces in this case is considered to be 0.1.

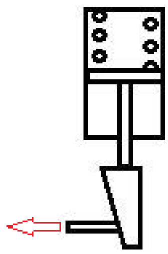


Figure 88 Concept draft of wedge power amplification

Considered as most appropriate for the current thesis the following setting:

alpha	friction	Fin necessary to provide Fout=35kN	distance
75	0,1	13,23	18,66

Table 23 Necessary input force into the power amplifier 'wedge' and required moving distance

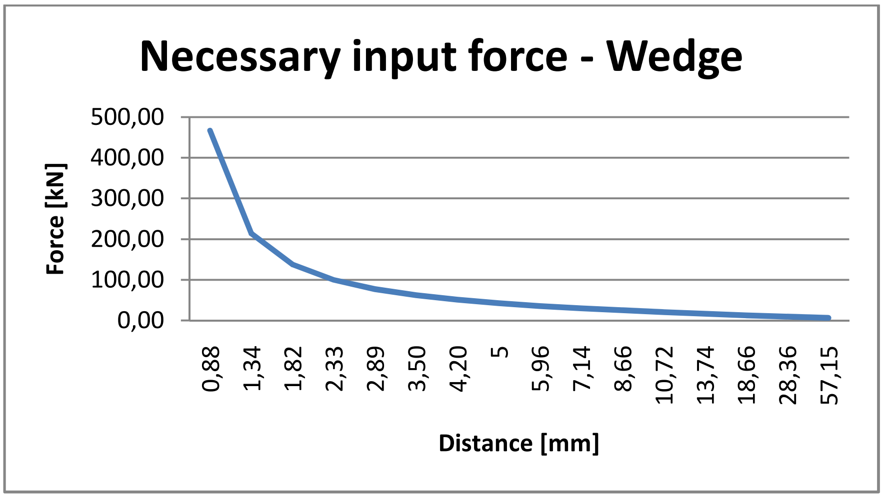


Figure 89 Necessary input force - Wedge

Toggle joint

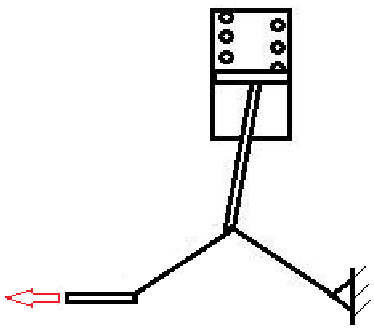


Figure 90 Concept draft of toggle joint power amplification

For the toggle joint it is reasonable to use the following setting:

alpha	length [mm]	Fin [kN] necessary to provide Fout=35kN	distance [mm]
15	50	18,76	12,07

Table 24 Necessary input force into the power amplifier ‘toggle joint’ and required moving distance

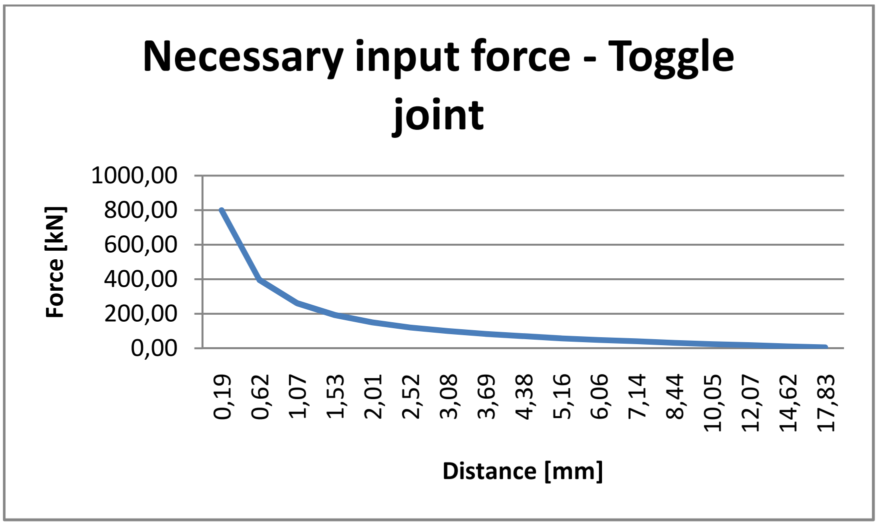


Figure 91 Necessary input force – Toggle joint

The toggle joint, though providing amplification with low required moving distance, still has the disadvantage to reduce the input force only to 18.76kN. In comparison to the other amplification possibilities, this is quite high. Additionally a rather large installation space is needed, compared to lever and wedge. Therefore the toggle joint is dismissed as single-tier power amplification.

Multi-tier power amplification

The effectiveness of power amplifications can be increased using series connections. The multiplication factors greatly increase in this case. Experience shows that combinations with levers are very good to handle. It is also reasonable to consider the output displacement of the first and the initial displacement of the following multiplication tier match when the amplification unit dimensions are equal. (Roth, 1994, p. 94)

It is most reasonable to consider combinations of hydraulic cylinder and the other three possibilities. This is caused by the fact that the hydraulic cylinder can provide the output force in any direction, while with lever, wedge and toggle joint a certain input direction causes a certain output direction of the force.

Considered are two representative combinations of the hydraulic amplification. The one provides low pressure but causes a large moving distance, the other has a lower moving distance and therefore a higher force.

Case 1

The fluid transmission requires:
Input force 10kN
Input distance 17.5mm

Case 2

The fluid transmission requires:
Input force 16.67kN
Input distance 10.5mm

Second amplifier	output force hydr.	input force hydr.	input distance	input force spring	moving distance spring	single input force	single moving distance
lever	35	10	17.5	6.67	52.5	8.75	20
lever	35	16.67	10.5	11.11	31.50	8.75	20
wedge	35	10	17.5	8.60	24.99	13.23	18.66
wedge	35	16.67	10.5	9.90	22.52	13.23	18.66
toggle joint	35	10	17.5	9.33	20.42	12.34	14.62
toggle joint	35	16.67	10.5	5.88	22.93	12.34	14.62

Table 25 Necessary input force when different amplification types are connected in series with the fluid transmission

As pointed out by the table above, there is no significant improvement in case of combination of two power amplifiers. In this case it is questionable whether the additional weight and installation space needed for a second amplification step reduces the spring size to an extent which justifies the second step.
However, in detail some information can be derived from the table above. First, compared to the single amplification step, it is possible to decrease the input force, except for the lever in Case 2. However, considering the moving distance of the spring, all distances increase, some more than double.
From this perspective it is reasonable to not use one of these combinations.

4.5.2 Restrictions on the source spring type

In order to figure out if the spiral spring is applicable, a basic assessment is conducted.

For the calculation the X10 CrNi 18-8 is used. It possesses of an tensile strength of 600-2200N/mm^2 and is available with a thickness t of 3 mm. (IKS - IndustrieKooperationen Kai Spachmann, 2012)

The wheel diameter is supposed to more less the same size as the diameter of the Faiveley Transport pneumatic BFC spring loaded parking brake, 178 mm. Therefore a r_{wheel} of 100mm is expected.

The resulting width b is supposed to be about 363 mm.

This width is very large and therefore the spiral spring cannot be used in the present thesis.

Considering this result, the disc and coil spring are the springs that are applicable for the thesis on hand.

Application	Mechanism	Component
Type of energy storage for braking	Spring deformation	Disc spring Coil spring

Table 26 Usable components for the energy storage with respect to presented constraints

4.5.3 Restrictions on the working media supposed to charge the energy source

Several components were found using the mechanisms possible to implement. Those listed below were dismissed due to different reasons.

Electro magnet

In order to figure out if the electro magnet is an appropriate component a simplified calculation to find the necessary overall size is conducted. Certainly several simplifications were taken. However, all of them are considered to be very positive for the design, therefore it can be considered that a detailed calculation will come up with an even larger size.

The very high value of 1T is used as magnetic flux density. Furthermore it is considered that 1T is constant for any position during the stroke. It is further known that flux density creates a force of 40N/cm^2 between the pole faces.

Additionally leakage field and magnetic resistance of the magnetic circuit are not considered.

The most appropriate design for the magnet is the pot design. The following figure provides the reason. The streamlines of the field have to pass only one big air gap.

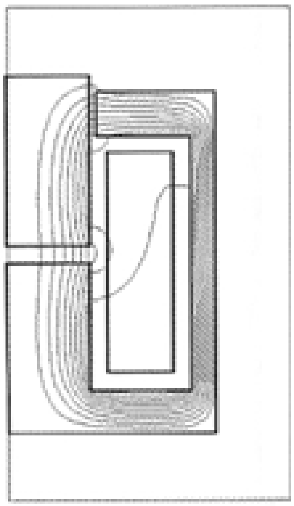


Figure 92 Streamlines of the field (Kallenbach, Eick, Quent, Ströhla, Feindt, & Kallenbach, 2003)

In order to clarify the nomenclature of the dimensions of an electro magnet with pot design the following figure is provided:

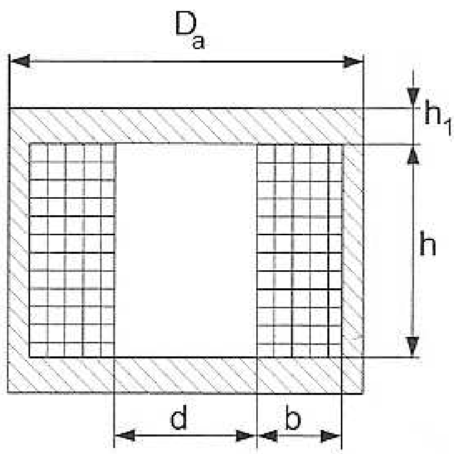


Figure 93 Nomenclature of the dimensions of a pot magnet (Kallenbach, Eick, Quent, Ströhla, Feindt, & Kallenbach, 2003)

It is taken that the coil design is subject to the following conditions:

$$x = \frac{b}{d} \tag{4.12}$$

$$y = \frac{h}{d} \tag{4.13}$$

While x_{opt} = 0.35 …. 0.48 and y_{opt} = 1.5.

Neccessary force [N]	air gap [mm]	active diameter [mm]	overall diameter without housing [mm]	hight of coil [mm]	wigth of coil [mm]	power [kW]
12000	17.5	195	383	293	94	56
10750	20	185	363	277	89	77
15230	18.66	220	432	330	106	56
20760	12.07	257	504	386	123	20
35000	5	334	654	501	160	2.67

Table 27 Dimensions of electro magnets with respect to the necessary force to be vanquished

The simplified calculation uses the power amplification types: fluid transmission, lever, wedge, toggle joint and none in the mentioned order.

As visible from the results of the simplified calculation, the size of the electro magnet is certainly not usable for the current application; especially when comparing the size to the size of the current spring loaded parking brake of Faiveley Transport.

Dimensions	width	200 x 200 mm
	Height	164,190,224,244 mm

Table 28 Dimensions of the Faiveley Transport BFC spring loaded parking brake

Additionally a very high power can be considered to be needed. This is caused by the air gap, which is used squared in the power calculation.

The implementation of an electro magnet used to charge the power source is not feasible.

Linear electric motor

The linear electric motor would provide the necessary translational movement for the application without any additional converter. What seems like the perfect advantage is actually a disadvantage. The travel speed and the feeding force cannot be adjusted using a gear. Still the force needed to charge the energy source could be provided. Forces up to 20kN are possible. Especially the combination of several primary parts with one secondary can increase the force to a large extent. Pitifully, the secondary part is quite heavy due to the large number of excitation windings. According to (A-Drive Technology GmbH, 2011) solely the weight of the secondary part is between 32kg and 45kg for forces between 2.498kN and 9.99kN. This is considered to be too high for a light weight design.

Pneumatic / hydraulic Motor

As described in Chapter 3.2.3, fluid motors need to be powered by a fluid under pressure. Considering the availability, both compressed air and hydraulic oil could be used. Compressed air can be provided by the already installed air compressor and an additional electro-hydraulic unit could provide hydraulic pressure. However, if using a rotational motor, there would be some sort of converter to transform rotational into translational energy necessary. This causes additional weight. Therefore a rotational motor is dismissed.

Concerning the translational it is understandable that no transformation of rotational to translational energy is necessary. However, the effect of a translational motor is comparable to a simple cylinder. Certainly the cylinder is given preference in a decision between motor and cylinder.

As visible the application of fluid motors is not a good choice despite the high provided force density.

Shape memory alloy

The most popular alloy is NiTinol. Basically shape memory alloys are highly applicable, compact, posses of low weight and have a high force-to-weight ratio. Nevertheless, to use shape memory materials heating and cooling needs to take place. This process requires time. Therefore the limiting factor for dynamic applications are heating and cooling rates. Currently 3Hz can be achieved.

As expected temperature, stress, strain and the number of cycles have limiting influence on the lifetime of on shape memory material component. (Vessonen, 2003, pp. 12-13)

Basically, with the current development, the application of shape memory alloys is not considered as recommendable.

Piezo-actuator

Piezo-actuators are available in several different designs. It appears that the stack design is most appropriate for the application in railway vehicles, considering the operating force.

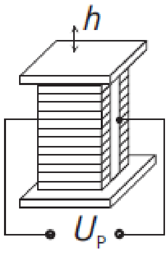
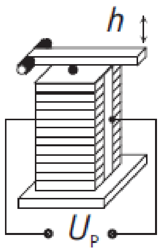
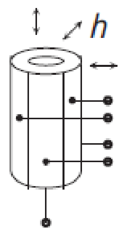
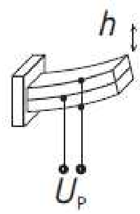
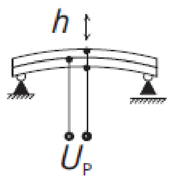
Design	Stack design	Stack design with lever	Tubular design	Bending design	Bending plate design
					
Operating displacement	< 200µm	< 1mm	< 50µm	< 5mm	< 500µm
Operating force	< 30kN	< 3,5kN	< 1kN	< 5N	< 40N
Operating voltage	< 1kV	< 1kV	< 1kV	< 400V	< 500V

Figure 94 Piezo-actuator designs based on (Mehlfeldt, 2006)

The values derived from the table above, point out that the operating displacement in relation to the operating force appears as not sufficient. Therefore the usage of piezo-actuators is not possible for a railway brake application.

Application	**Mechanism**	**Component**
Working media to charge energy source	Magnetic induction effect	Rot. Electric motor
	Compression	Pneumatic/hydraulic cylinder

Table 29 Components usable to charge the energy source with respect to the presented constraints

4.6 Combinations

Having defined the different possibilities for energy storage, power amplification and working media it is necessary to consider the real applications of the mechanism and their possible combination.

Application	Mechanism	Component
Working media to charge energy storage	Magnetic induction effect Compression	Rot. Electric motor Pneumatic/hydraulic cylinder
Type of energy storage for braking	Spring deformation	Spring accumulator
Power amplification converting rotational into translational movement	Resolution of a force Spreading of pressure	Lever Wedge Fluid transmission Screw type gears (threaded spindle; recirculating ball screw) Wheel – rod gear

Table 30 Compilation of the different components related to their applications

A combination of the defined components shows the different possibilities to design an actuator of a brake.

<table>
<tr><th>Working media to charge energy storage</th><th>Type of energy storage for braking</th><th>Power amplification</th></tr>
<tr><td rowspan="4">Rot. Electric motor + rotation to translation converter</td><td rowspan="2">Coil spring</td><td>Wedge</td></tr>
<tr><td rowspan="2">Lever</td></tr>
<tr><td rowspan="2">Disc spring</td></tr>
<tr><td>Fluid transmission</td></tr>
<tr><td rowspan="4">Pneumatic cylinder</td><td rowspan="2">Coil spring</td><td>Wedge</td></tr>
<tr><td rowspan="2">Lever</td></tr>
<tr><td rowspan="2">Disc spring</td></tr>
<tr><td>Fluid transmission</td></tr>
<tr><td rowspan="4">Hydraulic cylinder</td><td rowspan="2">Coil spring</td><td>Wedge</td></tr>
<tr><td rowspan="2">Lever</td></tr>
<tr><td rowspan="2">Disc spring</td></tr>
<tr><td>Fluid transmission</td></tr>
</table>

Table 31 Combination possibilities presented in detail

Not surprisingly literature research found that most of these principles are already used in the current available actuators. Except for the power amplification using a fluid transmission, all components were found to be used.

4.7 Design approaches

Considering the actuator there are two different design approaches defined in this thesis. On the one hand the linear improvement and on the other hand the new actuation principle. Both will be considered in the following sections.

4.7.1 Basics

In order to decide on a certain actuator it is necessary to have in mind the following criteria.

Criteria	Name
Must-have	Necessary braking force achievable Force creation without external energy supply Emergency release Slack adjuster Fit into installation space Weight Reaction time sufficient Controllability
Nice to have	Maintainability easy to accomplish
Additional	Costs as low as possible with respect to quality

Table 32 Different criteria to be considered when choosing an actuator

No matter which actuator is used it has to possess of the must-have criteria mentioned above. Certainly it is necessary to provide a sufficient braking force. The standard EN 13452-1:2003 further asks for a force creation which is independent from external energy for parking brake applications.

The existence of an emergency release prevents wheel flats in case the vehicle needs to be towed.

A slack adjuster is necessary to provide the right brake shoe clearance and therewith the right braking force. It is especially helpful to use a self acting slack adjuster, because this decreases maintenance costs.
A very important point concerns the installation space. The actuator needs to fit into the so-called black box, which covers the installation space. In case the actuator is too large; it is not possible to use it. Additionally the weight has to be considered. In order to find an improvement for the current actuator, the weight of the alternative should be lower.
It is also necessary that the actuator does not react too slowly. It is always the part of the braking system that acts the slowest, has major influence on the reaction time. In case the reaction time increases, the braking distance increases as well. Additionally it is important that the chosen braking system is controllable by the SIB.
Easy maintainability is considered as useful and therefore belongs to nice to have. Finally the costs are important. Though they are very important it is difficult to obtain reliable information on them. However, in case a certain actuator should be considered from a cost perspective, it is advised to use the VDI 2225. This guideline offers a simplified calculation on the costs of a product (Verein Deutscher Ingenieure, 1997).

4.7.2 Evaluation of improvement potential at linear enhancement

For the linear improvement three considerations are made. Firstly it is evaluated whether a change from pneumatic to a hydraulic system is reasonable. Secondly, the spring is evaluated and an alternative to the coil spring is considered. Finally the alternative for an overall design approach is provided.

Actuator medium

Considering the actuation media, it is possible along with the linear enhancement to change from a pneumatic to a hydraulic system. Certainly using a hydraulic system a higher pressure can be used to create the force which compresses the spring.

In the present case, the Hanning & Kahl hydraulic power unit HZY-K100 is considered to provide the hydraulic pressure. A pressure of 100bar can be provided by this unit (Hanning & Kahl, 2011).
Concerning the pneumatic pressure, usually there are two pressures available: 5bar for the service brake and 6bar for the parking brake cf. (Faiveley Transport, 2007, pp. 4-5).

The following diameters are necessary for the different forces using compressed air or oil.

Amplification type	Spring loead F1 [N]	diameter with oil [mm]	diameter with air [mm] / 5 bar	diameter with air [mm] / 6 bar
Lever	8750	40	166	152
Wedge	13230	45	200	180
Fluid transmissio	10000	40	175	160

Table 33 Comparison of the necessary piston diameter with respect to necessary amplification typ and actuation

It is evident that there is a big difference in the required diameters for hydraulic and pneumatic systems. From the first point of view it is reasonable to change from a hydraulic to a pneumatic system. The current diameter for the Faiveley- Transport BFC is 178mm. Yet the active piston area is only 0.021 m^2. The calculation results on hand do not consider a decrease of the active piston area due to a piston rod, because the new design is not chosen thus far. Overall a decrease in the diameter can be expected in case the media is changed.

However, the decision on the media can additionally be based on a different perspective. The spring parking brake considered in this thesis, is supposed to be used underground. Down there oil is of an important disadvantage. On the one hand it is flammable and on the other hand it is harmful for the environment in case of leaking. Actually, the environmental contamination can already be countered using synthetic oils. Products like the Panolin HLP Synth are widely consumed by microorganisms in soil or water (Panolin, 2002, p. 8). Yet the problem of flammability is still not solved.

Especially since the incident in Kaprun where the Gletscherbahn Kaprun 2 caught fire and 155 people died on the 11th of November 2000 hydraulic is a critical topic if used in tunnels. Back then a faulty fan heater caught fire. Additionally the surrounding was due to some reason in contact with the hydraulic oil used in the train. Pitifully the oil caught fire as well and finally after the train had entered a tunnel the hydraulic system broke down, while the oil fuelled the fire. The stack-effect and massive smoke emission made the situation even worse for the passengers. Altogether a dramatic accident took place. (Stieldorf & Stieldorf, 2012)

Additionally due to information from Siemens, no subway train using a hydraulic brake system was sold within the last ten years. Due to this reason the idea of changing the working media to hydraulic is rejected.

Spring

The next thing to consider is the spring. A calculation of coil and disk springs is conducted in order to find possible improvement potential. As known from the considerations concerning the actuation media, it is reasonable not to try to reduce the mean diameter of a spring but to reduce the block length in order to obtain compact design.

Coil spring

The calculation as described in Chapter 3.2.3 was conducted for a dynamic calculation of a single spring and a calculation of a spring set including two springs.

Single spring

Amplification type	Spring load F1 [N]	block length [mm]	length under loaded condition [mm]	mean diameter [mm]	lenght without load [mm]	wire diameter [mm]	approximated mass [kg]
Lever	8750	114	129	140	236	19	0.98
Wedge	13230	147	171	150	313	22	1.4
Fluid transmission	10000	128	145	140	250	20	1.1

Table 34 Coil spring dimensions with respect to the different power amplifications

Set of springs

Amplification type	max. spring load [N]	block length [mm]		length under loaded condition		mean diameter [mm]		lenght without load [mm]		wire diameter [mm]		approximated mass of set [kg]
Spring #		1	2	1	2	1	2	1	2	1	2	
Lever	10750	80	80	89.7	90.5	140	105	197	198	16	12	0.99
Wedge	15230	106.4	106.4	119.9	120.9	160	102	262	263	19	12	1.5
Fluid transmission	12000	90.1	90.1	101.2	102.2	140	98,8	206	207	17	12	1.1

Table 35 Coil spring set dimensions with respect to the different power amplifications

Considering this result there might be improvement potential in the block length compared to the benchmark. However, due to the fact that the currently used spring brake provides 35kN parking brake force using a wedge as power amplifier, it can be

expected that the detailed calculation of Faiveley-Transport offers an optimal spring design for a coil spring used in a parking brake actuator.
However, from the current result is seems to be most appropriate to use the lever power amplification.

Disc spring

As alternative to the coil spring the disc spring is considered.
Using the equations known form Chapter 3.2.3 it is possible to derive a block length and additionally the weight of a disc spring pack. The calculation is conducted for the most promising 'connection in series' of disc springs and the power amplification lever, wedge and fluid transmission. As basis for the calculation the product portfolio of (Adolf Schnorr GmbH + Co. KG, 2004) is used. The most promising results are presented in the table below.

Amplification type	Spring load F1 [N]	block length [mm]	diameter [mm]	weight [kg]	disc springtype number
Lever	8750	154,4	50	1,98	013800
Wedge	13230	63,6	125	4,4	020200
Wedge	13230	204,6	63	3,7	016200
Fluid transmission	10000	87,4	80	2,9	017500

Table 36 Disc spring set dimensions with respect to the different power amplifications

Again considering the results the combinations of disc spring with lever seems most appropriate.

Choose type of spring

In order to decide on the coil or the disc spring, two results are of interest. On the one hand the length under loaded conditions, which refers to the length of the spring in case the brake is released, and the weight of the spring or spring set.

First the weight is considered. The weight of the single spring is the lowest for any amplification type. From a lightweight design perspective this opportunity is the most appropriate. Due to the big difference concerning the weight, the disc springs can be considered as less suitable for the purpose in hand.

To choose between the coil spring set and the single spring, it is reasonable to consider aside the mass the length under loaded condition. This length defines the size of the housing which causes weight as well. The length of the springs under loaded condition is always smaller with the spring sets. Due to this fact and the usage of a coil spring set in the Faiveley Transport BFC, which can be considered as elaborate design, it is reasonable to choose a coil spring set for the present actuator.

Compact design

A good possibility to reduce weight and size of the actuator is redesign. As known from the description of the current benchmark, the Faiveley-Transport BFC combines active service and passive parking brake. In the current situation the passive parking brake is sufficient. The reduction of size and weight can be realized if the active parts of the service brake are removed.

For the current benchmark it is hardly possible due to the stack design, having parking brake (1) on top of the service brake (2). Additionally both use the same wedge to achieve the movement of the rod connected to the brake tread.

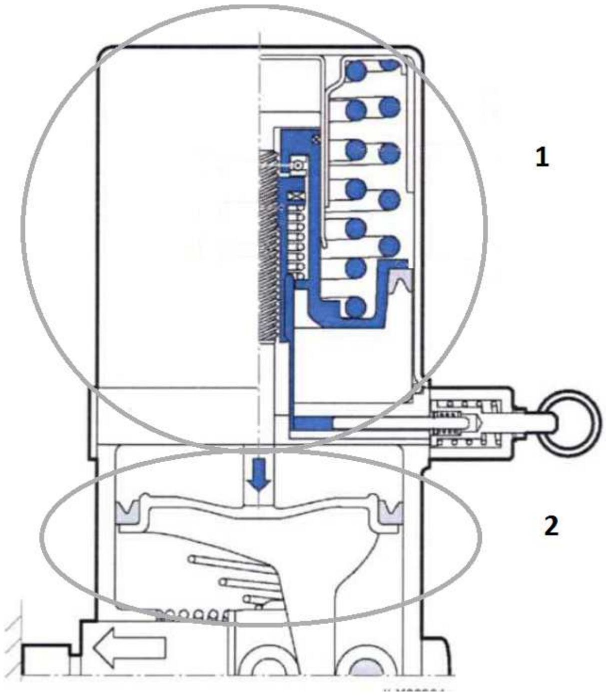


Figure 95 Faiveley Transport BFC stack design (Faiveley Transport, 2007)

A conducted patent research offers an alternative actuator. The alternative provided by a patent from Knorr-Bremse and invented by (Staltmeir & Wosegien, 1985).

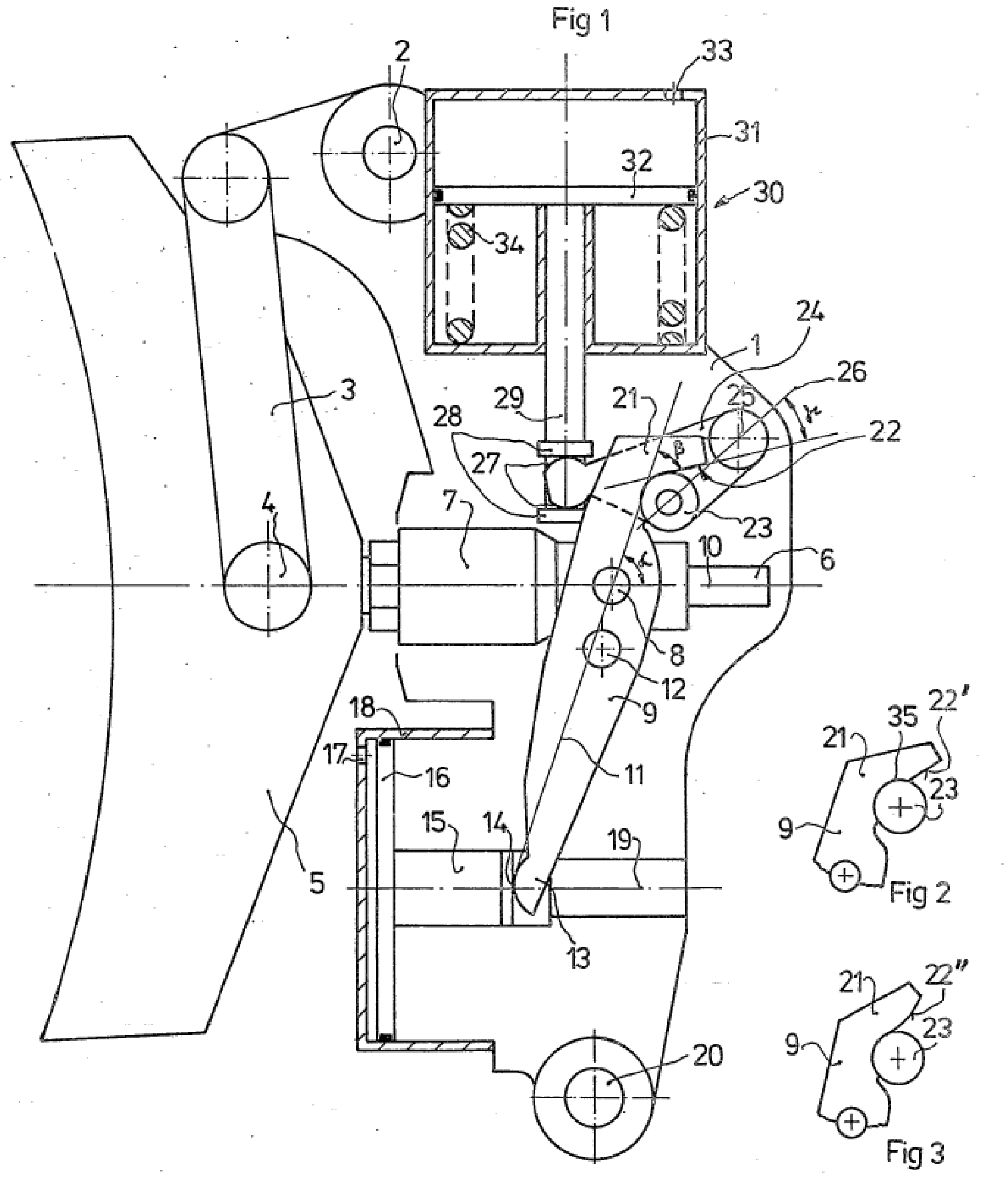


Figure 96 Tread brake actuator by (Staltmeir & Wosegien, 1985)

As visible the spring loaded parking brake is on the upper end of the actuator and the service brake is at the lower end. The basic idea is to reduce the lower end.

The following paragraphs should provide basic understanding of the design of the tread brake actuator in Figure 96 and are based on (Staltmeir & Wosegien, 1985, pp. 5-12).

The actuator consists of a service brake. For the application the piston (16) is forced to the right of the page by e.g. compressed air. In doing so the lever (9) rotates around supporting point (12). The lever (9) is also connected to the brake shoe (5) by the connection rod (6) including a slack adjuster (7). This connection causes the brake shoe (5) to move to the left of the page and apply force to the wheel.

Additionally the actuator consists of a parking brake. For the application the pressure in the cylinder chamber is reduced. The parking brake piston (32) moves to the top of the page due to the force created by the coil spring (34). This causes the piston rod (29) to move in the same direction. The piston rod (29) is connected to the parking brake lever (24) which in turn is connected to the lever (9) via a wedge transmission (22), (23). The rotation of parking brake lever (24) around bearing point (25) causes the movement in the wedge transmission (22), (23). This transmission provides several advantages. On the one hand the due to the usage of the suitable wedge shape a constant, progressive or decreasing force ratio can be maintained or, on the other hand a fast feeding option can be implemented. Fig. 2 provides the wedge shape for fast feeding and Fig. 3 a continuous change of the transmission ratio. In the provided convex shape the transmission ratio changes with the stroke.
However, the lever (9) moves just, as with the service brake the connection rod (6) to apply the braking force.

The following image shoes the new design concept using the approach of (Staltmeir & Wosegien, 1985). The simplified actuator possesses only of a parking brake.

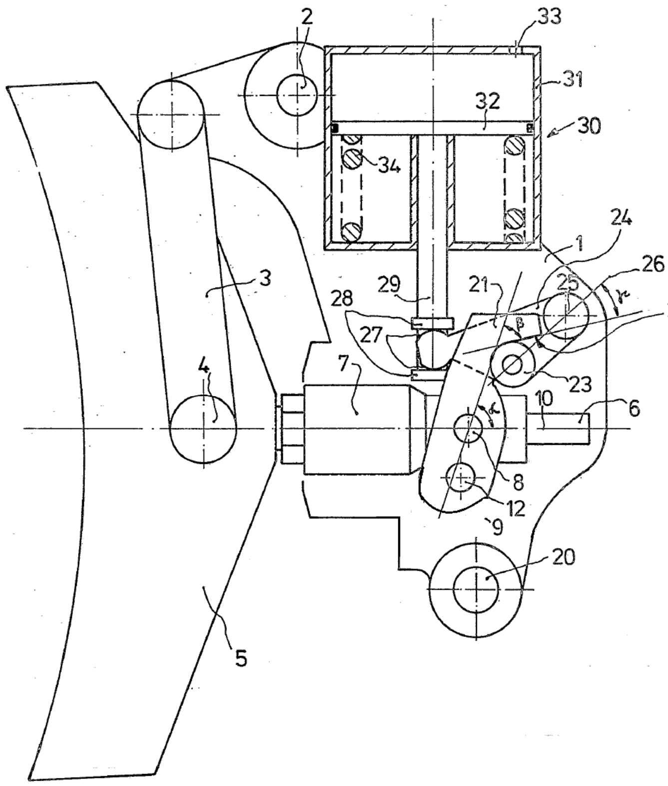


Figure 97 Pneumatic brake actuator without an active braking force creation based on (Staltmeir & Wosegien, 1985)

Consistent with the result of the spring calculation a combination of two levers is used as power amplifier.

Considering the 'must haves' mentioned in section 4.7.1, it is possible to realize all the 'must haves' due to the fact that the whole detailed design would have to be made by Siemens. The necessary force can be provided by sufficient design of the energy source, the spring. This spring additionally fulfills the criteria of applying force without external energy. The installation space can be considered with low effort, because it is an in-house development. Moreover the space for the slack adjuster is already available in the present draft and also the emergency release can be implemented. Reaction time and controllability are provided to the same extent as with the current actuator, due to the usage of the same actuation media.

In the current thesis it is quite hard to provide reliable information on the maintainability effort. However it can be considered, that it will not be very different to the one of the current actuator.

Concerning the costs it is not possible to provide reliable information with the current draft.

4.7.3 Compare new actuation possibilities to benchmark

Not all actuators presented in Chapter 3.2.5 new actuation possibilities are salable products at the moment. Offered for sale are the Faiveley Transport and the Raco Schwelm electro- mechanic brake actuator. These two are now compared to the benchmark the pneumatic Faiveley Transport BFC. Concerning the other possibilities no products are available at the moment, though a lot of research is conducted on that issue.

	Faiveley Transport BFC	Faiveley Transport EMB	Raco Schwelm GBMIII
Parking brake force	35 kN	32 kN	36 kN
Service brake available	yes	yes	yes
Stroke	7 mm	10 mm	11 mm
Max. slack adjustment	125 mm	80 mm	-
Emergency release installed	yes	yes	no
Weight	63 kg	37 kg	40 kg
Length (without brake shoe)	200 mm	282 mm	135 mm
Largest width	463.5 mm	305 mm	428 mm

The table above shows the result of the most important values for the current issue.

Considering the must haves mentioned in section 4.7.1, it cannot be taken for granted that all the must haves are fulfilled by the actuators, due to the fact that these are products provided by other companies for a general application. However they were chosen because they fulfill the criteria of applying force without external energy. Further the controllability is provided due to the electrical triggering. Additionally is taken, that a sufficient reaction time is provided because both are applied in vehicles that have to fulfill the EN 13452-1 already.

The maintainability is difficult to assess. Further to fathom the costs a vital detail is missing. Without a precise number of actuators, no company will offer a realistic retail price.

Considering the force, the Faiveley Transport EMB needs an additional lever in order to provide 35kN. However, installing this lever is not supposed to be a big problem. Furthermore the stroke – necessary are at least 5 mm, is provided by all the actuators. Pitifully the Raco Schwelm actuator does not possess of a slack adjuster and an emergency release. This can be considered as major disadvantage, because it has to be installed additionally. Concerning the weight the Faiveley Transport BFC is way heavier than the two other options. Concerning the installation space it is necessary to consider the space provided between the physical parts of the bogie; the wheel, the longitudinal beam and the cross beam. Yet this is not sufficient for railway applications, the structure gauge has to be considered. This structure gauge is defined by the customer or by a legal authority. However, the installation space provided is considered as black box. The length refers to the distance between cross beam and wheel, the width to the installation space between longitudinal beam and the end of the cross beam, and the height is measured from the structure gauge to the upper edge of the longitudinal beam.

The top view is limited by wheel, longitudinal beam, cross beam and structure gauge.

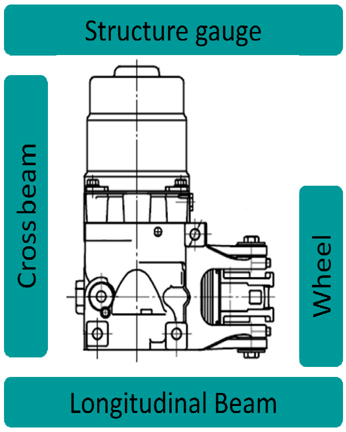


Figure 98 Schematic configuration of the Faiveley Transport BFC tread brake installation

As already mentioned, the Faiveley Transport BFC is currently installed in the in Syntegra bogie. Therefore it is taken, that an actuator of nearly the same size can be installed as well. The length of this actuator including the brake shoe and lining is usually 339mm, according the assembly drawing.

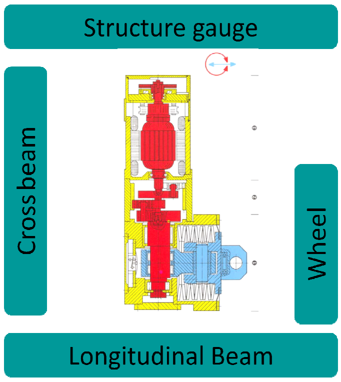


Figure 99 Schematic configuration of the Raco Schelwm GBM III installation

Concerning the top view of the Raco Schwelm GBMIII, it can be considered as possible to install this actuator. However, the disadvantage is based on the weight. Even without slack adjustment and emergency release, the actuator is heavier than the Faiveley Transport EMB. Due to this reason, this actuator is dismissed.

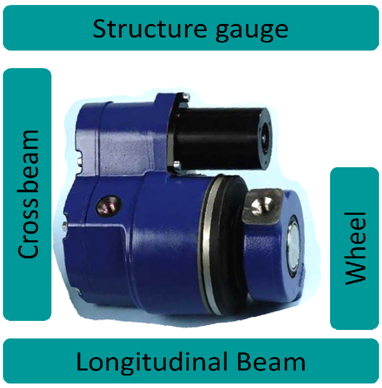


Figure 100 Schematic configuration of the Faiveley Transport EMB installation

Considering the Faiveley Transport EMB, it is also possible to install this actuator. It is lighter and quite small. Also the height of the actuator is no problem, the distance between the structure gauge and upper edge of the longitudinal beam is sufficient, even if the primary suspension deflection is considered. However, a problem can occur due to the length of the actuator. Compared to the Faiveley BFC, which is about 200mm long without the brake shoe and lining, the EMB is 82mm longer. It depends on the size of the brake shoe whether this actuator can be installed. If the brake shoe is to be used, the EMB is 29mm too long. Aside this problem, the EMB is still a promising solution, due to the very low weight, the slack adjuster and the emergency release.

5 Conclusion

The task of searching for improvement potential and alternatives for the currently used Faiveley Transport BFC electro-pneumatic brake actuator provides several results.

Firstly, the calculation of braking force found that the current actuator provides exact the amount of force needed for the parking brake application. The dimensioning of the springs in the section 4.7.2 proves that there is hardly any improvement potential.
Additionally it is recognized that the braking distance demanded by the EN 13452-1:2003 for a full fledged braking system to perform emergency brake applications is not achieved by the tread brake. It is certainly not possible to consider the SIB as having two fully-fledged separate braking systems. Still the current standard of the EN 13452-1:2003 demands this. However, this standard represents regulations for the current available solutions. As the SIB is considered to be a new development, it is necessary to clarify the compulsory conditions. The introduction of the SIB to approval authorities will bring new input on the whole surrounding conditions. Nevertheless, it has to be expected that extensive negotiations need to be done; with authorities and customers. Both need to be assured that the new system does not bear potential additional harm for passengers, the infrastructure and the vehicle itself.
Concerning the thermal stability, a positive statement can be made. A single tread brake per axle can endure the stresses applied according to the MMC performance specification. Certainly higher temperatures occur, but there are brake shoe linings available which can deal with the situation. As expected the tread brakes of the 100% motorized bogies have to endure the highest temperatures during two successive emergency brake applications. Yet even these temperatures are 10°C lower than the maximum temperature during steady-state performance for the chosen brake shoe lining.
In relation to the thermal calculation it was necessary to prove, that the application of about 35kN to a single wheel per axle does not cause any critical deformations on the bogie. Neither the displacement pitch nor the yaw angle reaches critical values. Therefore the lightest design can be used without a doubt.

This thesis also covers a structured approach to search for alternatives and improvement potential. Therefore it was evaluated which types of power amplification, energy storage and energy source are offered by physical means.
It was found that for the power amplification lever and wedge are most suitable. Concerning the energy storage the coil spring set possess of the biggest advantages and the charging of the energy source can be done using a pneumatic or hydraulic cylinder or a rotational electric motor.
It was further found that the presently used principles in actuators cover all those possibilities offered by the design catalogues. Certainly some new ideas like shape

memory alloys might offer alternatives in the future. However, currently there was no extraordinary new approach found.

Concerning the presently used actuator in detail it was evaluated whether the change from the electro-pneumatic system to the electro-hydraulic system would bring about a significant benefit. Certainly the necessary piston diameter decreased a lot. Yet it was found that the application of hydraulic braking systems in metros is most critical. Especially with respect to past incidents, additionally the flammability of the fluid is a major issue.

Further the spring was considered in detail. It was found that the present spring set can be considered as most appropriate for the application in an electro-pneumatic spring loaded brake. Yet the overall design of the actuator can be changed. The active brake part can be reduced. A patent from Knorr Bremse is presented which appears as very promising. It can be considered that a new in-house development based on this patent will fulfill all the requirements defined.

In case a complete switch from the electro-pneumatic solution is aimed, the electro-mechanical is an opportunity. Currently there is lots of research done in this field and many things are kept secret. The available electro-mechanic alternatives are provided by Faiveley Transport and Raco Schwelm. Both actuators offer advantages and disadvantages. The Faiveley Transport EMB seems most promising, because it consists of a slack adjuster and an emergency release. Nevertheless, a brake shoe needs to be designed which can be installed in the limited space between wheel and actuator. Moreover an additional lever transmission is necessary to achieve the obtained force; 3kN are missing. Yet probably Faiveley Transport could redesign the actuator to obtain the additional spring force. Certainly this is a question of quantity and investment.

Furthermore, it is necessary not to forget about the recent developments concerning the self amplifying brakes. As soon as those are ready for serial production even smaller and lighter actuators are available. Yet the introduction of a product was postponed many times in the past and the time necessary for the final development is uncertain. Additionally the control of the self amplification can cause quite big problems. The suspension of the wheelset will allow a relative movement between the brake, fixed to the cross beam and the running surface of the wheel. This movement cannot be considered as small. This causes problems for the self amplifying brake concerning the providing of a constant brake force. Lots of variables need to be considered in case a sufficient control should be provided. Yet it is not impossible to achieve a control like this.

Basically it gets obvious that a strategic decision is necessary to define the next steps.

5.1 Decision support & Future Outlook

It is obvious that, several different strategies can be applied.
Firstly, the company can stick to the electro pneumatic actuator – either to the already implemented or to a new development based on the design opportunity provided in section 4.7.2. This makes sense because it is considered that compressed air will be present in the vehicles for some more years. The reason is the air suspension. The current state of technology does not provide another type of suspension which can, at comparable low height, provide as large movement especially in X and Y direction. Therefore pneumatic is indispensable in the bogie these days. Additionally a reduction of weight due to the reduction of compression unit and piping will not occur within short time. Nevertheless it makes sense to keep an eye on the development of electro-mechanical, especially self amplifying brakes for the long run.
Yet in case a new electro-pneumatic actuator should be provided, it is recommended by the author, to consider the arising costs due to the development using the VDI2225 for cost estimation.

Secondly the ideas to save weight and to promote the all-electric vehicle are considered as tempting, and there is no interest in investing money in a new development of an old braking system. In this case the solution is to switch from an electro-pneumatic system to an electro-mechanic. However, the available products do not fit perfectly. It is necessary to add some now developments no matter if the Faiveley Transport or the Raco Schwelm actuator is used. Further it is questionable whether the implementation of a product available these days is the best solution. There is massive research done on self amplifying brakes. However, there is no product offered yet and certainly it cannot be taken for granted that a self amplifying solution can be introduced to the metro vehicle.

Basically there is little detailed information concerning figures for the benefit of electro-mechanic brakes. However, the information provided by Faiveley- Transport covers the information provide below.
The added values of the electro-mechanic brakes are considered to be: little noise, environmental friendliness and the intelligent control.
Concerning the system cost, it is said that less or equal system costs compared to electro-pneumatic or electro-hydraulic are achievable. The same is valid for the LifeCycleCosts. Concerning the size Faiveley- Transport especially focuses on light rail vehicles and metros hence the weight is considered to be comparable to electro-hydraulic systems. (Faiveley- Transport, 2008)
Other sources provide different information. The costs of an electro-hydraulic system are supposed to be lower than those of an electro-mechanic brake (Gralla, 1999, p. 108).

From this perspective it is certainly a benefit to change from an electro-pneumatic to an electro-mechanic actuator. However, there is also a bad image related to the electro-mechanic actuators.

It is not the first attempt to use an electro-mechanic actuator in a metro vehicle. In the 1990ies Siemens offered a prototype, the so-called CitySprinter. The vehicle consisted of an electro- dynamic brake and an electro- mechanic friction brake. The concept seemed promising and the vehicle, equipped with the Faiveley Transport EMB, was handed over for real life trial runs to the transportation company of Köln on the 12.07.1999. Pitifully an accident took place on the 23.08.1999. The vehicle crashed non-braked against another train, which was standing at a station. Overall eight people were badly injured and several other slightly. However, the reason was not a brake defect of the EMB, but the electronic brake control system was defect. This would not have been a problem, but in addition a handling error by the operator occurred.
Nevertheless this not exactly spectacular accident caught the railway industry in a bad moment. After years of focusing on comfort in rail vehicles the braking system centered. (Rosenberg, 1999)
As a consequence to the accident, Siemens stopped the CitySprinter and no new attempt to use a similar concept was taken again.

As additional information, attempts to use dry (electro-mechanic actuated) brake-by-wire systems are not unique to railway vehicles. They are also very popular in automotive industry. However, their implementation is slow. There are concerns about their reliability and safety. Still there is a constant evolution toward fully electric braking systems. (Mandragon, Mandragon, Miller, & Mondragon, 2007)
The conventional automotive industry EMB systems are used in airplanes, concept cars and electric vehicles. Interestingly one of the oldest patents related to EMB is Nr. 121043 ‘Elektrische Bremse für Motorwagen’ submitted to the Deutsches Kaiserliches Patentamt in the year 1899. (Breuer & Bill, 2004, p. 253)
As a trend it becomes obvious that several sectors work on all-electric solutions. However, the reasons for development, are different between railway and automotive applications. While the electro-mechanic brake in a car has to provide a bunch of braking interventions, e.g. ABS, ASR or ESP (Breuer & Bill, 2004, p. 280), it is sufficient to provide wheel slide protection for railway vehicles. Therefore the EMBs in cars need to be optimized for control frequency which is unnecessary for railway vehicles. Due to this fact a one-to-one transfer of the solutions will not be possible. However this annexation should point out the overall goal to apply EMBs.

Additionally the result of a survey made by RTWH Aachen states the following for the potential of the actuation principles: Most of the currently used actuation possibilities can be considered as matured. Not only electro-pneumatic or hydraulic systems, also the electro- mechanic system, because the major parts, the spring and the electro motor are fully developed. It is not considered that major additional benefits will occur due to new improvements. It is recommended to focus on the self amplifying wedge brake to accomplish a large improvement. (Hermanns & Müther, 2005)

Summarized, a strategic decision is necessary to decide which actuation possibility is most appropriate for the current issue. It is not only about making a short term decision on make or buy. It influences the degree of innovation applied in the Syntegra bogie. Additionally, due to the fact that the self amplifying brake actuators are subject to research in several companies it can be expected that new possibilities can pop up within the next years.

Many research results are kept secret and only patents or conference reports are available. It is nearly impossible to figure out how long it will take exactly until the first self amplifying brake reaches series-production readiness. Not speaking of the actual implementation possibility of such a brake in a metro vehicle.

Additionally, some manufacturers of braking systems try to keep their cash cows, the electro-pneumatic systems, because along with them they can sell compression units. For example, Knorr Bremse owns patents on an electro-mechanical actuated brakes. These are known since several years; however, they do not have a product yet.

It is vital to keep a close eye on the development in this sector in the next years. Revolutionary products might be introduced soon.

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7 List of figures

8 List of Tables

9 List of abbreviations

ED	Electro dynamic
EL 6	Six standing passengers per square meter
EL-E	Train ready for operation
EMB	Electro mechanic brake
Rm	Tensile strength
RWB	Rotating eddy current brake
SIB	Safe Innovative Brake
UIC	Union Internationale des Chemis de Fer