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A Digital Test Bench for Pneumatic Brakes

Simulating the Braking Behaviour of Freight Trains

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Abstract

This master's thesis covers the structuring and implementation of a digital test bench for the air brake system of freight trains. The test bench will serve to further improve the existing brake models at Transrail Sweden AB. These are used for the optimised calculation of train speed profiles by the Driver Advisory System CATO. This work is based on the research of the technical background, as well as the methodical approach to physical modelling and a modular implementation of the test bench. It gives full flexibility for the simulation of customised train configurations using the European UIC brake system. Train length and vehicle arrangement can be adapted to the user's specific needs. For example, the test bench could be used for the simulation of a train with distributed power. The system parameters are stored in a vehicle library for the convenient generation of train configurations. This vehicle library is freely expandable.

The simulation is based on an equivalent electric circuit model which is completed with nozzle flow modelling. This model involves monitoring the main pipe, brake cylinder and reservoir pressure. Linear approximation is used to obtain braking forces for the individual wagons and for the whole train. The depiction of the brake system behaviour is mostly accurate in the operational scenarios, which is validated with measurement data. Additional calibration is required for further reduction of the simulation errors and an extension of the model's domain of validity. The test bench is developed by incremental and iterative modelling and prepared for further improvements and variations, for example the adaption to the American AAR system variant.

The presented work can also be used as a basis for similar implementations such as driving simulators. The methods are transferable to other applications of modular simulation.

Keywords

Compressed air brake, Freight train, Modular simulation, Pneumatics, Railways, System modelling

Sammanfattning

Det här examensarbetet omfattar formgivningen och implementeringen av en digital provbänk för tyckluftsbromssystemet på godståg. Provbänken ska användas för att vidareutveckla befintliga bromsmodeller hos Transrail Sweden AB. De används för beräkningen av optimerade hastighetsprofiler för tåg i förarassistanssystemet CATO. Arbetet baserar sig på undersökningen av den tekniska bakgrunden, samt ett metodiskt angreppssätt för fysikalisk modellering. Verktyget är implementerat på ett modulärt sätt. Provbänken ger full flexibilitet för simuleringen av skräddarsydda tågkonfigurationer som använder det europeiska UIC-bromssystemet. Tåglängd och fordonsanordning kan anpassas enligt användarens behov, till exempel för simulering av fördelad traktion. Systemparametrarna lagras i ett fordonsbiliotek som förenklar inmatningen av tågkonfigurationer. Fordonsbiblioteket kan utvidgas enligt behov.

Simuleringen är baserad på en ekvivalent strömkretsmodell, som kompletteras med modellerad dysströmning. Simuleringen beskriver trycket i huvudledningen, bromscylindern och förrådsluftsbehållaren. Bromskrafterna approximeras linjärt efter trycken för de enskilda vagnarna såväl som hela tåget. Simuleringen återger beteendet av bromssystemet i alla driftsituationer på ett verklighetsnära sätt, enligt validering med mätdata från Knorr-Bremse:s testanläggning. Ytterligare kalibrering behövs för att minimera avvikelserna i simuleringen och för att utvidga modellens giltighetsdomän. Provbänken har utvecklats i stegvis modellering och är väl förberedd för vidareutveckling och anpassning. Ett exempel är anpassningen för att simulera det amerikanska AAR-bromssystemet.

Arbetet som presenteras här är lämplig för användning i liknande applikationer, såsom körsimulatorer. Metoden kan tillämpas allmänt på övriga användningsområden av modulär simulering.

Nyckelord

Tryckluftsbroms, Godståg, Modulär Simulering, Pneumatik, Järnväg, Systemmodellering

Zusammenfassung

Diese Masterarbeit umfasst den Entwurf und die Umsetzung eines digitalen Prüfstands für Druckluftbremsen von Güterzügen. Der Prüfstand soll der Weiterentwicklung bisheriger Bremsmodelle der Firma Transrail Sweden AB dienen. Diese werden zur Berechnung optimierter Geschwindigkeitsprofile für Züge durch das Fahrerassistenzsystem CATO genutzt. Die Arbeit baut auf der Untersuchung des technischen Hintergrunds, als auch auf einem methodischen Ansatz zur physikalischen Modellierung auf. Das Simulationswerkzeug ist modular umgesetzt. Der Prüfstand ist anpassungsfähig für die Simulation individueller Zugkonfigurationen, die das europäische UIC-Bremssystem nutzen. Die Zuglänge und Anordnung der Fahrzeuge können an die spezifischen Bedürfnisse des Nutzers angepasst werden, zum Beispiel ist die Simulation eines Zuges mit verteilter Traktion möglich. Die Systemparameter sind in einer Fahrzeugbibliothek hinterlegt, um die Erstellung der Zugkonfiguration zu vereinfachen. Diese Fahrzeugbibliothek ist beliebig erweiterbar.

Die Simulation basiert auf einem äquivalenten Stromkreismodell, das durch die Modellierung von Düsenströmungen ergänzt wird. Sie beschreibt den Druck in Hauptluftleitung, Bremszylinder und Vorratsluftbehälter. Die Bremskräfte werden daraus linear angenähert und für die einzelnen Wagen sowie den gesamten Zug ausgegeben. Die Simulation gibt das Verhalten des Bremssystems in allen Nutzungssituationen realitätsnah wieder. Dies wurde mit Messdaten aus der Testanlage der Firma Knorr-Bremse überprüft. Weitere Kalibrierung ist für die Verringerung von Abweichungen und zur Erweiterung des Anwendungsfeldes des Modells erforderlich. Da der digitale Prüfstand in einem schrittweisen Modellierungsprozess entwickelt wurde, ist er besonders zur Weiterentwicklung und Abwandlung geeignet. Ein Beispiel dafür wäre die Anpassung der Simulation auf das amerikanische AAR-Bremssystem.

Die hier präsentierte Arbeit eignet sich auch für ähnliche Anwendungen wie zum Bespiel Fahrsimulatoren. Die Methode ist generell auf Anwendungsgebiete modularer Simulation übertragbar.

Schlüsselwörter

Druckluftbremse, Güterzug, Modulare Simulation, Pneumatik, Eisenbahn, Systemmodellierung

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List of Acronyms and Abbreviations

AAR	Association of American Railroads
AD	Accelerator device
ANR	Atmosphere Normale de Reference
BC	Brake cylinder
CFL	Courant–Friedrichs–Lewy condition
DAS	Driver advisory system
DBV	Driver brake valve
DE	Differential equation
DV	Distributor valve
EEC	Equivalent electric circuit
EN	European Standard
EU	European Union
FVM	Finite volume method
KTH	Royal Institute of Technology
MP	Main brake pipe
R	Brake supply reservoir
RK4	Runge-Kutta method of 4th order
SDG	Sustainable Development Goal
SI	International System of Units
UIC	International Union of Railways
UN	United Nations

Chapter 1

Introduction

In times of climate change and environmental challenges, the drive towards sustainable engineering solutions is stronger than ever, particularly in the transport sector. Even though railways already are one of the most energy and resource efficient means of transportation, there are margins for making rail transport even more efficient.

One possible means of increasing efficiency is the implementation of a Driver advisory system (DAS) that calculates optimised speed profiles and recommends an optimal running strategy to the train driver. By following these recommendations, the train's energy consumption can be reduced by 15 to $30\,\%$, compared to a train driven by an inexperienced and unassisted driver [1]. One measure to achieve this is for example the reduction of braking manoeuvres by as much as possible. An example output for the driver is shown in Figure 1.1, where the black line indicates the speed profile. The bars on the left show the recommended braking and acceleration levels.

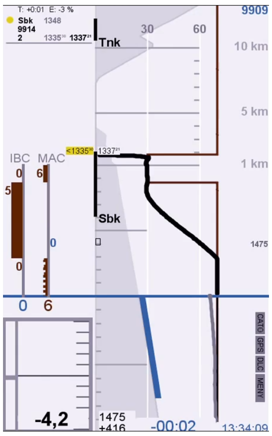


Figure 1.1 – CATO driver interface. Source: Transrail

The optimised speed profiles are generated by simulating the longitudinal dynamic behaviour of the train driving as a whole, including variables such as motor power, rolling resistance and braking force. Most trains have a pneumatic brake system installed, and the phenomena generated by this system are most significant for long freight trains with little or no installation of electronic support systems. Modelling this behaviour in a precise way must thus include fluid dynamics simulations.

1.1 Background

On freight trains, the brake system still follows the original principle that was established with the first implementation of indirect automatic brakes. These brakes have a purely analogue and fail-safe function principle, which is based on compressed air passing through the Main brake pipe (MP) along the entire train. The pressure in this pipe is used both to transfer the control signal for the brake system and to supply the energy required for brake application. The system is called indirect and automatic, because the pressure for applying the brakes is not taken directly from the MP. Instead, each wagon in the train is equipped with an auxiliary reservoir or Brake supply reservoir (R), that is charged with pressure from the MP. This can be seen, in a simplified way, as a loaded air spring that is retained as long as the MP is pressurised. When the MP pressure drops, the compressed air from the R is released into the Brake cylinder (BC), which converts the air pressure into braking force at the wheels. By using this setup, it can be assured that the brakes even are applied in case of a technical failure or train separation leading to a pressure drop in the MP, which means that the brake system has a very high safety level. This transition between the three stages of charging, application and release is controlled by the Distributor valve (DV). The DV is a complex assembly of many chambers and valves, that translates the pressure reduction level in the MP to a BC pressure application level and makes sure that the transition between stages is performed in a reliable and safe manner [2]. An example of these components mounted on a freight wagon is shown in Figure 1.2.

Due to the purely analogue nature of this system, its performance is directly linked to the speed with which the pressure changes can travel along the MP. This means that the brakes are applied with a delay, dependent on their distance from the locomotive or other source of pressure change. Especially for freight trains, the large diversity in length and wagon composition leads to many different pneumatic constellations and therefore different braking performance [3].

On top of the pneumatic system, there is the mechanical system that forwards the braking force from the BC to the wheels, where it is applied via disc brakes or braking blocks acting directly on the wheel rim. This system often consists of a complex arrangement of levers, in order to achieve a braking force that is well adapted to the wagon's characteristics [4].

As a DAS aims to make as accurate predictions of the train behaviour as possible, its braking calculations should be matched well to each individual train's characteristics. For this, very general mathematical models are

Figure 1.2 – Ore wagon with visible BC, lever mechanism, R and coupling hose. *Photo: Fredric Alm / Kiruna Wagon*

available [5], but no fully parameterised model for individual trains has been established. This creates the need of developing own calculation models that ideally are based on or verified by pneumatic simulations.

1.2 Problem

The main objective of this work is to develop a detailed simulation model that describes the dynamic pneumatic and mechanical principles involved in train and vehicle braking, in particular during brake application and release. Transrail's objective is to use the model as a baseline for a real time model implementation in the CATO system and its verification. This is not only a task of implementing an already well-defined system. Instead, the problem includes general conceptual and methodological matters that need to be resolved, leading to the following research questions:

- Which are the crucial components in the brake system and how do they behave?
- Which modelling approach is the most reasonable choice for the given scope?
- How can a modular air brake simulation model be constructed in order to produce pressure profiles that describe brake behaviour for specific train configurations?
- Which lessons can be learned from the simulation results?

1.3 Purpose

The central purpose of this work is to improve the available knowledge on pneumatic brake behaviour in freight trains, as well as to present possible ways to make this knowledge easily available and improve it further in the future. This work therefore is of interest for both for Transrail and the railway industry in general. At Transrail it shall be used to verify and further improve the existing models. Better knowledge of the system behaviour will then allow to study a wide range of braking scenarios in detail and to refine optimisation strategies.

Looking further into the future, the potential winnings of using DASs will increase with more advanced optimisation strategies, ideally leading to a broader use of such systems in operation. A broader use of optimal driving and braking strategies will drastically reduce energy consumption in the railway sector and even decrease the use of resources for maintenance, making railway transportation even more climate friendly than it already is. By increasing the competitiveness of railways, it will open the way for an increasing market share and a sustainable development for large-scale transportation with minimal emissions. This is a necessary step towards having a chance to fulfil climate goals in the coming decades, while maintaining a high availability of transport services.

This work can also give a guideline for other actors in the railway sector and beyond on how to implement a simulator for their own use. For them, possible applications could have another focus than this work and cover areas such as simulators for driver training or simulation of the pneumatic systems on ships.

1.4 Goals

The central goal of this project is to lay out a digital test bench for the air brake system of freight trains, hence the following modelling goals are pursued:

1. Research of the technical domain to ensure that the air brake model is based on the state of the art technology.
2. Accurate physical modelling of the pneumatic phenomena.
3. Establishing a structure for the flexible handling of train configurations, including examples covering the most common types of freight trains in the Swedish railway context.
4. A high level of modularity to allow further improvements and adaptation to future needs.

In a broader perspective, the goal of this project is to provide an overview on air brake modelling that is oriented towards the available information and demands of an operational perspective. The idea is to give engineers a tool at hand to easily obtain brake performance data that are good enough to meet their requirements for practical implementations. This is achieved by fulfilling the following secondary goals:

1. Providing an overview of modelling techniques from research and industry that are currently used in order to suggest one method that can be implemented with a limited volume of resources, i.e. the scope of a master thesis.
2. Providing an overview of the relevant subsystems from a modelling perspective, their interaction as well as implementation options and the choice between physical and empirical modelling.
3. Providing a summary of important principles for the brake control logic.
4. Providing a recommendation on how to achieve high modularity combined with a simple user interface, guaranteeing high accessibility of the test bench.

In terms of physical deliverables, this work shall provide a ready-to-use simulation tool, that produces result files for a train setup that is defined in an input sheet. The format of these inputs and outputs shall be well defined, in order to guarantee usability of the test bench. Furthermore, a set of recommendations and brief descriptions of further improvements of the test bench shall be delivered.

1.5 Research Method

The research for this work is based on a step-wise approach, covering four areas of interest consecutively, with each of them building on the insights of the previous ones.

- The first research area is the domain analysis, including identification of relevant components, their functions as well as typical configurations per region. This is done starting with the review of related course literature, moving on to technical literature in the field of railway engineering. The analysis of the technical background is completed with specifications provided by brake component manufacturers and the applicable standards.
- The second area is simulation theory, covering simulation theory of fluid flow and pneumatic systems, as well as general modelling assumptions used in air brake simulation. This is again based on course material, and then consolidated with specialised literature in the general field of simulation and the application to railway-specific systems.
- The third area covers and overview of available simulation tools or suitable programming environments for the creation of an own tool. Most of this is based on personal experience or formal circumstances, regarding the expected implementation efficiency, knowledge of appropriate features and availability of licenses.
- The fourth and final area is the previous work done in the field of freight train brake simulation. Most of this is published in scientific articles and papers, but there are also full project reports available from projects conducted at KTH.

There is a considerable theoretical background available, and much research and development have been conducted previously. Therefore, concepts and simulation methods can be assumed to be already existing, and the aim is not to deviate radically from that background. Instead, the ambition is to pick the right available option for each part of the simulation tool, which in combination produce the desired results. One important premise in this context is the fact that the most advanced option may not be the best choice for this work. The limited scope and specific use cases for this tool instead make it necessary to choose simplifications wherever they are suitable. Despite the big volume of previous work done, there are sometimes limitations to the description level of the proposed solutions, leaving space for own interpretation and custom solutions. But even for these missing links, there is often a fitting solution described elsewhere.

1.6 Delimitations

The focus of this project is on the modelling and simulation of the pneumatic components of the brake system. This means, that mechanical braking force generation is not within the scope of this project. The study of the system therefore ends at the mechanical behaviour of the brake cylinder, caused by pressure changes within it. No kinematics or longitudinal dynamics of trains shall be simulated or discussed within this project.

Only regular service braking application and release are relevant for the main purpose of this test bench. Others, such as emergency, parking, and shunting brakes, are not controlled by DASs but comply with safety standards. They are controlled either automatically by train protection systems or manually by the driver. Nevertheless, as emergency braking of regular freight trains is also facilitated by the pneumatic system, it can be added to the simulation without much effort. This allows for an additional test case in model validation.

Regarding the different local variants of the freight brake system, this work shall focus on the implementation if the European system and specifically variants that are used on the Swedish market. Nevertheless, it shall also present an overview of the differences to the system used in North America and give suggestions for a future implementation of this variant in the delivered simulation tool. Wherever possible, the model is built in a way that allows the simple adaptation of modules to simulate different systems as well.

These limitations shall ensure that the time frame of this project allows the complete implementation and incrementation of the pneumatic model as well as model validation. The clear interfaces that this delimitation creates are suitable for a direct approximate use of the results, as well as a later addition of extra capabilities to the implemented tool, such as mechanical force generation or implementation with a simulation model for longitudinal dynamics.

1.7 Structure of the Thesis

Chapter 2 presents relevant **background information** about the physical domain of railway air brakes that shall be modelled, as well as the established modelling and simulation techniques. It also gives an overview of other work done in this area and how this project relates to it. Chapter 3 presents the **methods** used to solve the problem and achieve the goals set for this project. This includes the choice of simulation tool and modelling methods, as well as model evaluation. Chapter 4 covers the actual **definition of the model** and its **implementation** in a simulation tool. It also describes the simulation process and adaption. Chapter 5 presents the **results** produced by the simulation tool and their validation against reference data. Chapter 6 contains the **discussion** of the choices made in the development process and the possible interpretation of the results. Chapter 7 finally presents the **conclusions** that can be drawn from this project and its results. It also points out limitations and gives suggestions for further improvement and topics to proceed with in future work on the matter.

Chapter 2

Background

This chapter provides background information about the technical domain of the freight train air brakes that shall be modelled. More specifically, it covers the pneumatic parts of it. Additionally, this chapter describes established modelling and simulation techniques. The chapter also describes related work, mainly the development of the simulation tool *TrainDy* and the European research project *DYNAFREIGHT*. Furthermore, a couple of articles on the implementation of models for railway air brakes have been published. Their assumptions and choice of modelling techniques are evaluated to find a suitable complexity level for the modelling process of this project.

2.1 Air Brake Systems

Pneumatic brakes have been in service since the early days of railways, with the current automatic indirect functional principle being established globally by the end of the 19th century [2]. From there, the system was developed and adapted to local demands, leading to separate standardisation. The two major standardised types in the EU and the western hemisphere are the systems defined by the International Union of Railways (UIC) and the Association of American Railroads (AAR).

For freight trains, the simplest brake systems are still the most common in use, as they are controlled and operated by compressed air only. Therefore, they do not require any electric equipment on the freight wagons. Especially for these purely pressure-dependent systems, the detailed behaviour and the set of additional components strongly depends on what is established and standardised locally. Beyond the general framework of behaviour set by the standard, the details of the complex system and functional features are only known by a small group of specialists. A table with typical values and measures for freight train brakes according to UIC and AAR can be found in Appendix A.

2.1.1 European Standard (UIC)

The main purpose of the catalogue of standards set by the UIC is to ensure freight train interoperability in Europe. This means that all freight wagons that comply with the standard shall be compatible with each other and a safe braking function shall be guaranteed. This is important, because there is a free exchange of rolling stock within Europe, especially in long-range and intermodal freight transportation. The brake systems are adapted to relatively short freight trains, which can be up to 600 to 750 m long, depending on the country. The basic layout of all UIC freight train brake systems is outlined in Figure 2.1.

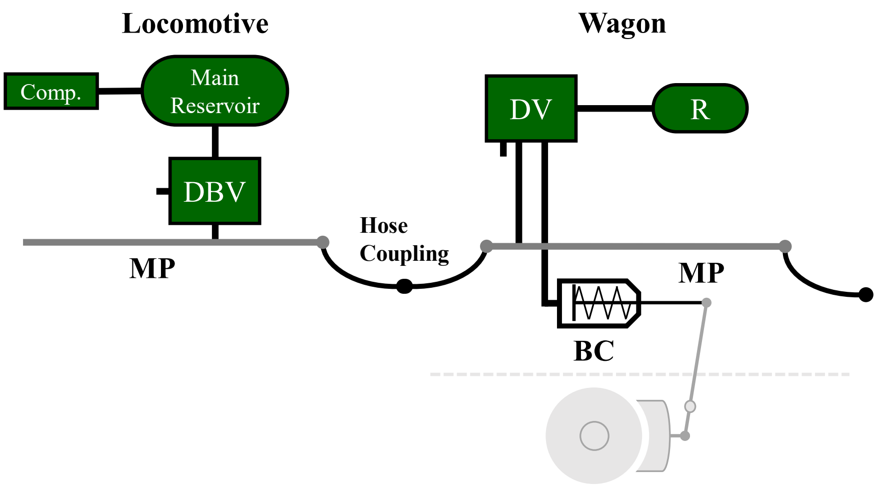


Figure 2.1 – System layout of the UIC air brake system for freight trains.

Even within the set standards, there are margins for detailed behaviour and the use of different control patterns and parameters in the brake components [6]. This is to ensure interoperability while still allowing manufacturers to maintain their individual features and to keep national well-established functional preferences. But as a consequence, even the operators running these mixed trains cannot be sure about the exact characteristics of the rolling stock, merely that they are within the range specified by the standards. In the past years, there has been an initiative that published a number of European

Standards (ENs) in addition to the established UIC standards. They are coherent with the existing standardisation, but in some cases they are more specific in their descriptions [7].

Distributor Valve (DV)

The heart of the brake system is the Distributor valve. It controls the gradual application and release of the BC pressure, which is proportional to the pressure decrease in the MP. A pressure decrease of $1.5\,\mathrm{bar}$ leads to full brake application, which usually corresponds to a BC pressure of $3.8\,\mathrm{bar}$. This is facilitated by the interaction of three pressures on a common piston set, namely the R pressure, the BC pressure and the control pressure set by the MP in an own control chamber. The exact working principle during the procedure of application and release within this complex component is not discussed in this work, but can be found in specialised literature [2], [8].

During the many years of use of the brake system, many improvements and additions have been made to the original concept. The first and most important one is the definition of brake modes or types. These types describe the timing with which the DV fills the BC to full pressure, or releases the pressure from it. As the air is channelled through the DV, an adapted nozzle regulates the flow precisely and therefore controls the timing of pressure changes within the BC. This feature is important because a too fast local change in pressure would lead to high longitudinal forces within the train, which become more severe with increasing train length. Therefore, long freight trains are usually operated in slow-acting mode G, which ensures slow and therefore more even brake application throughout the train. Shorter freight trains and passenger traffic are operated in fast-acting mode P, which allows for higher performance. As it became common practice to install a handle on the DV to manually switch between those modes, they are also commonly referred to as brake positions [9]. Even though all vehicles in a train usually are set in a common brake position, there are cases where they are mixed. Most prominent is the case of short but heavy freight trains, where the locomotive is set in position G and all the wagons are set in position P. This is for example common practice in Sweden for trains over $800\,\mathrm{t}$ [10].

Filling the BC in the exact right time is quite challenging. The filling time with a given nozzle is affected by stroke length and leakage, which increase as the brake components wear and age. A reliable solution for this is the Universal Action feature, that adds a relay valve to the DV. This incorporates a defined precontrol volume, which can be pressurised with the precisely right

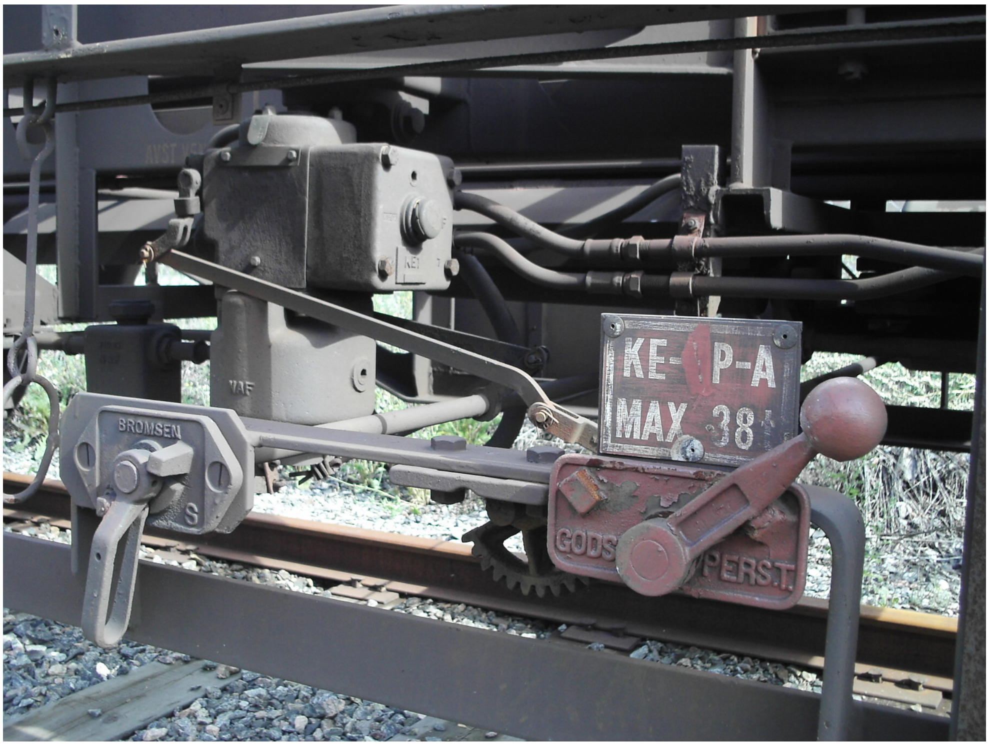


Figure 2.2 – DV, switch handles and piping mounted on a freight wagon. *Photo: Mats Schedin, License: CC BY 3.0*

gradient. The relay valve controls the BC pressure according to this time gradient, without being affected by the BC volume [2].

Another important feature for freight trains is the consideration of changes to the wagon weight. For some types of wagons, the difference between loaded and empty weight can go up to a factor of 6. The braking force must be adapted to that, because a too high braking force would make the wheels lock and slip. A too low braking force in turn leads to safety issues and an excessive braking distance. The technical solution to this is to either change the braking force between two fixed levels, or automatic continuous load-controlled braking. In the former case, setting a handle scales the braking force according to the typical wagon weight for the empty or loaded state, which is facilitated either mechanically in the lever system or pneumatically in a separate relay valve. In the latter case, weighing valves are connected to a load brake valve which scales the maximum pressure in the BC proportionally to the load [4].

There is a number of physical limitations that impair the performance of this purely analogue system. One example is the limited speed of a pressure wave in a pipe. Another factor is that for the sake of reliability, the brake application in the DV is triggered only when a certain pressure gradient is reached in the MP. A third example is a coil spring in the BC that together with friction in the leverage must be overcome before any braking force is applied. All of these effects delay the brake application. To remedy these issues and at least ensure a quick initiation of the braking process throughout long freight trains, two more features were added to the system. The first is the inshot response or initial application feature, that ensures a quick initial rise of the BC pressure, not limited by a nozzle, to bring the brake in contact with the wheel [2]. A second feature that is added to the system is the so called Accelerator device (AD) or quick-service chamber. The AD is a chamber included in the DV which is activated by the initial pressure drop in the MP when a new brake application is made. Opening it extends the equivalent volume of the main pipe, decreasing its pressure by the level required to trigger the brake application process in the DV. By implementing these improvements, the propagation speed of the signal is increased and the response time of the brakes is drastically reduced, leading to decreased internal forces in the train [8].

Along with and after the release of the brake, the DV also controls the refilling of the R from the MP. This means a conflict of interests between filling reservoir for brake availability and filling the MP as quickly as possible, because the air consumed by refilling the R delays the release of the brakes, especially towards the end of long trains. Therefore, a compromise is required, which is handled differently by manufacturers and local operators' specifications. All solutions must comply with the UIC standard, demanding inexhaustibility to an extent that emergency braking at 85% of maximal force must be guaranteed at any time [6]. Practically this implies that the DV may not release more pressure from the BC than what can be replenished from the R. A typical solution is to perform R refilling in two steps: The first step of fast refilling lasts until reaching a pressure level that guarantees the required safety. Then the valve is partially shut for a second step of limited refilling to allow the MP pressure to rise more quickly, while ensuring an eventual full recovery of the R pressure. The detailed sequence and switching threshold depends on the DV model [11].

Brake Cylinder (BC)

The Brake cylinder is the component that converts the air pressure into mechanical braking force to be applied via the friction brakes. Two-axle wagons usually have one BC that is connected to the axles via a lever system, whereas bogie wagons often have one BC per bogie. Brake cylinders on freight wagons have a diameter of 6 to $16\,''$, which produce a piston force of 6 to $50\,\mathrm{kN}$, and have a stroke length of 35 to $220\,\mathrm{mm}$. The BC volume is therefore in the range of 0.6 to $30\,\mathrm{l}$. Most BCs contain a coil spring that has the function to retract the brake shoes from wheels when brakes are released. The BC pressure must rise high enough to overcome this spring force before any braking force is applied, which lead to the initial application feature [8].

Brake Supply Reservoir (R)

The Brake supply reservoir is the volume that stores the energy for applying the brakes in form of pressurised air. This is important for the automatic indirect principle, which allows brake application upon power loss or technical failure. One or two of these are installed in each wagon in form of a steel tank. The volume is adapted to the number and size of BCs on the wagon and is usually in a range of 40 to $300\,\mathrm{l}$. The large volume compared to the BC plays a key role in the inexhaustibility of the system [4].

Main Brake Pipe (MP)

The Main brake pipe runs along the entire train's length and consists of a steel pipe that is mounted onto the frame of each wagon. Between the wagons there are rubber hose connectors with a standardised length of $620\,\mathrm{mm}$. The standard diameter for all freight trains is $1^1/4''$ or $32\,\mathrm{mm}$. The pressure wave of each control change travels ideally with the speed of sound through the pipe, which is $343\,\mathrm{m\,s^{-1}}$ at standard conditions. If the pipe has many tight bends and irregularities, the speed can go down to half of that, but ADs in the DV increase it again. Therefore, a propagation speed of around $250\,\mathrm{m\,s^{-1}}$ can be assumed to be a realistic value [4], [10].

The described issues with R refilling are present on freight trains because they are still mostly equipped with a single-pipe system, where control signal and pressure supply are transmitted to one and the same pipe. Passenger trains, which also use pressurised air for auxiliary functions such as door opening and toilet flushing, usually have a second pipe that is used for pressure supply. Then there is no such conflict [2].

Driver Brake Valve (DBV)

As mentioned before, the pressure changes to control the brakes originate in the locomotive. There they are created in the Driver brake valve (DBV), which translates the inputs from the driver into a control pressure. The application level is dependent on the system either set by the brake handle position or by the time that it is pulled down. The control pressure is converted either mechanically or electrically and then set inside a pilot valve, that controls the MP pressure. Analogously with the Universal Action feature in the DV, this makes sure that standardised pressure gradients are followed even in this part of the system. The standardised relative pressure for a fully charged and released system is $5\,\mathrm{bar}$ in the MP and R. Consequently, as the BCs reach maximum pressure at a MP pressure decrease of $1.5\,\mathrm{bar}$, the control pressure for full service application is $3.5\,\mathrm{bar}$ [12].

In addition to that, an emergency brake application can be initiated by either the driver or the safety system on the train. In that case, the control pressure is set to ambient pressure, and the MP is vented through a larger nozzle. This ensures fast pressure decrease throughout the entire train, but does not affect the maximal pressure in the BCs. It remains constant at $3.8\,\mathrm{bar}$ when the MP pressure has passed full application level. Releasing the brakes from an emergency application takes considerably longer than usual, because the entirely emptied pipe must be refilled first. For a long freight train, this would then be far beyond the $70\,\mathrm{s}$ that are required for a service release according to the UIC standard [10].

Standardised Brake Configurations

The European freight train brake systems are well-adapted to the local conditions with fast, short trains such as the one in Figure 2.3, operating on a system requiring fast control action. The force levels following these requirements are expressed by brake weight percentages in the range of 65 to $125\,\%$. The brake weight percentage is a standardised measure for calculating braking distances, which is based on the ratio between a vehicle's maximal retardation force and its weight [13]. Operating longer trains with these specifications would lead to longitudinal dynamics issues due to the high compressive forces between the wagons, as well as an amplification of the phenomena described above for regular trains. Being a hard limit in European freight vehicles nowadays, this could likely be handled by distributed traction and optimised brake control routines in the future [3].

Figure 2.3 – An example for a Swedish freight train: *Stålpendeln* or steel commuter. *Photo: Kasper Dudzik*

2.1.2 North American Standard (AAR)

This framework of standards, established by the Association of American Railroads (AAR), is followed by most freight haulers in USA and even used in other countries, for example individual lines in Australia. It is adapted to the long freight trains that are typical for North American traffic, with totals lengths in the range of 1 to $3\,\mathrm{km}$. Even though it follows the same functional principle of an indirect automatic brake as the UIC system, it differs significantly in a number of features.

Distributor Valve (DV)

The DV which is even called Triple Valve or Three-way Valve, fills the same role as in UIC system. A radical distinction is that it allows gradual application, but only a full release command. This direct release leaves no possibility for correcting the application level of the brakes, but leads to very quick reaction times and low longitudinal forces. Such a function is necessary to be able to handle trains of such lengths, and implies that reservoir refilling can start directly when the release signal is given. On the other hand, brake availability cannot be guaranteed by the DV logic, but functionality is improved if handled correctly. Fully recharging the brake system of a train with a length of $1500\,\mathrm{m}$ from full service application may still take 15 minutes or more [10].

The pressures are not controlled within defined pressure intervals, but are based on the proportions of pressure and volume between MP, R and BC. During brake application, pressure is released from the R to the BC until the R-pressure has reached the reduced level in the MP. Due to a defined volume proportion of $2/5$ between BC and R, each brake application follows the proportions of $2/5/7$. This means that the full braking pressure is reached when all three pressures are equalised at $5/7$ of reference pressure, when the MP and R pressure have dropped $2/7$ from reference [14]. This relation is the only fixed pressure parameter, meaning that the reference pressure level is flexible and operators can choose to increase the pressure if a higher braking capability is required. Pressures of 65 to $90\,\mathrm{psi}$ (4.5 to $6.2\,\mathrm{bar}$) are common, but can even reach $100\,\mathrm{psi}$ ($6.9\,\mathrm{bar}$) on lines with high gradients.

The fixed volume ratio also means that there can be no feature comparable to the Universal Action in the UIC system. This makes the BC pressure output very sensitive to leakage and changes of the cylinder stroke length. Slack adjusters are mounted to compensate the wear of the brake blocks, but this is still one of the biggest weaknesses of the AAR system [10].

Auxiliary and Emergency Reservoir (R)

The air reservoir in the AAR system is divided into two separate chambers, called auxiliary reservoir and emergency reservoir. The auxiliary reservoir has the same function as the R in the UIC system and was referred as such above. It is relatively smaller than its UIC counterpart though. The emergency reservoir is as the name implies reserved for emergency braking. When the DV detects a drop of MP pressure below full application level, it connects both reservoirs to the BC. The emergency chamber is the bigger part of the R, with a volume ratio of $7/5$, which allows the AAR system to reach a considerably higher BC pressure during emergency application, up to $6/7$ of reference pressure [14].

The pressure equalisation function in combination with the direct release leads to the exhaustibility of the auxiliary reservoir, a negative characteristic to be explicitly avoided in the UIC system. For example, two consecutive decreases of MP pressure would not cause any BC pressure application the second time, and further decreasing the pressure would only transfer the remaining pressure from the reservoir, leading to a drastically reduced braking force [10].

Additional Distinctive Features

Four additional distinctive features are worth mentioning for the AAR system. *Quick Action* adds a valve functionality in each DV that detects emergency brake applications from quick pressure drops and accelerates them by venting pressure from the MP to the atmosphere. The *Quick Service* feature, similar to the AD in the UIC brake, accelerates and increases regular service brake applications by quickly filling a chamber in the DV. Its volume corresponds roughly to a pressure decrease of $6\,\mathrm{psi}$ $(0.4\,\mathrm{bar})$ in the MP of each wagon. As this feature reacts on successive pressure reductions, no brake level adaptation smaller than that can be made. The *Quick Release* feature supports the refilling of the MP. When brakes are released, the MP is fed with pressure from the emergency reservoir, therefore accelerating the release signal speed through the train. Finally, *pressure retainers* are traditional manually controlled components that are still in used on some lines. They can be set to the BC on each wagon in order to keep some minimal pressure or to slow down the pressure reduction upon brake release. A typical use case would be as a safety and support feature on long downhill sections [10].

As a measure to decrease internal forces, the braking force levels also are considerably lower for AAR brakes. The brake weight percentages range between $11\,\%$ for loaded and $40\,\%$ for empty trains, and even do not exceed $80\,\%$ at emergency brake applications. This leads to considerably longer braking distances and a limitation of operating speeds. All in all, the AAR brake system contains more risks at the hands of an inexperienced driver, but this is necessary to allow the operation of trains with lengths up to $3\,\mathrm{km}$.

2.2 Numerical Simulation of Pneumatic Systems

This section gives an overview on the modelling and numerical simulation of pneumatic system. The basic approach for simulating continuous physical systems such as gas-filled pipes is the method of splitting them into discrete elements. Then the state variables, which for a pneumatic system are quantities such as pressure, flow and temperature, can be aggregated into a single value for each element. This is done by means of interpolation or averaging, dependent of the method. The interaction of these state variables and their change over time contain the information about the system that the simulation is attempting to give by means of modelling.

2.2.1 Basic Stability Requirements

Whenever this type of discretisation is performed, it is important to keep in mind how the purely numerical phenomena caused by this affect the physical phenomena that shall be simulated. One example for this coupling is the concept of numerical viscosity, that is caused during the discretisation of a model, and hinders it from depicting changes as quickly as it should according to the physical reality. For the explicit methods used in this project, this is covered by the Courant–Friedrichs–Lewy condition (CFL), that describes a stability requirement for unstable numerical methods that model convection or wave phenomena. For an element length of Δx, a time step length of Δt, and the velocity magnitude w, the Courant number C is calculated.

$$C = w\frac{\Delta t}{\Delta x} \tag{2.1}$$

Not meeting the condition $C \leq 1$ implies that a flow signal should travel through two or more elements per time step according to the physical example. The numerical model can however only transfer signals through one element per time step. Therefore, the flow would be slowed down artificially in the model. This negative numerical viscosity also leads to other stability issues in the model and must therefore be strictly avoided [15].

2.2.2 Equivalent Circuit Models

The most radical approach of aggregating state variables is to make the elements as long as possible by dividing them only where there is an interaction with an external effect. For a gas-filled pipe, this would mean that it is divided into line elements that are assigned some properties and one state variable each. Wherever there is some flow exchange with the outside or the pressure should be monitored, the elements are split and a connecting node is placed.

This model has a close analogy with electric circuits. By assigning line properties to each element that are equivalent to the basic electrical quantities of resistance, conductance and capacitance, such an equivalent electric circuit can be solved in the same way as electric circuits. The basic rules such as Kirchhoff's rule and Ohm's law make it then rather straightforward to solve the system. The relevant part of the modelling is that the equivalent quantities are correctly modelled, as the rest of the calculations follows conventions [16].

2.2.3 Finite Volume Method (FVM)

This method applies the approach of aggregating state variables by dividing pipe sections into discrete volumes. All system changes can then be described by the flow through adjacent surfaces of these finite volumes [15].

For the train brake system, only flow characteristics on a coarse scale are of interest, allowing to discretise the system in 1D. This means that each volume spans the entire diameter of the pipe and that each volume only has two adjacent volumes, which drastically simplifies system modelling setup and calculations. This also means that an average flow velocity over the pipe diameter must be approximated, as the velocity is zero at the walls and then increases towards the middle of the pipe [17].

Flow can be described by a set of differential equations, derived for this special case from the Navier-Stokes equations of fluid dynamics. For the one-dimensional case assuming a constant pipe diameter, the first two equations, covering conservation of mass (continuity) and conservation of momentum, can be expressed as the DEs below. The third DE, covering conservation of energy, can be neglected by assuming constant temperature in the system. The equation is then replaced by the ideal gas law [8].

$$\begin{cases} \text{Continuity} & \dfrac{\partial p}{\partial t} + u\dfrac{\partial p}{\partial x} + p\dfrac{\partial u}{\partial x} = 0 \\ \text{Momentum} & \dfrac{\partial u}{\partial t} + u\dfrac{\partial u}{\partial x} + \dfrac{1}{\rho}\dfrac{\partial p}{\partial x} + F_R = 0 \\ \text{Gas law} & \rho = \dfrac{p}{R_s T}, \qquad\qquad R_s T = \text{const.} \end{cases} \tag{2.2}$$

Here the three system variables are flow speed u, pressure p and gas density ρ, relative to time t and longitudinal position x. F_R denotes the flow friction term, R_s and T are the specific gas constant and temperature. The assumption of an isothermic process is a valid engineering assumption as a basic approach for this large and slow system [8]. This is generally valid for the modelling of pneumatic systems of this kind [18]. In order to simulate the system, the DEs must be discretised and linearised according to the finite volume structure.

2.2.4 Modelling of Nozzle Flow

For modelling the air flow between two pressure chambers that are connected by a restriction, the entire flow resistance can be modelled with a single equivalent nozzle diameter that regulates the flow rate between the chambers. The flow through this nozzle is compressible, which means that the flow chokes as soon as it reaches a mach number of 1, and cannot become any larger than this. Therefore, the flow through the nozzle is characterised as laminar, turbulent or choked flow. Laminar flow is present at conditions close to pressure balance, choked flow is reached as soon as the ratio between upstream and downstream pressure goes above the critical pressure ratio b. Everything in between is covered by the turbulent flow case. The mass flow ratio $\dot{m}$ for the different cases can be modelled mathematically as [18]

$$\dot{m} = \begin{cases} k_1 \cdot p_1 \cdot \left(1 - \frac{p_2}{p_1}\right) \cdot \sqrt{\frac{T_0}{T_1}} & \frac{p_2}{p_1} \geq 0.999 \\ \text{(laminar)} & \\ p_1 \cdot C_S \cdot \rho_0 \cdot \sqrt{\frac{T_0}{T_1}} \cdot \sqrt{1 - \left(\frac{\frac{p_2}{p_1} - b}{1 - b}\right)^2} & 0.999 > \frac{p_2}{p_1} > b \\ \text{(subsonic)} & \\ p_1 \cdot C_S \cdot \rho_0 \cdot \sqrt{\frac{T_0}{T_1}} & \frac{p_2}{p_1} \leq b \\ \text{(choked)} & \end{cases} \tag{2.3}$$

where p_1 and T_1 denote upstream pressure and temperature, p_2 denotes the downstream pressure, and ρ_0 and T_0 are density and temperature at reference conditions. The governing parameters are the sonic conductance C_S and the critical pressure ratio b. For a general sharp-edged restriction, they can be approximated mathematically from the restriction diameter d and pipe diameter D,

$$b = 0.41 + 0.272\sqrt{\frac{d}{D}} \tag{2.4}$$

$$C_S = 0.128 \cdot d^2 \tag{2.5}$$

where the unit of C_S is [dm^3/(s·bar)]. The linear gain factor k_1 for laminar flow is given by [18]

$$k_1 = 1000 \cdot C_S \cdot \rho_0 \sqrt{1 - \left(\frac{0.999 - b}{1 - b}\right)^2}. \tag{2.6}$$

The modelling of equivalent nozzles with a characteristic diameter and cross-section area is very common in the related literature [8], [19], but most methods do not have the benefit of efficiently handling all flow states, which is offered by the method presented here.

2.2.5 Numerical Integration Schemes

All of the modelling methods above require the solving of Differential equations (DEs), which is done numerically by using discrete integration schemes. The straightforward approach is the use of explicit methods, with Euler forward being the most basic 1st order method [20]. It approximates the change of the function value per time step on its derivative at the start of this time step:

$$x_{n+1} = x_n + hf(t_n, x_n). \tag{2.7}$$

A more advanced and very popular alternative is the Runge-Kutta method of 4th order (RK4). It achieves a higher accuracy by splitting the original time step in half and performing four sub-steps on these parts. Each sub step is similar to Euler forward [20], but uses different points of origin. The change of the function is then approximated by a weighted average of these sub steps:

$$\begin{aligned}
k_1 &= f\left(t_n,\ x_n\right)\\
k_2 &= f\left(t_n + \tfrac{1}{2}h,\ x_n + \tfrac{1}{2}h\, k_1\right)\\
k_3 &= f\left(t_n + \tfrac{1}{2}h,\ x_n + \tfrac{1}{2}h\, k_2\right)\\
k_4 &= f\left(t_n + h,\ x_n + h\, k_4\right)\\
x_{n+1} &= x_n + h\left(\tfrac{1}{6}k_1 + \tfrac{2}{6}k_2 + \tfrac{2}{6}k_3 + \tfrac{1}{6}k_4\right)
\end{aligned} \tag{2.8}$$

The higher order of this method implies that the error of the approximation decreases faster with a decrease of time step length h, but stability criteria must be met independently from the integration method order r. The local error can best estimated to be [16]

$$x\left(t_{n+1}\right) - x_{n+1} = \mathcal{O}\left(h^{r+1}\right). \tag{2.9}$$

2.3 Related Work

This section summarises the previous work done in the field of pneumatic train brake simulation. The brake system is a safety critical component of the train and has a very high operational influence on it. Therefore it is not surprising that a great deal of research and development have been performed in the field of rail brake simulation. The timeline of this work ranges from early implementations in the 60s and 70s [21], [22] to industry and application oriented software in recent decades. All of them rely to a large extent on experimental data or field measurements. Most prominent is the simulation tool *TrainDy*, which was developed by Cantone et al. in a broad cooperation within the UIC [23], [24], [25]. Another interesting development was performed by Melzi et al. as a part of the EU research project *DYNAFREIGHT* [26], [27]. Relevant individual contributions are Abdol-Hamid's doctoral thesis [28], as well as work done by Bharath et al. [29].

2.3.1 DYNAFREIGHT

The *DYNAFREIGHT* project was performed as a part of the *Shift2Rail* innovation programme. It was based on a cooperation of universities with KTH Stockholm as technical coordinator. The investigation scope of this project was on *Safety Precautions in Train Configuration and Brake Application* in the context of dynamic forces and derailment risk. Therefore, a simulation model for the pneumatic brake application at each wagon was required, which was implemented with an equivalent electric circuit model, as discussed in Section 2.2.2. It models the MP with two elements per wagon, where the middle nodes are used for the interaction of the MP with the DV and the end nodes are connection points that allow interaction with a locomotive [26]. These interactions are modelled as equivalent nozzle flow, an example for which is presented in Section 2.2.4. The arrangement of the equivalent electric elements is shown by Figure 2.4 below.

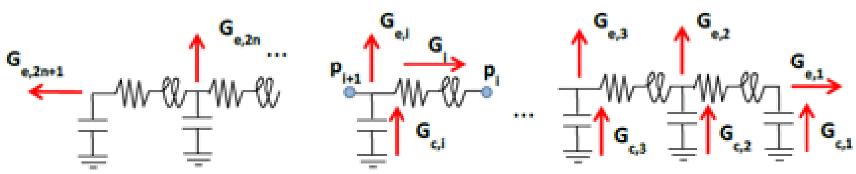


Figure 2.4 – Equivalent electric circuit setup of the pneumatic model used in the *DYNAFREIGHT* project. Source: [27]

The equivalent quantities for this lumped parameter model are based on 1D flow equations from fluid mechanics as in [17]. The internal variables are the pressure drop Δp across and the mass flow G through the pipe elements. Equivalent resistance is described by flow resistance and the pressure drop it causes can be expressed as

$$\Delta p = \lambda \frac{8l}{\rho \pi^2 D^5} G^2, \tag{2.10}$$

with the pipe section length l and its diameter D [27]. ρ is the gas density. This equation can be seen as the equivalent representation of Ohm's law, with the difference that contains a square term of the flow G due to the non-linear nature of fluid flow. For the dimensionless friction factor λ, the equation

$$\lambda = \frac{1}{\left(2\log\left(0.5625\,\mathrm{Re}^{\frac{7}{8}}\right) - 0.8\right)} \tag{2.11}$$

is used, where Re is the Reynolds number, describing the level of turbulence in the flow. It is calculated as

$$\mathrm{Re} = \frac{4\mathrm{G}}{\pi \mathrm{D} \mu}. \tag{2.12}$$

The air viscosity μ and air density ρ must also be updated continuously. For μ the empirical relationship in Equation 2.13 is used, whereas ρ is calculated according to the gas law in Equation 2.14.

$$\mu = \left(1.84 - \frac{300 - T}{300} \cdot 10^{-5}\right) \tag{2.13}$$

$$\rho = \rho_N \frac{p}{p_N} \frac{T_N}{T} \tag{2.14}$$

For both equations above, T is the air temperature in K inside the pipe, which in turn is updated under the assumption of adiabatic flow, based on p. The subscripts N denote values in standard conditions ANR. The equivalent inductance L is based on the inertial effects in the flow of compressed air that should be modelled, and is given by

$$L = \frac{4l}{\pi D^2}, \tag{2.15}$$

whereas the equivalent capacitance C aims to represent the fluid elasticity within the volume of one pipe section with the gas constant $\bar{R}$. As there are three capacitance elements connected to the pressure nodes of each wagon, its length term l in Equation 2.16 is divided into $50\,\%$ in the middle element and $25\,\%$ in each of the two other elements.

$$C = \frac{\pi D^2 l}{4} \frac{1}{\bar{R} T} \tag{2.16}$$

An equation system can then be assembled and solved in accordance to solution schemes for electrical circuits. For n wagons there are $2n$ pipe elements, leading to $4n+1$ unknown variables. They are assembled in the state vector x in the following form:

$$\boldsymbol{x} = \{p_1,\ p_2,\ \dots\ p_{2n+1},\ G_1,\ G_2,\ \dots\ G_{2n}\}^T \tag{2.17}$$

This requires the same number of equations for the system to solve. For the pressure drop $2n$ equations are assembled in the form of Equation 2.18, and for the flows, corresponding to Kirchhoff's law in an electrical circuit, $2n$ equations are assembled in the form of Equation 2.19. In addition, the total sum can be expressed as Equation 2.20 [27].

$$p_{i+1} - p_i = R_i G_i + L_i \dot{G}_i \tag{2.18}$$

$$G_i + \sum_{k=1}^{i} \left(-C_k \dot{p}_k\right) = \sum_{k=1}^{i} G_{ek} \tag{2.19}$$

$$\sum_{k=1}^{2n+1} \left(-C_k \dot{p}_k\right) = \sum_{k=1}^{2n+1} G_{ek} \tag{2.20}$$

G_e denotes the external flows mentioned earlier, which arise at the nodes that are coupled to a DBV or where there is a flow exchange with the DV. The DBV is modeled with two different cross-section areas, for service and emergency braking, and the AD in the DV is assigned one fixed cross-section that is opened as soon as MP pressure drops [26]. The system is set up in the form

$$\left[\boldsymbol{A}\left(\boldsymbol{x}\right)\right]\,\dot{\boldsymbol{x}} = \boldsymbol{Q}\left(\boldsymbol{x}\right) \tag{2.21}$$

and solved for $\dot{\boldsymbol{x}}$, which can be integrated per time step to calculate the pressure change. The pressure distribution in the DV is modelled in two steps. The pressure drop in the MP is first translated into a proportional command signal which is then processed according the initial application feature and limited gradient imposed by the braking type set in the DV.

Already before calibration is performed, the pipe length parameter l is scaled up by $7.5\,\%$ to consider bends and other means of concentrated pressure loss. Full scale validation and calibration of the model is performed using experimental data from Faiveley Transports test facility. Calibration is done by adapting the diameter in the DBV model and by reducing the pipe diameter parameter D to $70\,\%$ specifically in Equation 2.10. After these adaptations, the models shows high performance despite the relatively simple calculations.

Still, some systematic deviations dependent on the simulated train length do not seem to be removable with the calibration methods presented [27]. Refilling of the MP and brake release simulations are only mentioned very briefly in the published material [26] or not at all [27].

2.3.2 TrainDy

The development of the simulation tool *TrainDy* was a joint research project with high industry participation. It covers general longitudinal train dynamics. The simulation of the pneumatic system therefore takes an important role in this tool. It is modelled similarly to the approach presented in Section 2.2.3, but using a more accurate method including energy conservation for the exact simulation of temperature changes. It also allows the implementation of a variable pipe diameter, modelling quasi-1D flow [23]. The governing system of differential equations is therefore:

$$\begin{cases} \dfrac{\partial \rho}{\partial t} + u\dfrac{\partial \rho}{\partial x} + \dfrac{\rho}{S}\dfrac{\partial (uS)}{\partial x} = -\dfrac{\dot{m}}{Sdx} \\ \dfrac{\partial u}{\partial t} + \dfrac{1}{\rho}\dfrac{\partial \rho}{\partial x} + u\dfrac{\partial u}{\partial x} = \dfrac{\tau}{D} + \dfrac{u}{\rho}\dfrac{\dot{m}}{Sdx} \\ \dfrac{\partial q}{\partial t} + u\left(\dfrac{\partial q}{\partial x} + R\dfrac{\partial T}{\partial x}\right) + R\dfrac{T}{\rho S}\dfrac{\partial (\rho u S)}{\partial x} \\ \qquad\qquad = 4\dfrac{\phi_T}{\rho D} - \dfrac{\tau u}{D} - \dfrac{\dot{m}}{Sdx}\dfrac{1}{\rho}\left[(c_v + R)\, T_l + \dfrac{1}{2}u_l^2 - q\right] \end{cases} \tag{2.22}$$

The new terms introduced here, compared to Equation 2.2, are the pipe cross-section S, the specific heat q, the thermal flux ϕ_T and the specific heat c_v. The suffix l denotes the lateral contributions. The friction term τ contains both the distributed and concentrated dissipative sources in the system [24].

The interaction of external components with the MP pressure is calculated also in this model via equivalent nozzles, the mass flow contribution is included through the term $\dot{m}$ in Equation 2.22. This is due to the excessive computational effort that would be required to model them explicitly. In addition to the DBV and AD, even the Rs and their refilling upon brake release are included in the model. For the DBV, three equivalent areas for the service, emergency and release cases are identified from experimental data. The pressure control of the BC, which happens in the DV, is modelled even here in two transfer functions. The first represents the proportional control level applied by the MP pressure drop, the second is the time-dependent limitation curve set by the brake mode. The final output is practically the minimal or

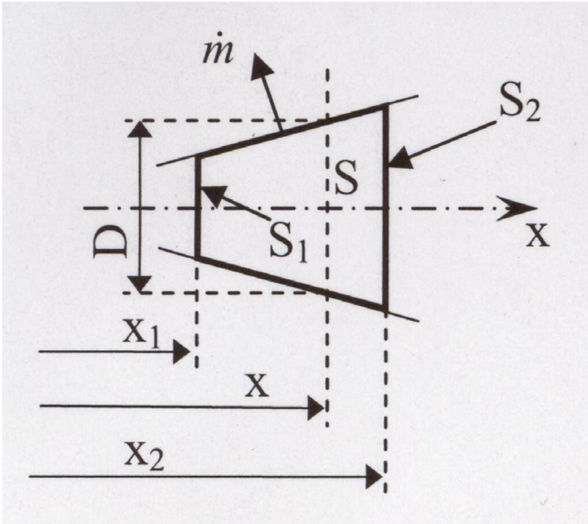


Figure 2.5 – Schematic view of the pipe model used in TrainDy. Source: [24]

maximal value from those two functions, dependent on if the system is in application or release [25].

Validation and calibration of this simulation tool was based on a considerable amount of reference data. In total 28 test runs and simulations performed by several operators as well as Faiveley Transport were used. These show the very high adaptability and accuracy of the produced results. Therefore, this tool has been adapted by the UIC as base for the revision of standards and guidelines [23].

2.3.3 Independent Research

In addition to the publications related to the projects mentioned above, two more research contributions shall be mentioned. One of them is Khaled Sayed Abdol-Hamid's doctoral dissertation, *Analysis and Simulation of the Pneumatic Braking System of Freight Trains*, which was published in 1983 [28]. The modelling method used there for the MP is similar to the one used in TrainDy, but isothermal flow is assumed. This work contains extensive modelling of additional components such as DBV and DV, which may improve the accuracy of the results but exceeds the scope of this work. Particular attention is also paid to modelling leakage and its effects on system behaviour.

Another alternative approach to pneumatic brake modelling is proposed by Bharath et.al. in [29]. There a lumped parameter model of brake pipe sections used, which connects chambers containing an equivalent spring and piston system that models the DV behaviour. The system is discretised using a finite differences scheme. Even though this approach may not be as practicable as the ones proposed recently, the modelling of friction within the flow through the MP using the Colebrook formula may be an interesting alternative to the methods used by others.

2.4 Summary

The most important features and characteristics of the of UIC and AAR systems are summarised in Table 2.1. Even though only the first line describing the UIC system is directly relevant for this work, the comparison shows the distinction between the systems and also shows what the important features are. This is the reason why the study of the AAR system remains in the scope if this work, as it shows what must be kept in mind for the sake of modularity. It can be seen as a baseline to know which other features should be possible to implement in the model structure and to deliver basic recommendations for how these could be implemented.

Table 2.1 – Comparison of the UIC and AAR systems.

System	Brake Release	Exhaustibility	Pressure Control	Additional Features	Braking Modes	Braking Percentage
UIC	Gradual release	Inexhaustible	Based on pressure ratio	Universal Action, Accelerating Reservoir	Switchable G/P	High (65-125%)
AAR	Single release	Exhaustible	Based on volume ratio	Quick Action, Quick Service, Quick Release, Pressure Retainers	Fixed	Low (10-40%)

The different model implementations in related work are summarised in Table 2.2. It shows that a wide range of methods has been used in the past an that up to today there is no clear method that has become prevalent for this type of simulation. Its structure also shows the most important modelling areas that need to be tackled and questions that must be answered before and while developing a model of a pneumatic brake for freight trains.

Table 2.2 – Summary of the modelling methods used in related work

Related work example	**MBP model**	**Additional components**	**Temperature modelling**
DYNAFREIGHT (Melzi) [26][27]	Equivalent Electric Circuit	DBV, AD as equivalent nozzle, DV empirical	adiabatic
TrainDy (Cantone) [23][24][25]	Finite Volumes	DBV, AD, R as equivalent nozzle, DV empirical	full energy equation
Abdol-Hamid [28]	Finite Volumes / Finite Differences	DBV, DV, BC explicitly modelled, leakage model	isothermal
Bharath [29]	Finite Differences	DV, R and BC as equivalent chamber-piston system Colebrook friction	isothermal

This base of knowledge of the technical system details gives a guideline for which components and features need or can be modelled and how they should behave. Based on the established modelling and simulation techniques placed in the context of the previous work, choices of modelling methods can be made for this work and implemented into the test bench.

Chapter 3

Methods

Considering the purpose and goals of this project, this chapter mainly focuses on modelling methods and how these methods are supposed to fulfil these goals. Section 3.1 describes the basic methodical approach towards system modelling, whereas Section 3.2 details the choice of modelling methods used for the individual system components or modules. Section 3.3 focuses on the options available for model implementation. The acquisition of reference data and methods for calibration and validation of the results produced by the model are explained in Section 3.4. Finally, Section 3.5 describes the documentation of the simulation tool that is delivered to Transrail at the end of the project.

3.1 Modelling of Dynamic Systems

A well-established approach for the modelling of dynamic systems is divided into three phases. As proposed in [16], these phases are:

- Phase 1: The problem is structured.
- Phase 2: The basic equations are formulated.
- Phase 3: The state-space model is formed.

The objective during the first phase is to structure and divide the system into subsystems, to define their variables and how they interact. The decision where to place the division points and how to connect the modules must be made carefully in order to make them exchangeable. It is important to already have the intended use of the model in mind during this phase. The result can then be visualised, for example in a block diagram. In this phase the level of complexity is determined as well, and connected to that, the degree of approximation.

The second phase translates this structure into a set of basic equations which describe relationships and constraints between the modules. This is also closely related to the intended level of approximation and deals with the question of which idealisations should be assumed. The question is to which extent the physical procedures actually must be calculated with the full background of physical equations, if some of them can be disregarded, or if they can be replaced by an empirical model based on the known functional logic.

The final and third phase then covers the setup of the full state-space model and its organisation in order to be solved and analysed as easily as possible.

3.1.1 Modular Setup of the Model

The intended modularity of this test bench shall enable user and developer to rearrange and replace individual parts with as little effort as possible and without risking to cause errors in the system. It can be divided into three different areas:

- Modularity by means of train configuration:
  - How many vehicles does the train consist of?
  - Which properties does each vehicle have?
  - Where are the locomotives located within the train?
- Modularity by means of modelling:
  - How is the contribution of each module calculated?
  - How can they be exchanged without needing to adapt the entire system?
- Modularity by means of system type:
  - Which regional standard do the system components and the control logic follow?

For the input of specifications about the train configuration that shall be simulated, a table format is defined. As sketched in Figure 3.1, this table is used by the model assembler algorithm, to bring it into a standardised form.

The train brake model can then be used by the model solver to produce the desired brake performance data in an optimal way.

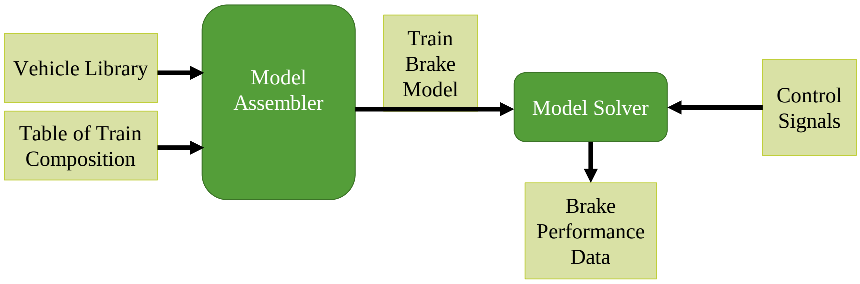


Figure 3.1 – Functional framework structure of the test bench.

As a result of the first modelling phase described above, a model library is created, which is a list of all relevant components. For the sake of modularity on the modelling level, each entry in the model library is assigned with several variants, instead of strictly defining the modelling method. In phase two, all equations for the individual components, and those equations that describe their connections, are specified for each variant. To simplify the exchange of module variants and even enable the parallel implementation in the test bench, the number and type of connections must be kept consistent between the variants of each module. The approach for the choice of modelling method for all modules is presented in Section 3.2. The model library should even contain special features of components as submodules, with the condition of widespread adaption in operation and that they give a significant contribution to the system behaviour.

Block diagrams are used for all visualisation of the model, such as the example in Figure 3.1 above. This seems like the natural choice as it is most relatable and well suited for this case: Both the system to be modelled and the model itself have clearly dividable units, represented by the individual vehicles, with the consistently arranged brake components as sub-modules.

For all modelling of pneumatic systems, the basic links between the submodules must be set to the pressure as the across variable and the flow as a through variable. This means that the interaction between the submodules is governed by the pressure difference between the two points that are connected, and by the mass flow through the connections. The product of those two variables then describes the transfer of power through the system, as pointed out in [18]. This means that a dynamic model is created, which is based on several DEs. Their number and complexity depends on the choice of modelling method made in Section 3.2.

3.1.2 Assembly of the Train Brake Model

The model assembler algorithm translates the train configuration table into a structure, called train brake model, that forms the base for the equation systems. These "inputs" to the assembly of the train brake model are no inputs in the real sense, but system parameters set via the train configuration and design parameters chosen by the modeller or user of the test bench. Examples for design parameters are the activation of features, setting of calibration factors and choice of simplification level for those modules where several options are retained in the test bench. According to the configuration sequence, system parameters are fetched from the vehicle library, in which types of vehicles and submodules are defined. A very important step in preparing the model is to assemble this list of system parameters that are required for depicting all the intended system variants, as well as the behaviour of additional sub-modules.

Each type of object in the vehicle library, such as wagons, locomotives, or DVs, is set to have the exact same sub-structure of parameters. Therefore vectors can be assembled for each parameter, which are complemented by auxiliary state vectors covering information about the status of the sub-modules. They also function as toggles, forwarding information about the locations in which of the modules shall be active.

3.1.3 Solving the Train Brake Model

As introduced previously in this chapter, the model to be simulated is a hybrid model composed of several sub-models. Each of them can be expressed as an individual set of DE, with varying interdependence between each other. In case of one-sided dependence, this means that their integration can be evaluated alternately for each time step. The case of interdependent systems in a multi-step integrator is more difficult, as all interdependent equation systems must be solved in parallel in order to preserve the mathematical correctness of the higher order method. Explicit schemes are used for the time integration, due to simplicity and familiarity with the implementation. Implicit integration schemes could become interesting if no stable simulation can be achieved with explicit schemes [20], but they are not the initial choice.

Some of the interactions between the sub-modules are steered by the control signal, which is the only real input to this model. It introduces all changes to the otherwise stable system. Its format must be defined as one of the first steps of model implementation. It is assigned during the assembly step as well, where it is adapted to fit the time resolution of the model solver.

In addition to the evaluation of the main equation systems, state machines take an important role in the model. They are needed to control the empirical model parts which are known to show different behaviour dependent on the system status. By updating the auxiliary state vectors after each time step, they define which optional parameters and equations should be used for the calculation in each position of the model, and if a value should be assigned at all.

The model outputs are assembled after performing all integration steps for the intended simulation length. The most important output are the brake application curves at each individual wagon.

3.2 Choice of Modelling Methods

A set of most reasonable modelling options is assembled for each module and ordered from simplest to most advanced and promising. The module interfaces are then designed in a way that they match all options and models are interchangeable. This makes it possible to switch between different options during the development process. It starts with the creation of the most simple baseline implementation that proves the validity of the general modelling approach. The modularity is then used to iterate and increment towards a more advanced model. Even in the finished test bench, there shall be some options left to allow its adaption to specific user needs in a simple way.

The backbone of the brake system, the MP, is a continuous pneumatic system, which must be aggregated in some way to be able to handle the number of state variables. The connection points with other subsystems are the first positions where the pipe must be divided, but in between them the aggregation scheme is dependent in the modelling method. Based on the approach described above, the intention is to keep the choice open between an EEC and FVM implementation. Arguments in support of the FVM are its high capabilities and its thorough mathematical description of the MP flow. Nevertheless, first implementation tests show that the required derivations for the FVM variant, in form of discretisation and linearisation to make the computational effort somewhat limited and efficient, exceed over the scope of this work. Therefore, the well-established and validated EEC approach is chosen instead, for which a more detailed description if the implementation is available. The option of an even simpler modelling variant for the first implementation is rejected as it eventually would have to be replaced. For a module that large this would take away a substantial amount of time from other parts of the project, as it would require an entirely different solver layout. Some of the simpler options would also not be as suited for the modelling tool chosen in Section 3.3. Even the choice of modelling method for other modules is somewhat dependent on simulation tool and framework of implementation. Some tools limit the possible options, or at least suggest the use of specific methods by being especially adapted to them or offering ready-made modules that can be reused.

An important principle that must be established for the modelling in the very beginning is that the average and not the extreme shall be modelled. This is related to the purpose of creating a model that depicts the behaviour most likely to be expected from the system. The data shall not be used to design safety criteria based on the most extreme premises, but to give an accurate idea of how an average system would behave in the most probable case. But still, the incentive to interpret the results with a tendency to the conservative side remains, when estimating uncertainties.

3.2.1 Physical vs. Empirical Modelling

As mentioned above, the most typical means of simplifying a model is the introduction of empirical elements. These are modelled by defining a known behaviour in terms of an equation or logical clauses, leaving the actual physical phenomena in a "black box". This minimises the effort required to establish the first functional level of the model, but also poses a risk of neglecting important relationships and details. In a complex and interdependent system, this can also lead to issues with interaction between logic boxes that produce unforeseen and incorrect behaviour. Therefore, physical modelling is often the final goal, even though it can require excessive computational effort. Another limiting factor with physical modelling can be that it requires detailed information that is not available to the modeller.

This trade-off makes the use of hybrid models common, which incorporate both empirical and physical parts. The development process in this project is also based on a hybrid model, which is iterated to gradually include more and more calculations based on the physical background of the system. As discussed above, the purely empirical approach is skipped for the MP model. For the DV, empirical transfer functions are used, which are presented as a common way to go in the literature. For them, the limiting curve shape is gradually improved to be closer to the real example. A similar process is followed for the DBV. The results are different stages of a hybrid model as recommended in [19] and as implemented in *DYNAFREIGHT* and *TrainDy*.

For the early calibration of system parameters, identification and fitting are performed towards values defined in industry standards. This is ideally done in a step-response experiment for parameter estimation, as described in [16].

3.3 Choice of Implementation Tool

This train brake model is not only developed in theory, but is also implemented as a test bench. Therefore, an implementation environment must be chosen that will be valid for its future use, maintenance, and further development. The process towards this decision is described in this section.

For this decision, it is important make a good choice as it will affect the whole development decisively. Therefore, different aspects of the choice must be weighed against each other and all important characteristics for the system must be identified. Familiar options are naturally favoured from the start, which is reasonable as they can be considered a safe choice if they fulfil basic requirements. On the other hand, there may be less familiar options that are much better suited for the application.

3.3.1 Tool Options

Below follows a list with brief descriptions of a number of reasonable and available tools, that are eligible for the implementation of the train brake model.

Java with GAMS

Java is a programming language that has been used to a large extent by Transrail before, therefore it is the natural choice from the company's perspective. Is is an object-oriented language which is beneficial for modular modelling applications. It is not intended for performing demanding calculations, which is why it would have to be used in combination with the Apache Commons Mathematics Library. Even more powerful would be the use of the computation tool GAMS, which is a system especially developed for physical modelling. There is previous experience at Transrail with this combination of tools.

Python with NumPy

Python is another popular programming language that could be used with a viable adaptation effort. Is supports object oriented programming and there are several toolboxes for numerical applications available, most prominently the program library NumPy that enables basic vector algebra up to efficient numerical applications. Nevertheless, dedicated modelling libraries for physical systems usually use different programming environments.

MATLAB

MATLAB is a commercial software offering a programming environment for scripting of calculations. It is well-established within research and education for numerical calculation and simulation, based on vector and matrix algebra. On top of its basic functions there are several specialised toolboxes available that contain ready-made functions for different application areas. For this project, MATLAB comes with the benefit of very high level of implementation practice, which not only increases efficiency but also promises a more profound progress in experience through this project. By mainly relying on basic functions, the license cost for MATLAB can be kept within a limit.

Simscape

The most advanced multiphysics toolbox offered within the MATLAB software is Simscape. It offers an environment for object-oriented modelling of physical systems, where ready-made object blocks are arranged in a graphical user interface. These modular blocks are available in form of several libraries for types of physics. The system parameters can be set for each block individually or in bulk, but the underlying physical modelling is fixed. By building larger block sections out of several blocks, a modular vehicle library can be created, but a rearrangement would require accessing the model within Simscape and some manual effort. The implementation for this project could be built on basic previous experience with the toolbox, but it is coupled to a high additional license cost for commercial applications.

3.3.2 Tool Selection

Pugh matrices are a well-established method to support decisions of all kinds, and are especially suitable when there are several similar options to serve a purpose that can be divided into distinct aspects. They are formulated in seven points in this case. Most important is the level of familiarity with the tool, as this increases the efficiency of model implementation, and therefore frees up time for model development. As important is availability in terms of licensing at Transrail. The final aspect is also of central importance, regarding the competence at Transrail with the tool for an easy delivery of the test bench, and continued development beyond this project. Other aspects of importance are how well the tools are suited for the specific implementation goals of this project, such as simulation of a multiphysics system with high computational effort. An object-oriented programming environment can support the modular setup of the model. The second last point covers the aspect of the intended learning progress through this project, both in terms of multiphysics simulation and programming techniques.

According to the priorities explained above, each aspect is given a weight factor, meaning that the most important aspects contribute with triple weight to the final score. Then, ratings are assigned to each tool option and aspect, ranging from very good "++" over neither good nor bad "0" to very poor "– –". The sum of each column is taken based on the weight factors and the options are ranked according to their final score.

Table 3.1 – Pugh matrix for the four possible simulation tools

	Weight factor	Java	Python	MATLAB	Simscape
Familiarity, expected efficiency	3	–	0	++	+
Availability, license cost	3	+	0	–	– –
Object oriented programming	1	++	+	+	+
High computational capability	2	+	+	++	++
Multiphysics background provided	1	0	0	0	++
Possible theoretical understanding outcome	1	+	+	++	0
Continued use of results (Transrail competence)	3	++	+	+	0
SCORE		11	7	13	4
Rank		2	3	1	4

The final score shows that there are two clear favourites for the simulation. Ranked second is Java, which would not have been an option without the strong background at Transrail. MATLAB ends up on first rank by a small margin, and is chosen for the implementation of the simulation tool. This choice is especially motivated by the highest expected efficiency and learning outcome.

3.4 Reference Data Collection

In this section, the options for the acquisition of reference data are presented and their use for validation is evaluated. A reasonable approach for a company like Transrail is to execute an experiment or field measurement in collaboration with one of its industry partners. But due to the limited time and the ongoing Covid-19 pandemic, this is not possible for this project. Therefore, it is necessary to rely on existing measurements, and to perform simulations with parameters that are adapted to match this available data.

Trainsrail has access to a measurement conducted in 2006 on a LKAB iron ore train. It does not consist of continuous time measurements of pressures but instead noted the times required for the BC to reach different pressure levels at the first and the last wagon of a $500\,\mathrm{m}$ train.

In addition to that, there is a lot of reference and calibration data published for the developments and research projects mentioned in Chapter 2. All of this data is only available in graphical form, which means that approximate samples can be taken from the data, but a systematic analysis would be very time-consuming and inaccurate.

Fortunately, it was possible to establish contact with Knorr-Bremse *Systeme für Schienenfahrzeuge*, one of the leading manufacturers of train brake systems in the world. Knorr-Bremse operate a test facility in Munich, which is an indoor facility containing a full-scale model of the pneumatic train brake system. Its wagon length is fixed to $15\,\mathrm{m}$, but the number of wagons and DV type can be adapted. Out of a list of published measurements for component approval, a couple of cases that show the most characteristic pressure profiles were chosen. The original data for these measurements was provided by Knorr-Bremse, as well as one additional test case performed after specifications made for this project.

On top of that, Stefano Melzi contributed with one data set of simulation results from the validated tool used in the *DYNAFREIGHT* project.

3.4.1 Evaluation Framework

The general evaluation method discussed in this section is used in the analysis of the results in Chapter 5. In order to utilise the available reference data in the best possible way, simulations must be performed based as closely as possible on the system configuration of the reference case. Even the time step length of the simulation should be chosen to be synchronous or in an even multiple with that of the reference data, in order to create as manly direct points of comparison as possible. This is easy to do with the test bench developed in this project due to the modular setup. Simplest is then to calculate the absolute pressure error per time step and to create an average over the hole simulation time. This aggregated error can then be compared between different calibration iterations, to find the optimal parameter values. The sign of the error can also be of interest, in order to know in which direction calibrations should be made. Taking the average then poses the risk of evening out the errors though, if the simulation for example is lags behind the reference in both directions. Therefore, a graphical analysis is more reliable for that latter purpose.

Another possibility is to evaluate the time it takes the system to reach certain pressure levels, such as the $95\,\%$ of target pressure specified in the standards [6]. This time can then be compared to the corresponding time in the reference data, and a timing error is obtained. This is much more relatable as pressure errors, and therefore especially suitable for presenting the final relative error estimations. Even the time it takes for the first pressure change to advance through the system after a stable state, as well as the first reaction to it, can be relevant to compare.

The final purpose of this evaluation is to estimate the level of accuracy as well as the domain of validity of the developed test bench.

3.5 System Documentation

The main means of system documentation is this report covering the methodological background, the general implementation as well as results and their evaluation. The program code for the simulation tool is delivered in a set of commented MATLAB functions and scripts, which give a more thorough understanding of the implementation and use. The spreadsheets and their defined format used in support of the MATLAB code are set up to be self-explanatory, but a brief instruction for starting a simulation and interpreting the results is included in them as well.

Chapter 4

Development of the Digital Test Bench

This chapter covers the realisation of the simulation tool, following the methods laid out in the previous chapter. This practical stage of the project is divided into three sections, starting with the modelling of the system according to the methodical principles established in Sections 3.1 to 3.2. The second section describes the implementation of this model in MATLAB. This chapter is concluded by a section about simulation, covering the work done to ensure the basic functionality of the test bench and to obtain the desired outputs. For the calculations in this chapter, SI-units are used. This means that a pressure which by standard is defined to $3.8\,\mathrm{bar}$ from now on is referred to as $380\,\mathrm{kPa}$.

4.1 Modelling

The practical execution of the first two modelling phases is covered in this section. For the division of the system according to modelling phase 1, there are two perspectives. The first and most intuitive perspective for the model definition is the external one, in which a train would be pictured as an string of vehicles. These vehicles are connected via the hose coupling of the MP and each vehicle produces a brake force output from a control signal input conveyed via one or more locomotives in the train. This perspective is used for structuring and assembling the system parameters taken from the vehicle library, and is illustrated as a block diagram in Figure 4.1.

The second perspective is the internal one, which focuses on the structural patterns in the train. It is oriented at the functional perspective as well, grouping the submodules of each vehicle to one new array or lumped submodule which allows handling all of them in bulk. This is directly apparent for the MP, being one actual continuous body of air, but can even be transferred to the other sub-modules of each vehicle. When the assembly of the train brake model is finished and passed on to the solver, it follows the structure dictated by this second perspective. This new structure is illustrated in Figure 4.2.

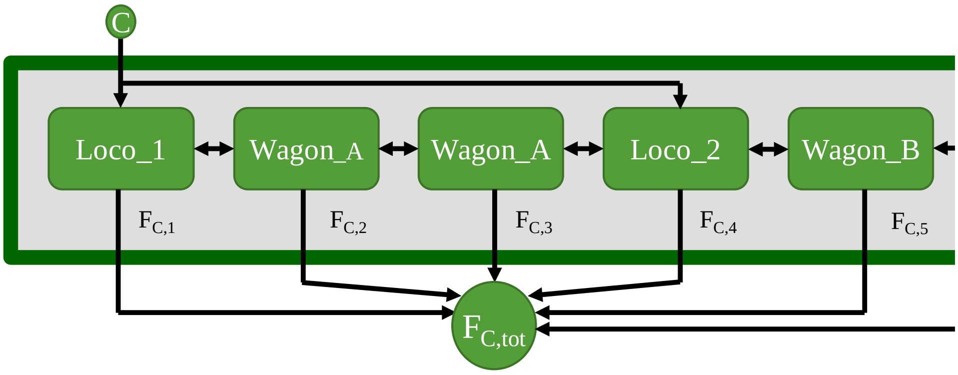


Figure 4.1 – Modular structure from an input perspective.

4.1.1 Model Architecture

The functional framework of the model is already presented in Figure 3.1. The remaining work of modelling phase 1 is then to construct the first level of the inner layout of the train brake model. It consists of three main modules which are the MP, the BC and one so-called auxiliary module. State vectors are assigned to these modules, which are tracking their respective pressures as seen in Figure 4.2. These state vectors are called MP pressure p, BC pressure p_C and the auxiliary pressure y. The first state vector forms together with the internal flow G the combined state vector x. The latter one contains all remaining pressures that interact with the MP module, including most features of the DV and the control pressures set by the DBV. The interaction between those three state vectors can be expressed mathematically as

$$\begin{cases} \dot{\boldsymbol{x}} = f\left(\boldsymbol{x}, \boldsymbol{y}\right) \\ \dot{\boldsymbol{y}} = f\left(\boldsymbol{x}, \boldsymbol{y}\right) \\ \boldsymbol{p_C} = f\left(\boldsymbol{x}, \boldsymbol{y}\right). \end{cases} \tag{4.1}$$

Equation 4.1 shows that x and y are interdependent whereas p_C only is dependent of the other two. This is related to the real interfaces in the system.

The pressure change in the main pipe and its surrounding are dependent of the flow through their interfaces, which is modelled as the internal variable G_e. G_e is in turn dependent on the pressure difference between x and y in each interface. In contrast to that, the circuit that is filling the BC is separate from the MP, so that even in the physical model there is no flow between the two. The BC pressure p_C is nonetheless controlled by the MP pressure x and may be dependent on sufficient R pressure contained in y, which leads to the third equation.

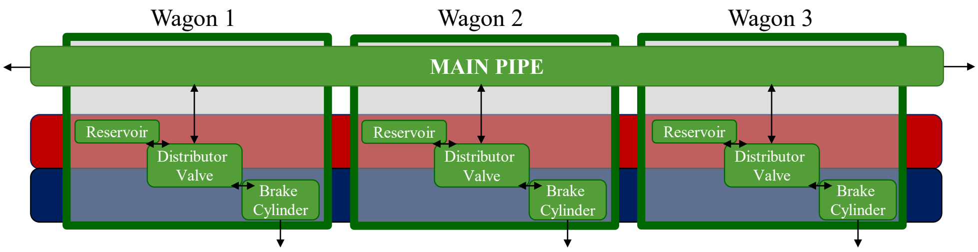


Figure 4.2 – Modular structure from a solution perspective.

4.1.2 Model Library

This list of modules is assembled from the core functional components in the system. In addition to that, it is completed with auxiliary features that are found to be common in service and considerably affecting the system behaviour. Table 4.1 below shows a summary of this list and its division into implementation stages. The first stage consists of the minimal basic implementation, in which only absolutely necessary modules for the MP are included. Its purpose is to proof the functionality of the basic modelling concept and to form a stable backbone which all additions can be build upon. The second stage is the implementation of the full model. In the third increment, optional modules are added and model complexity is increased. The fourth and fifth stages are not implemented in this work, but constitute an important part of the recommended improvements in further work, which are discussed in Chapter 7.

As seen in Table 4.1, both the actual DV module and the auxiliary module *AUX* exist. *AUX* contains several additional features that are physically located inside the real DV. A division is made in order to handle the different functions of the DV separately, according to their way of interaction. All submodules of *AUX* interact with the MP in some way and contribute a flow component to G_e. The module DV on the other hand explicitly does not affect the MP pressure when transferring the pressure signal. Therefore it can be modelled separately with its own interfaces.

Table 4.1 – Summary overview on the Model Library and its implementation stages.

<table>
<tr><th>Name</th><th>Interfaces</th><th>Sub Modules</th><th>Description</th><th>Stage</th></tr>
<tr><td rowspan="3">MP</td><td rowspan="3">Pressure nodes x,
Flow exchange G_e</td><td rowspan="3"></td><td>EEC, isothermal</td><td>1</td></tr>
<tr><td>FVM, isothermal</td><td>4</td></tr>
<tr><td>FVM, non-isothermal</td><td>5</td></tr>
<tr><td rowspan="3">DV</td><td rowspan="3">Pressure input x,
control output p_C</td><td rowspan="3">BC</td><td>Linear transfer function
and limiting curve</td><td>2</td></tr>
<tr><td>Proportional transfer
Improved limiting curve</td><td>3</td></tr>
<tr><td>Nozzle flow to
pre-control chamber</td><td>4</td></tr>
<tr><td rowspan="3">BC</td><td rowspan="3">part of DV
pressure level p_C,
supply flow from y</td><td rowspan="3"></td><td>No BC modelling
Pressure set by DV</td><td>2</td></tr>
<tr><td>Constant cylinder volume
Pressure transfer from R</td><td>3</td></tr>
<tr><td>Pressure dependent volume
Flow from R according to
Pre-control pressure</td><td>4</td></tr>
<tr><td rowspan="3">AUX</td><td rowspan="3">Pressures y,
Flow exchange G_e</td><td rowspan="3">DBV, R</td><td>DBV</td><td>1</td></tr>
<tr><td>DBV
AD</td><td>2</td></tr>
<tr><td>DBV
AD
Added components to G_e:
R-filling, venting, leakage</td><td>3</td></tr>
<tr><td rowspan="3">DBV</td><td rowspan="3">part of AUX,
control pressure y_{DBV}</td><td rowspan="3"></td><td>Set control pressure
Fixed nozzle diameter</td><td>1</td></tr>
<tr><td>Set control pressure
State dependent nozzle</td><td>3</td></tr>
<tr><td>Linear control pressure
State dependent nozzle</td><td>4</td></tr>
<tr><td rowspan="2">R</td><td rowspan="2">part of AUX,
reservoir pressure y_R
supply flow to BC</td><td rowspan="2"></td><td>Pressure transfer to BC
Refilling flow from MP</td><td>3</td></tr>
<tr><td>Full pressure-dependent
flow to BC
Nozzle function for
filling flow from MP</td><td>4</td></tr>
<tr><td>Loco
Brake</td><td>control pressure y_{DBV},
pressure level p_C</td><td>–</td><td>Synchronous with control,
time delay from brake mode</td><td>2</td></tr>
</table>

4.2 Implementation

This section describes the implementation of the test bench in MATLAB code. It follows the array of stages which are outlined in the previous section. For the sake of conciseness, no code is explicitly laid out. Instead, the implemented features and calculation methods are explained.

4.2.1 First Basic Model

This first stage of implementation only includes the MP module and a simplified DBV module, which is required to set the boundary conditions of the model. The MP module is modelled strictly according to the EEC approach proposed in [27], but assuming isothermal flow.

This means that the MP module is internally divided into two sections per wagon. The node pressure p is the across variable in the system, governing the mass flow G that is the through variable causing pressure changes. This structure is followed both within the EEC core of the model and along all interfaces with the auxiliary pressure vector y. These interfaces are in this case modelled as equivalent nozzles with a fixed diameter.

The equation system to solve the model is assembled as suggested in Equations 2.17 – 2.20. In order to make the solution process clearer, the equivalent resistance term can be isolated from Equation 2.10. All three equivalent electric quantities are then summarised as

$$R = \lambda \frac{8l}{\rho \pi^2 D^5} G, \tag{4.2}$$

$$C = \frac{\pi D^2 l}{4} \frac{1}{RT}, \tag{4.3}$$

$$L = \frac{4l}{\pi D^2}, \tag{4.4}$$

which means that the pure equivalent resistance is dependent on the mass flow G. This stands in contrast to electric resistance, which is usually modelled as not current-dependent. From the assumption of isothermal flow follows that R and C are calculated with a constant temperature T. The model description in [27] states the use of the universal gas constant $\bar{R}$, but dimensional analysis shows that the specific gas constant R should be used. For air, R is $287\,\mathrm{J/(kg \cdot K)}$, whereas $\bar{R}$ is $8.31\,\mathrm{J/(mol \cdot K)}$ [17]. By inserting Equations 2.17 – 2.20 and Equations 4.2 – 4.4 into Equation 2.21, it can be expressed as

$$\overbrace{\begin{bmatrix} -C_1(k,T) & 0 & \cdots & 0 & 0 & 0 & \cdots & 0 \\ -C_1(k,T) & -C_2(k,T) & & 0 & 0 & 0 & & 0 \\ \vdots & & \ddots & & & & \ddots & \\ -C_1(k,T) & -C_2(k,T) & \cdots & -C_{2n}(k,T) & 0 & 0 & \cdots & 0 \\ -C_1(k,T) & -C_2(k,T) & \cdots & -C_{2n}(k,T) & -C_{2n+1}(k,T) & 0 & \cdots & 0 \\ 0 & 0 & & 0 & 0 & L_1(k,T) & & 0 \\ \vdots & & \ddots & & & & \ddots & \\ 0 & 0 & & 0 & 0 & 0 & & L_{2n}(k,T) \end{bmatrix}}^{A(x)} \cdot$$

$$\cdot \overbrace{\begin{bmatrix} \dot{p}_1 \\ \dot{p}_2 \\ \vdots \\ \dot{p}_{2n} \\ \dot{p}_{2n+1} \\ \dot{G}_1 \\ \vdots \\ \dot{G}_{2n} \end{bmatrix}}^{\dot{x}} = \overbrace{\begin{bmatrix} G_{e1} - G_1 \\ G_{e1} + G_{e2} - G_2 \\ \vdots \\ G_{e1} + \ldots + G_{e2n} - G_{2n} \\ G_{e1} + \ldots + G_{e2n+1} \\ p_2 - p_1 - R_1 G_1 \\ \vdots \\ p_{2n+1} - p_{2n} - R_{2n} G_{2n} \end{bmatrix}}^{Q(x)} \tag{4.5}$$

from which $\dot{x}$ is obtained. For calculating the full development of x over time, $\dot{x}$ is integrated with the RK4 routine. In addition to the MP module, the only required module for this stage is a simplified DBV module, switching the reference pressure from release pressure to full application or emergency level. An example for such an early simulation result is given in Figure 4.3.

4.2.2 Full Model Implementation

The full model is implemented in the second stage, which means that the DV module is added, covering the pressure translation between the MP and the BC. This stage even includes the implementation of the auxiliary state vector, as well as the systematic handling of train configuration and outputs.

Modelling the DV

In this stage, the BC is not modelled explicitly at all. Instead, an empirical model consisting of two transfer functions is implemented, as recommended

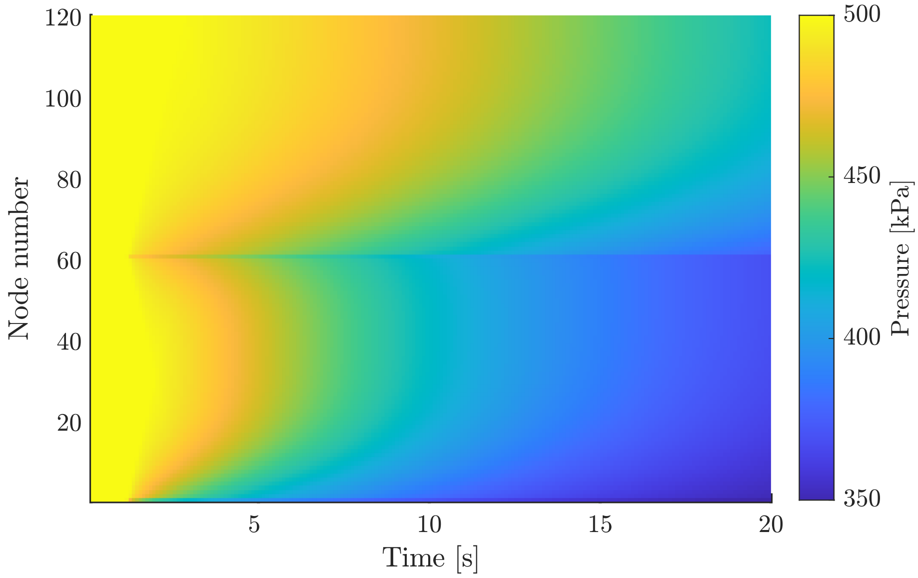


Figure 4.3 – Pressure reduction in the MP of a $900\,\mathrm{m}$ long train consisting of 60 wagons. A second locomotive is located in the middle of the train.

in [25]. The first of those is the linear transfer function, with the typical pressure ratio of $380\,\mathrm{kPa}$ BC pressure for $150\,\mathrm{kPa}$ pressure reduction in the MP. The second one is the time-depending limiting function. It is fitted to match the middle of the interval specified for brake application and release times in the UIC standard [6]. Three variants of limiting curves for the delayed brake action are shown in Figure 4.4. In the first implementation, this limiting function is fitted as a straight line up to $p_{C,max}$. The two additional variants are attempts to bring the empirical model closer to the profiles of the real DV. One attempt is to apply limited exponential growth and decay, the other one is based on equivalent nozzle flow. In the latter case, the flow profile is calculated by identifying an equivalent nozzle diameter that fulfils the time requirement for filling or emptying an arbitrary volume. Such a filling and emptying is then simulated with Equation 2.3 and the Euler-forward method. The resulting pressure profile is saved as a lookup-table for the rest of the simulation. This third variant is chosen as the most realistic one for the average case, and is used for all further simulations. It is also edited to cover the initial application feature, that allows a quick rise of the BC pressure to typically $80\,\mathrm{kPa}$, resulting in the green pressure profile in Figure 4.4. It is interesting to note that the exponential decay function almost exactly depicts the nozzle

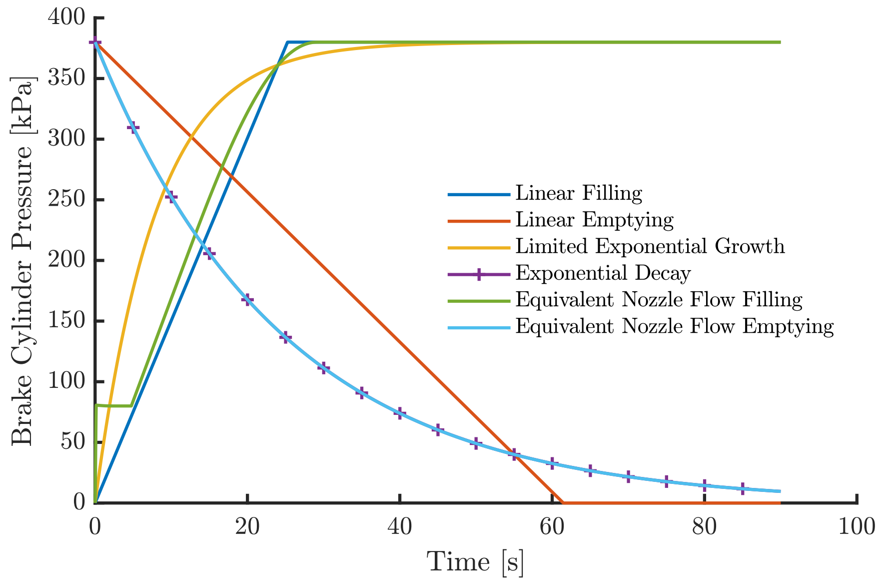


Figure 4.4 – Three variants of limiting curves for delayed brake application and release, in brake type G.

flow emptying profile.

For the assignment of p_C, both transfer functions are evaluated for the current MP pressure and progress in the timer, which is started in the beginning of each brake action. p_C is then set to the result corresponding to the smaller change from the previous time step.

The pneumatic variant of load-controlled braking is one common exception to the typical pressure ratio mentioned above. In the empirical model, the resulting lower BC pressure can be set manually by adapting the system parameter $p_{C,max}$. All transfer functions are then calculated to fulfil the requirements for this new target pressure instead.

As soon as the output p_C is calculated, it can also be used to estimate the corresponding braking force F_B acting on each wagon. It is obtained by multiplying the nominal maximal braking force $F_{B,max}$ for each vehicle with the fraction of $p_{C,max}$ that is currently applied in the BC. This value is then corrected for the effect of the retraction spring, which means that F_B is 0 below a BC pressure level of about $5\,\%$. F_B must still be seen as a rough approximation though, as there are many non-linear phenomena present in the mechanical transmission of the braking force [5], [8], which are neglected for simplicity.

The values of p_C and F_B for the locomotive brakes are calculated separately, using the limiting curve for the brake type set in the locomotive and the control pressure y_{DBV} as reference. They are added to the corresponding arrays when the outputs are assembled and saved.

Auxiliary state vector for the ADs

Another important part of this second stage is the systematic implementation of the auxiliary state vector and the external flow G_e entering and leaving the MP. G_e is modelled as equivalent nozzle flow and calculated using the equations presented in Section 2.2.4. The first auxiliary feature to be implemented are the ADs, because their large effect on the system behaviour can be identified in both simulation results and measurements that are presented in the literature. Each AD is modelled with a small volume that causes a pressure decrease of $30\,\mathrm{kPa}$ in the MP when filled. They are controlled by a simple state-machine that activates and opens the AD as soon as the MP pressure derivative, starting from reference pressure, falls below an assigned level. The AD chamber is then filled with air from the MP. It is closed as soon as the MP pressure falls below the pressure in the AD. All ADs are reset as soon as the brake is fully released. The pressure change caused by the flow $G_{e,AD}$ into the AD with a volume V_{AD} is modelled as an isothermal process, leading to

$$\dot{y} = \frac{G_{e,AD} \cdot R \cdot T_0}{V_{AD}}. \tag{4.6}$$

With these additions made, the internal structure of the train brake model is fully implemented, as illustrated in Figure 4.5. The illustration also shows the hybrid nature of the model, as it contains both the equivalent electric elements of the MP and the equivalent nozzles connecting the MP to *AUX*. The *AUX* module is symbolised by the red bar at the top of the diagram and contains at this stage the ADs and the DBVs, but more parts are added to it at the next stage.

Handling of train configuration and outputs

There are two more important means of fulfilling the basic goals of this project. The fist of them is the systematic handling of the train configuration and the assembly of the system parameters, which is described in Section 4.2.4. The second one is the recording of the outputs, with the purpose to provide organised data in large quantities for further analysis and as a base for development. The output of each simulation run is saved in a spreadsheet

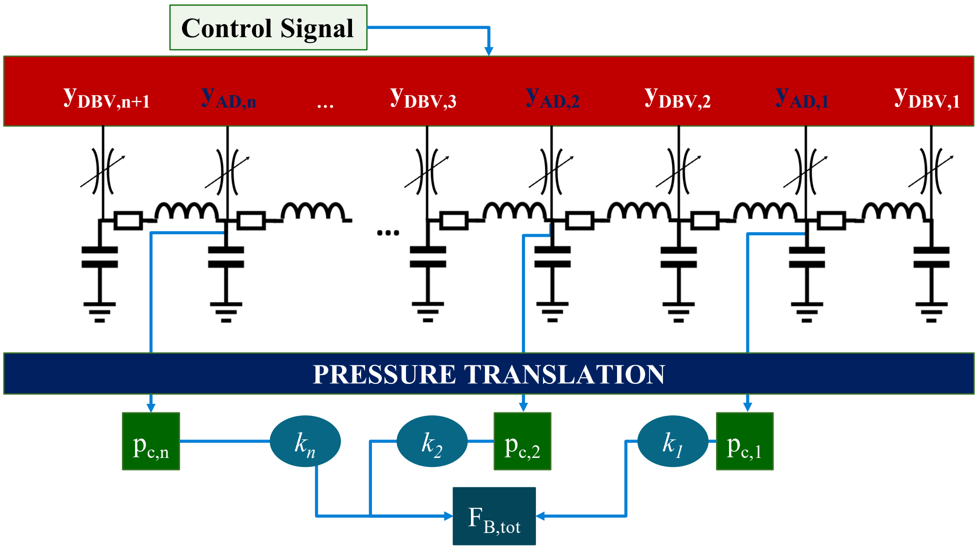


Figure 4.5 – Model structure of the equivalent electric circuit hybrid, *n* wagons.

file with four sheets. In the first three sheets, the values of p, p_C and F_B are saved, each with one data point per wagon and second, completed with the longitudinal position per wagon. The total braking force per train is written with one data point per second in absolute and relative terms in the fourth sheet.

The system parameter values presented in this section are a good example for the adaptability of the system due to the system assembly procedure. All of them can be adapted in the vehicle library to customise the simulation and depict individual systems as precisely as possible. This is also the reason why not all system parameters are mentioned explicitly, as they would still only depict one special case.

4.2.3 Further Module Increments

The third stage of model implementation does not introduce any further additions to the layout of the model, but it introduces additional features to the *AUX* module and takes the modelling method for some submodules one step forward. This includes the improved limiting functions for the DV pressure transfer already shown in Figure 4.4.

The additional features included in *AUX* are the monitoring of the R-pressure y_R, and an optional venting or leakage flow from the MP to the atmosphere. For the sake of clarity, y is split into four separate parts, one for each feature in *AUX*. G_e is calculated for each of them separately, and then assembled in the way it is illustrated in Figure 4.6. The placement of the DBV elements is on the outer connection nodes of the wagons, with indices $k \in \{1, 3, ...2n+1\}$, whereas the other elements are grouped on the central nodes, with indices $l \in \{2, 4, ...2n\}$, where the DVs are placed.

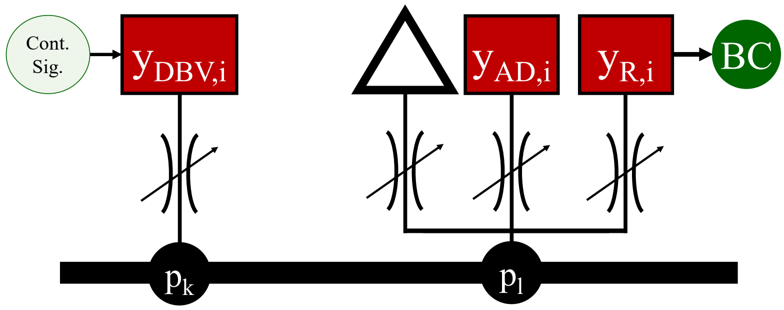


Figure 4.6 – Sketch of the models included in the *AUX* module.

The monitoring of y_R includes both the air consumed by filling the BC, and the refilling from the MP upon brake release. The transfer of pressure to the BC is modelled in a simplified way: For each time step, an increase of p_C is translated to a decrease in y_R based on their volume ratio and assuming an isothermal process.

$$\Delta y_R = -\Delta p_C \cdot \frac{V_C}{V_R} \tag{4.7}$$

All new features are implemented with a variable nozzle diameter, which is controlled by state machines. A variable diameter is also added to the equivalent DBV nozzle, with the three states; service, emergency and release. The assignment of these states is controlled according to the control pressure level and its relation to the adjacent node pressure p_k, which reveals the flow direction. At every time step, the reference chamber pressure is set according to the control signal, so that it is not affected by the flow $G_{e,DBV}$.

The feature of local venting of the MP does not represent any common feature on UIC-DVs, but enables the simulation of leakage and a modern variant of AD, where no chamber is used. Furthermore, local venting of the MP is used as emergency feature of AAR systems, so that no structural adaption of the system will be required to simulate the AAR variant. The downstream pressure for venting is handled similarly to the DBVs, with the difference that it is set to atmospheric pressure. The state activation of the larger venting nozzle diameter follows the same logic as the one for the AD, whereas a very small leakage diameter can be assigned to be permanently open.

The state machine controlling the nozzle for the refilling of R is slightly more complicated. In order to model the two steps of refilling described in Section 2.1.1, the state machine follows the logic visualised in Figure 4.7.

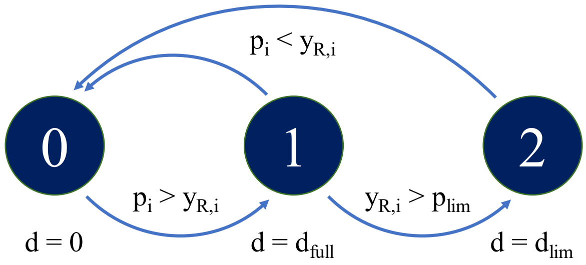


Figure 4.7 – Graphical representation of the state machine logic for the filling of R.

Many of the features and modules described in this chapter are optional or only appear on a limited number of vehicles in the train. During the train brake model assembly, this is captured in a vector called *set*, which is nonzero for each component that actually is present in the train. For those components which can be switched on and off during the simulation, this is handled by their state machines. By including the *set* and *state* vectors in the calculations of the state variables, it is made sure that only the active components contribute to the system behaviour.

4.2.4 Train Brake Model Assembly in MATLAB

The concept of assembling submodules of each vehicle into arrays for bundled handling, as introduced in Section 4.1.1, is consequently continued down to the system parameter level. For that purpose, the *Train Configuration* spreadsheet is read column by column. Each column represents one definition of a vehicle, containing either a locomotive or wagon type number, a valve type number and the number of vehicles k of that kind. The assembler algorithm then reads all system parameters of the vehicle and the valve in the *Vehicle Library* spreadsheet, and adds them to the corresponding vectors. This is repeated k times for each column. As soon as the whole configuration array is read, dependent parameters are calculated and the data is processed in order to fit the solver. The use of spreadsheets makes train configuration a quick, easy and clear task. Apart from the arrangement of vehicles, the *Train Configuration* spreadsheet also contains some general parameters, as well as the specification which control signal to use.

These control inputs are read from a separate *Brake Control* spreadsheet. Each signal consists of one time vector and a vector with the application levels. The application levels are given in relative terms, which in reality often correspond to the fixed brake handle positions. A common example would be positions 1–8 for minimal to full service application, and 9 for emergency braking. In the *Train Configuration* spreadsheet it is specified if the application levels shall be adapted with linear interpolation or step interpretation, in order to make the time vector fit the simulation time step. The application levels are then translated to corresponding target MP pressures.

The formatting and some example entries of the three spreadsheets used for the train brake model assembly are shown in Appendix B.

4.2.5 Solver Code Structure in MATLAB

The structure of the solver links together all the calculation steps and parts of the model that have been discussed so far in this chapter. One time step calculation is visualised by the block diagram in Figure 4.8, going from top to bottom.

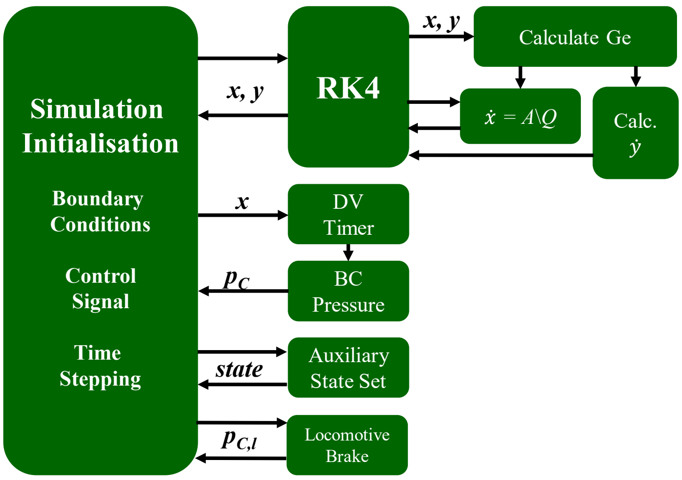


Figure 4.8 – Solver structure implemented to solve the model.

The initialisation function on the left acts as the backbone of the code, by calling and connecting the other functions. Before the first time step, it also sets the time step t_0 from the initial values defined for the simulation. The time step calculation then starts with performing one RK4 integration step based on the values of the previous time step. After this parallel calculation of new x and y values, p_C is calculated from x. Then all states for the submodules are updated. The time step is concluded by calculating the locomotive brake levels.

4.3 Simulation

This section describes the implementation steps that are directly based on simulation. The work process of model implementation depends strongly on the monitoring of the model behaviour as well as on the evaluation of changes and additions. This means that each iteration or increment of the code is tightly connected to a thorough analysis of the simulation results, based on the comparison to previous results. This procedure of testing and debugging ensures that the model correctly depicts the qualitative behaviour of the pneumatic brake system.

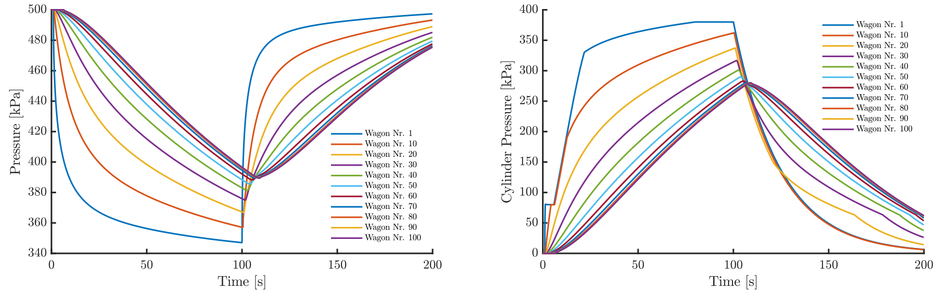
Three areas of simulation testing are described in this section, namely the variation of input signals, the tuning of parameters and tests with the temporal and spatial resolution.

4.3.1 Variation of Input Signals

The intention for the model is that it not only is able to handle on/off inputs by performing the full transition. Instead, switching between filling and emptying must always be possible amidst application or release, as shown in Figure 4.9. For that purpose, it is controlled if the MP pressure has changed direction before each p_C translation. If that is the case, the translation direction is reversed and the timer is reset to fit the current BC pressure on the other limiting curve.

The handling of gradual application and release of the brakes is also very important for depicting the correct behaviour of the brakes in service. Again, the timer function for the empirical pressure translation curve must be adapted to make that possible. Whenever a stable level of p_C at target pressure is identified, the timer is interrupted until the signal changes again.

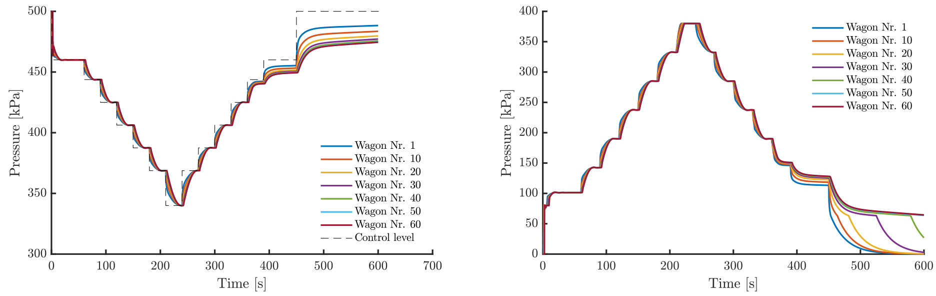
Other details affecting the input are the minimal allowed command pressure levels at application and release. A thorough study of the UIC standard [12] reveals that the brakes always must be applied with a MP pressure drop of at least $40\,\mathrm{kPa}$ and fully release when the MP pressure drop becomes smaller



(a) MP pressure (b) BC pressure

Figure 4.9 – Example of brake release during ongoing application on a $1.5\,\mathrm{km}$ train.

than $25\,\mathrm{kPa}$. This is one of the many examples where additional system parameters are included into the *Vehicle Library* during the simulation and testing, in order to maintain correct results without decreasing the adaptability of the model. The result of the last additions is visible in the pressure profiles in Figure 4.10.

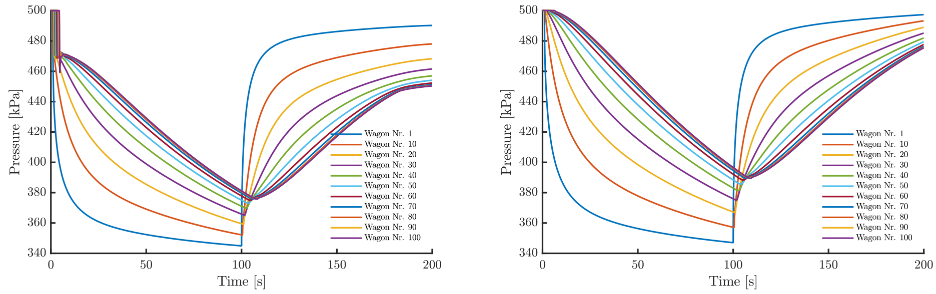


(a) MP pressure (b) BC pressure

Figure 4.10 – Example of gradual brake application and release on a $746\,\mathrm{m}$ train with brake type G.

4.3.2 Tuning of Parameters

Some system parameters can be calibrated already during the model implementation, because their effect on the pressure is clearly visible, and the intended effect is defined in the literature. One example is the identification of AD nozzle diameters and volumes. The equivalent diameter is identified with step-response experiments to be $5\,\mathrm{mm}$, which agrees well with values from related work [25]. After some attempts to identify AD volumes, a system parameter for the pressure drop caused by their filling is added as an optional override. If the pressure drop is set, the AD volume is calculated according to it during the parameter assembly, dependent on the MP volume and therefore the wagon length. With this variant the ADs can easily be controlled to behave as intended, even in a train with varying wagon lengths. A comparison of the MP pressure profiles with and without AD is shown in Figure 4.11.



(a) MP pressure, active AD

(b) MP pressure, inactive AD

Figure 4.11 – Comparison of brake application with and without AD on a $1.5\,\mathrm{km}$ train.

Another important system parameter to be estimated is the equivalent nozzle diameter for the DBV module. This diameter was identified to be in the range of $10.9\,\mathrm{mm}$ to $18.7\,\mathrm{mm}$ in related work, dependent on the type of application and DBV variant [25]. However, in that model a different flow coefficient is used, so a direct adaptation of those values is not reasonable. A value in the range above is used as a starting point for development, which produces a qualitatively right model behaviour. The analysis and calibration of the DBV diameters is covered in Chapter 5.

4.3.3 Temporal and Spatial Resolution

The temporal and spatial resolution describes the division of the model into discrete steps in time and space. The spatial resolution (state variable aggregation) is fixed by the current EEC MP model. The temporal resolution (time step length) is strongly related to the integration method and can lead to stability issues. Except for that, there is no explicit rule for the choice of the temporal resolution, it is merely a trade-of between computation effort and accuracy.

The standard in-built RK4 function in MATLAB uses a variable step length. This function is not suited for the given model though, because parallel integration of interdependent functions and the switching of parameters during the run are not directly supported. Instead, an own implementation of a fixed step RK4 is used in order to guarantee a correct and reliable base for the integration. This solution is clearly not computationally efficient, and means to remedy this deficiency are suggested in Chapter 7.

The only cause of instability issues that is encountered for this model is the calibration of the equivalent fluid resistance R. A reduction of R during calibration tests make a decrease of step length from $10\,\mathrm{ms}$ to $1\,\mathrm{ms}$ necessary. The model outputs are otherwise stable for a step length of $10\,\mathrm{ms}$.

The influence of time step length on the output accuracy is examined in one case of service brake application and release. The case is based on a $746\,\mathrm{m}$ long freight train in with brake type G. The required times for pressure changes in the first and the last wagon are noted. The pressure levels to be reached are $95\,\%$ of full pressure reduction and $80\,\%$ of full pressure restoration. For a step length of $10\,\mathrm{ms}$, the pressure release time in the MP is $29.52\,\mathrm{s}$ to $39.78\,\mathrm{s}$ and the pressure refilling time is $24.72\,\mathrm{s}$ to $72.53\,\mathrm{s}$. The corresponding values with a step length of $1\,\mathrm{ms}$ are $29.50\,\mathrm{s}$ to $39.76\,\mathrm{s}$ for MP release and $24.70\,\mathrm{s}$ to $72.42\,\mathrm{s}$ for refilling. The deviations between the two cases are in the range of $0.02\,\mathrm{s}$ to $0.11\,\mathrm{s}$ or $0.08\,\%$ to $0.15\,\%$. In accordance with Equation 2.9 for the estimation of the local numerical error, it can be assumed that the remaining numerical error is considerably smaller in scale than the calculated deviations. This allows to assume that the total numerical error in the calculations does not exceed the magnitude of the values shown here, showcasing the accuracy of the method and implementation.

A brief analysis is also made regarding the fulfilment of the CFL-condition presented in Section 2.2.1. For the fully developed flow in the main pipe, the maximal flow velocity w is estimated from the mass flow G to be around $30\,\mathrm{m\,s^{-1}}$ to $50\,\mathrm{m\,s^{-1}}$. In order to fulfil the CFL-condition with that w and a minimal wagon length of $5\,\mathrm{m}$, the time step length may not exceed $50\,\mathrm{ms}$. When instead considering the propagation speed of a pressure wave at $250\,\mathrm{m\,s^{-1}}$, the maximal allowable step length goes down to $10\,\mathrm{ms}$, which means that the standard simulation settings closely fulfil the condition. See Equation 2.1 for the calculation process.

The effect of the step length on the limiting curve calculations presented in Section 4.2.2 is also analysed, and its influence is found to be marginal. Due to the identification of an optimal nozzle diameter for the curve calculation, only the curve shape is affected by the step length. As the intersection with the standardised pressure point is fixed, there is only little room for deviation. The pressure difference at each time step is in the magnitude of $0.1\,\mathrm{kPa}$ and therefore not noticeably affecting the results.

Chapter 5

Results and Analysis

This chapter presents the results produced by the test bench, their analysis, as well as their comparison and calibration with reference data.

The main purpose of a test bench is to produce accurate outputs for a specific input. It shall allow to draw conclusions from the variation of parameters, to make comparisons to reference data, and to test new configurations.

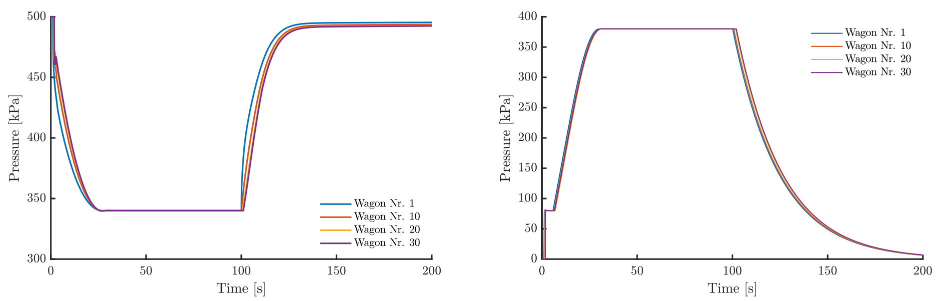
Considering the specific characteristics of a test bench for pneumatic train brakes, the metrics can be further narrowed down: From a result analysis perspective, the main interest is how the system transfers from one state to another. However, these states only depend on the pressure in the MP and BC, which vary within strictly defined ranges. Therefore, the important metrics are the time which is required to transfer from one state to another, and the shape of the pressure profiles during the transfer. Another relevant question is how far the state transfer can progress before a certain time, when the transition may be reversed again.

Beyond that, individual BC pressures and subsequent braking forces allow further analyses. The analysis of the brake system's impact on the dynamics of the train as a whole is most immediate. For that purpose the sum of the forces acting on the train are considered. In this context, it can be studied how pressure profile shapes vary with the position in a train, and how these characteristic shapes compare to the average pressure profile. In addition, the individual braking forces on each wagon can be used to analyse longitudinal forces transferred between wagons, a use case that the test bench is prepared for within this work.

5.1 Major Results

In this section, the outputs that can be produced by this test bench are presented. As they vary greatly dependent on the choice of train configuration, they are exemplified with two typical configurations for Swedish freight traffic: The first one is a short but heavy freight train carrying steel plates, which is referred to as steel commuter. It is composed of 30 flatbed wagons with a length of $13.9\,\mathrm{m}$ each and drawn by two locomotives in the front, each $19\,\mathrm{m}$ long. The train has a total length of $455\,\mathrm{m}$, due to which it is operated with brake type P. Still, the total loaded weight of $3168\,\mathrm{t}$ imposes that the locomotives are operated in brake type G. The second example is a long and heavy freight train for the transportation of iron ore. It is assembled from 68 ore dumper wagons with a length of $10.3\,\mathrm{m}$ each. This train is drawn by a twin-unit IORE locomotive, bringing the total train length to $746.4\,\mathrm{m}$. Due to the length and total loaded weight of $8520\,\mathrm{t}$, the entire train is operated with brake type G.

The most basic simulation scenario is the full service application of the brakes followed by a full release. It is shown for the steel commuter in Figure 5.1 and for the iron ore train in in Figure 5.2. The application times

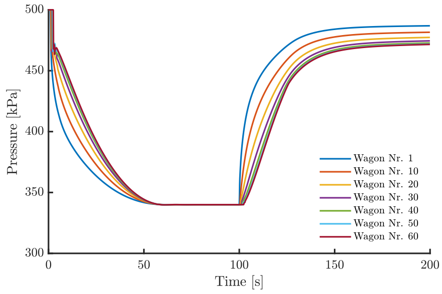


(a) MP pressure (b) BC pressure

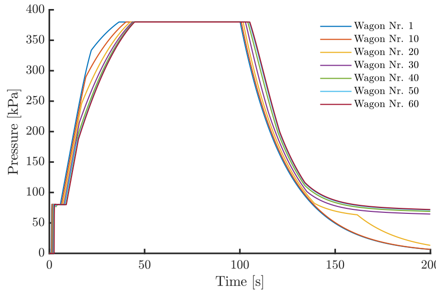
Figure 5.1 – Pressure profiles for full service brake application and release on the steel commuter train.

to reach $95\,\%$ of BC pressure in the sum of the entire train are $26\,\mathrm{s}$ for the steel commuter and $38\,\mathrm{s}$ for the iron ore train. The release times down to $15\,\%$ BC pressure are $48\,\mathrm{s}$ and $74\,\mathrm{s}$ respectively.

The monitoring of the R pressure and its implications are illustrated by the example in Figure 5.3. It shows the refilling delay that can arise in long freight trains and cause longer release times for the brakes. The figure shows how the R pressure goes below the safety level after 2 successive full applications



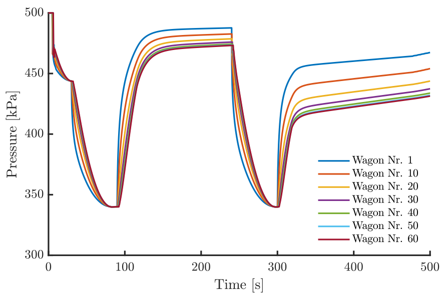
(a) MP pressure



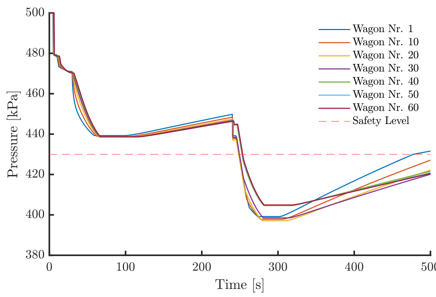
(b) BC pressure

Figure 5.2 – Pressure profiles for full service brake application and release on the iron ore train.

in short time. At that point, the refilling of R is prioritised over releasing the brakes to ensure inexhaustibility, leading to a release sequence that takes drastically longer than usual. The exact refilling rate of the R and the effects on MP pressure are currently not modelled to full satisfaction. As it affects many other parts of the system, this challenge is addressed in the next section and in Chapter 7.



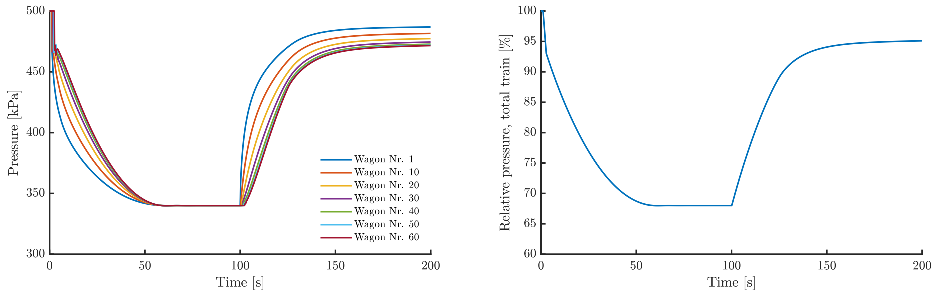
(a) MP pressure



(b) R pressure

Figure 5.3 – Pressure profiles for two consecutive service brake applications and releases on the iron ore train.

A comparison between the MP pressure profiles for different positions in the train and the average profile is shown in Figure 5.4. There are the profiles in the beginning of the train with a distinctive concave shape, which could maybe be approximated with an exponential function. In contrast to that stand the profiles towards the end of the train with their almost linear increase of pressure. These can be approximated with small deviations as two or three linear sections. The weight of the more linear shapes is increasing with the train length, as more of these latter pressure profiles are added, being almost identical to the ones for the wagons before.

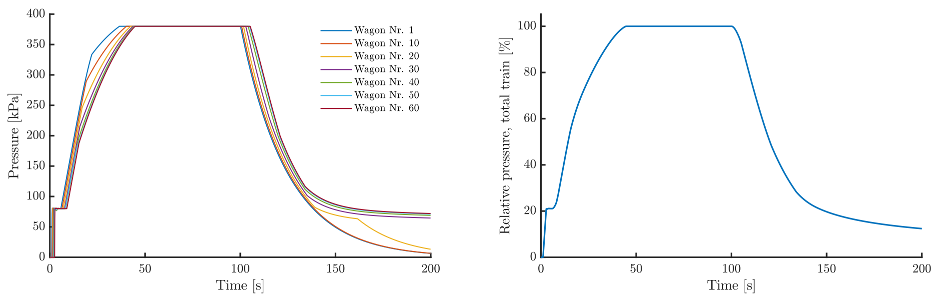


(a) MP pressure, individual

(b) MP pressure, sum

Figure 5.4 – MP pressure profiles for full service brake application and release on the iron ore train, individual and sum of all pressures in relative terms.

The same phenomenon can be observed for the BC pressure in Figure 5.5, where the pressure profiles become very close to linear towards the end of the train. This means that a linear approximation of the total pressure profile becomes increasingly appropriate with the increasing length of a train. The distinct convex shape towards the end of the BC pressure release may not be fully accurate as mentioned earlier, and is therefore not discussed any further.



(a) BC pressure, individual

(b) BC pressure, sum

Figure 5.5 – BC pressure profiles for full service brake application and release on the iron ore train, individual and sum of all pressures in relative terms.

5.2 Model Validation

This section describes the validation of the simulation results, and thereby the validation of the entire model. It is performed in accordance with the description in Section 3.4.1. In order to draw coherent conclusions, the reference data to be used must be chosen carefully, allowing systematic comparisons. For the sake of efficiency and accuracy, only time series data is used for this first analysis. For the sake of coherence, seven measurements performed by Knorr-Bremse are used, which offer a direct physical reference in contrast to the simulation results from the *DYNAFREIGHT* model. The measurements form a large base of data together with a set of system parameters that allow drawing reliable conclusions.

For the simulations, the system parameters are matched exactly to those of the test facility. The only difficulty for implementing the exact same parameters is faced with the inner diameter of the MP: It is historically standardised to be $32\,\mathrm{mm}$, which back then corresponded to an outer diameter of $42\,\mathrm{mm}$. As modern pipes are manufactured with thinner walls, they have an effective inner diameter of $36\,\mathrm{mm}$ [30]. The hose couplings, which impose a local flow restriction at each coupling of wagons, still have the standardised inner diameter of $32\,\mathrm{mm}$. The structure of system parameters only allows the assignment of one single pipe diameter per wagon. Therefore, the most restrictive value of $32\,\mathrm{mm}$ is used for the pipe diameter. Regarding the remaining system parameters of the test rig, it is equipped with DVs of the latest model *KEf*. For each "wagon" there is a $15\,\mathrm{m}$ long pipe in the test facility, and each is equipped with a single BC with a diameter of $406\,\mathrm{mm}$.

The comparisons are visualised as pressure profiles as in Figure 5.6–5.12, which make it possible to detect deviations in their shape. This is important, as even in case of perfect matching for single points in time, there might still be deviations in the shape of the profile. As an alternative to pressure profiles, timing curves indicate the time needed for each wagon in a train to reach a certain pressure level. They give a good overview on the timing accuracy along the train in Figure 5.13 and 5.14.

For the first calibration step, all parameters except one system parameter and one design parameter are locked. The remaining parameters to be adapted are the DBV nozzle diameter and the friction factor λ. Direct calibration of the friction factor λ as in [25] is preferable to the adaption of the term D in the friction Equation 2.10 as in [27]. This makes the adaption of the parameters more intuitive than the inverse effect present with D, which is important for the future use of the test bench.

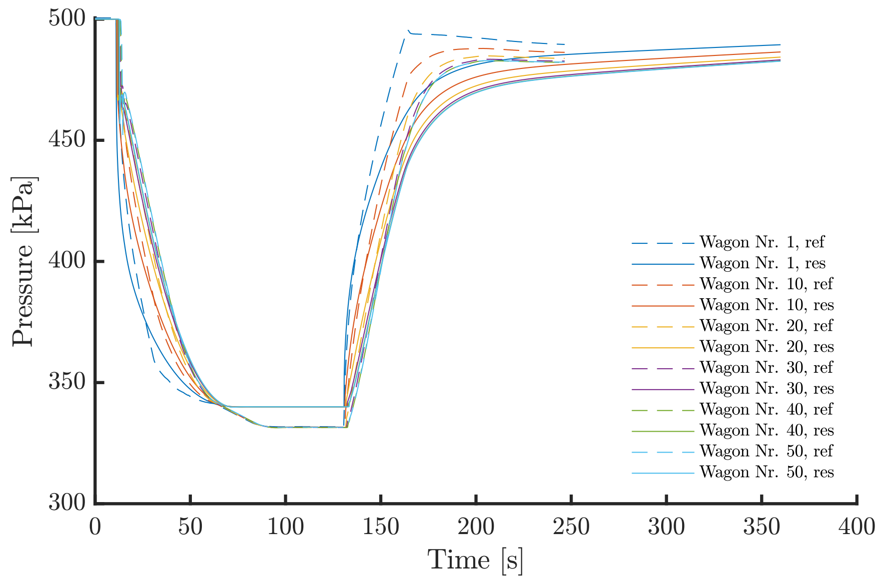


Figure 5.6 – MP pressure profiles of full service application measurement and corresponding simulation after first calibration step, test train length 750 m.

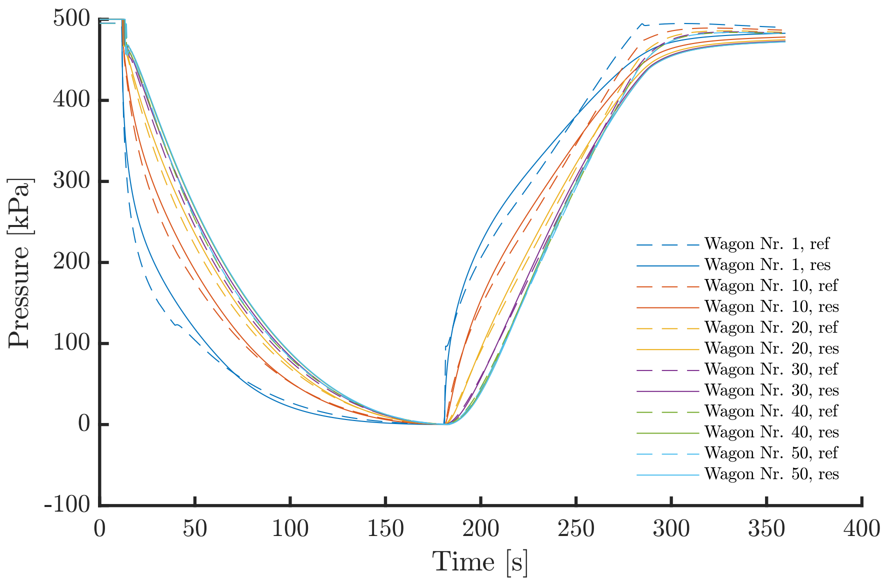


Figure 5.7 – MP pressure profiles of emergency application measurement and corresponding simulation after first calibration step, test train length 750 m.

As a simple optimisation approach, one parameter is varied at a time with rough steps, which are halved whenever the result pressure profile passes the

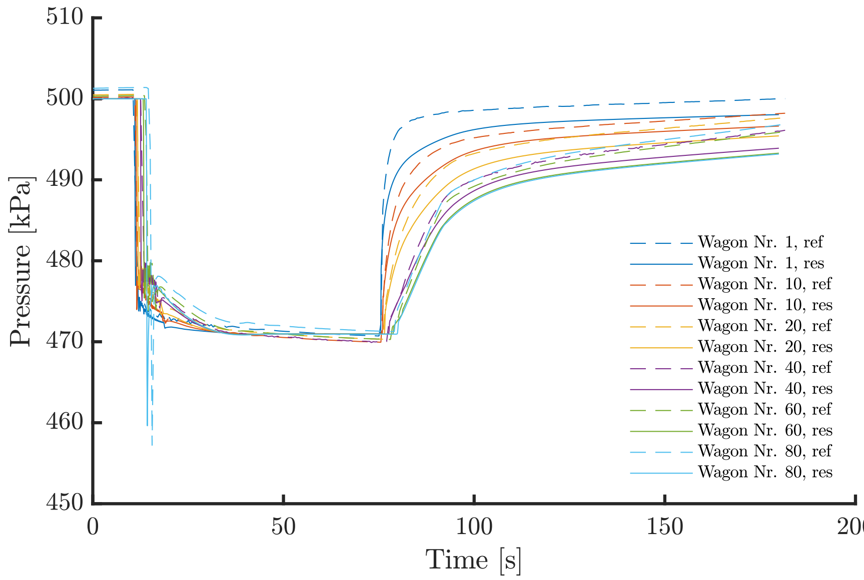


Figure 5.8 – MP pressure profiles for measurement of minimal brake application and simulation after calibration, train length $1200\,\mathrm{m}$.

reference profile. The first adaption is made to the additional friction factor λ_{cal}, based on the reference data for MP pressure at full service and emergency brake application. An optimal value for λ_{cal} within steps of 0.05 is reached at 0.45. Then, the DBV nozzle diameter is adapted in the same manner. Individual values are identified for service application as in Figure 5.6 and emergency application as in Figure 5.7 respectively. The diameter for release, meaning refilling of the MP, is adapted for both cases. Best alignment of pressure profiles within steps of $0.5\,\mathrm{mm}$ is reached with $8\,\mathrm{mm}$ for service and $10.5\,\mathrm{mm}$ for emergency application. The best match for the early stage of the release profiles is achieved with $6.5\,\mathrm{mm}$ release diameter. However, the deviation of the profiles at a later stage shows that much higher values would be required towards the end of brake release. For this reason $8\,\mathrm{mm}$ are used as standard value even for the release diameter in the general simulations. The adaption of the DBV nozzle diameters also affects profiles in general, therefore a further reduction of λ_{cal} to 0.40 is made.

Due to the direct relation of the BC pressure on the MP pressure, the former is not calibrated before accurate results for the MP pressure have been achieved. After that, the profiles in Figure 5.9 and 5.10 show good agreement with the reference data, especially for application and the early stage of release. This proves that the timing parameters of the DV are valid. What remains to

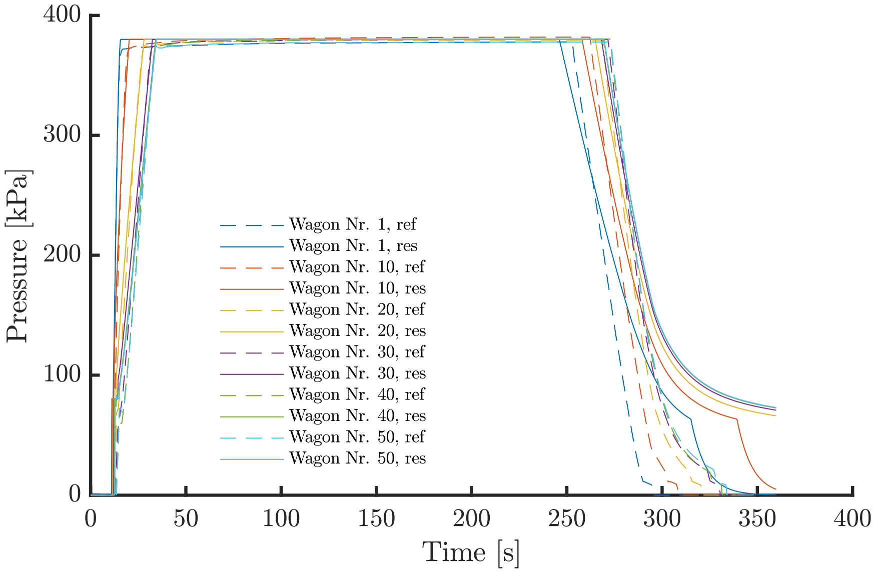


Figure 5.9 – BC pressure profiles of emergency application measurement and simulation after calibration, test train length 750 m, brake type P.

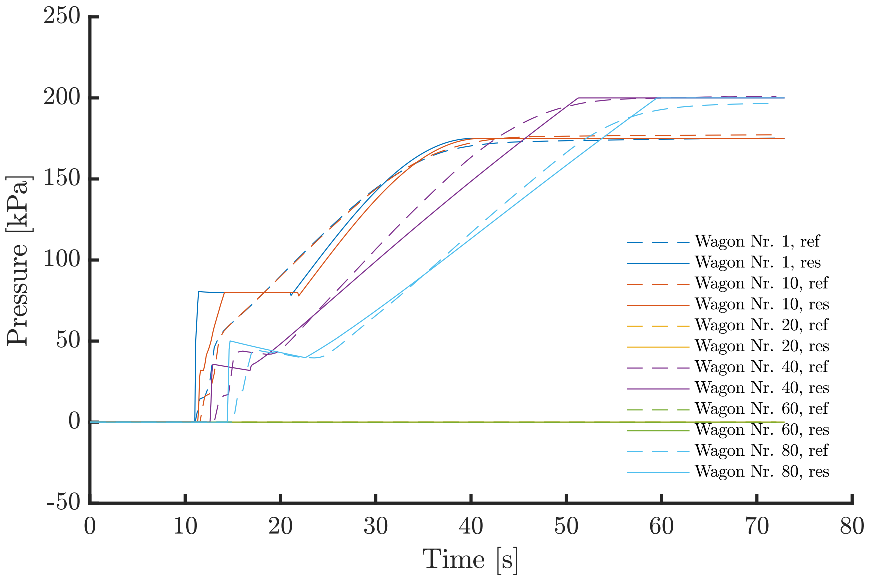


Figure 5.10 – BC pressure profiles for measurement of pressure propagation and simulation after calibration, train length 1200 m, brake type G.

improve are the empirical curve shapes of the DV model. There is room for improvements of the initial application and mainly towards the end of release, which becomes clear for example for the time interval of $300\,\mathrm{s}$ to $350\,\mathrm{s}$ in Figure 5.9. It is important to note though, that the majority of these deviations is directly caused by deviations in the MP pressure.

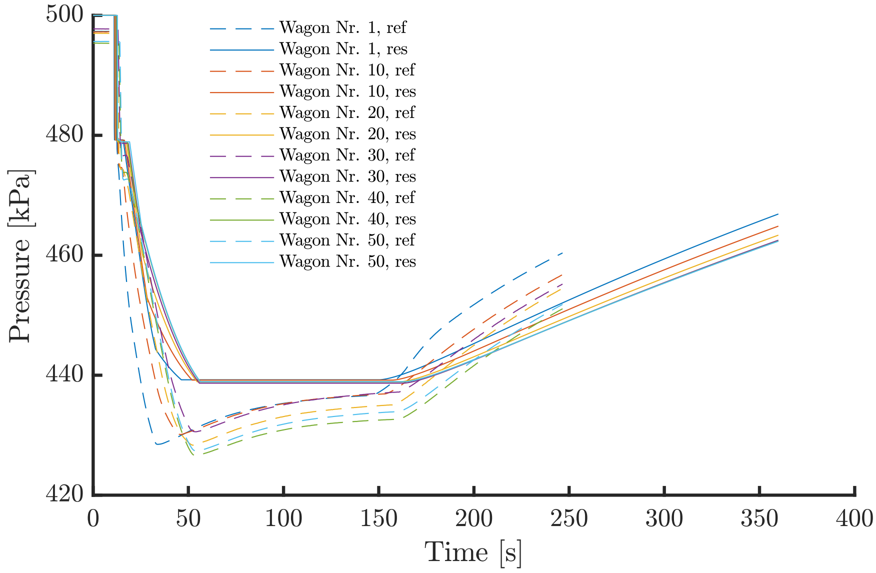


Figure 5.11 – R pressure profiles of full service application measurement and simulation after calibration, test train length $750\,\mathrm{m}$, brake type P.

For the calibration of the R pressure, the first focus is on the consumption of pressure for BC filling. There a systematic error is detected. Comparison of different BC filling levels reveals that it is no proportional, but a constant error in that is connected to initial brake application. An additional pressure reduction of $10\,\mathrm{kPa}$ is included in the model, compensating the systematic error and thereby bringing the depiction of the consumption to an accurate level. The initial difference that is clearly visible in Figure 5.11 is caused by thermal assimilation of the air in the R, whereas the simulation is isothermal.

The refilling of the R is not correctly calibrated in the model, as seen in both Figure 5.11 and 5.12. It is closely related to the air flow from the DBV in a late stage of release, where both the DBV and the R-filler nozzle diameter limit the increase of the MP pressure. An increase of the R-filler diameter causes large flows leaving the MP. Due to the empirical modelling of the DV control, these flows can cause an unstable behaviour of the control logic that is currently implemented for the DV, especially if they are not compensated with an increased flow into the MP from the DBV.

The correct implementation of this relation requires a new calibration

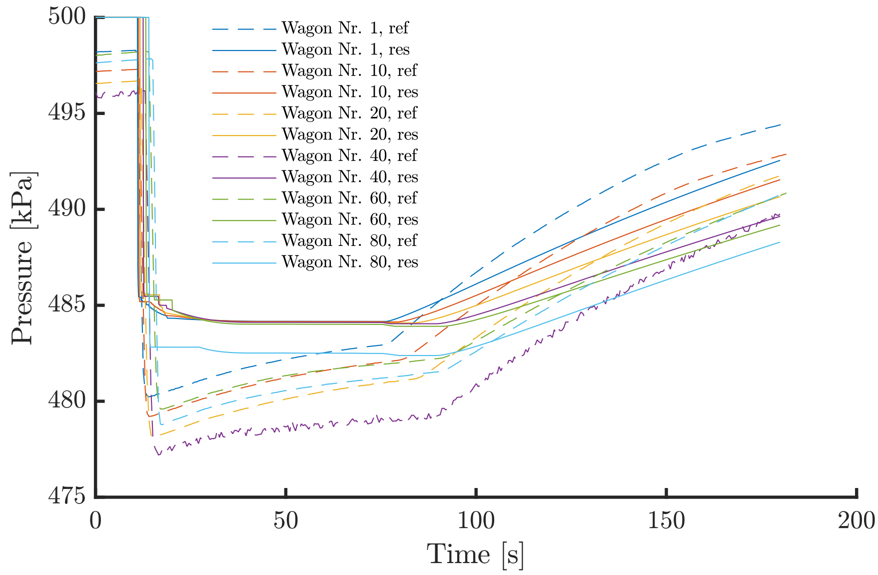


Figure 5.12 – R pressure profiles for measurement of minimal brake application, simulation after calibration, train length $1200\,\mathrm{m}$, brake type G.

step for DBV and R-filler diameter, which most likely must include another iteration of their control. As this iteration is outside the scope of this project, it is covered in Chapter 7.

Another important detail to note in this context is that these results are achieved with parameters from the latest type of Knorr-Bremse DV, *KEdv* or *KEf*, which have characteristics that support the quick restoration of the MP pressure, such as a low threshold for filler limitation and a small diameter of the limited filler nozzle. General conclusions about the refilling of R from the results presented here must be drawn very cautiously, because these parameters vary greatly for older types and other manufacturers, which is discussed in Chapter 2.

An estimation of the modelling errors is made based on the comparison of transition times. In Figure 5.13, the red line shows the times for $95\,\%$ MP pressure release, followed by the green and black lines for pressure restoration to $80\,\%$ and $95\,\%$, respectively. The dashed lines show the corresponding reference times from the measurement, and the dotted lines indicate that a transition command is given. Figure 5.14 follows the same principle, but the red lines show $95\,\%$ BC pressure application, followed by the green and black lines for BC pressure release to $20\,\%$ and $5\,\%$, respectively. As indicated by the good alignment of the red lines in both figures, the error for the brake application timing is relatively small. The time deviation is in the range of $1\,\mathrm{s}$ to $7\,\mathrm{s}$ for both pressures, which means an average error of $9\,\%$ for the MP and $11\,\%$ for the BC pressure. The best alignment is achieved for the MP pressure restoration to $80\,\%$, with deviations of $-0.4\,\mathrm{s}$ to $0.2\,\mathrm{s}$ and an average relative error of $2\,\%$. For the other timings, the deviations are much larger, with release times up to a factor 2 longer than the reference. Summary results of this and one additional comparison are available in Appendix C.

The general level of accuracy meets the goals of the project to the extent that it proves the validity of the basic approach in modelling and structural setup. Nevertheless, the numbers above also clearly show the need for additional calibration of the MP refilling towards the end of release, in order to draw quantitative conclusions from the results.

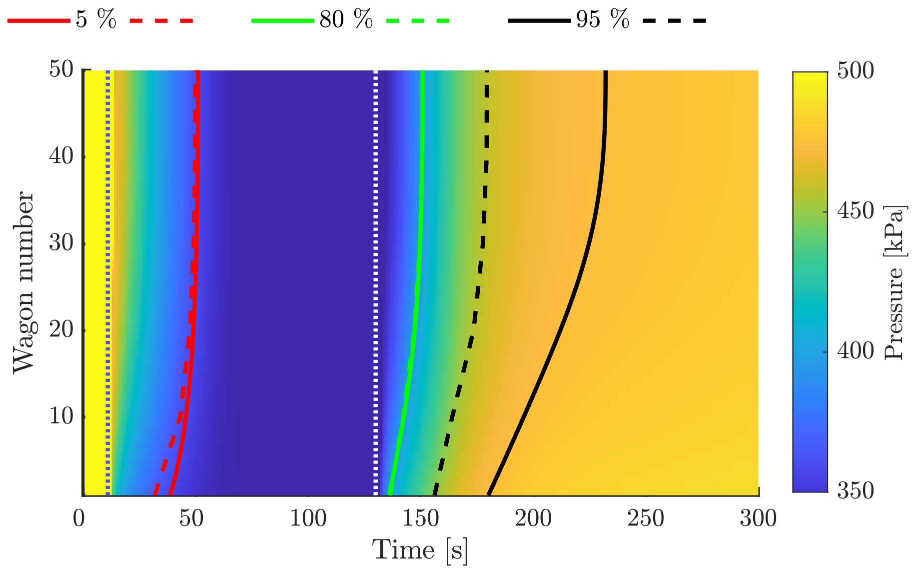


Figure 5.13 – Timing curves for MP pressure, full application and release, train length $750\,\mathrm{m}$, brake type P. Dashed lines indicate reference values.

The leakage feature which is included during the last implementation stage of the model is not used for the simulations that are shown in this chapter. One reason for this is the lack of sufficient data for its calibration. Another reason is that there currently are no known errors in the model behaviour that could be reduced by the introduction of leakage. To the contrary, it would further delay the refilling of the MP and therefore require additional calibration of other parameters to reach accurate results. If there is an interest in the specific effect of leakage, this can be analysed in a separate study using this test bench.

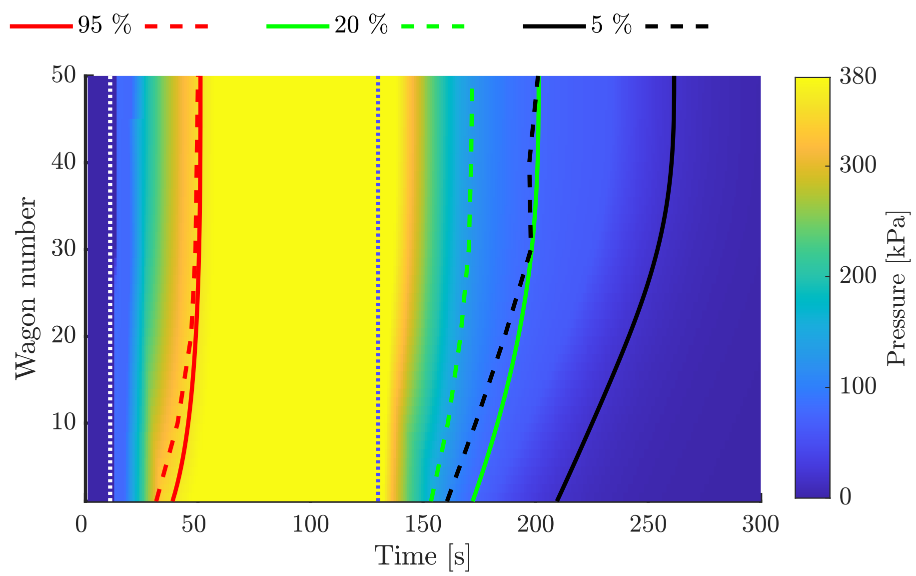


Figure 5.14 – Timing curves for BC pressure, full application and release, train length $750\,\mathrm{m}$, brake type P. Dashed lines indicate reference values.

Chapter 6

Discussion

In this chapter, a critical perspective on the project is taken with respect to the methods and outcomes. This includes choices that were made during the working process and seeks to assess if different decisions could have led to a better fulfilment of the project goals. The second section takes a somewhat more methodological view on the goals of modelling and their application to the given problem.

6.1 General Discussion

During the working process, the focus of this project gradually shifted from simulation theory to system studies. It became increasingly apparent that the complex system background and functional logic, as well as the structure of the simulation are at least as relevant to the project. As it was necessary to deliver results in a given time frame, the balance between the areas shifted, with the former topics gaining more and more importance. The expectation of the author to become an expert of fluid simulation with some necessary knowledge on train brakes, turned out becoming a brake system expert with a consolidated basic knowledge within the simulation of fluid flow.

In this context stands the use of a modelling method that is based on making notable approximations, which makes it simpler to implement than an FVM model. This nonetheless means that the model goes to its limits with the assumptions of electrical equivalence. It is proven that the model fulfils its purpose by means of calibration. The user must still be aware of the limitations of the equivalent circuit model, especially if the further needs of Transrail require even more advanced analyses that this model is not capable of achieving. In that case, it would be required to replace the model used for the main pipe module with one based on the FVM approach.

The modelling process of this work is based on a thorough study of the technical background of the brake system. This initial study comprises to a large extent specialised literature and standards, but not as much practical experience or measurement data. If the latter would have been available in an earlier stage, empirical models could have been created with a higher accuracy from the start. For this project, the contacts first had to be established and the theoretical knowledge was prioritised in order to build a base for discussion. The final outcome proves this to be a valid approach as well.

The delivered model is shown to be highly versatile, especially regarding train configuration. It is not following the strictest theoretical definition of modularity on a modelling level though, as its logic and equations are located in the solver. To achieve full modularity on a modelling level, an object-oriented implementation of the solver would be required, which moves storage of both logic and equations into the brake component objects. This approach is not reasonable to fit into the project scope, and is therefore referred to in Section 7.2.

In the current model, nozzles are assigned an equivalent diameter which is calibrated in order to produce correct results. For those cases where values for the physical diameter become available, it might lead to confusion to use a different diameter in the model. It would therefore be better to use the physical diameter and calibrate the flow coefficient instead.

6.2 Precision vs. Universality

Throughout the development process of this project, the drive has been towards capturing more and more details and covering special features. This is, as one would assume, the general aim of all modelling and simulation. In case of the air brake system, this means attempting to depict the exact behaviour of individual components as precisely as possible. For the train brake system, this includes most prominently the DV. The behaviour of the DV depends on a lot of external factors though. Key parameters such as timing, leakage and reaction patterns to pressure inputs, are affected by the variant, manufacturer, wear and ageing of the DV. Every variant also has a range of tolerance for its behaviour, as discussed in Section 2.1.1. Some of these factors could be defined in parameters for a simulation case, but that would limit the domain of validity of the results to that specific scenario.

This is in stark contrast to the demand towards the test bench and its expected use cases. The exact configuration of individual vehicles is often not known to the user of this test bench, who is interested in the general behaviour of a certain train type. For those few cases where a well-defined fleet of unit trains is performing specialised transports, such as the iron ore trains in northern Sweden, the creation of a detailed model of the specific components could be attempted. A great number of external factors will still be unknown. In the more general case though, even train operating companies or train drivers do not have detailed information about each car in a train, as also mentioned in Section 2.1.1.

A single fact known for sure is the compliance with the local set of standards, such as UIC and EN standardisation. This uncertainty makes it necessary to find a reasonable level of generalisation, which nonetheless allows for an accurate simulation. As shown in Chapter 5, this can be done for the test bench by using average values of the intervals set by UIC standards, which are refined with measurement data. Nevertheless, the results presented in Section 5.2 prove that further improvements by calibration would be appropriate. This should ideally be done with reference data for different types of components or based on additional practical knowledge from train operation.

Such calibrations will also handle the issue that the current model and default parameters most likely are closest to components by Knorr-Bremse, more specifically the latest DV variants *KEdv* and *KEf*. Development has to a large extent been based on their literature and functional information, in addition German and Swedish literature, where equipment delivered by Knorr-Bremse is predominant. The reference measurements were made with these DV variants as well.

With that said, it is also important to mention that there is a source of considerable uncertainty for braking behaviour beyond the pneumatic system, namely in the mechanical transmission and application of the frictional braking force. This part of the braking system is also affected by a lot of external factors such as climate and maintenance, and the uncertainty was lifted to an entirely new level with the latest introduction of new brake block materials [31]. As these materials behave differently and are mounted on a fraction of the vehicle fleet, more and more parameters would be needed, if all eventualities were to be covered. But this opens up an entirely new topic that needs to be addressed elsewhere. As a final point, this still raises the question of what level of precision the pneumatic part of the system does need to achieve.

There is no question about the importance of making accurate predictions in order to gain a correct understanding of the system. After all it is vital to have the right expectations about the general system behaviour when operational strategies are based on them. It is the level precision though that should bear proportion to the precision that can be achieved in the other areas of braking calculations. In others words, it should be kept in mind that the overall error of a system model is always as high as the highest subsystem error, even if the others are small.

Chapter 7

Conclusions and Future Work

This final chapter wraps up the elaboration of this work, starting with a recap of the purpose and goals of the project. After the discussion of the questions posed there, remaining conclusions drawn from this work are summarised. The limitations faced during the project are addressed at the end of the first section, together with those present in the final outcome. Based on that, possible areas for further work and the recommendable topics to be tackled are outlined in the second section. Finally, some reflections are made regarding the results and the contributions of this work in a wider point of view.

7.1 Conclusions

The overview on the technical background and the behaviour of the crucial brake system components in Chapter 2 shows how complex the modelled system is. One important aspect of this is the variety of solutions on the market fitting local standardisation, brought about by decades of refinement. It also describes the number of special features added during this refinement, making a long list of possible additions the model. Finally, this chapter shows the wide set of equally valid options available, and how the choice between these depends on the scope of the application.

The methodical perspective on modelling techniques taken in Chapter 3 presents a structure of iterative and incremental progress. It also discusses the handling of interdependence in complex models and the possible difficulties caused by empirical modelling, even though it simplifies calculations. This chapter also shows the first step in establishing a structure for handling train configurations with a simple interface.

An example of implementation is given in Chapter 4, fulfilling full flexibility for the simulation of UIC brake systems. Most importantly this chapter shows how the implementation of a digital test bench for the air brake system of freight trains is accomplished.

Chapter 5 then proves that the test bench accurately depicts most aspects of the brake system behaviour in operation. The presented results already allow a first study of system behaviour, for example how its reaction to control inputs is situation dependent. The results also show some functional limitations of the brake system, which are important to consider for optimisation.

The final result of this work is a tool that is a stepping stone in the hierarchy of train brake simulation. When comparing tools with respect to their levels of accuracy and complexity, the one delivered by this project is a clear improvement compared to the simple and very general models in common use. Still, it is not as mature as the most advanced applications currently adapted by authorities or leading manufacturers.

The outputs that this test bench creates are suitable for a direct use as a rough development guideline. Further calibration using the available reference data will increase the accuracy of the outputs to a level at which they can be used for validation of other brake models. The output format with individual wagon forces allows the continued use of the results for simulation models of longitudinal train dynamics.

7.1.1 Limitations

The domain of validity delineates the limitations of the test bench on a general level. Its limits run along the lines that are sketched in Chapter 6 and in the first part of this section. The model structure is qualitatively valid for all simulations of UIC brake systems. Quantitative validity is dependent on the adaption of parameters.

The factors limiting the possible outcome of this work are connected to the conflict between the desired generality of the model and the vast diversity of component variants in operation. As only compliance with vague standards can be assumed certain for system behaviour, there remain some uncertainties that limit the expected accuracy for general applications of the test bench.

In that context the available reference data was a limitation. Data for other component types or manufacturers within the UIC would have given a better understanding of the possible differences. Nevertheless, it is questionable if there would have been sufficient time to analyse the data, a limitation which also is reflected by the long list of future work. The topic of pneumatic train brake simulation easily provides material for several projects of this volume.

7.2 Future Work

As mentioned throughout this work, the scope of the project only allows to tackle a limited number of aspects to the problem. Even for those aspects within the scope, only a limited number of iterations can be made, so that there still are many improvements and variants left to be addressed in future work. The most urgent or promising ones are listed and outlined in this section.

7.2.1 Detailed Calibration

The most important aspect to improve is that the nozzle diameter of the R-filler should be calibrated together with the refilling diameter in the DBV. The refilling flow from the DBV is too small in the late stage of refilling, but a general increase of diameter may impair accuracy in the early stage of refilling. Instead, it may be necessary to also include modelling improvements and an improved control logic to the model, as mentioned further down.

In order to increase the accuracy of the other results compared to the level shown in Section 5.2, a more systematic calibration against the measurement data would be required. To do this, a two-dimensional optimisation algorithm should be used varying the DBV nozzle diameter d_{DBV} and the friction factor λ for minimising the average pressure error. The results from Section 5.2 indicate that d_{DBV} should be bigger and that λ should be smaller than the current default values. For obtaining a more universal model and broadening the domain of validity, more reference data is required. As a first step in this direction, graphical samples of the validation data from other projects can be used for gaining a general understanding of the deviations.

An alternative approach, which requires even more data and effort, would be to create context- or operator-specific sets of parameters. The prominent sources of deviations in behaviour are the shape of pressure transfer profile dictated by the DV, the pilot pressure profiles and flow capacity through the DBV, and the parameters for R-filling.

7.2.2 Modelling Improvements

This section lists further possible module iterations according to Table 4.1.

MP module

If future improvements aim at lifting the accuracy level of the test bench to the most advanced level of modelling, then the MP module should be implemented as a non-isothermal FVM model. An intermediate step in implementation would be the isothermal variant. A FVM model may produce more universal and reliable results, if it is brought to an advanced implementation level and calibrated properly.

DV module

For the modelling of the DV, the final missing addition to achieve the full implementation of the logical-physical layout presented in [19] is to model the pressure transfer as equivalent nozzle flow which constitutes the precontrol chamber pressure for a relay valve. This precontrol chamber pressure is then directly used as BC pressure, which would be the explicit modelling approach for the relay valve feature described as Universal Action in Section 2.1.1.

This model variant would simplify the code needed to cover the logic in the DV significantly, even though it requires its own state-machine and another implementation of a higher-order integrator. All exceptions and requirements added to the timer of the limiting function to make it behave correctly could be replaced with the physical flow over a pressure differential. This contradiction is an apt example of how modelling a complex system empirically can require a noticeable effort to achieve a stable system, as discussed in Section 3.2.1. This might in some case even out the additional effort needed for a simplified physical model. Only the implementation of both variants and direct comparison can give an answer regarding the accuracy of both variants.

BC module

One improvement option for the BC module includes a variable volume considering the pressure dependent piston movement. Another one is the modelling of flow from the R according to the precontrol pressure set by the DV.

DBV module

In addition to the state-dependent nozzle diameters that are introduced in the final implementation stage of the project, a linear function for control pressure variations can be implemented. Replacing the direct signal input currently used with the variant described in [25], would likely lead to a more accurate depiction of the pressure profile at the DBV.

In order to increase the model's accuracy for the refilling of the MP, the flow from the DBV must be increased specifically in the late stage. This might even include techniques that are used in operation, such as high pressure refilling or an overcharge of the MP to accelerate refilling towards the end of the train.

R module

As already mentioned for the BC model, a pressure-dependent flow interface between R and BC would model their pressures more accurately. The modelling of the variable nozzle diameter that limits the refilling flow from the MP can according to [25] also be refined. For this, a pressure-dependent function is adapted.

Mechanical force generation

In order to obtain reliable results even beyond the scope of the pneumatic system, the very approximate proportional assignment of BC pressures to braking forces should be replaced with a model considering nonlinearities as well as the individual characteristics of force transmission from piston to rail for each wagon.

7.2.3 Implementation Improvements

For the functionality of the test bench, not only perfect modelling is requested. The adequate handling of data and computational efficiency are also important characteristics for making the test bench convenient to use. Some improvements can be made in this field as well.

More robust control logic

The control logic that is currently implemented for the empirical model parts is very efficient while the state variables have an asymptotic behaviour, but not as much in quasi-stable state. This can lead to oscillations which require high computational effort as the system switches states and timers are reset. These oscillation can be reduced by implementing a robust state machine that tracks the overall system state and synchronises the modules.

Implementation of variable time step length

The increase of computational efficiency is the most urgent issue to address in this field. It can be achieved by using a variable time step length. In general, this means that the time step length is increased when there are little changes to the system and vice versa. The first and simplest option is to detect stable time sections during the simulation and to skip calculations until a change of input occurs. This can, dependent on the input signal, directly reduce computational effort by half or more. As this also makes calibration less time-consuming, it is recommendable to make this adaption in an early stage of continued work on the test bench.

Higher order integration

In general, the implemented RK4 integrator can be assumed to be of sufficient order. Using an explicit method is fine, as there are no stability issues. The lower-order Euler forward integrator works where it is implemented, but the accuracy of the outputs would be increased by using the RK4 for all instances of time integration in the code.

Fully object-oriented solver

If full modularity on the solver side is desired, it can be implemented by an object-oriented programming implementation of the model. All information is then saved in the vehicle library, by linking model objects to each vehicle object, which contain the logic and equations that the calculations of the output depend on. The single remaining content of the solver itself is then the structure of calling object contents to produce the outputs.

7.2.4 Model Variant for the AAC System

Based on the summary of system characteristics and the adaptable structure of the test bench, a model of the AAR brake system can be implemented. In addition to the adaption of system parameters, this must include a new variant of the DV module with its pressure transfer logic. More specifically, changes must include the direct brake release and the pressure translation following the pressure equalisation principle. The logic for the state machines controlling auxiliary features should be adapted as well. This includes the nozzle for filling the R, and the AD which in the AAR system reacts to any pressure reduction.

The use of direct release brakes in the AAR system makes DASs very relevant, even for safety purposes. Giving the driver a tool at hand that supports long-term planning of brake applications will increase brake availability in possibly difficult or even dangerous situations.

7.3 Final Reflections

The test bench that is delivered through this project add to the knowledge and understanding of pneumatic brake behaviour at Transrail. As discussed in Section 1.3, this work presents ways to make this knowledge easily available to a broader audience and to improve it further in the future. If this work can increase the number of trains that are driven in an optimal way, either by promoting the advancement DASs or by its use for driver training, it makes a long-term contribution to higher efficiency in the operation of rail freight transportation. This will ideally increase the market share of rail freight, which plays a decisive role for the environmental impact of the transport sector.

Thereby, this thesis directly contributes to the United Nations (UN) Sustainable Development Goals (SDGs) number 9 for industry, innovation and infrastructure as well as number 13 for climate action.

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Appendix A

Typical Brake System Data and Parameters

Feature Description	Value			Unit	
	EU UIC 540 mode G	EU UIC 540 mode R	US AAR AB- / DB60		US ref value
System Characteristics					
Release time from full application	70	25	>0	s	
Main Line					
Regular signal transmission speed	150-250		107-183	m/s	350-600 fps
Emergency braking transmission speed	250		290	m/s	950 fps
Main line working pressure - p_ref	400 - (500) - 600		450 - (620) - 760	kPa	65-110 psi
Main line inner diameter	32		32	mm	1 1/4"
Distributor / Control Valve					
Main line pressure reduction for full brake application	150		2/7*p_ref	kPa	
Brake control sensitivity (main line pressure)	60 in 6s		5	kPa	1-1.5 psi
Supply / Auxiliary Reservoir					
Auxiliary Reservoir Volume	40 - (150) - 300		41	dm^3	2500 in^3
Emergency Reservoir Volume	-		57	dm^3	3500 in^3
Brake Cylinder					
Max brake cylinder pressure - service	380		5/7*p_ref	kPa	
Max brake cylinder pressure - emergency	380		6/7*p_ref	kPa	
Filling time to 95 % pressure	18-30	3-5	-60	s	
Pressure release time to 0.4 bar	45-60	15-20	>0	s	
Brake Cylinder Applied Volume	5 - (10) - 20		20	dm^3	1000 in^3
Brake Cylinder Piston Stroke Length	140		203	mm	8"
Brake Cylinder Diameter	150 - (300) - 406		305	mm	12"

Appendix B

Train Configuration Spreadsheets

Vehicle Library: Locomotives sheet

Variable	Unit	Description	Re/Traxx	IORE	E402B	Dummy
l_l	[m]	vehicle length	19	46	9.7	1
D_l	[m]	main brake pipe diameter	3.20E-02	3.20E-02	3.20E-02	3.20E-02
m_l	[kg]	vehicle static mass	8.40E+04	3.60E+05	7.30E+04	0.00E+00
F_B_max_l	[N]	maximum braking force	1.05E+06	4.50E+06	9.13E+05	0.00E+00

Vehicle Library: Wagons sheet

Variable	Unit	Description	FAMMRR / SMMNPS		SGDMS	KnorrRig
l	[m]	vehicle length	10.3	13.9	18.3	15
D	[m]	main brake pipe diameter	3.20E-02	3.20E-02	3.20E-02	3.20E-02
m_we	[kg]	vehicle empty static mass	2.00E+04	2.12E+04	2.05E+04	0.00E+00
F_B_max	[N]	maximum braking force	1.20E+05	1.50E+05	1.40E+05	4.50E+04
V.BrC	[m3]	total brake cylinder volume	4.00E-02	2.00E-02	2.00E-02	2.00E-02
V.res	[m3]	total supply reservoir volume	3.00E-01	1.50E-01	1.50E-01	1.50E-01

Vehicle Library: Driver brake valves sheet

Variable	Unit	Description	UIC_G_cal	UIC_P_cal
d.DBV_service	[m]	equivalent nozzle diameter for service brake applicat	8.00E-03	8.00E-03
d.DBV_emerger	[m]	equivalent nozzle diameter for emergency brake app	1.05E-02	1.05E-02
d.DBV_fill	[m]	equivalent nozzle diameter for brake release (MBP fil	6.50E-03	6.50E-03
dPmin	[Pa]	minimal control pressure drop upon brake applicatio	4.00E+04	4.00E+04
L_t_app	[s]	application time until 95% pressure in BC	24.00	4.00
L_t_rel	[s]	release time until BC has dropped to 0.4 bar	55.00	18.00
dPext	[Pa]	additional pressure drop in full service brake commaı	1.00E+04	1.00E+04

Vehicle Library: Distributor valves sheet

Variable	Unit	Description	KE_G	KE_P	KE_Gempty	KE_Gbemp
PC_max	[Pa]	maximum BC pressure	3.80E+05	3.80E+05	2.00E+05	1.75E+05
PC_rf	[Pa]	MBP pressure reduction for full bral	1.50E+05	1.50E+05	1.50E+05	1.50E+05
t_app	[s]	application time until 95% pressure	24.00	4.00	24.00	24.00
t_rel	[s]	release time until BC has dropped to	55.00	18.00	55.00	55.00
IAP	[Pa]	initial application pressure level, un	8.00E+04	8.00E+04	8.00E+04	8.00E+04
d.acc	[m]	equivalent nozzle diameter to accel	5.00E-03	5.00E-03	5.00E-03	5.00E-03
V.acc	[m3]	equivalent accelerating chamber vo	0.00E+00	0.00E+00	0.00E+00	0.00E+00
DPA	[Pa]	pressure drop per accelerating chan	3.00E+04	3.00E+04	3.00E+04	3.00E+04
trig.acc	[Pa/s]	pressure derivative to open accelera	1.00E+04	1.00E+04	1.00E+04	1.00E+04
d.vent	[m]	equivalent nozzle diameter to venti	0.00E+00	0.00E+00	0.00E+00	0.00E+00
trig.vent	[Pa/s]	pressure derivative to open venting	0.00E+00	0.00E+00	0.00E+00	0.00E+00
lim.vent	[Pa]	lower limit at which venting valve cl	0.00E+00	0.00E+00	0.00E+00	0.00E+00
d.leak	[m]	equivalent nozzle diameter correspo	1.00E-04	1.00E-04	1.00E-04	1.00E-04
d.fill	[m]	equivalent nozzle diameter for rese	5.00E-03	5.00E-03	5.00E-03	5.00E-03
d.fill_lim	[m]	equivalent nozzle diameter for limit	8.00E-04	8.00E-04	8.00E-04	8.00E-04
lim.fill	[Pa]	lower reservoir pressure limit above	4.30E+05	4.30E+05	4.30E+05	4.30E+05
trig.app	[Pa/s]	pressure derivative to activate distril	1.00E+04	1.00E+04	1.00E+04	1.00E+04
lim.rel	[Pa]	minimal control signal level (MBP pı	2.50E+04	2.50E+04	2.50E+04	2.50E+04
V.addp	[Pa]	additional R-pressure drop upon init	1.00E+04	1.00E+04	1.00E+04	1.00E+04

Brake Control Cases

time [s]	**brakingPos []**
0	0
1	2
30	4
60	6
90	8
120	6
150	4
180	2
210	0
250	0

Train Configurations

Variable Name	Unit	Description											
		Locomotive Type	5	0	0	0	0	0	0	0	0	0	0
		Car Type	0	7	7	7	7	7	7	7	7	7	7
CVType		Valve Type	6	5	0	5	0	5	0	5	0	4	0
		Number of Vehicles	1	1	2	2	2	4	2	1	1	3	4
m_l	[kg]	vehicle loaded static mass (optional override)	0	0	0	0	0	0	0	0	0	0	0
F_B_max	[N]	maximum braking force (optional override)	0	0	0	0	0	0	0	0	0	0	0
t_delay	[s]	time delay for control signal implementation (e.g. due to signal transmissio	0	0	0	0	0	0	0	0	0	0	0

Additional Parameters Setup			
T0	[K]	ambient temperature for simulation (293.15)	293.15
P0	[Pa]	nominal brake system pressure (5E5)	5.00E+05
b0_leak	[-]	constant leakage coefficient (8.32875)	0.000
bT_leak	[-]	linear leakage coefficient (-0.025 for increased leakage at lower temperatur	0.000
h	[s]	simulation step length for integration (1E-2)	1.00E-02
t	[s]	total simulation time	73.00
CVlimType	[-]	flag, controlling which type of limitation curve to use [1-linear, 2-exponenti	3
l_cal	[-]	MBP length calibration factor (1.08)	1.08
Rcal	[-]	MBP friction factor calibration (0.4)	0.4
l_coupl	[m]	MBP connector length addition per wagon connection (0.3)	0.3

Brake Control Signal filename		Control Signal Properties	
filename	description	values	
Brakecontrol_Cases.xlsx	Brake Control Signal format [1-step (ı		1
	Sheet Number		7
	Full Service Level		8

DESCRIPTION

This is the test case for simulating comparison data for the reference data provided by Knorr Bremse. (test 6.3d).

BRIEF INSTRUCTIONS FOR USE

One vehicle definition per column, either Locomotive or Wagon, otherwise error.

Numbers correlate columns of the corresponding sheet in Vehicle_Library.xlsx, numbering starts from the frist data entry.

Important to **always save all used .xlsx-files** after making changes and **before running the MATLAB code**. It does not consider changes made in open documents.

Rows marked grey are placeholders and not used in the test bench so far.

To create a new simulation case, copy the latest case slide, adapt it and mark its number for calling it from MATLAB.

The simulation is started by running Psim_init.m and entering the sheet number.

Appendix C

Timing Comparison

Table C.1 – Summary of time difference and relative error after the first calibration steps. Test case full service application type P and G.

Brake Type	MP				BC			
	Press. Level	Time diff. [s]	Rel. diff. [%]	Avg. error [%]	Press. Level	Time diff. [s]	Rel. diff. [%]	Avg. error [%]
P	5 %	1 – 7	3 – 30	9	95 %	1 – 7	3 – 35	11
	80 %	-0.4 – 0.2	-6 – 2	2	20 %	18 – 30	69 – 78	72
	95 %	24 – 53	91 – 107	97	5 %	49 – 63	83 – 159	106
G	5 %	1 – 7	4 – 35	10	95 %	-1 – 3	-3 – 10	3
	80 %	-0.7 – 0.3	-10 – 2	3	20 %	-5 – 30	-11 – 55	33
	95 %	26 – +9	97 – 173	143	5 %	14 – 96	20 – 130	86

For DIVA

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