12120 - Department of Automotive, Combustion Engine and Railway Engineering

Diploma thesis

Consequences of reducing truck losses for vehicle braking and deceleration systems

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ZADÁNÍ DIPLOMOVÉ PRÁCE

I. OSOBNÍ A STUDIJNÍ ÚDAJE

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II. ÚDAJE K DIPLOMOVÉ PRÁCI

Název diplomové práce:

Důsledky snižování odporů nákladních automobilů pro brzdové a zpomalovací systémy vozidel

Název diplomové práce anglicky:

Consequences of reducing truck losses for vehicle braking and deceleration systems

Pokyny pro vypracování:

1) Proveďte rešerši brzdových a zpomalovacích systémů nákladních vozidel včetně rozboru legislativy a připravované legislativy s ohledem na důsledky pro brzdové a zpomalovací systémy budoucích nákladních vozidel. Proveďte zjednodušené výpočty brzdného potenciálu různých kombinací brzdových a zpomalovacích systémů.
2) Proveďte citlivostní studie potenciálu motorové brzdy na poskytnutém termodynamickém modelu motoru. Vytvořte pokročilý model jízdy nákladního vozidla s motorovou brzdou, případně dalším zpomalovacím systémem, na uživatelem definované trati.
3) Model rozšiřte o systém řízení - spolupráce hlavního brzdového systému, převodovky a zpomalovacího systému. Analyzujte výsledky.
Školitel specialista (GT-Suite): Ing. Ondřej Bolehovský

Seznam doporučené literatury:

1) Firemní literatura Eaton European Innovation Center
2) Firemní literatura výrobců nákladních automobilů
3) SAE Digital Library dostupné z počítačů v učebně 717 na https://saemobilus.sae.org/search/
4) DOBROVOLNÝ, L.: BRZDNÉ SYSTÉMY NÁKLADNÍCH VOZIDEL. Bakalářská práce dostupná online. https://www.vutbr.cz/www_base/zav_prace_soubor_verejne.php?file_id=53551

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Datum zadání diplomové práce: **18.04.2019** Termín odevzdání diplomové práce: **10.07.2019**

Platnost zadání diplomové práce: ______________

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III. PŘEVZETÍ ZADÁNÍ

Diplomant bere na vědomí, že je povinen vypracovat diplomovou práci samostatně, bez cizí pomoci, s výjimkou poskytnutých konzultací. Seznam použité literatury, jiných pramenů a jmen konzultantů je třeba uvést v diplomové práci.

Datum převzetí zadání Podpis studenta

ABSTRACT

This diploma thesis deals with consequences of reducing truck losses for vehicle braking and deceleration systems. In theoretical part is research about driving resistances and legislation of trucks and its direction to future. Theory about truck braking problematics and about service and auxiliary brake systems serves to better understanding problematics of this thesis. Practical part discusses study about impact of lower driving resistances in GT-SUITE truck model to auxiliary braking systems for dynamic drive cycles, for legislation test and for stopping distance. In static cases is shown comparison of different types of engine brakes for current and for theoretical future trucks on constant downhill grade. Study and sensitivity analysis of GT-POWER Two-Stroke engine brake model shows impact of lower engine to engine brake power and potential to use and improve in the future.

KEYWORDS

Auxiliary braking systems, Trucks, Brake, Service brake, Engine brake, Two-stroke engine brake, Compression release engine brake, Driving resistances, Lowering driving resistances, Aerodynamics of truck, Rolling resistance, Engine downsizing, Comparison, GT-SUITE, GT-POWER, Drivetrain retarder.

ABSTRAKT

Tato diplomová práce se zabývá důsledky snižování jízdních odporů nákladních vozidel pro brzdové a odlehčovací systémy vozidel. V teoretické části jsou zkoumány jízdní odpory a legislativa nákladních automobilů a jejich směr do budoucnosti. Teorie o problematice brzdění nákladních vozidel a o servisních a pomocných brzdových systémech slouží k lepšímu pochopení problematiky této práce. Praktická část pojednává o vlivu nižších jízdních odporů v GT-SUITE modelu kamionu na pomocné brzdové systémy pro dynamické jízdní cykly, legislativní zkoušku a brzdnou dráhu. Ve statických výpočtech je ukázáno porovnání různých typů motorových brzd pro současný a pro teoretický budoucí kamion na konstantním klesání. Studie a analýza citlivosti GT-POWER modelu Dvoudobé motorové brzdy ukazuje vliv menšího motoru na výkon motorové brzdy a potenciál pro použití a zlepšení v budoucnosti.

KLÍČOVÁ SLOVA

Pomocné brzdové systémy, Nákladní automobily, Brzda, Provozní brzda, Motorová brzda, Dvoudobá motorová brzda, Motorová brzda uvolňující kompresi, Jízdní odpory, Snížování jízdních odporů, Aerodynamika, Odpor valení, Snížování zdvihového objemu motoru, Porovnání, GT-SUITE, GT- POWER, Retardér hnacího ústrojí.

DECLARATION

I declare that I have elaborated this diploma thesis independently under the professional supervision of Ing. Miloslav Emrich, Ph.D, and under the professional supervision of Ing. Luboš Tomiška. I further declare that I have used only the documents listed in the attached list.

In Prague 10.7.2019 ....

Signature

Table of Contents

List of symbols and abbreviations used

Name	*Symbol*	*Unit*
Compression release engine brake	CR	[-]
Jacobs High Power Density brake	HPD	[-]
Compression release with exhaust brake	CR+EX	[-]
Braking gas recirculation	BGR	[-]
Variable geometry turbine	VGT	[-]
Coefficient of friction	CoF	[-]

Introducion

One of the main targets in heavy duty vehicles design nowadays is to have the best efficiency of vehicle as possible. However, in the freight transport, the truck is considered to be a thing mainly to transport goods at the lowest costs as possible even at the expense of higher emissions. Whole fleet with high fuel consumption means lower profit to owner of the transporting company or more expensive cost of transporting goods. But innovation of design for better efficiency and lower emissions is not for free neither. So the compromise between the technologies and costs needs to be found. Current trend of lowering vehicle emissions to have a cleaner environment and near future legislations would decide this compromise and push the vehicle designers to radical improvement of vehicle efficiency or more “cleaner” engines. Lower driving resistances could be reached by decreasing of air resistance, rolling resistance and losses in whole powertrain on braking systems in trucks. Lowering the driving resistances means more powerful and sophisticated brake systems. Therefore this diploma thesis deals with consequences of reducing this losses for vehicle braking and deceleration systems.

At higher speed, fully loaded truck is a huge amount of kinetic energy. In case of braking, this huge amount of kinetic energy must be transformed into heat the shortest time as possible using the vehicle's braking systems. For this purpose, service brakes are primarily required by legislation. For higher braking performance, especially for safety, trucks are equipped with auxiliary braking systems. Nowadays, the friction service brakes reach their maximum performance limits. Auxiliary braking systems are most commonly used on long mountain passes where frequent use of frictional service brakes would result to overheating and decreasing efficiency and possible risk of accident. This phenomenon is called "fading". In addition, the use of friction brakes produces abrasion particles, which can be harmful to the human body. More available braking power of auxiliary braking systems results in less use of friction brakes, that is, shortening the service intervals of the brake lining and reducing abrasion emissions. Because lower drive resistances means higher required braking power, in the future there will be requirements for more powerful truck brakes even for reasons of safety, decreasing brake dust emissions, cost reduction and speeding up freight traffic.

Theoretical part describes theory about braking systems in commercial vehicles. It discusses the functions of braking systems that are used in trucks, as well as their main advantages and disadvantages. The other part presents research about driving resistances, innovations and current trends in freight transport design which could improve the truck efficiency and decrease driving resistances. The end of the theoretical part is dedicated to legislation, because legislation determines the constraints and needs of development of auxiliary braking systems. Legislation pushes the manufacturers to innovate and to improve the efficiency of the vehicles.

In the beginning of the practical part are described the main parts of GT-SUITE tractor-trailer model, building strategy and limitations that were figured out in process of building the model. After that, two parametrical options of commercial vehicle in GT-SUITE were created. The first model represents typical current type of commercial vehicle and the second model represents the future commercial vehicle, which has the lowest driving resistances as is theoretically possible to have.

This thesis is focused in detail on engine brakes because the company Eaton is working on valvetrain mechanisms on heavy duty truck's engines in cooperation with CTU Prague. Diesel combustion engines of truck will have to be improved due to strict limitations of legislation in the future. Therefore it was decided to research the potential of the most powerful engine brake on diesel engine - Two-Stroke engine brake. All of the obtained knowledge and assumptions were applied to GT-POWER model of 6-cylinder diesel engine equipped with Two-stroke engine brake. But every engine has some limitations that have impact on maximum brake power of the engine and cannot be exceeded. To find compromise between variable factors and limitations of the engine was done by optimalisation. The result of this optimalisation is the maximum brake performance of Two-Stroke engine brake on downsized engine with respecting all of the constraints. Because innovations and improvement are expected in driving resistances, it is also expected in diesel combustion engines for example by progressive materials. Sensitivity analysis of the engine was used to find potential of Two-Stroke engine brake with increased limitations of the engine in the future.

However, recieved data of brake performance from GT-POWER Two-Stroke engine brake model engine brake needs to be divided to 4 modes (levels), because full brake power is not always needed. In the truck model are used 3 different types of brake systems, therefore control strategy has to be designed to synchronise all of the systems in order to minimise use of friction service brakes.

The end of the practical part discusses consequences of lower driving resistances for truck model. Firstly, in the static calculations is shown comparison of different types of engine brakes with data from company Jacobs vehicle systems for current and for theoretical future truck on constant downhill grade. Secondly, the recieved data from GT-POWER model of Two-stroke engine brake are used in the parametric GT-SUITE model of commercial vehicle to simulate dynamic tests on the prescribed routes and comparison to Compression release engine brake and to ZF Intarder. The results of dynamic tests shows the limits of auxiliary brakes, how much energy is dissipated through friction brakes, how much abrasive emissions is released into air from friction brakes during the dynamic tests.

1. ANALYTICAL PART

“What is good to know before Practical part”

1.1 Freight Transport - Commercial Trucks

Since the first commercial truck was released (*Fig. 1*), many things have changed. An amount of transported goods has been still increasing. Transportation of goods changed practically the whole world as we know today. Improvements in design and powerful engines have allowed the construction of ever larger trucks. Nowadays, maximum amount of transported goods is limited by legislation of maximum gross weight. Economic pressure on transport industry is increasing year by year. Emission restrictions force vehicle designers to constantly improve their vehicles. The only way to improve trucks, considering current infrastructure and mass limitations, is to increase its efficiency - by reducing driving resistances and improving driving aggregate to achieve the lowest possible emissions and the lowest fuel consumption.

Fig. 1 - Daimler Motor-Lastwagen from 1898 [1]

It's important to realise right from the start, that requirements of the truck customers are totally different than passenger cars customers. In trucks, an emphasis is on reliability, robustness and proven technologies. The main purpose for trucks is to transport as much amount of goods as possible at the shortest time, and at the same time to keep the costs to the lowest. That is not an easy task for truck designers. The proportion of freight transport continues to grow in order to meet the needs of humanity with the least possible enviromental impact. That's the reason, why it is so important to innovate and improve trucks.

"*The European Commission has set ambitious emission targets for the transport sector in order to reduce global warming, slow climate change, and improve air quality. In 2013, the transport sector contributed about one-quarter of the EU's GHG emissions. Moreover, this is the only sector whose contribution to GHG emissions is growing. The growing demand for freight transport caused by globalisation is one of the driving forces* *behind this trend. The European Commission Directorate-General for Energy and Transport (EC 2008) projects that road freight transport activity will increase, accounting for 75.4% of total freight transport by 2030 (Fig. 2). It is clear that intrinsically more efficient logistics are needed. Furthermore, according to the EU, a modal shift from road to rail, short sea shipping, and inland waterways-i.e. optimum multimodality-is necessary."* [2]

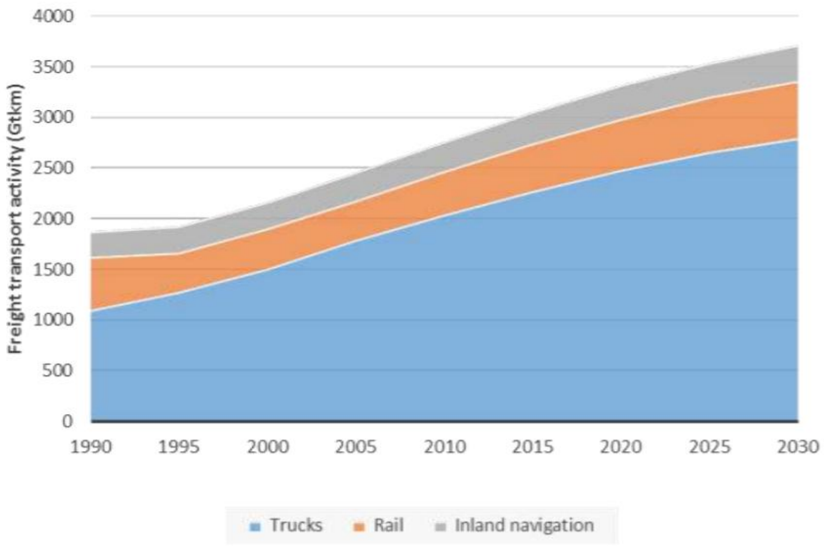


Fig. 2 - Freight transport activity, 1990–2030 [2]

But there are many reasons why is possible to transport goods only by road. The modal shift is not possible for reasons such as: receiver of goods is not located on an inland waterway or at a railway line, transport on shortest time or small amount of transported goods Road transport is still the fastest. Moreover, even if resources are invested to new infrastucture in rail transport, it could take minimal 20 years to be ready for use. In order to be able to absorb the growth of freight transport by road and to meet societal needs and strict emission targets, freight transport by road should be more efficient. [2]

2.1 Braking Fundamentals

"Why is it so hard to safely stop the truck"

The large amount of kinetic energy contained in a loaded truck which is usually driving at highway speeds requires equally large amount of braking system energy to slow down the vehicle or to completely stop. Braking systems have to be reliable, especially in case of this huge amount of kinetic energy. Current braking systems are integrated to a lot of emergency features, such as: antiblocking system, vehicle stability control, collision avoidance, rollover protection, and traction control systems. To control of these features is needed a large element of sophisticated electronics control operating the braking system. The heat, produced by combustion engine to bring the vehicle into a motion is stored in vehicle as a kinetic energy, which is in case of braking converted back into heat through friction brakes as shown in Fig 3. [3]

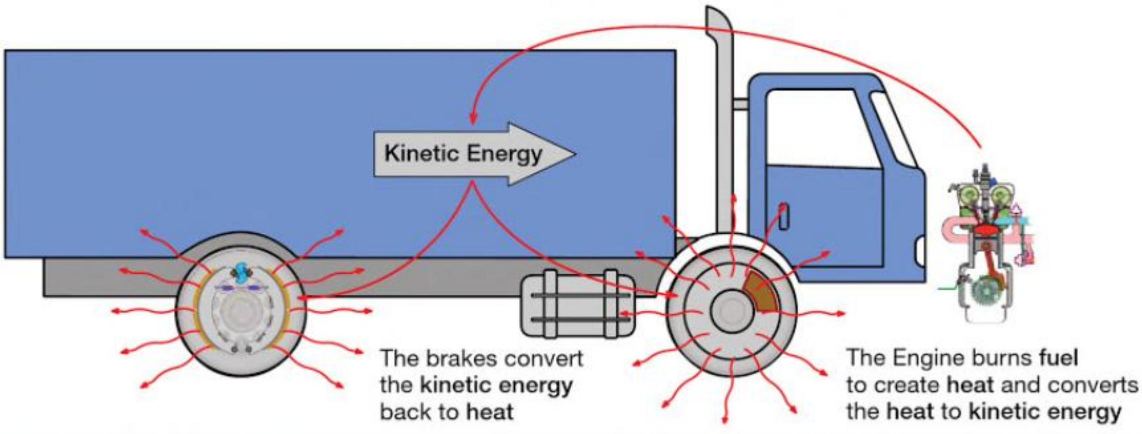


Fig. 3 -Transforming energy in truck [3]

For example, In Czech Republic, the maximum gross weight mass of truck is 48000 kilograms. When we compare the kinetic energy of this truck (Chart 1) and passenger car with 1340 kilograms at same speed 90 kilometers per hour, the kinetic energy of truck is approximate 37 times higher. Only for the interest, the 15 megajoule of energy is equivalent of approximate 3.3 kilograms of TNT.

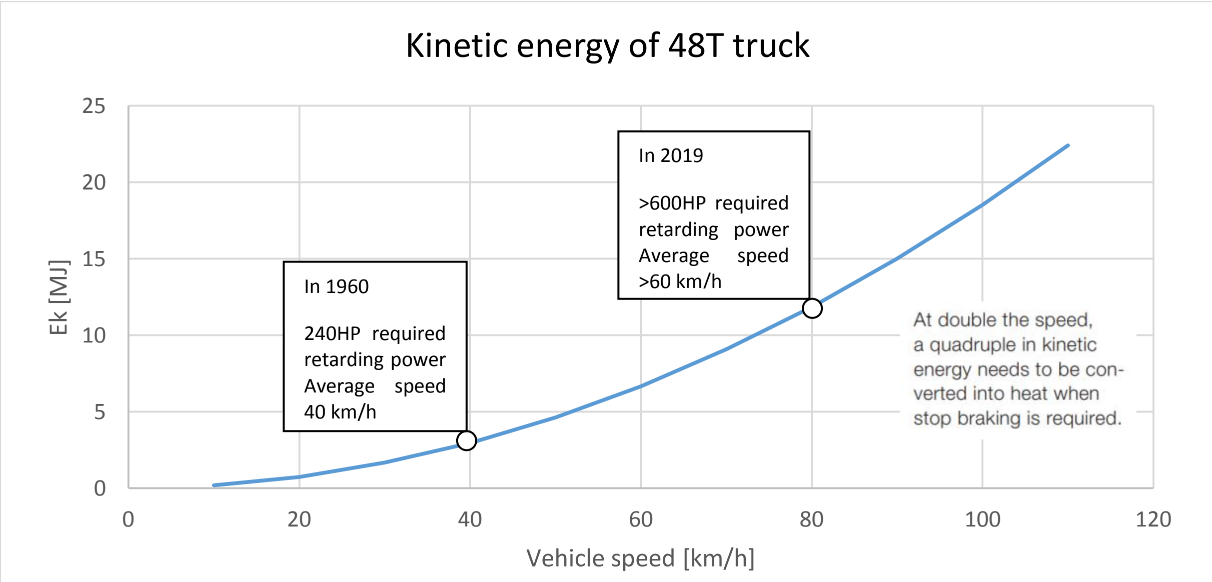


Chart 1 - Kinetic energy past vs. today

Heat created in brake parts is dissipated to the air through brake design factors as is shown in Fig. 4. Unless the design of brake parts is build-up for minimizing accumulations of heat, created heat will occurs to a loss of braking efficiency and damage to braking parts.

Legislation for braking systems specify brake performance standards. The most latest changes in rules requires shorter stopping distances as shown in *Fig. 5*, with a high level of vehicle stability of braking. Antilock braking, Stability and Traction control, or more advanced disc and more sophisticated materials of brake parts are technologies used to meet the latest braking legislation requirements. [3]

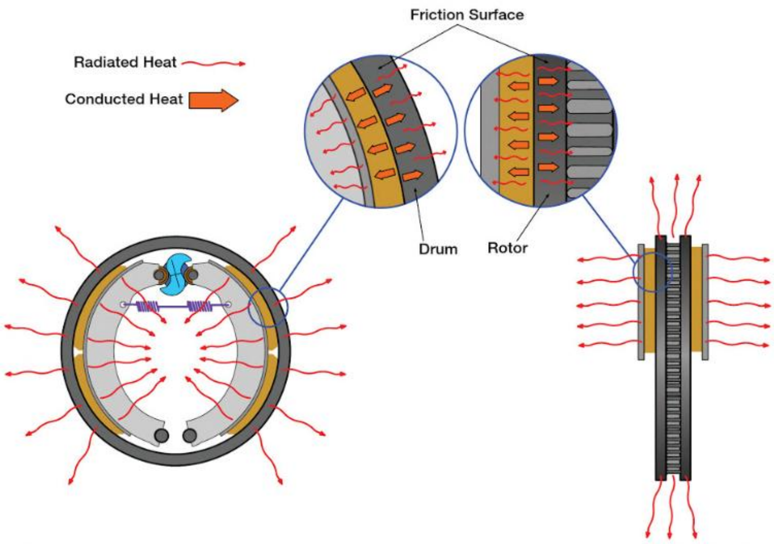


Fig. 4 - Heat in friction brakes [3]

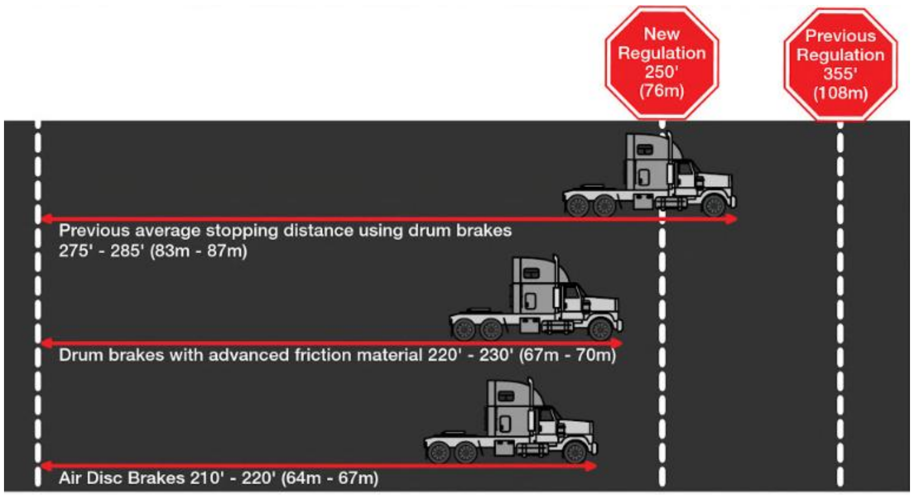


Fig. 5 - Brake performance standards [3]

Innovations in truck design have placed even more demands on braking systems. Improved drag resistance of the vehicle, radial tires and superśingle tires would reduce rolling resistance and more efficient drivetrains with lower friction mean that higher demands to slow a vehicle are placed on the brake system. Vehicle safety is controlled by legislation standards and is improved year by year by use of the more sophisticated ABS control, automatic slack adjusting mechanisms or by automatic brake systems controlled by cameras. Even with still inceasing numbers of trucks, collisions of trucks were declined close to 20% between 2008 and 2009 by advanced brake technologies. [3]

HOW BRAKES WORK

Using brake drums or discs attached to the wheels, friction is produced by forcefully applying heat-resistant braking material against these rotating components. Friction's by-product heat is then dissipated into the air. Because a vehicle must be capable of stopping faster than it can accelerate, a tremendous amount of braking force is needed. With the heavier weight and the speed commercial vehicles travel, the power generated by the brakes must be several times that of the engine. *Fig. 6* shows that many times power is required to slow and stop a vehicle than is required to accelerate the vehicle. [3]

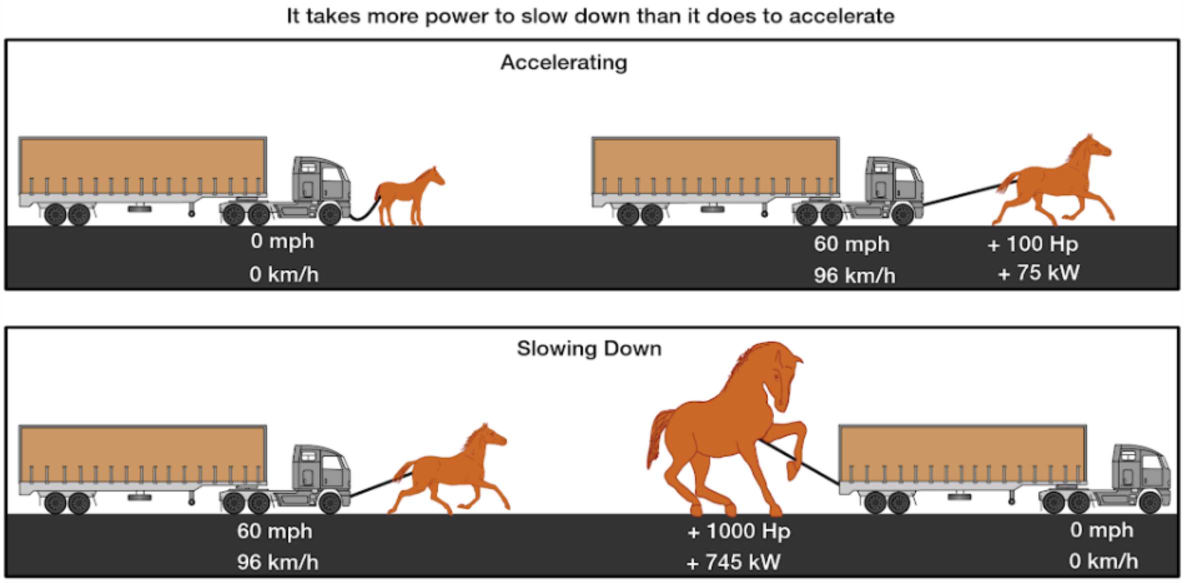


Fig. 6 - Comparison of performance between acceleration and deceleration [3]

Just a horsepower is used to measure the energy used to bring a vehicle up to speed, the retarding force required of the braking system can by estimated using horsepower to get an idea of the energy required by the braking system. [3]

To better understand this concept, consider a truck with a 350-hp engine used to accelerate a loaded vehicle to 60mph in sixty seconds. To slow and stop the vehicle in an emergency condition would take approximately 6 seconds a tenth of the time used to accelerate the unit. The vehicle is also decelerated at 10 times the rate of acceleration. Calculated together, the **brakes** would require **1000 times** the power of the engine. Or, in this example the brakes would exert 350000 hp of retarding force! [3]

INFLUENCE OF VEHICLE WEIGHT AND SPEED

The forces involved in decelerating a vehicle are considerable. Looking more closely at the factors influencing brake system capabilities, it is important to note increasing amounts of energy are required as a vehicle’s weight and speed increase. Using the engineering formula used to calculate the energy of motion, kinetic energy, it can be demonstrated that, as the weight of the vehicle is doubled, the kinetic energy converted into heat energy is also doubled. *Fig. 7* shows the influence of vehicle speed and weight on required braking force. Doubling vehicle weight or speed needs twice the braking power for the same deceleration rate. When weight and speed are both doubled, braking force must increase by a factor of eight. [3]

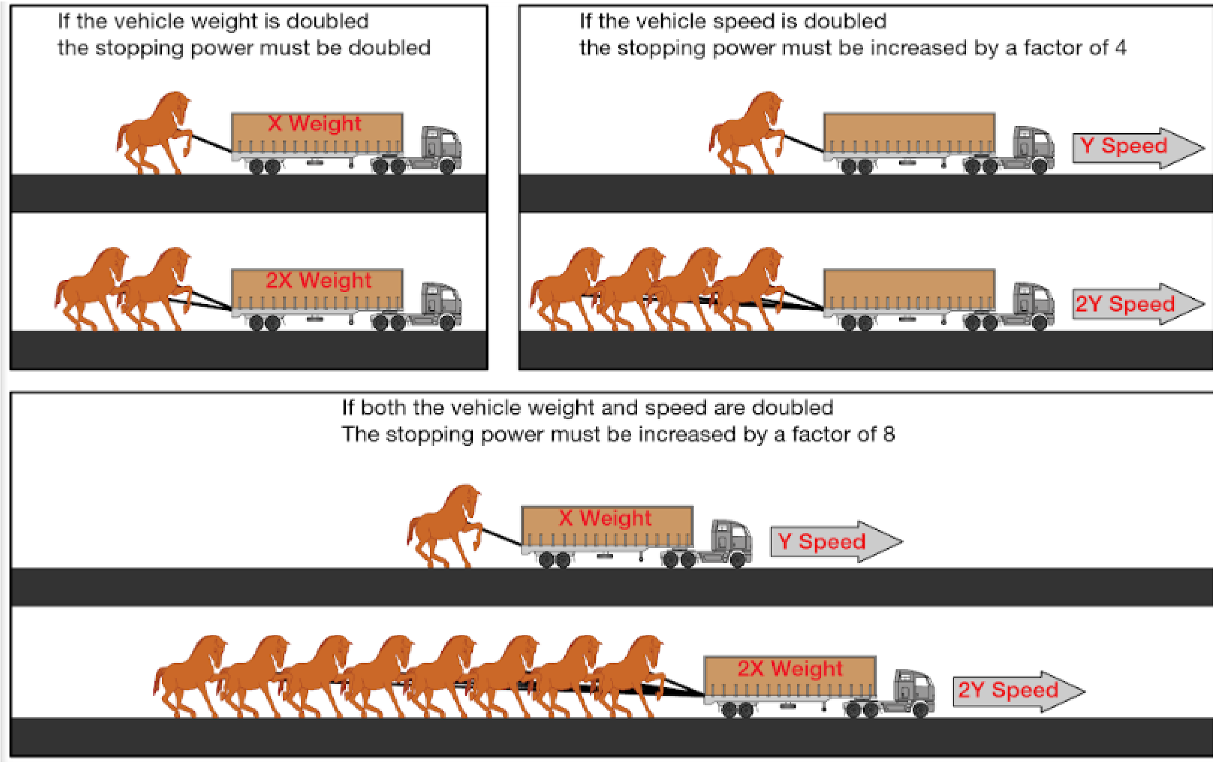


Fig. 7 - Influence of vehicle speed and weight [3]

If the brakes cannot adequately absorb and dissipate the additional heat produced by increasing vehicle weight, braking performance and safety suffers. This explains why heavy-duty brakes are specified not by the type of vehicle but the weight carried by an axle and its location on the vehicle. *Fig. 8* shows how vehicle weight is distributed over each axle. [3]

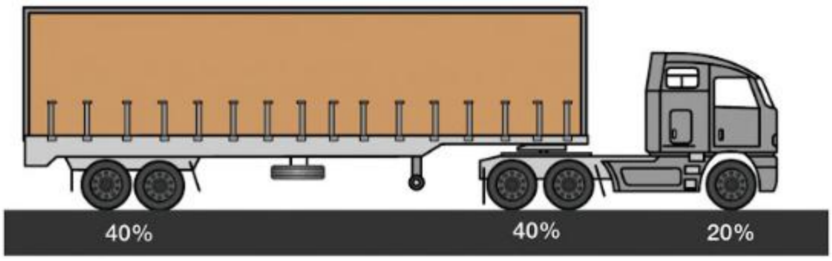


Fig. 8 - Braking load distribution [3]

When is too much brake power on the front axle, it can cause *jackknifing**. However, too much brake power on the rear axles can cause the trailer or rear axles to swing out.

Increasing vehicle speed has a greater effect than vehicle weight on the braking system power. Calculations of kinetic energy show braking power requirements increase by factor of two speed as speed increases. This means four times as much power is required to decelerate from a speed of 40mph compared to 20mph. A stop from 60 mph which is three times the speed of 20 mph, needs nine times the energy. [3]

**Jackknifing refers to the folding of an articulated vehicle so that it resembles the acute angle of a folding pocket knife*

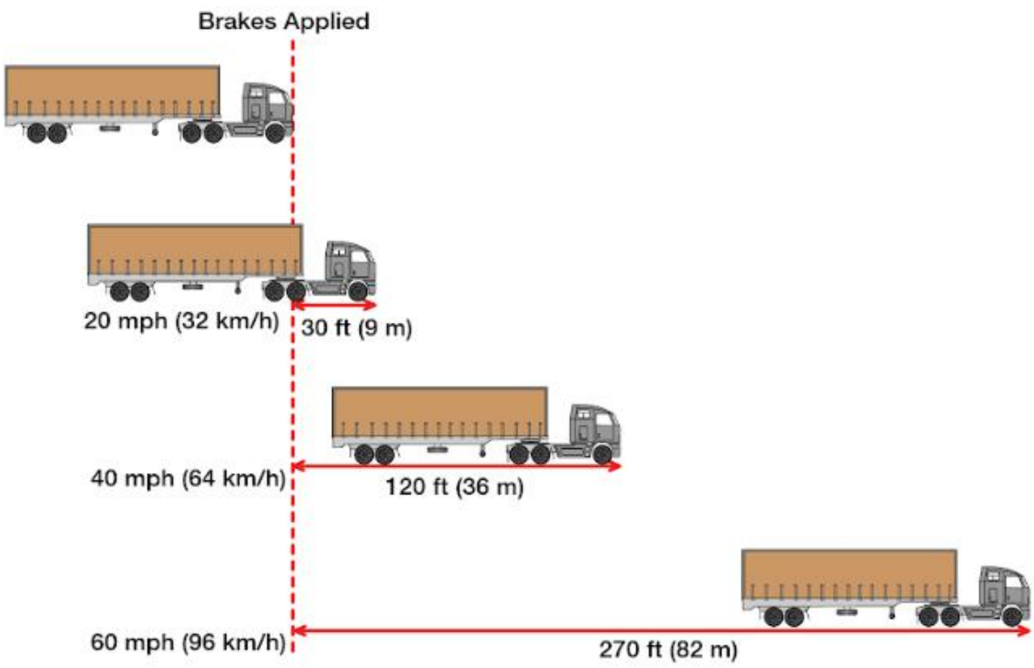


Fig. 9 - Stopping distances [3]

It is understandable, then, why commercial vehicles needs higher braking system pressure, larger friction surfaces and greater absorbing capacity of heat as for example commercial vehicles. In the *Fig. 9* is shown the influence of higher vehicle speeds on stopping distances, which are progressive increasing on higher speeds. That's the reasons, why are requirements for reliability, safety and performance of braking systems in commercial vehicles so strict. There are three factors, which can decelerate the truck:

- Service brakes
- Driving resistances
- Auxiliary braking systems

The next chapters discusses about these three factors.

3.1 Conventional Friction Brake (Service brakes)

Service brake system is composed of the following main parts: **the master cylinder** which converts foot pressure from brake pedal and converts this pressure into hydraulic pressure. Master cylinder is located under the hood which is directly connected to the brake pedal. The **Brake hoses** which connects the master cylinder to the slave cylinders which are mounted at each wheel. The system is filled by **Brake fluid** with high boiling point, specially designed to work high temperature conditions. Brake pads or Brake shoes are pushed by the slave cylinders to contact the drums in case of drum brake or discs in case of disc brakes thus causing drag, which slows the vehicle. There are two basic kinds of friction service brakes: disc and drum brakes. [4]

3.1.1 Drum brakes

Drum brakes consist of a heavy flat-topped cylinder, which is sandwiched between the wheel rim and the wheel hub. The inside surface of the drum is acted upon by the linings of the brake shoes *(Fig. 10).* When the brakes are applied, the brake shoes are forced into contact with the inside surface of the brake drum to slow the rotation of the wheels. [4]

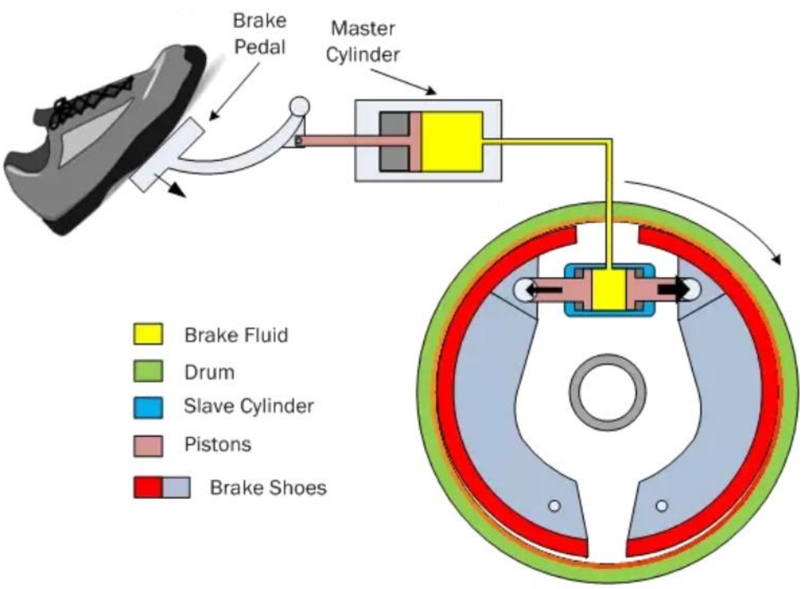


Fig. 10 - Hydraulic drum brake [5]

CHARACTERISTICS

- Self energizing effect requires less input force
- Less expensive to produce
- Construction protected from dirt inside the wheel recess (however, the dirt will still get there only to a lesser extent)
- Parking brake is easier to realised
- Long service life of brake lining
- Replacement of brake lining and maintenance are expensive
- Design size is limited by wheel recess
- Poor heat dissipation, the brakes are therefore prone to ***fading*** [6]

3.1.2 Disc brakes

Disc brakes use a clamping action to produce friction between the “rotor” and the “pads” mounted in the “caliper” attached to the suspension members (*Fig 11*). Disc brakes work using the same basic principle as the brakes on a bicycle: as the caliper pinches the wheel with pads on both sides, it slows the vehicle.[4]

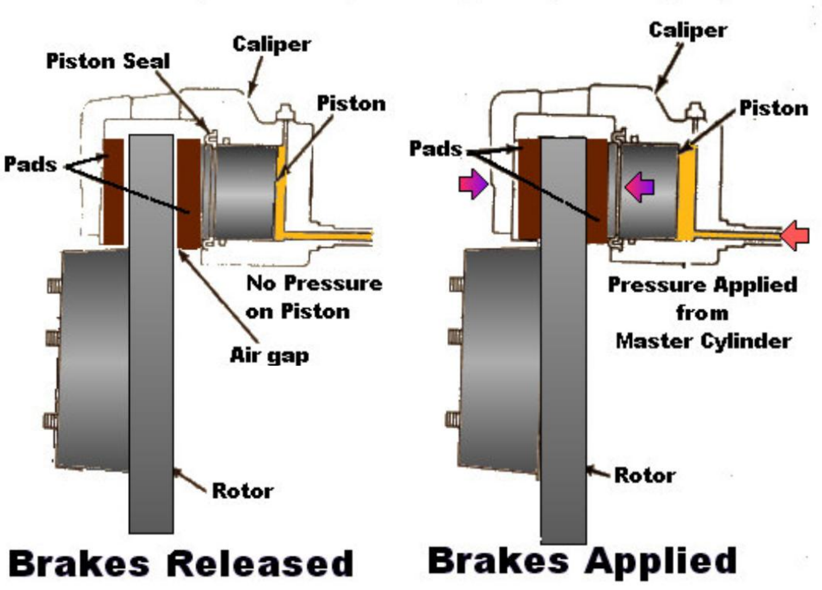


Fig. 11 - Hydraulic disc brake [7]

CHARACTERISTICS

- There is no reinforcement due to flat braking surfaces. These require larger thrust forces, so larger diameter brake rollers are needed than with drum brakes.
- There is no oblique tension and the braking force can be dosed well, because virtually no brake force fluctuation due to lack of self-reinforcement and small variations in friction coefficient
- Good cooling, low fading tendency, may be due to small braking surfaces, and higher temperatures may cause higher temperatures
- Greater wear on the lining caused by high contact forces
- Easy maintenance and lining replacement
- Automatic clearance adjustment
- The braking effect is independent of the direction of trave
- Good self-cleaning provided by centrifugal forces
- Tends to form vapor bubbles in the brake fluid because the brake pistons fit tightly against the brake lining [6]

It can be concluded that the main target for future truck is to lower the use of friction service brakes, mainly to increase safety at long mountain passes where the risk of fading occurs. The friction brakes should be used only in cases of emergency or at low vehicle velocities, because auxiliary brakes can't stop the vehicle steadily. The other target is cost reduction. Frequent use of service brakes means more frequent replacement of brake lining. The last target is cleaner environment. Lower use of friction brakes means less harmful particles from brakes.

Coefficient of Friction (CoF)

"CoF refers to the amount of force required to move an object while in contact in another. Stated another way, CoF describes how slippery the surface is between two objects. CoF is decreased by increasing temperature (Fig. 14) of braking parts, and wear of brake lining is more faster on higher temperatures."[3]

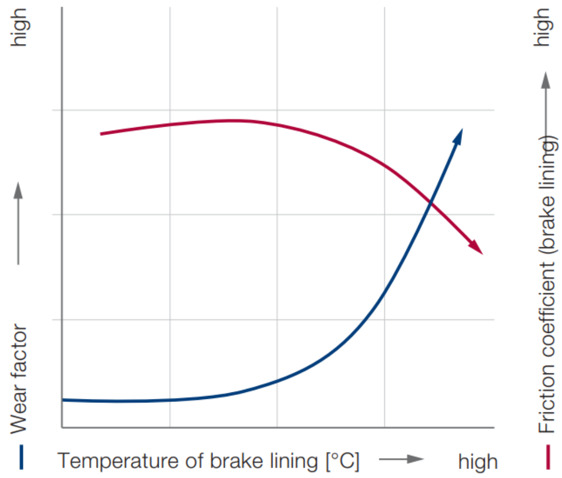


Fig. 12 - Change of the coefficient of friction and wear by temperature [8]

3.1.3 Brake fade

"Why the friction brakes won't brake for long time"

Brake fade is a service friction brake problem, which could be described as a reducing brake power or effectiveness during long time duration of braking due to factors such as overheated brake parts, contaminated lining, or changes to the friction material in case of glazing of brake linings. Decreasing of Cof is known as a "hard pedal". A driver of the vehicle would describe fade as the need to push to braking pedal with more force to get the same braking effectiveness what. Even advanced brake linings frim high quality materials will fade slightly upon each brake application but will immediately return to their initial state after cooling. However, the CoF of the brake lining is changing, especially by using only service brakes during its mileage life. Drum brakes are significantly less immune to fade than disc brakes. There is also the opposite case of brake fade what is known as an *anti-fade*. In the case of *anti-fade,* the *CoF* is increasing when the brake parts are getting hotter. [3]

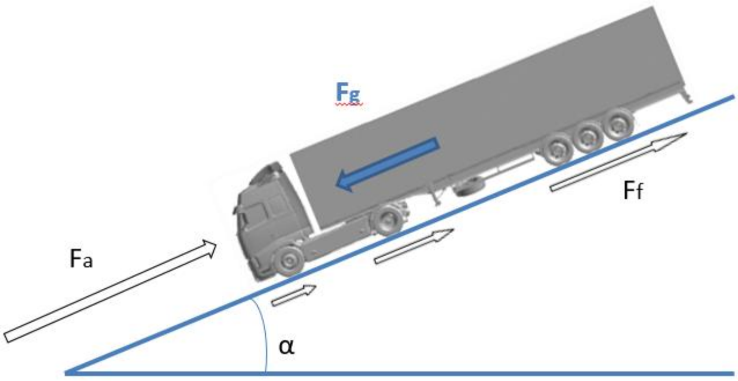
<u>Brake fade occurs for several reasons and the types of brake fade are:</u>

- Heat
- Water
- Mechanical
- Chemical fade, as well as glazing of lining

For more detailed description of fade types please see: [3].

4.1 Driving resistances – improving the efficiency

Driving resistances are actually forces, which counteract the movement of the vehicle – slow it down. For the target of most efficient vehicle dynamics, the current trend is to reduce these driving resistances to minimum level for reach the lowest fuel constumption and emissions at the same order to keep reliability of the truck. But on the other hand, lower driving resistances means higher demands on braking system to have same slowing characteristics of the vehicle.



Fa- Aerodynamic drag

Fg- Gravity force

Ff- Rolling resistance

(includes drivetrain losses)

Fig. 13 - Driving resistances

PRINCIPLES OF THE DRIVING RESISTANCES AND EFFICIENCY OF TRUCK

The actual CO_2 emissions of a vehicle depend on the vehicle's driving resistances, the powertrain's efficiency and the energy demand of potentially activated auxiliary consumers *(Fig.14).* The efficiency of the powertrain describes those parts of the total fuel's energy content that can be used for the mechanical propulsion of the vehicle. The majority of the employed chemical energy gets lost by heat dissipation and friction in the powertrain. [9]

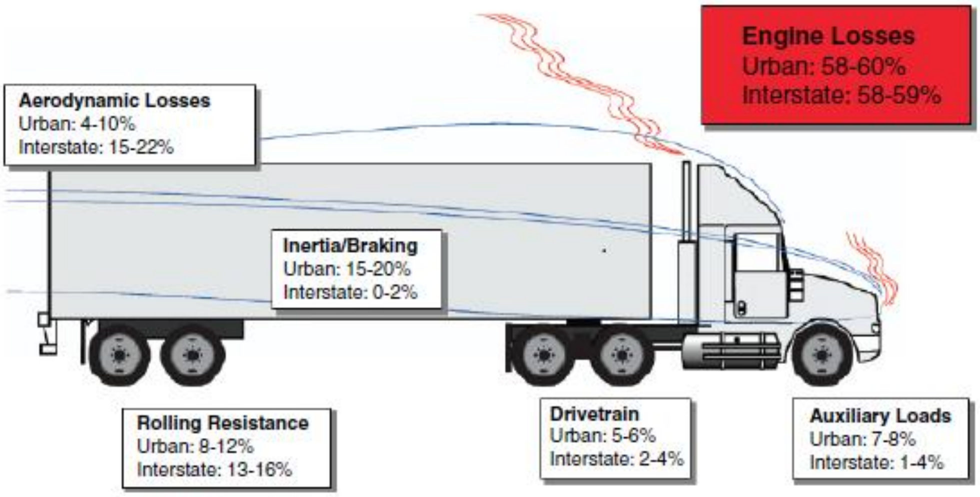


Fig. 14 - Percentage of impact losses to vehicle [10]

4.1.1 Drivetrain Losses

All components of the truck (except for engine) together which are transferring the power from the engine to the wheels are called drivetrain (Fig. 15). The power is transferring from the engine via transmission, differential, driveshaft and than to rear axle(s) where are the wheels mounted. Before the energy goes to the wheels, some part of energy is dissipated by friction. Drivetrain losses result from friction between bearings and rotating components and in calculations are usually included in coefficient of rolling resistance. Truck will always have a large mass. Therefore, is very hard to decrease the friction in drivetrain. In addition, automatic transmissions incur losses in the torque converter used to transfer power between the input and output shafts of the transmission. A manual transmission typically incurs fewer losses than an automatic transmission because a manual transmission has no torque converter. In fact, the most of heavy duty combination trucks currently use manual transmission. [11]

Fig. 15 - Truck drivetrain [12]

4.1.2 Acceleration(deceleration) resistance

The acceleration resistance represents the forces that counteract the movement of the vehicle when the speed changes. This resistance consists of two components and the total value of resistance is sum of:

- ***Acceleration resistance of sliding masses*** - This resistance is given by the response to the change of speed
- ***Acceleration resistance of rotating masses*** - Some parts of the vehicle (flywheels, clutch, transmission) are rotating during movement and have moment of inertia.

It has substanial impact in lower engaged gears and in low vehicle speeds, but truck is operating most of the time at highway speeds. In the future, mass of the trucks and of the rotating parts **will not be radically changed**. Therefore, there are not large possibilities to decrease deceleration resistance.

4.1.3 Aerodynamics od truck (air resistance)

“Why Enzo Ferrari wasn't quite right”

State-of-the-art methods for determining the aerodynamic parameters of vehicles allow designers much more creativity, because they can by means of numerical calculations and simulations to get needed parameters and confirm characteristics of prototype in aerodynamic tunnel. Up to now the designers were limited by legislation which will be changed in near future. This moves the possibilities of aerodynamics development to a completely different level. In the *Fig. 16* is shown the ideal aerodynamic truck shape of Volvo Supertruck concept. This type of truck will have a significantly lower air resistance that consequence higher demand on brake system.

Fig. 16 - Volvo supertruck concept [13]

Because the truck spends significant time by travelling at highway speed and aerodynamic drag is increasing by squared vehicle speed, the aerodynamic drag has a large impact of fuel consumption. Trucks hasn’t very good aerodynamic shape, there is an effort to transport that much amount of goods as possible, that’s the reason why truck’s trailers looks like a big “box”. Especially in Europe, truck has a big front area and sharp end of trailer.

Only for interest: In cold Canadian climates, the aerodynamic drag in winter can be nearly 20% greater than at standard conditions, due to the ambient air density.

<u>*More advanced aerodynamic shape can bring many benefits for example:*</u>

- Better stability and steering control due lower side wind sensitivity
- Lower noise of vehicle
- Lower fuel consumption

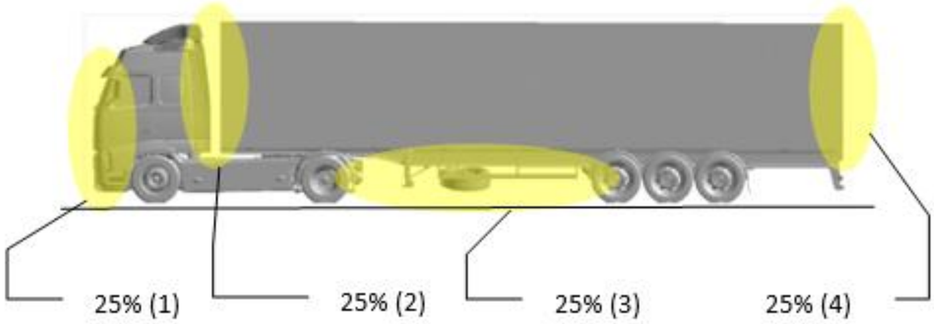


Fig. 17 - Main parts of tractor-trailer affects an aerodynamic drag

Aerodynamic drag on a tractor-trailer is affected by: (Fig. 17):

- The shape of the tractor cab, the transition between the back of the cab and the front of the trailer (1)
- The gap distance between the tractor and the trailer (2)
- The underbody of the truck (3)
- The rear edge of the trailer (4)

FRONTAL SURFACE AREA OF THE TRACTOR CAB (1)

Air resistance is affected by a vehicle's total frontal surface area and by its shape which has a large impact of total air resistance. A smaller frontal surface area will produce less drag at a given speed. Likewise, sloping shapes that "slice through" the wind produce less drag than flat surfaces directly perpendicular to the direction of travel. An important element of aerodynamics is also the chassis height. The air resistance coefficient is better, when the chassis is closer to ground, because less air is able to flow under the vehicle's chassis and there is turbulent flow as well. Another rule is the height of the cabin, which should be the same as the height of the trailer. If the tractor cab is lower than the semi-trailer, a roof cover is used which causes smoother airflow. [14]

GAP DISTANCE BETWEEN TRACTOR AND TRAILER (2)

The gap distance between cab and trailer should be a small as possible. In case of vehicles with rigid semi-trailers, the gap can be bridged, but in the vehicles with trailer is problem with necessary space in cornering. In general, larger gap distance means higher air resistance. To improve the turbulent flow in the gap between the cab and trailer is the deturbulator most commonly used. Deturbulator generates smoother flow, to less intense vortices that occur more naturally. Its cost of ownership will be returned after about 2-5 weeks to save fuel, but needs to be replaced once a year with a new one. [14]

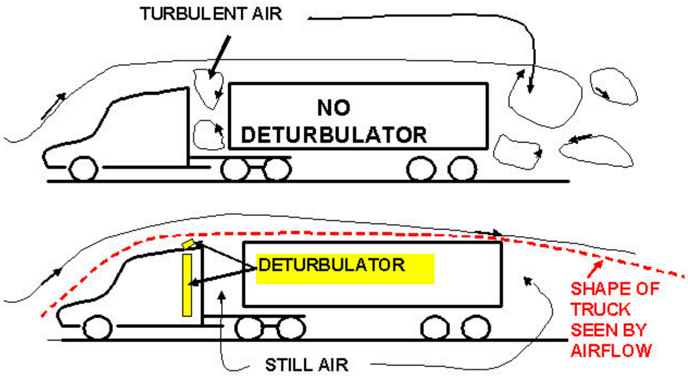


Fig. 18 - Impact of deturbulator [15]

UNDERBODY OF THE TRUCK (3)

Underbody of the truck (*Fig.19*) is very important mainly when the side wind is applied, because the side wind disrupts the flow and a large amount of air gets under the trailer, causing large turbulences. [14]

Fig. 19 - How aerodynamics contribute to save fuel [16]

THE REAR END OF THE TRAILER (4)

The rear part of the trailer has a large share of the air resistance because the air streamlines are closing behind the trailer (Fig. 20). The most known example of this trailer is *boat tail.* As a result of testing different types of trailer ends on a real semi-trailer, the best configuration is an open end including the bottom that confirmed a simulated **12% reduction** of air resistance. [14]

Fig. 20 - Boat tail [17]

On a heavy-duty combination truck total aerodynamic drag is affected by the size and shape of both the truck tractor and the trailer it is pulling. Interesting example for improving the trailer aerodynamics is Don-Bur Teardrop™ trailer*,* which can save about 11% of fuel.

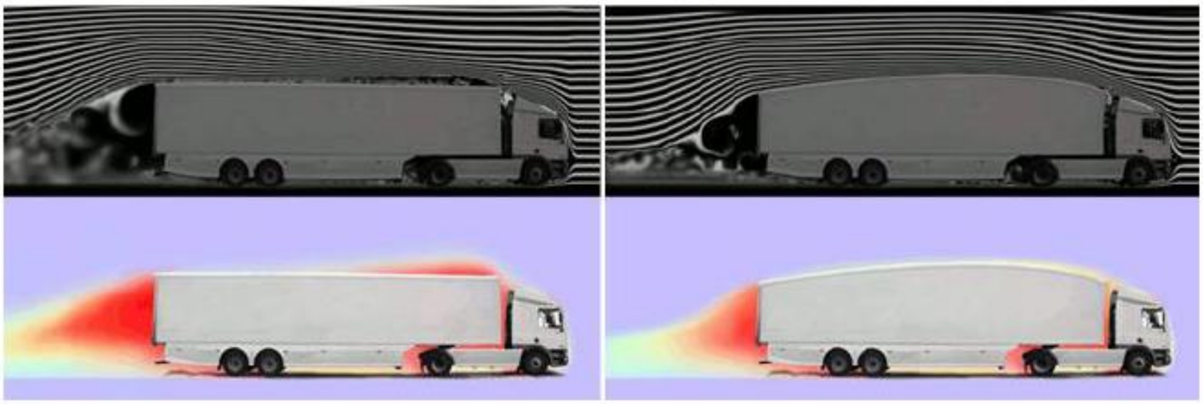


Fig. 21 - Don-Bur Teardrop trailer [18]

There are a lot of features, that can reduce the aerodynamic drag and their impact is shown in Annex 2.

4.1.4 Rolling Resistance

Rolling resistance is the friction exerted on a vehicle's tires by the roadway surface. Rolling resistance is proportional to a vehicle's mass, and is also affected by the material, configuration, and air pressure of the vehicle's tires. High air pressurised tire have greater rolling resistance than a tire with low air pressure. In general, when the tire is dense, the overall rolling resistance of the vehicle is smaller, but the overall contact area is reduced, thereby reducing comfort and, in particular, driving safety because the vehicle does not have sufficient ground contact (*Fig. 22*). Rolling resistance results from tire deformation. [11]

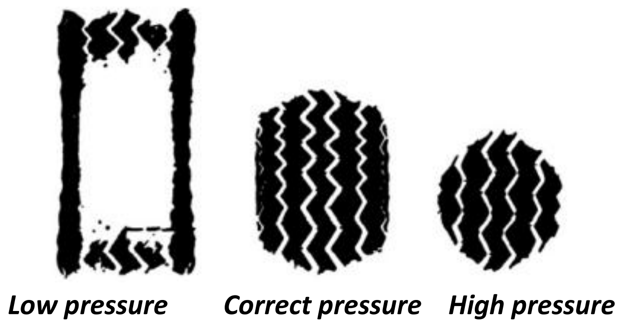


Fig. 22 - Comparison of different tire pressures [19]

The total rolling resistance of the vehicle is the sum of the rolling resistance of the individual wheels and also depends on the road surface. Rolling resistance change only small with increasing vehicle speed, but is proportional to vehicle weight. Therefore, Low rolling resistance tires provide greater benefits on heavier trucks. In the future, there is no much possibilities to radically improve rolling resistance, because of materials of tires and large weight of truck.

TECHNOLOGY	FUEL SAVINGS		Source
	% Reduction (l/100 km)	Annual Savings[1] (liters)	
Single Wide-Based Tires ("Super Singles")	3.8%	2,308	RMI 2005 [28]
	2% - 5%	1,176 – 2,857	EPA 2004 [35]
Michelin New Generation Single Wide-Based Tire (NGSWBT)	9%	5,117	ORNL 2008 [37]
Low Rolling Resistance Tires	2.9%	1,748	RMI 2005 [28]
Automatic Tire Pressure Control	1.2%	711	RMI 2005 [28]
	0.5% - 1.0%	299 – 594	EPA 2004 [35]

[1] Based on: 120,000 km annually and baseline fuel consumption of 47.6 Liters per 100 km. Baseline fuel consumption assumes regularly overloaded trucks

Note: Driving speed does not significantly affect rolling resistance, but rolling resistance is proportional to truck weight. These fuel savings estimates (% reduction) are based on the average weight of trucks in the U.S., and may understate the benefits of these technologies to Chinese trucks which typically operate at significantly higher weight.

Fig. 23 - Fuel savings of tire technologies [20]

4.1.5 Engine Losses

"It's all about the friction."

From the whole energy what is in fuel tank, is only about 33% of the input energy in the fuel is turned into useful work at the engine output. The other per cents of fuel input energy are lost to engine friction, goes out the vehicle tail pipe as heat, or is dissipated as heat by cooling circuit. Part of the energy is used to power engine accessories (i.e. alternator, air condition, hydraulic fans, air compressor for brakes) and part of the energy is dissipated to heat in the vehicle's drivetrain. Rest of the energy is used to overcome inertia, gravity, aerodynamic drag, and rolling resistance (*Fig. 16)* to accelerate the vehicle and keep it moving down the road and up and over hills. [11]

Potential for improvements can be made in net efficiency by modifying current engine equipment and control strategies, more significant gains may also be possible from the use of a completely different combustion cycle. Manufacturers have to date been able to mitigate part or all of this efficiency loss through various engine design improvements. Even more stringent emission requirements that take will require additional changes that might further impact net efficiency. [11]

Diesel engines are more used than gasoline engines because they have more pulling power and low-speed torque than other engine types with low fuel constumption. But diesel engines have a small disadvantage of smaller brake power compared with gasoline-fueled engines. Gasoline engines have throttle plates in inlet manifold that produce what is known as a pumping loss and that is the reason of higher natural brake power.

4.1.5.1. Downsizing of the engine

"Same power with smaller engine"

The larger engine means more vehicle weight and therefore more power. Downsizing has begun to grow with the advent of increasingly stringent emission standards. Downsizing means reducing engine volume while maintaining its original maximum power, for example, by direct fuel injection, variable valve timing, turbocharging or using supercharger. The main disadvantage of downsizing engines is the use of multiple components and more technologically sophisticated engines, which means lower engine life and higher service costs. By reducing the engine volume, we also reduce its weight and thus the inertia so that the engine consumes less energy to move and change the vehicle speed. Also, overall dimensions and mechanical losses are reduced. Smaller engines require less installation space. Past years, the engine displacement was not radically reducted as shown in *Fig. 24* and development of engine technologies even picked up the engine performance, but in the future will be harder to keep this target because is supposed much larger reduction of displacement. [21]

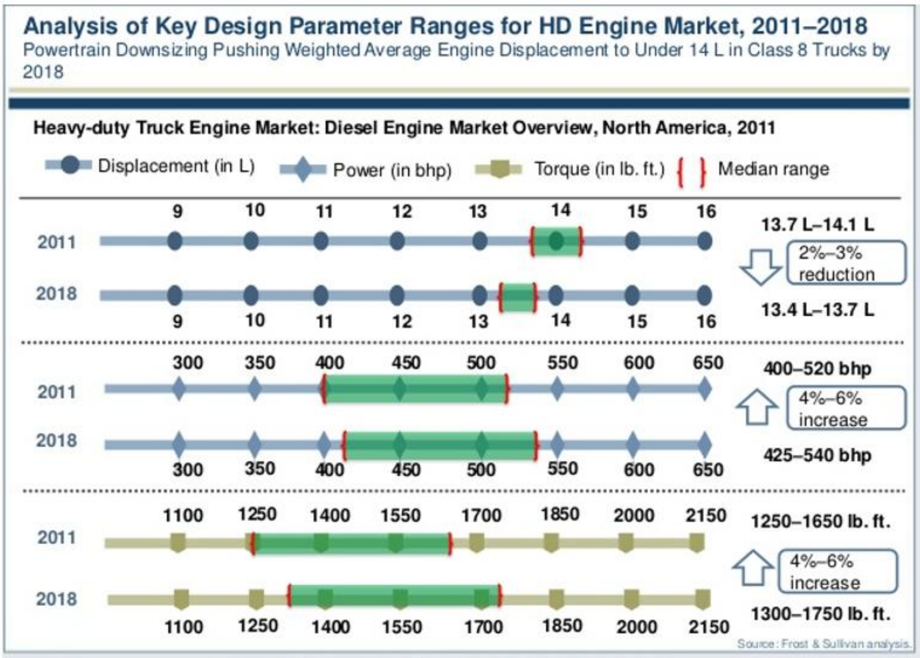


Fig. 24 - Analysis of downspeeding and downsizing [21]

For the engine braking, the downsizing means smaller displacement of pressurised air and smaller efficient area. There is a lot of limitation to obtain original brake power of larger engine such as maximum pressure cylinder pressure, maximum inlet turbine temperature and pressure drop between the cylinder pressure and exhaust manifold pressure. High pressure drop can cause unwanted valve opening and possible contact of valve with piston. But smaller engines allow increase some values of this limitation. For instance in smaller engine, there is a smaller turbocharger, but smaller turbocharger allows higher inlet temperature.

4.1.5.2. Downspeeding of the engine

"Do something slower can also have its advantages."

This technology expects fuel efficiency improvement in about 4-7%. It is expected that the engine operating speed would be reduced and the braking power is directly proportional to the engine speed, therefore reducing the maximum speed increases the additional braking requirements. However, lower-speed engines have lower fuel consumption. Engine speed and load have impact of friction. Engine friction contributes to nearly 20% of total engine losses. Lowering the engine speed range, and friction means higher demand on braking system, because "natural braking forces", which helps with slow down the vehicle, are decreased. Achieving the same power for lower revolutions means that the engine operates with higher BMEP at lower speeds. The required wheel speed could be adjusted by changing either the gear ratio of the transmission or of the differential gear ratio.

There are some advantages to have slower engine:

First, less energy is taken to turn the moving parts of the engine. All moving parts of the engine such as the pistons, rings, the crankshaft and camshafts, gears and timing chains, valves and all the engine accessories have some mass, and it takes a lot of energy to move up this mass. That means that slow the movement will save the energy.

Second, when are masses moving slower in relation to each other, heat transfers more slowly between these two masses. Less energy is wasted by reducing heat transfer to the block of the engine and coolant. [22]

Key technologies to lower driving resistances and fuel emmisions for commercial vehicles are as follows:

- Reduced Friction and Auxiliary Load Reduction
- Advanced Transmissions
- Downspeeding
- Engine Downsizing
- Aerodynamics
- Low-Rolling Resistance Tires
- Reduced Friction and Auxiliary Load Reduction

Potential efficiency improving from 2017 baseline is About 35%

These technologies would have to the highest impact for lower natural retarding forces of the vehicle and higher brake power of auxiliary brakes. Therefore, practical part is focused on these technologies and their impact for braking devices.

5.1 Auxiliary braking systems (Retarders)

"You can also brake without wear"

In truck system design, all the retarders belong to category of auxiliary brakes. Auxiliary brakes are essentialy continuous wear-free brakes which supplements the service brakes. Wear-free because the retarding torque is not generated by the friction between two sliding surfaces. In case of continuous braking by service friction brakes on long downhill passes, the temperature of brakes can reach high temperatures (above 1000˚C) where is the risk of fading and service brakes would be insufficient and wearing much faster. The main target to the future is to synchronize service and auxiliary brakes with the control unit to get maximum braking power with minimum wear (use) of service brakes at high level of safety.

As an example for advantages of auxiliary brake, are shown testing data of heavy truck by company Voith. There are 2 testing routes (modes):

Test route* 4,8 km: Guadix – Granada (Spain)

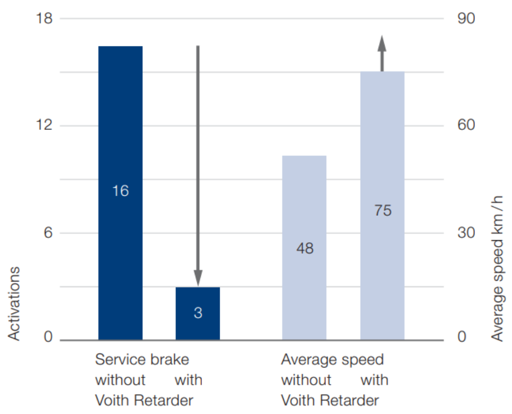


Fig. 25 - Test route with high decreasing [8]

Test route* 3 164 km: Italy – Germany

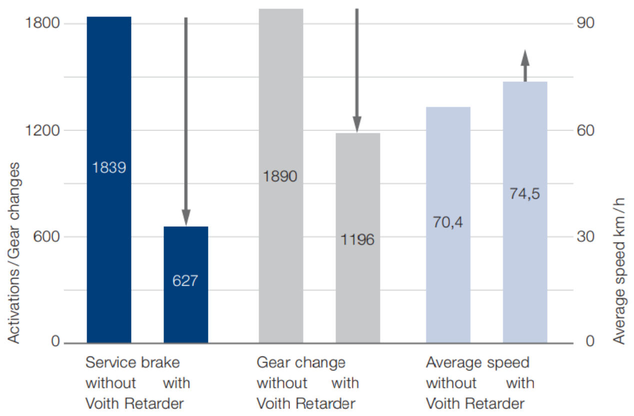


Fig. 26 - Long test route [8]

The maximum downward gradient of the 4,8km test route is 7% *(Fig. 26)*

- 85% service brakes reduction operations
- 56% increase of average speed [8]

<u>For long-distance haulage application *(Fig. 27)*:</u>

- 70% reduction of service brakes operations
- 36% fewer shifting operations
- 5,9% increase of average speed [8]

5.1.1 Classification of Auxiliary brakes

All the Auxiliary brakes or retarders can be classified into two categories. Basic difference between these two types is that the brake torque of engine brakes depends on engine speed and engaged gear of transmission and brake torque of drivetrain retarders depends only at driveshaft speed, because are bolted directly to the transmission output. In next chapters, I would like to describe principles how they works, their main advantages and disadvantages.

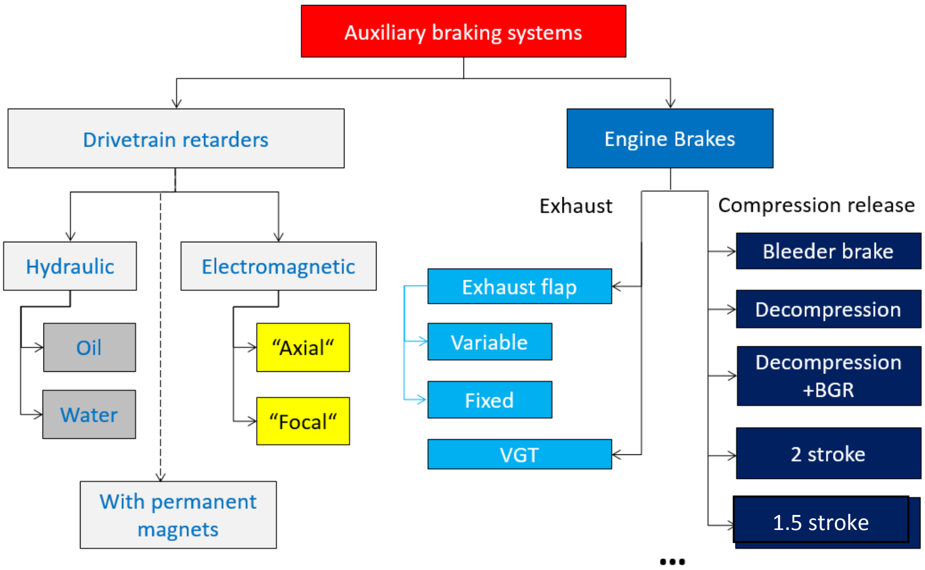


Fig. 27 - Classification of auxiliary braking systems

5.1.2 Drivetrain retarders

The drivetrain retarders include any retarding device that is located in the transmission or the driveline, essentially between the transmission and the rear axle.

LOCATION ON THE VEHICLE

Mounting of the drivetrain retarders can be divided to secondary retarders also called "offline" or direct retarders placed in the gearbox (called intarders or "inline")

„Inline"

Inline retarder (*Fig. 29*) is directly mounted on the transmission or freely in the driveline and is connected with the propshaft shaft of the vehicle. The speed of retarder rotor is same like a crankshaft speed. [8]

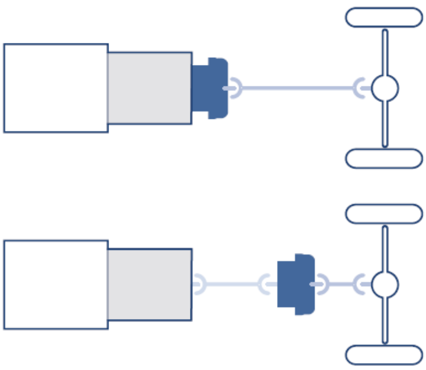


Fig. 28 - Inline retarder [8]

„Offline"

With offline retarder (*Fig. 30*), the speed is increased in relation to the propshaft speed with step-up gear. They are compact and provide high braking outputs eve at low driving speeds, but they may lose the braking performance during the gear shift in the case of manual transmissions. [8]

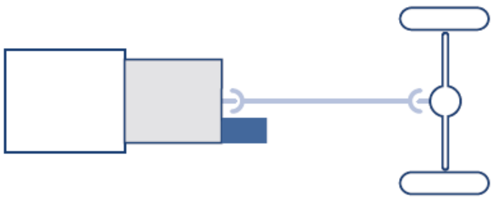


Fig. 29 - Offline retarder [8]

5.1.2.1 Electromagnetic

The working principle of the electromagnetic brake *(Fig. 31)* is based on the application of physical principles discovered by the French physicist J. B. L. Foucault (1819-1868): creation of eddy currents within a metal disc rotating between two electromagnets, which sets up a force opposing the rotation of the disc. When the electromagnet is energized, the rotation of the disc is retarded and the energy absorbed appears as heating of the disc. If the current exciting the electromagnet is varied by a rheostat, the braking torque varies in direct proportion to the value of the current. [23]

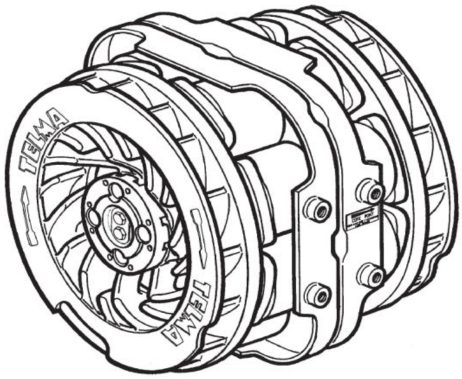


Fig. 30 - Telma electromagnetic retarder [24]

A typical retarder consists of stator and rotor. The stator holds 16 induction coils, energized separately in groups of four. The rotor is made up of two discs, which provide the braking force when subject to the electromagnetic influence when the coils are excited. Shape of the fins, which are integral to the disc, permit independent cooling of the retarder. The retarder is cooled only by air by shape of fins, therefore isn't suitable for long time braking applications. Characteristics of electromagnetic brake are shown in *Fig. 33*, where (a) use in cold condition, (b) average torque (power) over 3min and (c) continuous use (> 60min). The braking torque is controlled by adjusting the exciting current passing through the electromagnets. [24]

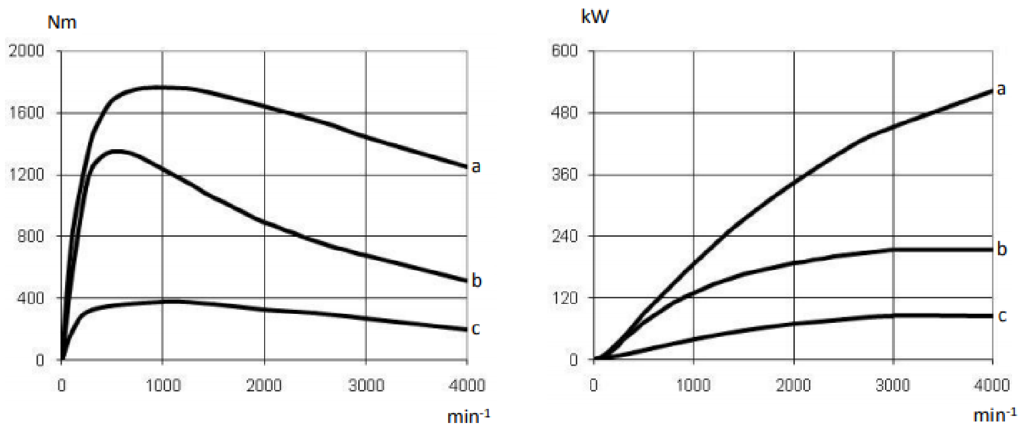


Fig. 31 - Characteristics of Telma Axial AC61-60 retarder [24]

5.1.2.2 Hydrodynamic (Oil)

The hydrodynamic retarder converts the mechanical energy from the wheels via driveshaft to the heat of the fluid inside the retarder via a rotor-stator design. Stator is fixed mounted on the retarder housing, rotor is coupled to drivetrain with wheels and rotates. The inclination of the stator and rotor blades relative to the plane perpendicular to the shaft axis is 45 °. Its retarding torque depends on the amount of fluid flow and the pressure of viscous fluid inside the retarder. The retarding torque is controlled by adjusting hydraulic pressure. The fluid under viscous damping is cooled by the engine coolant through the vehicle radiator or a separate cooler (in the case of a retarder-equipped trailer). Hydrodynamic retarder performs chiefly at higher vehicle speeds. Endurance braking may be impeded due to a high engine temperature. [25]

Legend:

1: oil tank;
2: oil filter;
3: oil pump;
4:two-position two-way pilot control valve;
5: pressure regulating valve;
6: stator;
7: rotor;
8: drive shaft;
9: hydraulic retarder;
10: electromagnetic relief valve;
11: radiator

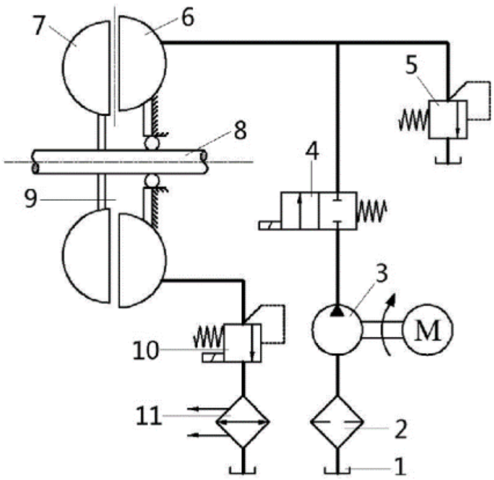


Fig. 32 - Hydrodynamic retarder [17]

5.1.2.3 Hydrodynamic (water)

The braking torque is produced in the same way like in oil hydrodynamic retarder but only difference is that the braking medium is replaced to water, using the cooling agent of the engine for the process. This means that it does not need an additional operating medium. In addition, SWR requires small instalation place. [25]

The *Voith Aquatarder SWR* is the first secondary retarder to using water to brake. There is no Power Consumption when the Retarder is disconnected. Since the rotor of the retarder is coupled to the drive train, it rotates permanently while driving. This consumes energy – even if no braking power is required. In the ECO-SWR, a coupling disconnects the rotor from the drive shaft in idling mode. This means that no further energy is consumed. [25]

5.1.2.4 Retarder with permanent magnets

This revolutionary concept of braking of the vehicle is used latest years in 7,5-16 tonnes commercial vehicles and in buses. The function of magnetic retarder is based on eddy current principle without the use of an additional energy. Permanent magnets are arranged in the stator so that their opposite poles are placed side by side. When the retarder is deactivated (*Fig. 35*), the magnets are bridged and the magnetic field lines flow through the pole piece. When the retarder is activated, the magnets are pressed. The air lines are connected directly to the pole piece and the magnetic field lines start to flow through rotor. Strong eddy currents are generated which counteract the rotation of the stator and slow down the vehicle [23].

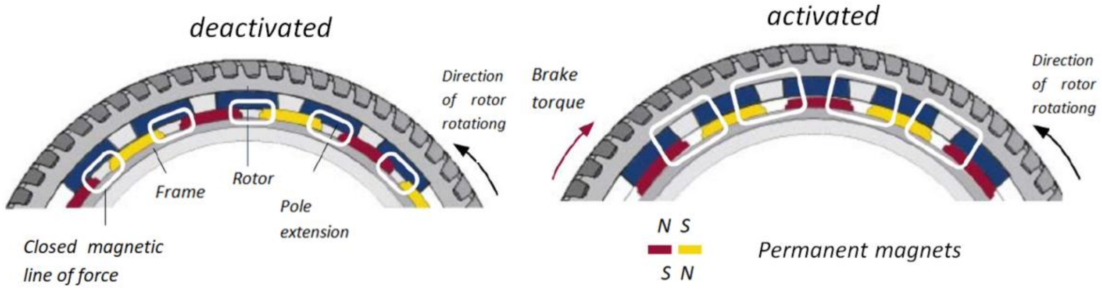
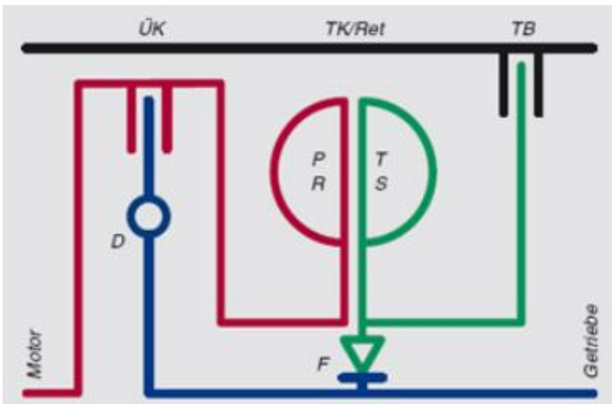


Fig. 33 - Retarder with permanent magnets [23]

5.1.2.5 Voith Turbo Retarder Clutch VIAB

The VIAB combines hydrodynamic coupling and retarder only in one hydrodynamic circuit. The VIAB connects the advantages of hydrodynamics when starting and braking to one device with high efficiency. While at engine motoring, the engine transmits power through the hydrodynamic circuit as well as the free wheel under power to the gearbox input shaft. While braking *(Fig. 35)*, after clutch for blocking convertor is coupled, the Turbine brakes are stopping the turbine wheel. The system becomes a high performance retarder.[25]



Legend

ÜK Converter lockup clutch
D Torsional vibration damper
TK Fluid coupling
P Impeller
T Turbine wheel
R Rotor
S Stator
F Free wheel coupling
TB Turbine brakes

Fig. 34 - Voith VIAB Principle of operation diagram [25]

5.1.3 Engine brakes

"How is possible to change the combustion engine to a powerful brake"

The following chapter is devoted to great attention, because practical part of this thesis is focused mainly on engine brakes. Engine brakes are part of the powertrain, so in comparison to drivetrain where is no engine included. It means, their braking performance directly depends on the **engine speed**, and further depends mainly on **engine displacement**, **layout** and **timing** of valvetrain. Braking power can be controlled, for example by cylinder deactivation or changing valve timing. Engine brakes are used only in diesel engines.

Engine brakes can be classified into two categories:

- Exhaust brakes
- Compression release brakes

5.1.3.1 Exhaust brakes

5.1.3.1.1 Conventional exhaust brake

Conventional exhaust brake was originated for purpose usage in diesel engine to compensate the natural losses of engine which are lower than in petrol engine. It is the most used type of auxiliary brake because is cheap and simple. It can be realized as a variable or fixed flap in exhaust manifold, controlled by actuator in engine non fueling mode. During the exhaust stroke, is exhaust brake creating backpressure against the piston (Fig. 36), which via crankshaft and drive train is transmitted on driven axle and slow down the vehicle. Usually, the exhaust brake is used in combination with another type of auxiliary brake. The power range depends on many parameters of the combustion engine, especially – the number of valves per cylinder and the stiffness of the exhaust valves and of course on engine displacement. [26]

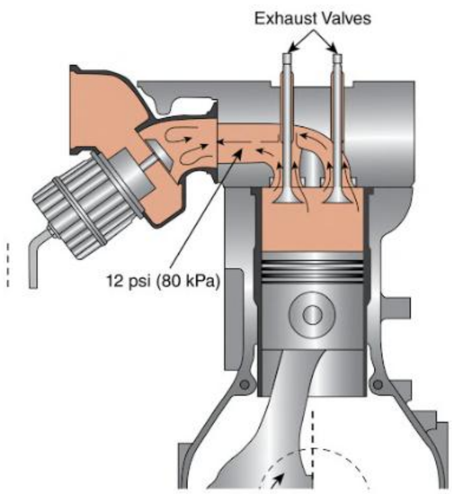


Fig. 35 - Exhaust brake [26]

5.1.3.1.2 VGT exhaust brake

To meet strict emission targets were created turborchargers with variable geometry *(Fig. 39).* This relative new mechanism is a more advanced alternative to turbochargers with fixed geometry. The main advantages of variable geometry are supplying pressurized air to the combustion chamber faster and driving EGR cylinders with faster response time to target to eliminate turbo lag. On the other hand in case of engine braking, the VGTs has gained popularity of using the VGT like a braking device (Fig. 37) and they can be used like an ideal and sophisticated replacement for variable or fixed exhaust flap in the exhaust manifold. [26]

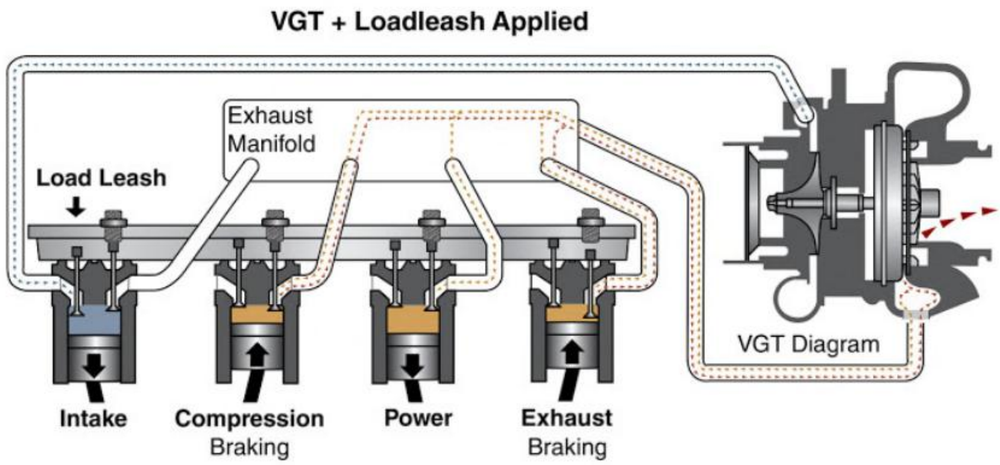


Fig. 36 - VGT operation [26]

A high exhaust and intake manifold pressure can be developed at the same time by closing the distribution vane opening of the VGT (*Fig. 38),* which can result increasing of the turbine speed. As a result, the pumping loss and an engine delta P increase as the VGT vane is closed. Compared with the conventional exhaust brake, the VGT exhaust brake usually has a lower engine delta P and retarding power, but this difference is compesated by higher turbocharger speed and engine flow rate. Reduced thermal load on the components during the braking is a result of the higher air flow which helps reduce the exhaust manifold gas temperature. [27]

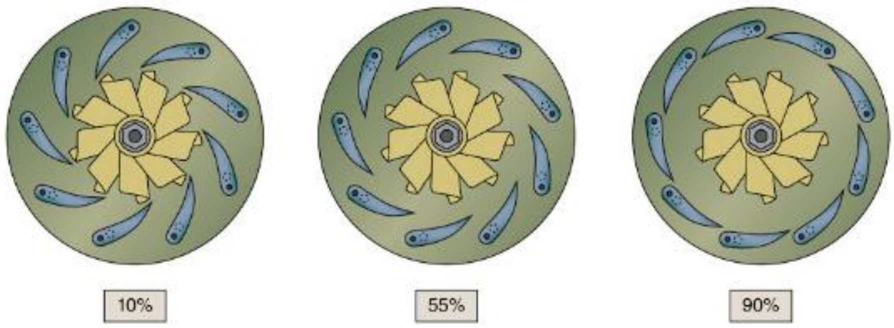


Fig. 37 - VGT Rack position [26]

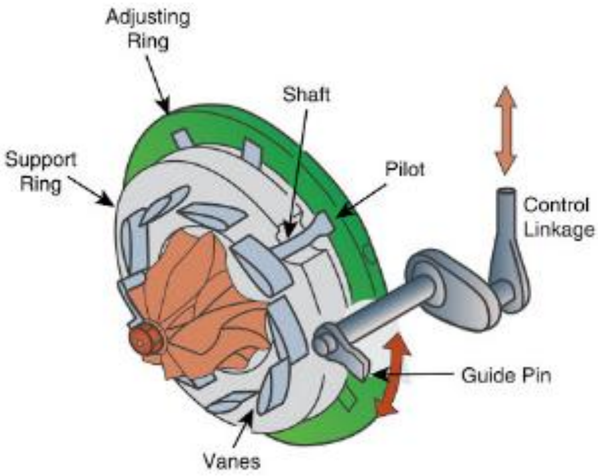


Fig. 38 - VGT assembly [26]

Operation of VGT

Low engine speeds. To provide high torque even at low engine speeds, the boost pressure must be high. To achieve the required high turbine performance, its distribution vanes are adjusted so that the inlet cross section is small. The constriction causes a high flow velocity of the exhaust gas and simultaneously exerts a dynamic pressure of the exhaust gas flow on the outer region of the turbine vanes (larger "arm" - higher torque). Turbine speed increases and thus the filling pressure increases. [6]

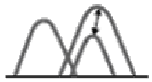
High engine speeds. Distribution vanes of the turbine release a large inlet cross section area, to accommodate large amounts of exhaust gas at high engine speeds. This achieves the required filling pressure, but is not exceeded. [6]

5.1.3.1.3 VVA exhaust brake

Variable valve actuation used as a braking technique provides another means to increase the retarding power. Generally, there are two methods to increase the pumping loss:

- increase *engine delta P*
- reduce volumetric efficiency

The VVA brake uses the reducing volumetric efficiency by throttling the valves to change the cylinder pressure. The cylinder pressure is decreasing during the intake stroke or is increasing during the exhaust stroke. The change of the cylinder pressure is effective for the exhaust valves due to the much greater potential than of an intake VVA brake what is just like the intake-throttle brake. There are many methods how to realise VVA. Every method has their advantages and disadvantages, choose suitable actuation method depends only on customer‘s demands. A general classification of different types of VVA technology is depicted below in *Fig. 40*. [6]

Type		Valve lift characteristics	Phase	Lift	Event	Deacti-vation	Conti-nuous control	Engine perfor-mance	Instal-lation	Cost
With cam	Valve timing control		○	×	×	×	○	Low	Good	Low
	Cam switching		△→ ○*	○	○	○	×→ △*	↓	↑	↑
	Variable valve event and lift control		△→ ○*	○	○	○	○			
Without cam	Hydraulic or electromagnetic drive		○	○	○	○	○	High	Poor	High

* Additional functions made possible by combining with valve timing control.

Legend: ○ Possible; △ Partially possible; × Not possible

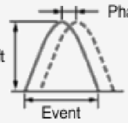


Fig. 39 - General Classification of VVA Technology [21]

5.1.3.2 Compression release brakes

The retarding power of compession brakes is sensitive to many factors including:

- Engine speed
- The number of cylinders which are braking
- Turbine area
- Compression ratio
- Braking valve effective flow area and the valve timing

In compression release engine brakes is used only one valve in engine braking which resulting reducing the load acting on the braking components such as the cam, the injector and the exhaust pushrod. In order to provide a sufficient braking valve flow area, one-valve braking needs to have basically twice the valve lift used in two-valve braking, thus demanding a larger valve-to-piston clearance. The optimum braking valve opening timing also depends on the engine speed. The optimum timing occurs earlier at higher speeds in order to assure adequate blow-down of the cylinder charge. [27]

The key for reach high retarding power is to achieve highest peak cylinder as possible which could be increase by the increasing of compression ratio, intake manifold boost pressure and the amount of gas in the cylinder. BGR which is described in the next chapter helps with achieve high peak cylinder pressure help with increasing of the turbocharger speed as well.

The braking gas recirculation (BGR) theory

The BGR mechanism of has not been clearly explained in the open literature because is relatively new mechanism discovered in recent years. Recirculation of hot gas from compression during the engine braking which is used to supplement to the another cylinder charge from the common exhaust manifold in order to target of reach higher retarding power. It is essentially equivalent of EGR technology used in the non fueling mode during engine brake operation. As shown in Fig. 41 in the exhaust port there are the creating pressure pulses which are trying to open the exhaust valve during the compression stroke to flow the exhaust gas into the cylinder. BGR technology utilizes this to increase retarding power. In BGR is utilizing a pressure differential between the exhaust port pressure pulse and the in-cylinder pressure, there is the braking valve opened in late intake stroke until the early part of the compression stroke to flow the exhaust gas which has increased temperature into the cylinder. The braking valve lift event for BGR can be generated by any of the following:

- a mechanical cam
- a bleeder brake
- a variable valve actuation [27]

"It should by noted the engine volumetric efficiency in compression braking with BGR becomes complex due to the change in the valve events, compared to the case in firing operation. Considering the requirements on both retarding power and component thermal durability, a good design objective for compression brake design can be stated as to achieve simultaneous high engine air flow rate and high exhaust manifold temperature within the design constrains. BGR is the mechanism to achieve this goal in a balanced manner. The ultimate retarding power limit is bounded by allowable peak cylinder pressure and exhaust manifold gas temperature." [27]

Using technologies of VGT or BGR helps move the maximum retarding torque to lower engine speeds. However, at very low engine speeds is the turbine pressure ratio lower or close to 1, therefore on a turbine outlet can be exhaust brake considered. A compromise between the engine constraints braking by change of the valve timing, engine and the turbocharger is required on order to keep all the design requirements. [27]

5.1.3.2.1 Bleeder brake

The bleeder brake is a simplified version of traditional engine brakes. In the bleeder brake, the braking valve is continuously opened by a small stroke either during entire engine cycle or during compression, expansion and exhaust strokes without precisely timed controls for valve opening and closing timing. [28]

When the bleeder brake is turned on, a piston extends to its full stroke and stays there, holding the exhaust open a small, fixed distance throughout the entire four-stroke engine cycle. Since the bleeder brake only holds the exhaust valve open a fixed distance (Fig. 41), it can be designed to not put any load on the camshaft and most of the overhead components. This makes the Bleeder Brake an ideal technology for smaller diesel engines.

- Exhaust valve is held open throughout all four strokes of the engine cycle.
- The compressed air ‘bleeds’ out through the slightly opened exhaust valve during the entire compression stroke.
- The engine pumps by pushing against the valve restriction and against the back pressure.
- 'Bleeding' off compressed air prevents the return of energy to the piston, which slows the vehicle down. [27]

The Bleeder Brake must be combined with either a VGT or an exhaust brake to provide back pressure and optimize the retarding performance. The result of this combination is retarding performance comparable to a traditional compression release engine brake with a noise level similar to an exhaust brake. [28]

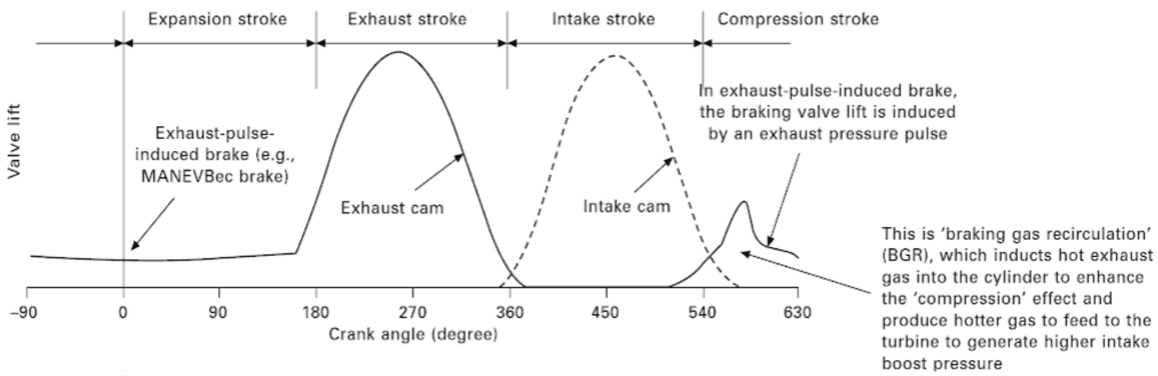


Fig. 40 - Bleeder brake operation [27]

5.1.3.2.2 Conventional CR brake

Compression release engine brakes also sometimes called Decompression brakes are most common on diesel engines with a larger engine displacement. The operation of an engine brake produces a release of the compression as depicted in *Fig. 43* as a CR Event. The compressed air is released from the engine cylinder by fast opening of the brake valve near top dead center TDC of the compression stroke, dissipating the energy used to compress air. This could be compared to dissipating energy of the spring. "*When compressing a spring, normally it will rebound after the pressure compressing it is released. In the engine, not opening the exhaust valve will allow the energy stored in the compressed air to simply push the piston downward during the power stroke, even if no fuel is injected. Opening the exhaust valve dissipates energy used to compress this air.* [26]

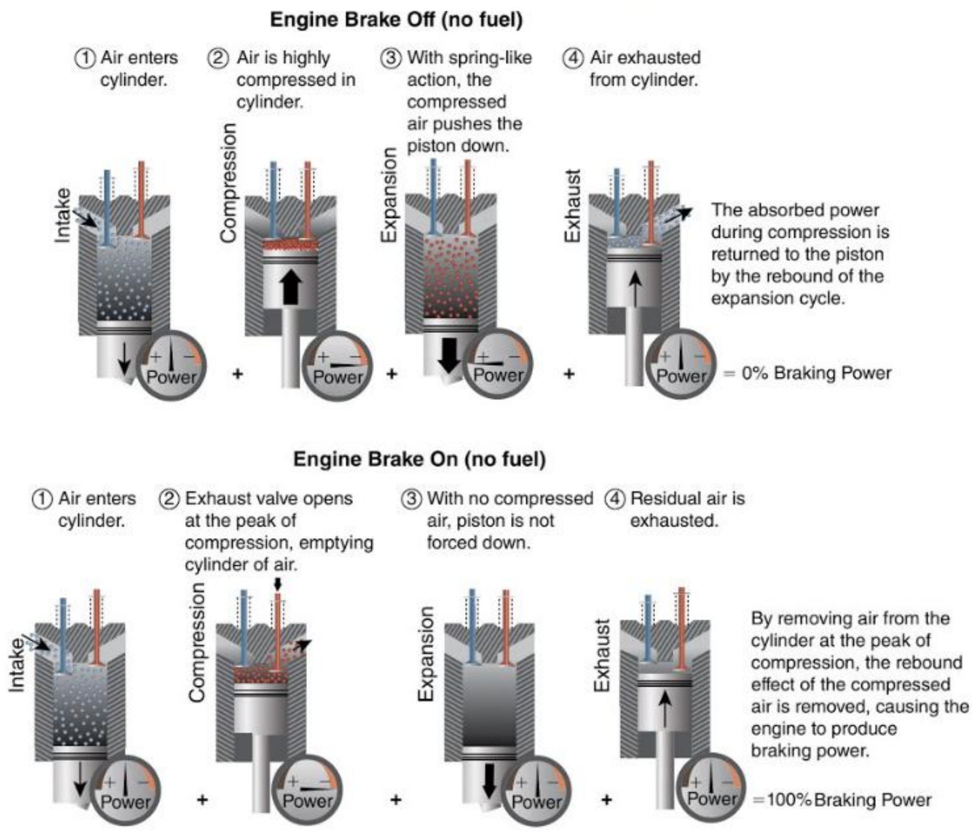


Fig. 41 - Compression release engine brake principle [26]

If the compression wouldn't not be released by opening of brake valve, than the energy from compression would be forcing on the piston during downward movement. During the downward piston movement (Fig. 42) in what would have been the power stroke, no fuel is injected and the engine pulls almost a vacuum in the cylinder as a result of brake the crankshaft instead of driving it. [26]

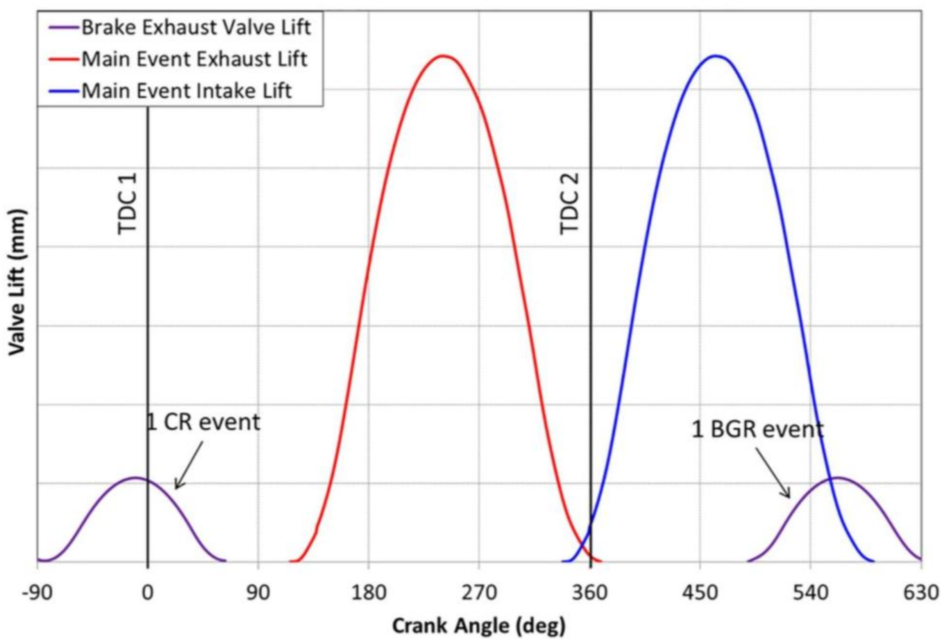


Fig. 42 - 4 Stroke Compression Release Brake Valve Motions [29]

5.1.3.2.3 Two-stroke compression brake

The conventional compression release braking power is limited by various design constraints. Two-stroke brake can be considered to increase the power without violating the design constraints such as the braking assembly loading. A conventional engine brake has one CR and one BGR event for each of the engine's cam rotations. By removing the normal exhaust valve event, the exhaust stroke in the four-stroke engine cycle becomes a second compression stroke from which more retarding power can be obtained via the strong compression-release effect. By using the Two-Stroke engine brake coulde more than 40% braking power increase was achieved compared with the conventional compression release brake. By cylinder deactivation mechanisms the main valve events could be eliminated, allowing the braking rockers to provide two intake, two CR and two BGR events per cam rotation as shown in *Fig. 44*. [27, 29]

The power of Two-Stroke engine brake is similar like of hydrodynamic retarder even larger in lower vehicle velocities, but still significantly with significantly smaller cost. Nowadays, nobody from truck manufacturers use this type of engine brake, but it is assumed, that in the near future, when will be power of convetional Compression release brake insufficient, because of lower resistances of truck, will be this type of engine brake an ideal solution for auxiliary brake of truck.

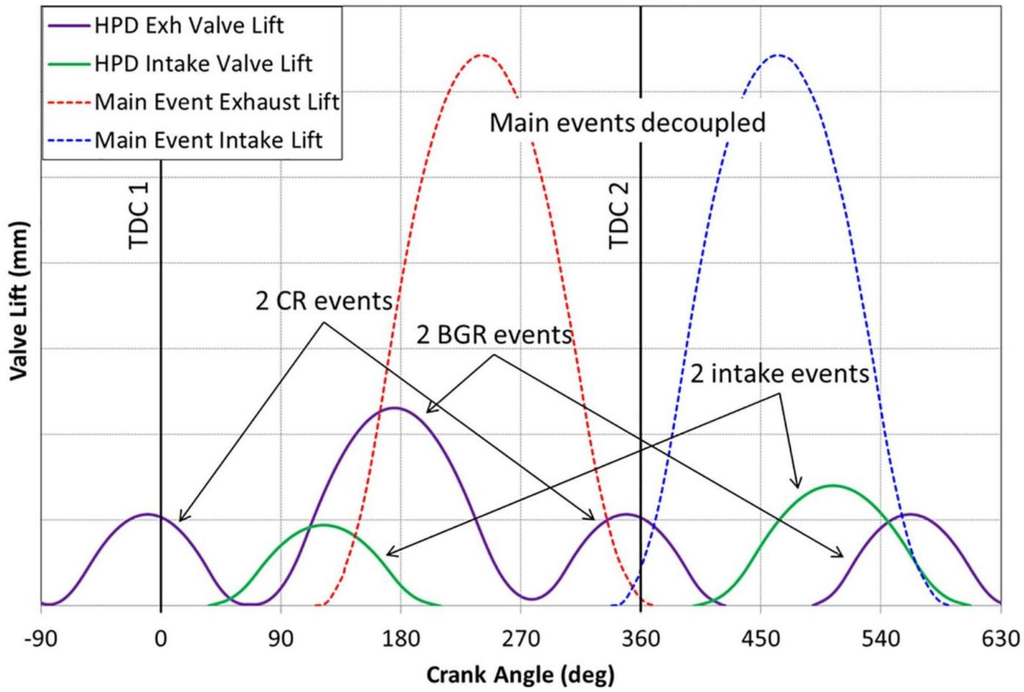


Fig. 43 - 2 Stroke Compression Release Brake Valve Motions [29]

5.1.4 Summary: comparison of auxiliary brakes

It's very difficult to say, which type of auxiliary brake is the best. Like everything in our lives, everything has some advantages and disadvantages, so most often it's up to customer to choose which type is the most appropriate for its purposes. Simple comparison between auxiliary brakes is shown below:

Table 1 - Comparison of engine brake and Drivetrain retarder

Engine/Exhaust brakes	***Drivetrain retarders***
• Practically only advanced valvetrain and exhaust system • Louder becauce of high-frequent exhaust pressure waves • Performance depends on engine speed • No performance in changing gears • More compactness • Worse regulation • Braking torque transmitted through transmission.	• Much higher braking performance than engine firing rated power • Independent of used engine (electromobility) • High acquisition and maintenance cost • Considerably higher weight • Silent • High cost • Takes a lot of space

Table 2 - Comparison of different auxiliary brakes

Type	***Advantages***	***Disadvantages***
Exhaust brake	Simple Activation/Deactivation time 100 to 200 milliseconds (depends on exhaust manifold volume) Loud popping noise when operating (but lower than compression brake) Can increase engine warp-up	Performance is 30 to 70% of rated engine horsepower Loud popping noise when operating Pulsations
Conventional compression release brake	High brake power Relatively low costs Well-tried by years on the market	Noisy due to the exhaust pressure pulses High peak loads of suddenly release of high pressure
Bleeder brake	Simple design and its flexibility Lower additional valvetrain loading than CR Less noisy than CR	Needs to be coupled with Exhaust Flap or VGT
Two-stroke brake (HPD)	Highest braking power from all of the engine brakes High braking torque at medium and low of engine speed range	Highest cost from all engine brakes Complicated valve timing
Hydro-dynamic	Performance can be 110-200% of rated engine horsepower High braking performance Braking performance from vehicle's speed 40km/h	Braking performance is limited by cooling capacity of engine Weight about 300kg Activation/Deactivation time is 1.6 to 2.3 seconds. Loss of fuel economy if acceleration occurs quickly after braking such as stop and go applications Braking performance from vehicle's speed 40km/h The fuel consumption is slightly affected by viscuous friction in the oil circuit.

Electro-magnetic	Performance can be 110-300% of rated engine horsepower Activation/Deactivation time 120 milliseconds Independent of vehicle's cooling system Braking torque available at any time (independent of engine speed) Compatible with ABS (progressive) Braking performance from vehicle's speed 5km/h Energy is dissipated directly to the air. Performance is not limited by the engine cooling system	Weight higher than other retarders (400kg) High energy consumption from battery Need to measure the distance between discs Overheating for long term-braking (braking torque reduction)
Brake with permanent magnts	Continuos use of permanent magnet generates low heat Does not require any auxiliary energy Weight 43kg No interference with cooling system No shortening of the propshaft Maintenance free Simple structure and compact, Almost no power (electromagnetic valve consumes only), saving the braking power, no external power generating exciting Continuous use of permanent magnet itself does not generate heat, well to maintain a stable and lasting braking performance	Compared with electric retarder disc with dual rotors, rotor permanent magnet retarder is a single action, its poor heat dissipation, resulting in difficult braking torque means bigger, permanent magnets may cause loss of excitation temperature is too high , resulting in brake failure; Cannot Binning adjust the braking torque according to road conditions or vehicle speed, the use of inconvenience; Structure is more complex, cost is not high. Currently used only in buses

6.1 Legislation

Legislation is fundamental part in design and innovation. For the reason of high weight of commercial vehicles is necessary to ensure traffic safety by legislation. In general, requirements for braking systems determine legislation *EHK-R13, ES 71/320*.

WEIGHTS AND DIMENSIONS DIRECTIVE

In every country is maximum gross vehicle weight different. Summary of maximum GVW in Europe is shown in **Annex 1.**

*"For international transport, the Weights and Dimensions Directive (96/53/EC) limits the maximum weight of heavy-goods vehicles to 40 tonnes (44 tonnes in combined transport) and their length to 18.75 metres in international traffic. It was amended by Directive (EU) 2015/719, which allows new heavy-duty vehicles with more **rounded** and **aerodynamic cabins** starting in 2022. Besides reducing CO2 emissions, this improves road safety and the visibility and comfort of drivers. On 18 May 2018, the Commission put forward a proposal for a decision to amend the directive so as to bring the starting date forward by three years – from 2022 to 2019."* [30]

AUXILIARY BRAKING SYSTEMS

The main regulation for auxiliary braking systems in trucks is ***Type-IIA test (endurance braking performance)***. This regulation is for every vehicle exceeded 26 tonnes. When the vehicle is able to keep an average speed of ***30 km/h*** on a ***7 per cent*** down-gradient for a distance of ***6 km*** only by engine without using the service brakes, vehicle don't need an auxiliary braking device. Detailed text of this regulation is shown in **Annex 3.** [31]

BRAKE DUST EMISSIONS

While exhaust emissions from vehicles have declined since 1990 due to vehicle technology advances, such as particulate filters, the exhaust emissions of particulate matter due to brake and tire wear are increasing. At present, non-exhaust sources account for a large proportion of total particulate emissions from vehicles - about half of PM10 and one third of PM2,5.

Every braking action creates brake dust through friction on the brake disc and brake pads. This mainly comprises particulates and due to the small particle size is harmful to health and the environment. The brake dust contributes considerably towards the pollution caused by particulates in road traffic. More than 90% of the brake dust consists of ultra-fine particles which have a negative effect on human health. A study from the World Health Organization came to the conclusion that 92% of humans live in areas where air pollution is above the permissible level. [31]

<u>Simple shown of calculation of saving brake dust from brake pads:</u>

24kg per service brake replacement by using hydrodynamic retarder

Dust produced over a lifetime of 1500000km and 40t truck:

- Without Retarder: 180 - 360kg
- With Retarder: 36 - 72kg [32]

Therefore, it is assumed that legislation will limit the emissions from brakes in the near future.

The specific brake lining wear per 1 km for heavy trucks is about 50 to 80 mg/km. For comparison, the specific passenger car abrasion is from 10 to 20 mg/km. [33]

CO_2 EMISSION STANDARDS FOR HEAVY-DUTY VEHICLES

The average CO_2 emissions from new heavy-duty vehicles in 2025 would have to be **15 % lower** than in 2019. The overall proposed target is translated into binding CO2 emission targets in grams of CO2/km for each manufacturer, taking into account the composition of its fleet, including technical and business characteristics. Manufacturers would have full flexibility to balance emissions between the different groups of vehicles within their portfolio. For **2030**, the proposal sets an indicative **reduction target** of **at least 30 %** compared to 2019. While the 2025 target could be met by deploying readily available cost-effective technologies, achieving a more ambitious 2030 target would require the implementation of new technologies that are not yet on the market. [30]

*"The European Automobile Manufacturers' Association (ACEA) welcomes the introduction of CO_2 standards for trucks, but considers that a realistic ambition level would be a **7 %** CO_2 **reduction** by **2025** and a **16 % reduction** by **2030**. ACEA's position paper on CO_2 standards calls for a technologyneutral regulation and advocates an integrated approach to reducing heavy-duty vehicles' CO_2 emissions, which addresses not only the vehicle but also freight logistics, driver training, maintenance, tyres and alternative fuels."* [30]

It will have impact for brake performance of engine brakes, because this estimated downisizing and downspeeding of combustion engines that has a smaller natural retarding power. In case of hybrid vehicles, there will be probably used small combustion engine, which has a small natural retarding power. But note, in case of hybrid truck is possibility to utilize recuperation of energy for braking. Electric engines have larger brake potential than combustion engines, but there are many problems with recuperation energy in batteries for long time but this issue goes beyond this thesis.

2. PRACTICAL PART

“What is the tax for improved efficiency”

In this part are apllyed all of aquired knowledge from teoretical part. The main target for practical part is study of near future of auxiliary braking systems and impact of lower drive resistances of truck. Mainly the study of potential of Two-Stroke engine brake on heavy truck with lower driving resistances because it is assumed, that conventinal Compression release engine brake will be insufficient in the future. Therefore, Two-Stroke engine brake could be ideal compromise between insufficient Compression-release engine brake and expensive Hydrodynamic retarder. Sensitivity analysis of Two-Stroke engine brake is done in inline 6 cylinder diesel engine brake model in GT-POWER. Data obtained from GT-POWER are used in parametrical tractor-trailer model in GT-SUITE to detect the responses on vehicle. Monitored parameter is usage of friction service brakes, therefore, the main target is minimize its usage because of expected brake dust limit legislation and for short service intervals as possible and of course, because of safety on road. For quick static comparison of different types of engine brakes was created “Braking power calculator” in MS-Excel.

Like everything in our lives, there is some balance *(Fig.45).* It's simple, when an efficiency of vehicle is improved for minimize the fuel consumption and lower emissions, vehicle has a lower driving resistances. But on the other hand of the scale are lower natural retarding forces and more performance braking systems of vehicle.

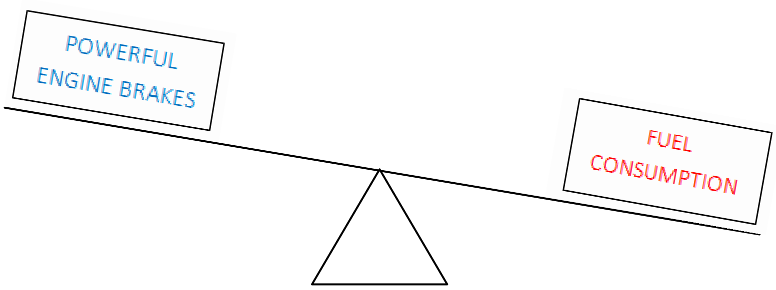


Fig. 44 - Balance in innovations of trucks

A typical heavy duty Truck from 1990 requires additional about **110 kW** of braking power to maintain a downhill control of highway speeds. How much braking power will be teroetically needed in the future? The theoretical study to find the answer is in the following chapters.

1.2 GT-SUITE model description

There is a theoretical potential for improve a truck efficiency of 35%. Theoretically future truck could be changed as follows:

- Drag reduction about 30%
- Rolling resistance about 9%
- Drivetrain efficiency about 3%
- Engine losses about 25%

GT-model (*Fig. 46*) is based on Tractor-trailer example, which was customized to demands of testing brake systems of the vehicle.

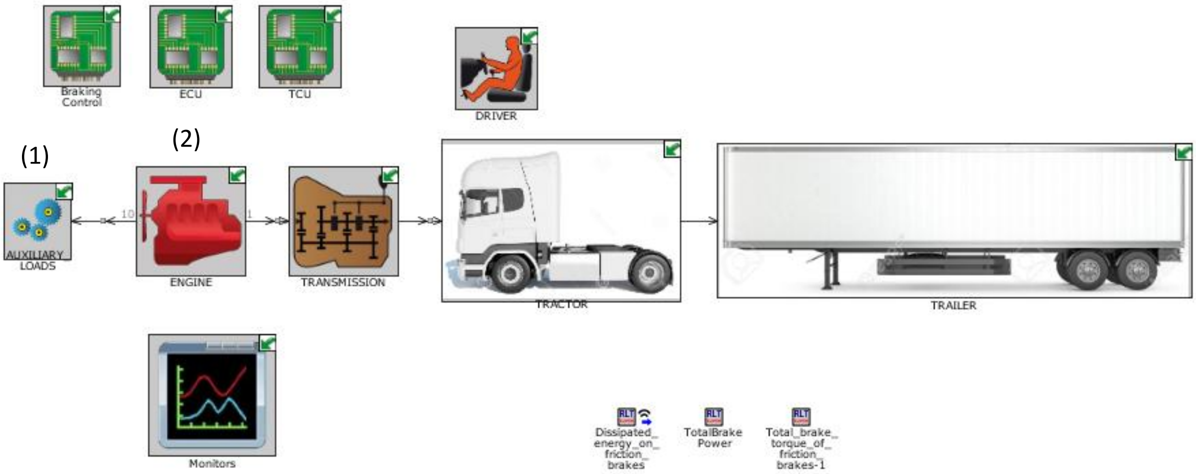


Fig. 45 - GT-SUITE model environment

The model is built that way, that the user change basic parameters and test types in case setup (*Fig. 47)* by super parameters, so user can choose from predefined super parameters like for example trailer type: (Classic/Tanker/Teardrop trailer) or fill the values of drag coefficient manually.

Main | EngineBrake | Engine | Retarder | Trailer | Tires | Test_type | Air_Conditioning | Calculations | Num_of_tire | Tractor | All

Parameter	Unit	Description	Case 1	Case 2	Case 3
Case On/Off		Check Box to Turn Case On	☐	☐	☐
Case Label		Unique Text for Plot Legends	Current 2-stroke	Optimized 2-stroke	HIGH Resistances Retarder ...
Engine		Engine type	Downsized_with_friction	Downsized_with_friction	Downsized_with_friction
Retarder		Retarder Toggle	Retarder_Off	Retarder_Off	Retarder_On
EngineBrake		Engine Brake type	2-stroke_current	2-stroke_future_optimized	Engine_brake_OFF
ret_on-off		Constant or Dependency Refer...	0	0	0
Tires		Truck tire type	Low_friction_tires	Low_friction_tires	Medium_tires
Trailer		Trailer type	DonBur_Teardrop	DonBur_Teardrop	Classic
Test_type		Initial parameters and paramet...	shifting_test	Type_IIA_(legislation)	Type_IIA_(legislation)
Roof_fairing		Improving the aerodynamics	YES	YES	NO
Deturbulator		Improving the aerodynamics	YES	YES	NO
Air_Conditioning		Air Conditioning On/Off	Off	Off	Off
diff_eff	fraction	Differential_Efficiency	0.99	0.99	0.97
trans_gear_eff	fraction	In-Gear Efficiency	0.97	0.97	0.96

Fig. 46 - Case setup in GT-SUITE model

TRACTOR-TRAILER (1)

Truck will be always the "box" on the wheels used for carrying maximum goods as possible. There is not much possibilities to achieve ideal aerodynamic shape and has a super low friction because truck will always have a large mass. The most important improving are shown below.

<u>In the future model, I consider follows improving:</u>

- More rounded cab edges and the front "nose"
- Deturbulator
- Roof fairing
- Instead of classic mirrors using the cameras
- Teardrop trailer
- Side fairing and wheel covers
- Single tires on drive axle with lower friction
- More efficient differential

SERVICE BRAKES in (1)

It isn't considered to change of service friction brakes in the future, because they are on their performance limit nowadays. In tractor-trailer example were brakes built only like a brake template which generates braking torque to wheel/axle depends on brake pedal position. It is used MS Excel *braking power calculator* and calculated brake torques as depicted in *Fig. 48*, that were used in GT-SUITE model to have approximal real brake characteristics. Air disc brakes are used in calculations and in GT-SUITE model.

Input parameters

Pressure of air brakes	*800000*	*Pa*
Area of brake lining	*0,117*	m^2
Coefficient of friction	*0,4*	-
Disc effective radius	*0,132*	*m*

$F_n = p \, . \, A \qquad F_b = 2 \, . \, \mu \, . \, F_n \qquad T_b = F_b \, . \, r_e$

Maximum brake torque on wheel

Front	*9884,16*	*Nm*
Rear	*9884,16*	*Nm*
Trailer	*19768,32*	*Nm*

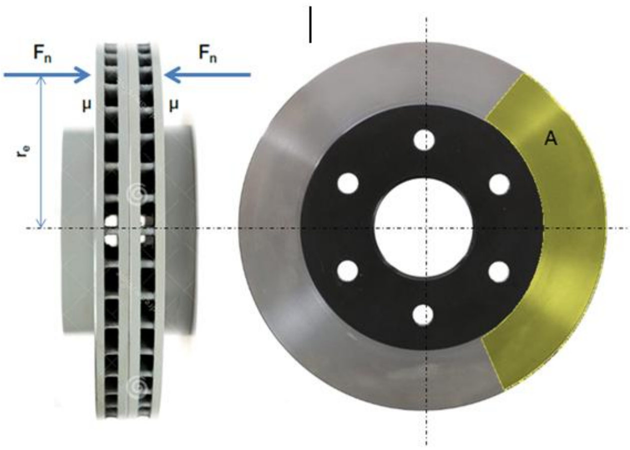


Fig. 47 - Calculation of service brakes

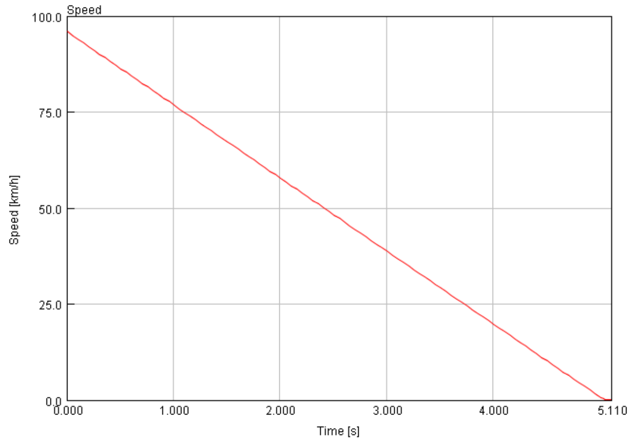


Fig. 48 - Vehicle speed on maximum brake torque

In Fig. 5 is depicted stopping distance of truck with disc air brakes from initial speed 96 km/h in range from 64 to 67 meters. Result of stopping distance in Fig. 49 of parametrical model approximately follows the reality. Result of this calculation is for ideal and new brake lining, it means is not taking into account the brake fade effect and wear of linings, but for for relatively short dynamic tests, which are used in this thesis is it sufficient. In braking distances calculations is used simple compensation of brake fade phenomenon. In this compensation (Fig. 50), are CoF and brake torque of friction brakes gradually decreasing with braking time (Chart 2) in addition of brake pedal position for simple approximate the brake fade phenomenon. Fig. 51 shows, assuming a given wear factor that the vehicle is near of the maximum limit of regulation (76m) after 80000km mileage without using an auxiliary brake during service life of brake linings.

Real brake torque = Brake torque · Wear factor(Mileage) · Fade factor(t)

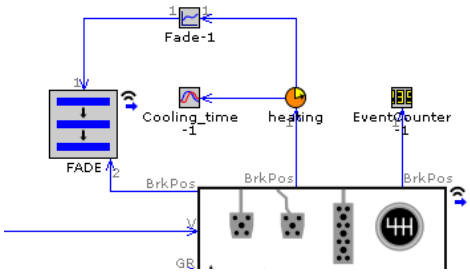


Fig. 49 - Brake fade approximate model

0km mileage (Wear factor 0,997*)

Distance-Speed		
Instantaneous Vehicle Speed	km/h	9.874888E-4
Instantaneous Vehicle Position	m	67.97857
Total Distance Traveled	m	67.97857
Average Vehicle Speed	km/h	47.426907
Maximum Vehicle Speed	km/h	96.0
Average Vehicle Acceleration	m/s^2	0.0
Maximum Vehicle Acceleration	m/s^2	-0.0041119014
Time to Max. Acceleration	s	5.16
Average Vehicle Deceleration	m/s^2	-5.1672297
Maximum Vehicle Deceleration	m/s^2	-5.599445
Time to Max. Deceleration	s	1.0835244E-4

80000km mileage (Wear factor 0,9)

Distance-Speed		
Instantaneous Vehicle Speed	km/h	7.4516603E-4
Instantaneous Vehicle Position	m	74.84646
Total Distance Traveled	m	74.84646
Average Vehicle Speed	km/h	47.18866
Maximum Vehicle Speed	km/h	96.0
Average Vehicle Acceleration	m/s^2	0.0
Maximum Vehicle Acceleration	m/s^2	-0.0012827681
Time to Max. Acceleration	s	5.71
Average Vehicle Deceleration	m/s^2	-4.669511
Maximum Vehicle Deceleration	m/s^2	-5.1167665
Time to Max. Deceleration	s	1.0835244E-4

Fig. 50 - Impact of wear and fade factor

**Wear factor of new lining is lower than 1 because new brake lining are not trimmed from usage.*

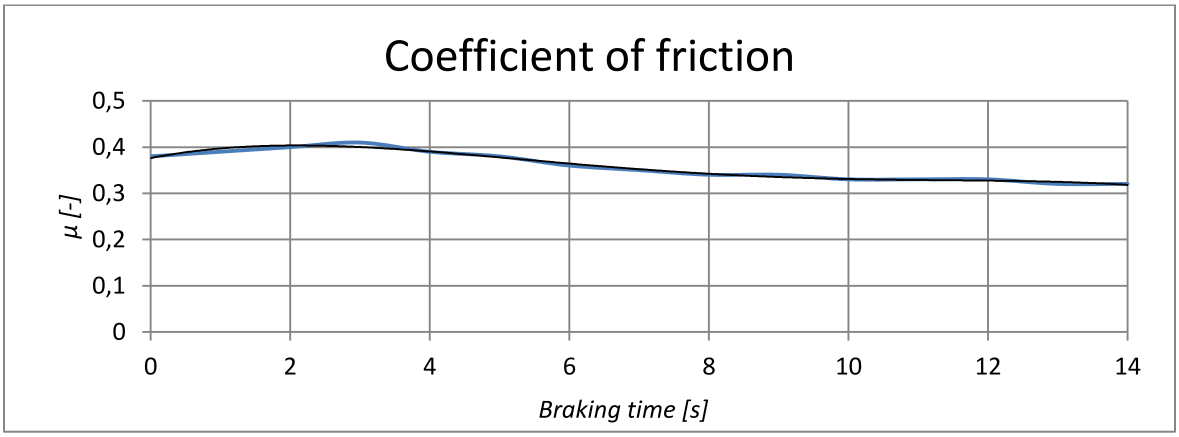


Chart 2 - Decreasing of Coefficient of friction during the braking time

TRANSMISSION (3)

Radical change in truck transmissions technology is not expected. Only parameters, which could be changed in the future are the gear shift duration time, that I suppose, it will be shorter in the future and the efficiency of gears will be higher.

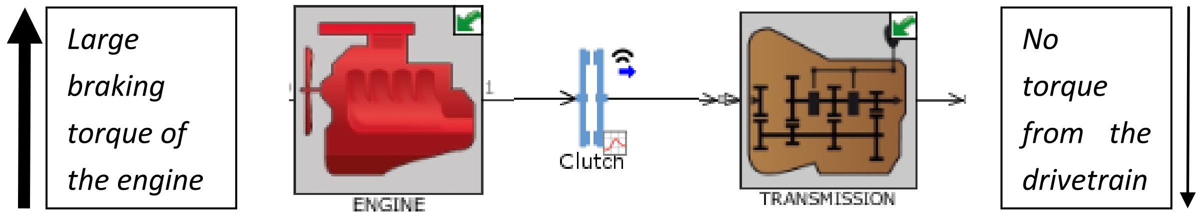


Fig. 51 - Engine brake has to be deactivated during gear shifting

The main problem for heavy duty transmissions without a torque converter or dual clutch is **engine brake deactivation during shift**. When the engine brake is activated and engine speed is decreasing, transmission is going to downshift and the clutch will be disengaged during shift. But when clutch is disengaged and engine brake is on, the large amount of load from drivetrain is disengaged (*Fig. 52*) from engine and engine speed is rapidly decreasing because of large braking torque of the engine. In the model is the engine brake deactivating when the engine speed is lower than 1600rpm or higher than 2300rpm and gear shifting range is set from 1500 to 2400 rpm.

It is chosen robotised 18-speed manual transmission *RT 18 (Fig.53)* from company *Eaton*, because more gears is better for vehicle efficiency and for use maximum potential from engine brake. It have to be used same transmissions in current and in future model because it will be inadequate to use different number of gears or gear ratios, then models would have a different engine speed at same vehicle speed, so different power from engine brake as well.

RT-18 Gear Ratios

Overall Ratio	Forward Gears																	
	LL	L	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16
19.72	14.4	12.29	8.56	7.3	6.05	5.16	4.38	3.74	3.2	2.73	2.29	1.95	1.62	1.38	1.17	1.00	0.86	0.73
% Step	17	44	17	21	17	18	17	17	17	19	17	20	17	18	17	17	17	--

Fig. 52 - Eaton RT-18 Transmission [35]

HYDRODYNAMIC RETARDER in TRANSMISSION (3)

It is chosen one of the most powerful hydrodynamic retarders nowadays. Intarder means “offline” retarder, where is added a gear to increase retarder speed for more power. Power of drivetrain retarder depends on vehicle speed, dynamic radius of tire and the gear ratio of differential. In GT-SUITE model in retarder map are brake torque and driveshaft speed the input values. So the data from Fig. are recalculated to driveshaft speed. There is no considered decreasing of brake power by longer use of intarder.

It has large brake power, but only at higher vehicle speed. Crusing on mountain passes with high road decreasing is usually an average vehicle speed about 50km/h. At this vehicle speed is the brake power of MX Engine brake power (*Fig. 56*) even similar like of ZF Intarder (*Fig. 54*).

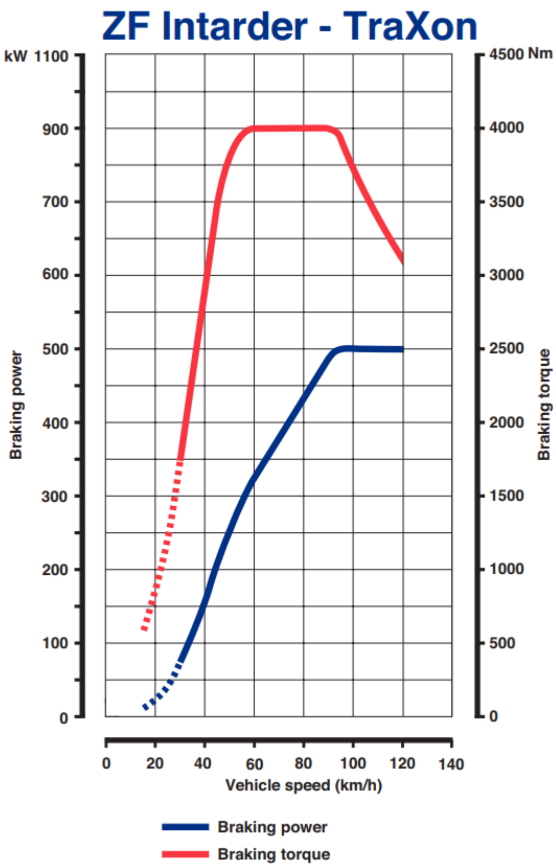


Fig. 53 - Characteristics of ZF Intarder [36]

ENGINE (4)

GT-POWER model of Two-stroke engine brake represents approximate PACCAR MX-11 (10,8l) engine *(Fig. 55)* that is modified in chapter 2.2. Nearest template of engine in GT-SUITE is inline 6-cylinder 10,4l turbocharged, so this engine template was used and engine maps, inertia, engine displacement and CPU were corrected to 10,8l engine. This engine is used in **current** model.

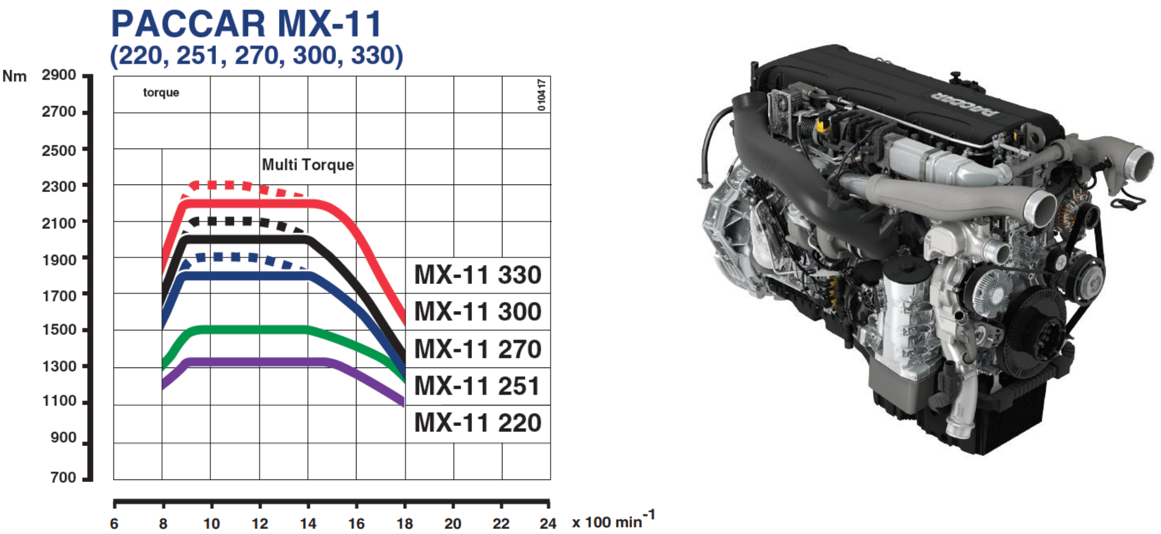


Fig. 54 - MX-11 Engine [36]

Representing the engine and engine brake in the **future** model is this engine downsized in GT-POWER to 8,4l engine displacement. This engine downsizing is detailed described in chapter 2.2. In GT-SUITE is the nearest template of engine 8,5l inline 6 cylinder with VGT so engine template was corrected same way like as mentioned above.

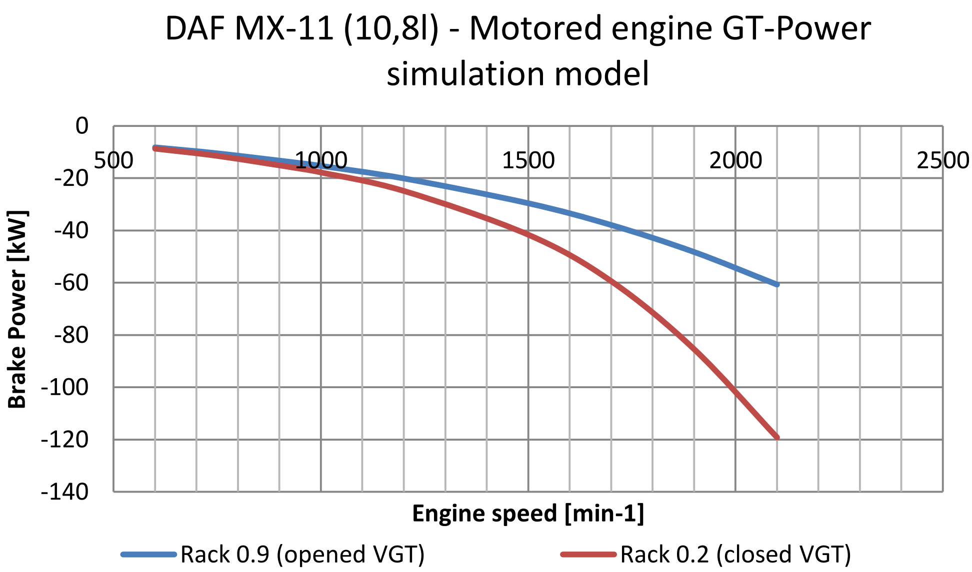


Chart 3 - Friction losses in MX-11 Engine

AUXILIARY LOADS (4)

In this assembly, there are modeled all of auxiliary loads which are consuming torque from engine to their own drive such as air conditioning, fan, alternator, steering etc. These torques are helping to slow down the vehicle when are active. So in the model is possibility to switch on/off air condition and for example. These auxiliary loads will be always in vehicle with combustion engine and will be not rapidly changed in the future. These auxiliary loads helps with braking with the maximum torque about **150Nm**.

ENGINE BRAKE in (4)

In same way, because of simplicity, I modeled engine brake torque like an auxiliary load torque of engine as well. Simulation of time delay of the onset of the torque is realized by time constant.

Nowadays, the most used type of engine brake in heavy duty vehicles is Compression release engine brake. Most often is used in combination with exhaust flap or with VGT for more performance. **So It was decided to use in current truck *MX-11 Engine Brake (Compression release with exhaust flap).***

The customers with large engines aren’t forced to have more powerful auxiliary brake, because for current driving resistances is CR sufficient. It could be seen, that the maximum brake power of MX-11 Engine Brake is even higher than in fueling mode.

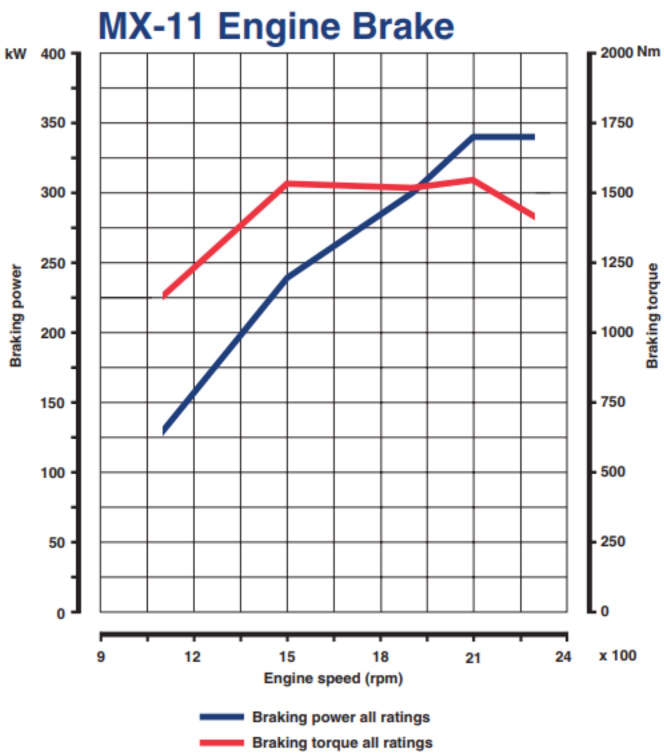


Fig. 55 - MX - 11 CR+EX Engine brake [36]

In the future truck, it is considered, in trucks will be used ***Two-stroke engine brake***, because Compression release brake will be probably insufficient. The market will be forced by requirement for more powerful brakes and it is more efficient to invest a bit more money to better engine brake, which is still significantly cheaper than drivetrain retarder.

It is neccessary to realise, that the data of engine brake power from manufacturers are measured on the output of the engine. That means sum of auxiliary brake power of engine brake and natural retarding power of engine. But in template **ENGINE (4)** is retarding power of engine included from mechanical and friction output maps.

To avoid carry double engine losses into model, is necessary, to set this maps in engine braking mode to **zero value** for zero accelerator pedal position, when are used data from manufacturers.

When are used data from GT-POWER model of Two-stroke engine brake, it are used indicated values of torque and the friction losses *(Chart 3)* are assumed in template **ENGINE (4)** maps in engine brake mode.

Summary of all parameters for current and future truck for quick comparison are shown below in the *Table 3*:

Table 3 - Values used for calculations in Current and Future truck

	Current	***Future***	
Vehicle mass	*40000*	*40000*	*[kg]*
Vehicle frontal area	*8*	*7,5*	*[m^2]*
Drag coefficient	***0,9***	***0,6***	*[-]*
Coefficient of rolling resistance	***0,012***	***0,008***	*[-]*
Drivetrain efficiency	*0,93*	*0,96*	*[-]*
Final-drive ratio	*2,64*	*2,64*	*[-]*
Dynamic radius of tyre	*0,49*	*0,49*	*[m]*
Engine displacement	*10,8*	*8,4*	*[l]*

2.2 Engine downsizing

Because of strict new regulations from European Union to decrease CO_2 emmisions of diesel combustion engines, it is assumed, that manufacturers in the future choose the way of extreme downsizing. It is used provided GT-SUITE model of Two-stroke engine brake, which was next modified. Parameter, which has to be respected in downsized engine is **bore/stroke ratio**. Engine was downsized *(Table 5)* to 77,7% of original engine displacement (*Table 4*). In this ratio I changed the main parts of the engine - the valves, pistons, connecting rod and turbocharger. This method is not entirely precise, but for the purposes of this diploma work is sufficient. Effect of downspeeding is taking account to lowering maximum engine speed in engine braking mode from 2400rpm to 2300rpm.

The lenght of connecting rod *(Fig. 57)* has to be changed about the value of difference of change stroke length to avoid the collision between cylinder and engine head. The position of valves (x,y) was changed too, to keep same thickness ratio of material in the cylinder head *(Fig. 57)*.

Table 4 - Original MX-11

Bore	0,123	[m]
Stroke	0,152	[m]
Compression ratio	18,5	[-]
Number of cylinders	6	[-]
Engine speed	2400	[min^{-1}]
Valve reference diameter (exhaust)	0,031	[m]
Valve reference diameter (intake)	0,035	[m]
Exhaust valve diameter	0,039	[m]
Intake valve diameter	0,042	[m]
Bore/stroke ratio	0,809	[-]
Engine displacement (1cylinder)	1,806	l
Engine displacement (engine)	10,837	l
Mean piston speed	12,16	[m/s]

Table 5 - Downsized MX-11

Bore	0,113	[m]
Stroke	0,14	[m]
Compression ratio	18,5	[-]
Number of cylinders	6	[-]
Engine speed	2300	[min^{-1}]
Valce reference diameter (exhaust)	0,027	[m]
Valve reference diameter (intake)	0,031	[m]
Exhaust valve diameter	0,034	[m]
Intake valve diameter	0,037	[m]
Bore/stroke ratio	0,807	[-]
Engine displacement (1cylinder)	1,404	[l]
Engine displacement (engine)	8,424	[l]
Mean piston speed	10,7	[m/s]

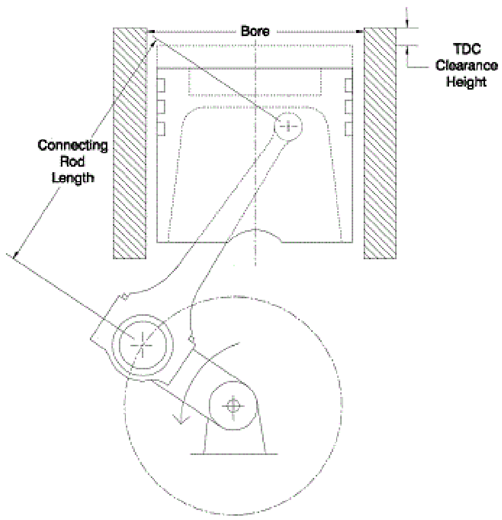


Fig. 56 - Cylinder geometry

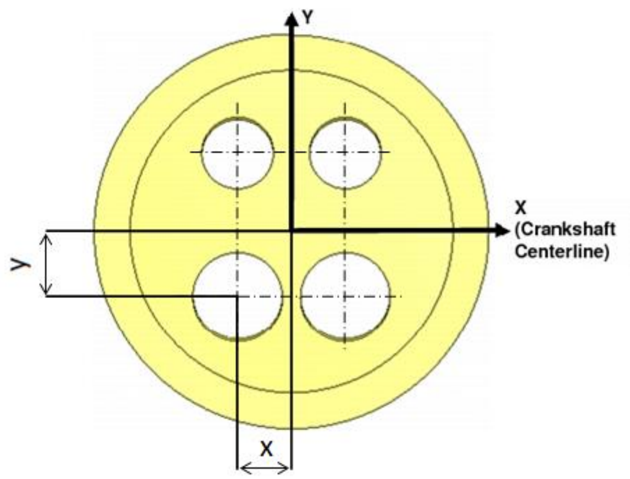


Fig. 57 - Valves position

To avoid destruction of parts of engine is maximum cylinder permissible pressure reducted. Exceeding pressure drop between cylinder and manifold could affect not wanted valve opening. By spontaneous valve opening in different cam angle like is predesribed could result engine destruction by contact of valve and piston. Maximum inlet turbine temperature is limited by material characteristics and exceeding of this value could damage the turbocharger. Exceeding safety distance between cylinder and pistok could. I take into account all of this constraints, because this constraints radically obtain maximum brake power of engine. The characteristics of Two-Stroke engine brake on downsized engine were obtained in GT-POWER model.

The Compression release engine brake on downsized engine characteristics was estimated by consultations and the result is shown below:

Assumptions of downsizing for MX-11 Compression release Engine Brake

For *MX Engine brake (CR+EX)* was used assumption, that downsizing will decrease brake torque about 18%.

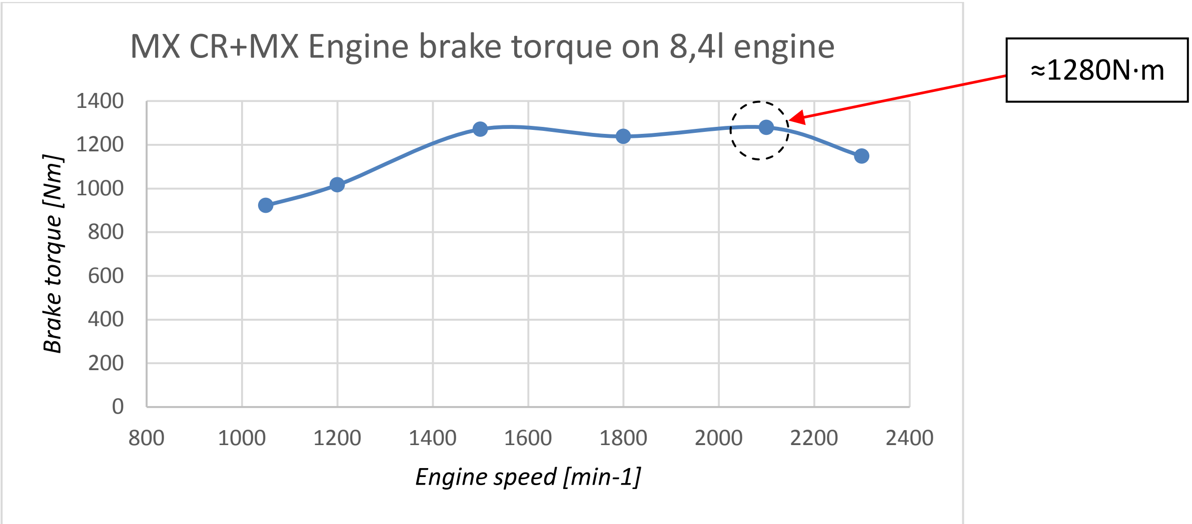


Chart 4 - Brake torque of CR+EX

Additional use of Exhaust brake (EX) change the course of the Compression release (CR) torque curve, especially in low to middle rpm range. The EX should increased the maximum torque about 20%. However, for verifying the correctness of the downsizing was used data of CR from Jacobs Vehicle systems shown in *Chart 5*. Comparison maximum brake torque from Jacobs Vehicle systems (CR) which would be improved of 20% should be approximately same like the maximum torque (CR+EX) from company PACCAR (*Chart 4*).

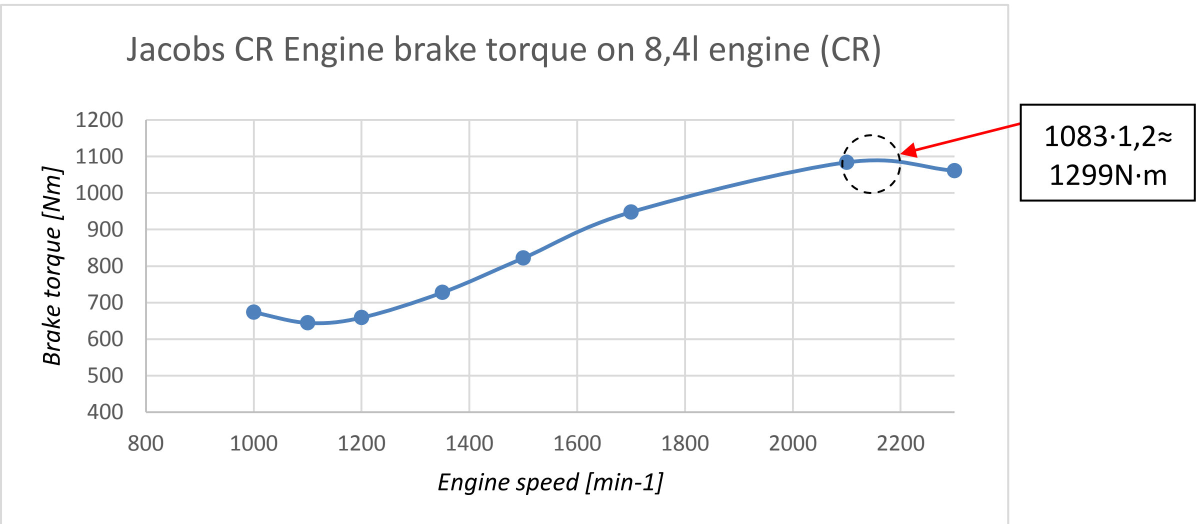


Chart 5 - Brake torque of CR

In case of MX-11 CR+EX *(Chart 4)* is the maximum brake torque about 2100rpm's in compare to *Two-stroke engine brake* to (Chart 7) where is the maximum torque in 1000rpm's.

Results of downsized 8.4l engine for Two-stroke engine brake:

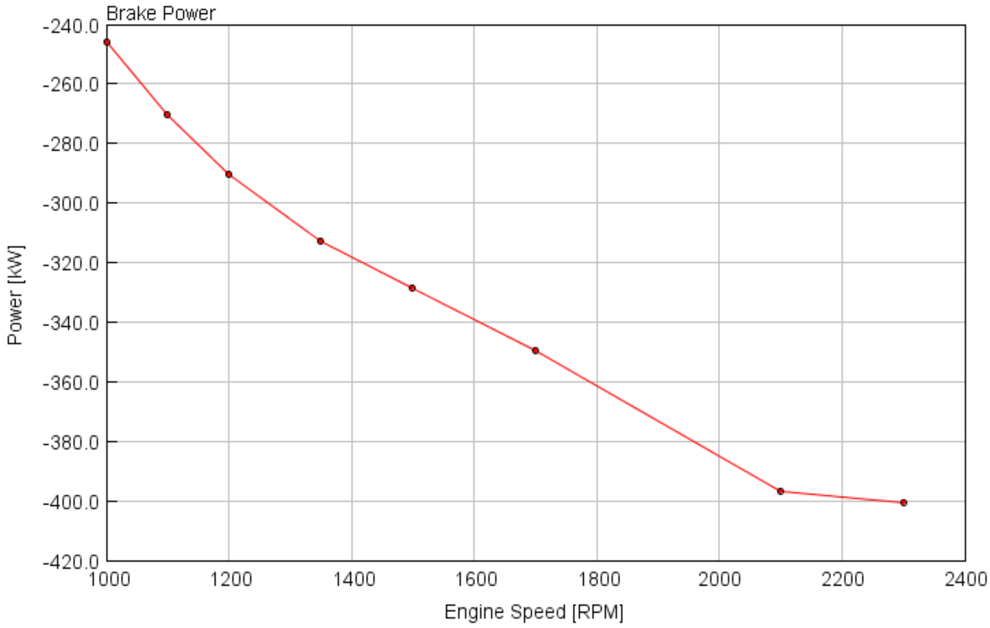


Chart 6 - Two-Stroke engine brake torque after downsizing

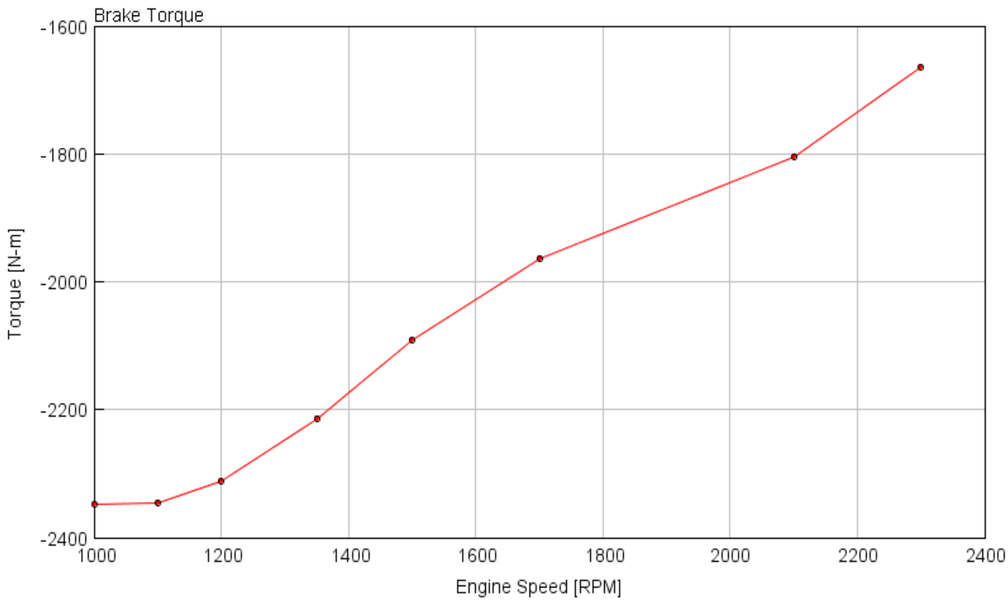


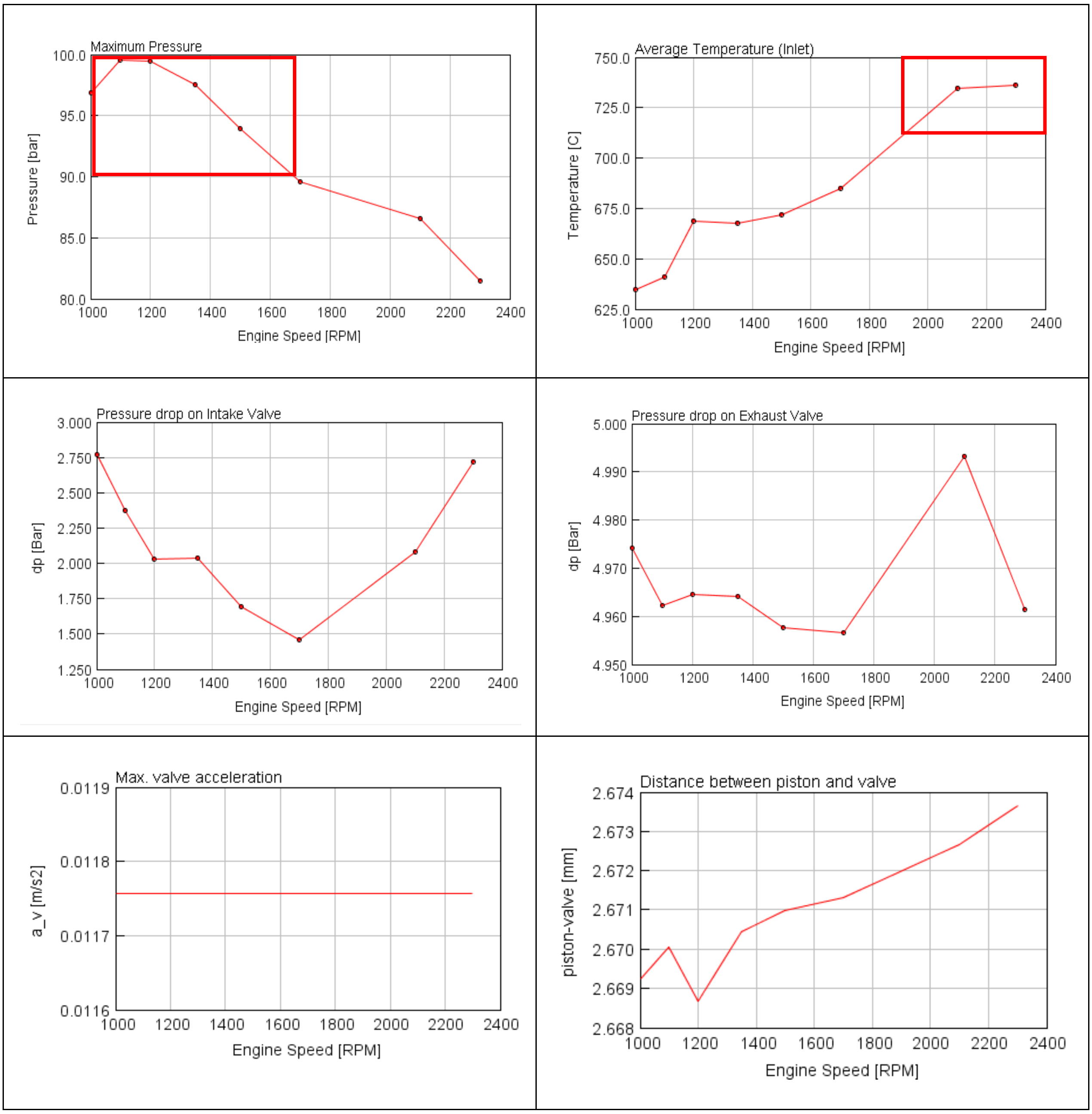
Chart 7 - Two-Stroke engine brake power after downsizing

After the simulation was obtain, that some of the permissible values were exceeded. It could be seen from the Table 6 that the maximum permissible pressure has been exceeded from engine speeds of *1700rpm* below. From *1100* to *1200 rpm's* was the maximum pressure about *10bar* higher than permissible value from the engine manufacturer. So engine brake power(torque) will be lower in these engine speed range and has to be optimalised (chapter 2.3) to find maximum brake power in regard to keep maximum cylinder pressure *90bar*. Another monitored constraints are satisfactory.

***Current Constraints** of tested engine that cannot be exceeded are as follows:*

1) Maximum cylinder pressure 90bar
2) Maximum average temperature on turbine inlet is 710˚C (for short time event 740 ˚C)
3) Maximum pressure drop on Intake valve is 3bar
4) Maximum pressure drop on Intake valve is 5,2bar
5) Maximum valve acceleration is 0,012mm/deg^2
6) Safety distance between piston and valve >1,2mm
   (1x angle gap between exh. cam lobes > 97 CA (6ms at 2700rpm))

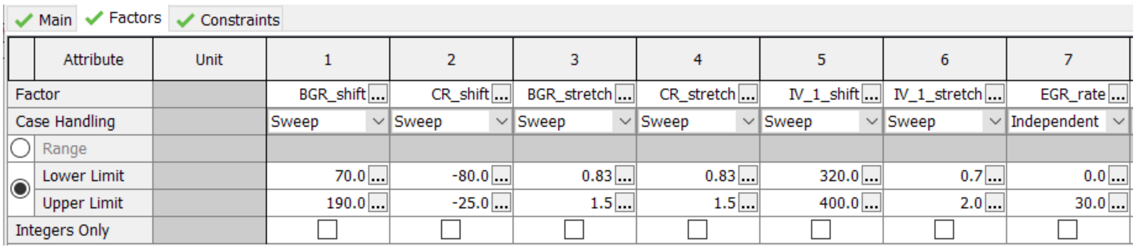
Table 6 - Monitored current constraints of downsized engine for Two-stroke engine brake



(The angle gap between exh. cam lobes = 117,5 CA)

3.2 Optimization of downsized engine (Two-stroke engine brake)

Integrated Direct optimizer finds out maximum brake power *(Chart 8)* like a best combination of factors and set limits which are shown in *Fig. 59* with respecting monitored parameters from *Table 6*. Search algorithm is Genetic and values for Population Size is set for 30 and Number of Generations for 10. Results of the best design are shown in Table 7.



Attribute	Unit	1	2	3	4	5	6	7
Factor		BGR_shift	CR_shift	BGR_stretch	CR_stretch	IV_1_shift	IV_1_stretch	EGR_rate
Case Handling		Sweep	Sweep	Sweep	Sweep	Sweep	Sweep	Independent
Range								
Lower Limit		70.0	-80.0	0.83	0.83	320.0	0.7	0.0
Upper Limit		190.0	-25.0	1.5	1.5	400.0	2.0	30.0
Integers Only								

Fig. 58 - Factors and limits in design optimizer

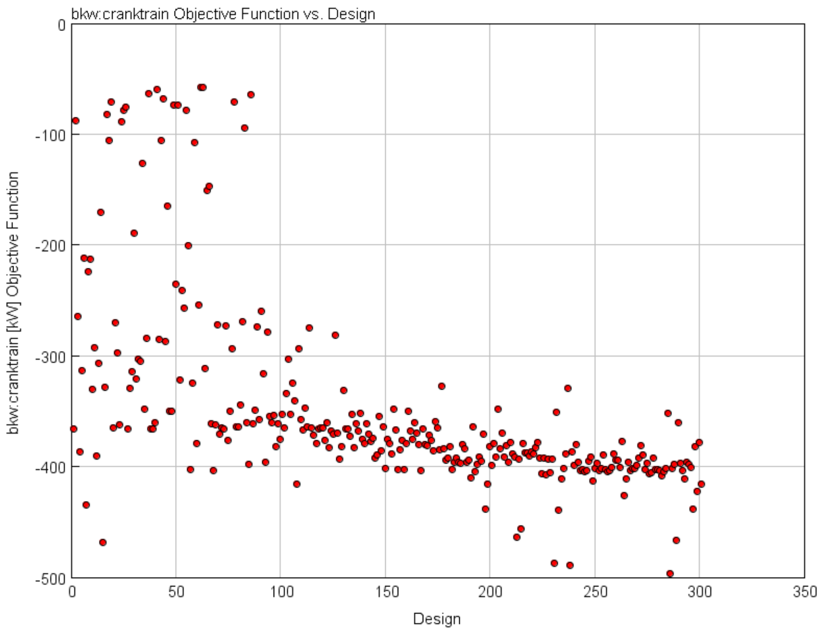


Chart 8 Result of design optimization for current constraints

Table 7 - Calculated factors for current constraints

	Variable factors	First event	Second event	
BGR	Stretch(angle multiplier)	0,918	0,918	[-]
	Angle Shift	184.15	544,15	[deg]
CR	Stretch(angle multiplier)	1,37	1,37	[-]
	Angle Shift	-59.37	300,42	[deg]
Intake valve	Stretch(angle multiplier)	1,62	1,62	[-]
	Angle Shift	393,19	747,01	[deg]
EGR	**EGR rate**	12,81	12,81	[-]

Consequence of downsizing (8,4l) to *Two-stroke engine brake*

Comparison between original and downsized engine shows substantial reduction of maximum brake performance of engine equipped with *Two-stroke engine brake*. The brake power of downsized engine is still relatively high, especially in middle engine rpm range. In the Chart is shown that the maximum of brake torque is in the 1000 rpm's where was the limit of simulation. Results of 8,4l engine optimisation for current constraints, where was all of monitored parameters respected is shown in *Chart 9* and *Chart 10.*

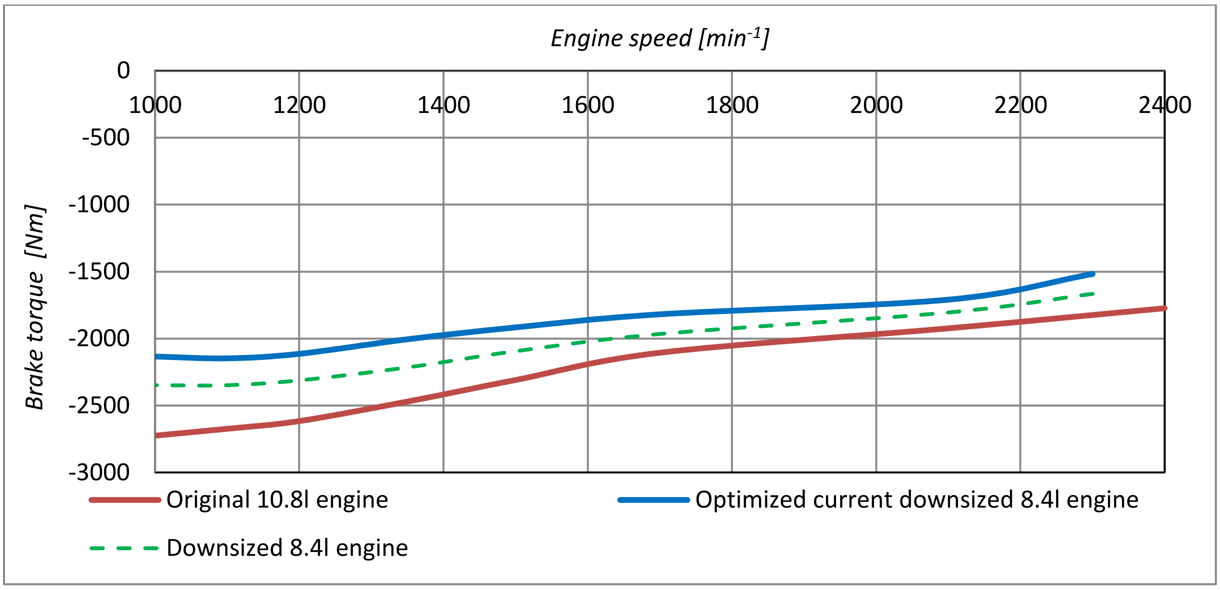


Chart 9 - Result of optimization for Brake torque

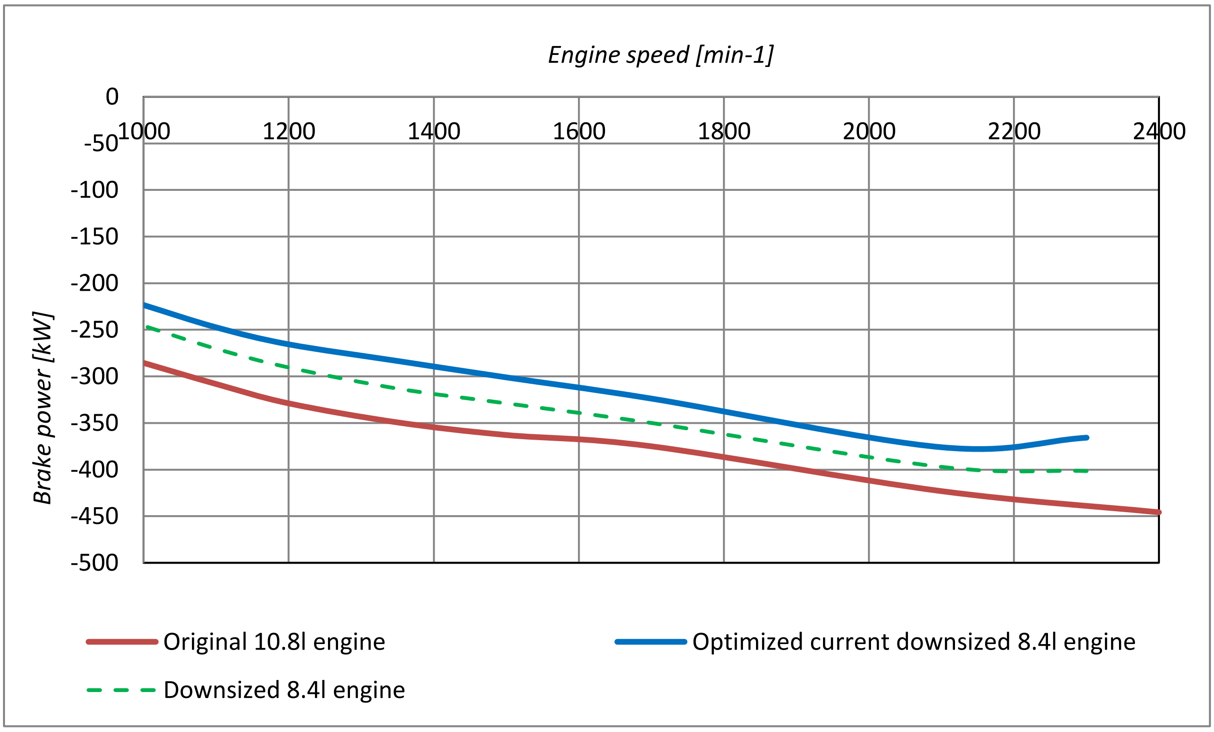


Chart 10 - Result of optimization for Brake power

Results of downsized engine are limited of current contstraints that are mentioned in previous chapters, because using more strict constuction and more sophisticated materials would be nowadays excessively expensive. But there is an assumption, that the direction of market will push on the manufacturers of components and engines to have more strict demands of engines because of trend lowering engine emissions. With more sophisticated materials could be inlet turbine temperature increased, with better construction of engine could be cylinder pressure increased and with more stiff springs increased the pressure drop so I done senstivity analysis of future Two-stroke engine brake which is described in the next chapter.

4.2 Sensitivity analysis of future Two-stroke engine brake

To find out potential of Two-stroke engine brake of downsized 8,4l engine was used Integrated Direct Optimizer as well with same factors and run setup like in previous chapter but with different constraints. Calculation of whole rpm range will takes about 170 hours of computing time. For long computing time of direct optimizer was done only 2 calculations for 1500 and 2300rpm, what is minimum and maximum of rpm range for engine braking. Found factors from optimization for one rpm's were after used for recalculation of whole rpm range.

***Future constraints** of tested engine that wouldn't be exceeded was changed as follows:*

1) Maximum cylinder pressure is increased to 120bar
2) Maximum average temperature on turbine inlet is incresed to 770˚C
3) Maximum pressure drop on Intake valve is incresed to 3,5bar
4) Maximum pressure drop on Intake valve is incresed to 5,7bar

Direct design optimizer done 301 designs, and in *Chart 11* is shown, that from design number 100 was results of maximum engine brake power converging to value about -400kW. Direct design optimizer founds best design for maximum engine brake power potential -425kW in 2300rpm's with following factors in *Table 8*:

Table 8 - Calculated factors for future constraints

	Variable factors	First event	Second event	
BGR	Stretch(angle multiplier)	0,92	0,92	[-]
	Angle Shift	184,15	544,15	[deg]
CR	Stretch(angle multiplier)	1,38	1,38	[-]
	Angle Shift	-50,5	309,49	[deg]
Intake valve	Stretch(angle multiplier)	1,62	1,62	[-]
	Angle Shift	386,61	443,7	[deg]
EGR	EGR rate	12,8125	12,8125	[-]

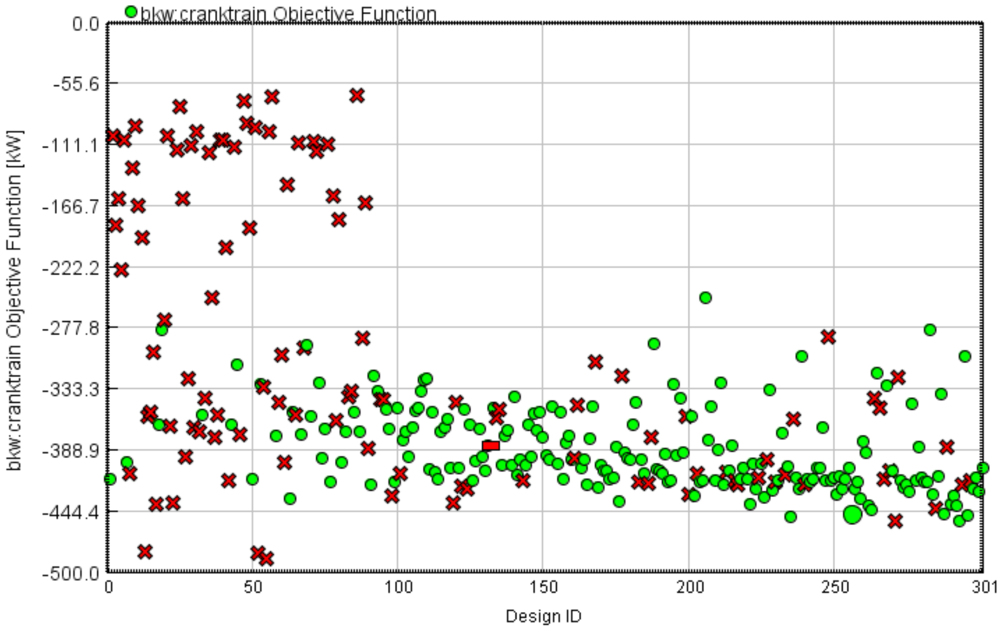


Chart 11 - Result of design optimization for future constraints

With all of these factors from *Table 8*, is whole rpm range recalculated and the result of brake characteristics is shown in *Chart 13* and *Chart 14*. However, the maximum cylinder pressure was low under the permissible value in mid rpm range. Factor EGR rate is independent from the other factors, therefore for the target to approaching to maximum permissible cylinder pressure 120 bar (*Chart 12*) is EGR rate set individually for each rpm's value as shown in *Table 9*:

Table 9 - EGR rate setting

Engine speed [min^{-1}]	**2300**	2100	1700	1500	1350	1200	1100	1000
EGR rate [-]	**12.8125**	12.8125	18	22	23	20	13	4

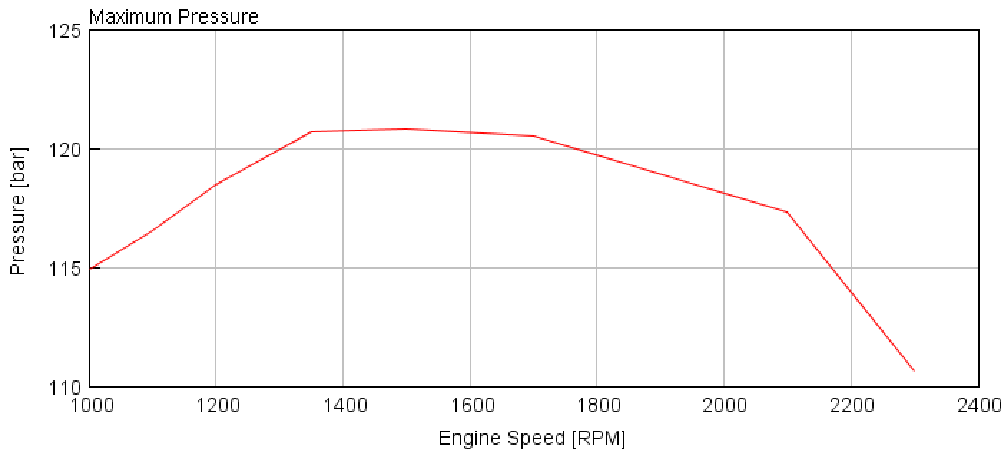


Chart 12 - Maximum cylinder pressure

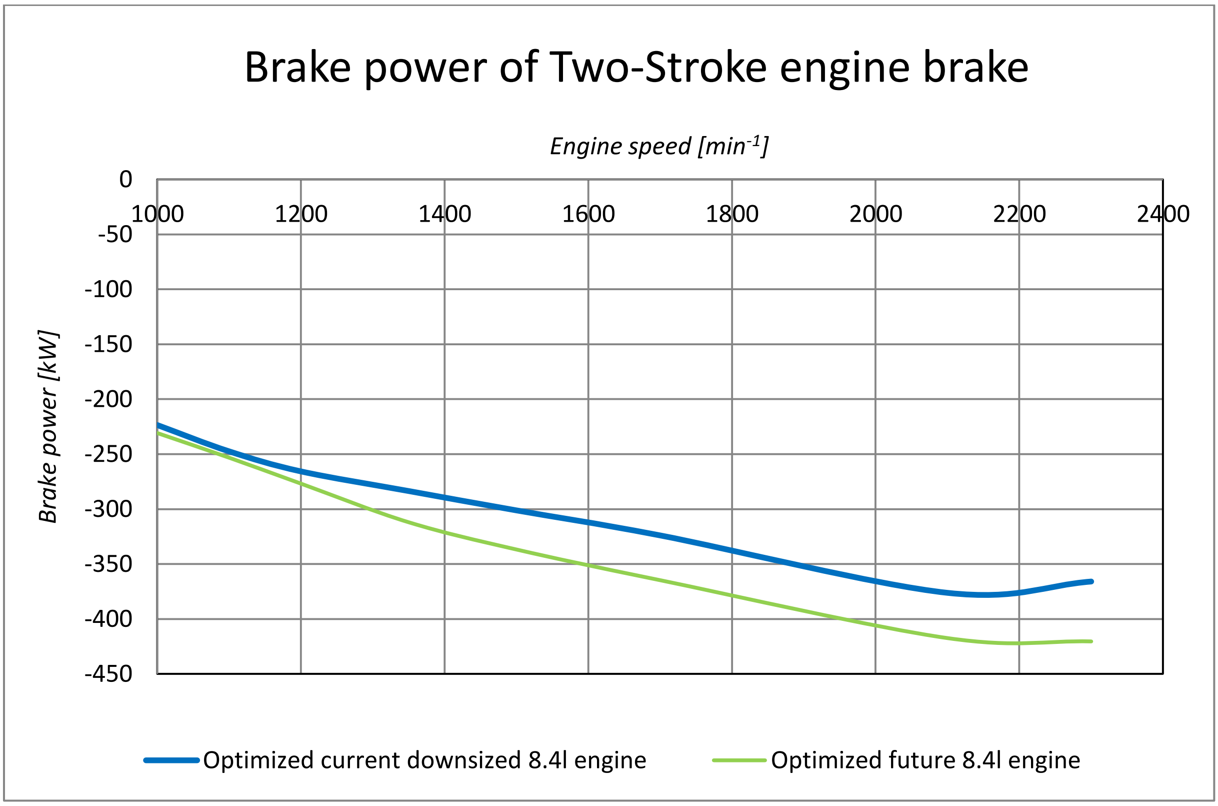


Chart 13 - Result of Brake power of sensitivity analysis

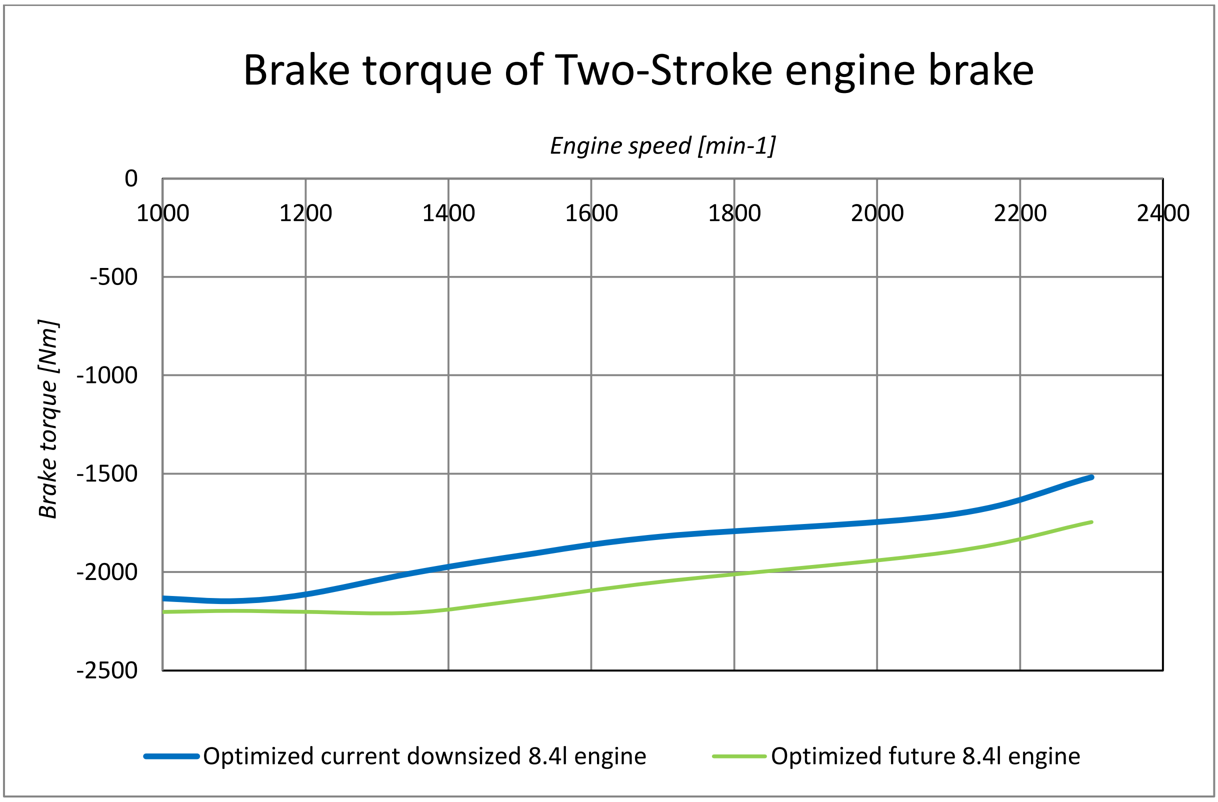


Chart 14 - Result of Brake torque of sensitivity analysis

5.2 Engine brake control

Auxiliary brakes have to have graded brake effect because not always is full brake power needed. Since the model has a 6 cylinder engine and advanced valve-train control, it is possible to "shut off" some of the cylinders and control the valvetrain mechanisms for each cylinder independly. For more advanced, continuous and sensitive regulation of engine brake torque could be used change of the VGT rack position, but for this purpose have to be connected the Engine brake model with Tractor-trailer model. Brake torque is controlled automatically independent to brake pedal position by switching between engine brake levels. Brake torque of the engine brake is divided to 4 modes (levels) as follows:

- Engine brake Level 3
- Engine brake Level 2
- Engine brake Level 1
- Engine brake OFF

*For **MX Compression release Engine brake control** (Chart 15) was used assumtions, that*

- *Level 3* is is the maximum Engine brake torque, obtained from the *Chart 4*
- *Level 2* will be 50% of *Level3*
- *Level 1* will be 25% of *Level 3*

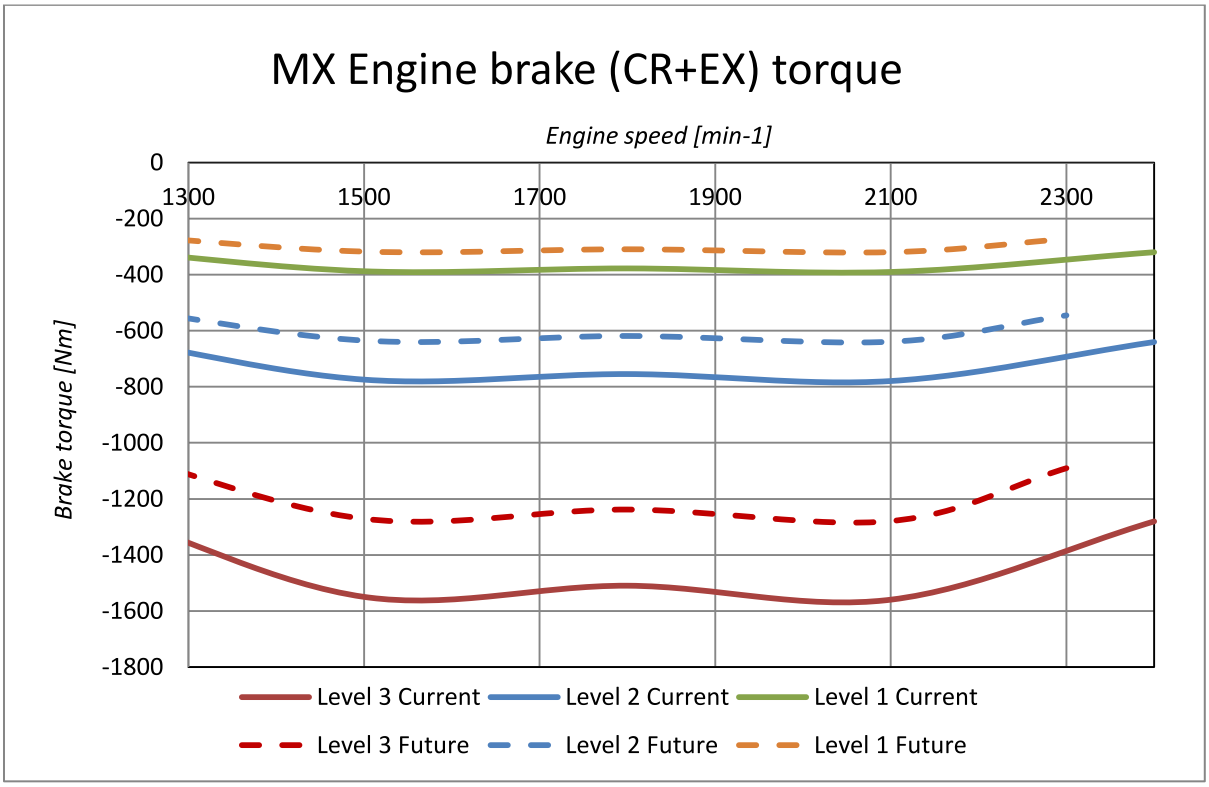


Chart 15 - Brake torques of CR+EX brake for different levels

*For **Two-stroke engine brake control** (Chart 16) (Chart 17) was data obtained from GT-POWER model as follows:*

Engine brake Level 3

This means maximum power of engine brake when all cylinders has a engine brake valve timing, and *VGT Rack* is not fully closed, because when will be fully closed, speed of turbine will be low and amount of saturated air in intake as well so brake power will be lower. Level 3 torque is the result of optimization.

Engine brake Level 2

Model was changed that way, that all of cylinders have engine brake valve timing, but *VGT Rack* si fully open in whole engine rpm range.

Engine brake Level 1

Model was changed that way, that 3 cylinders has a engine brake valve timing and 3 cylinders has a normal four-stroke cycle valve timing. VGT rack is fully open.

Engine brake OFF

All cylinders has a normal four-stroke cycle valve timing, rack is fully open and braking torque is only drawn by natural retarding forces in engine (mechanical and friction losses). All of this plots were used in braking strategy to switch between each levels.

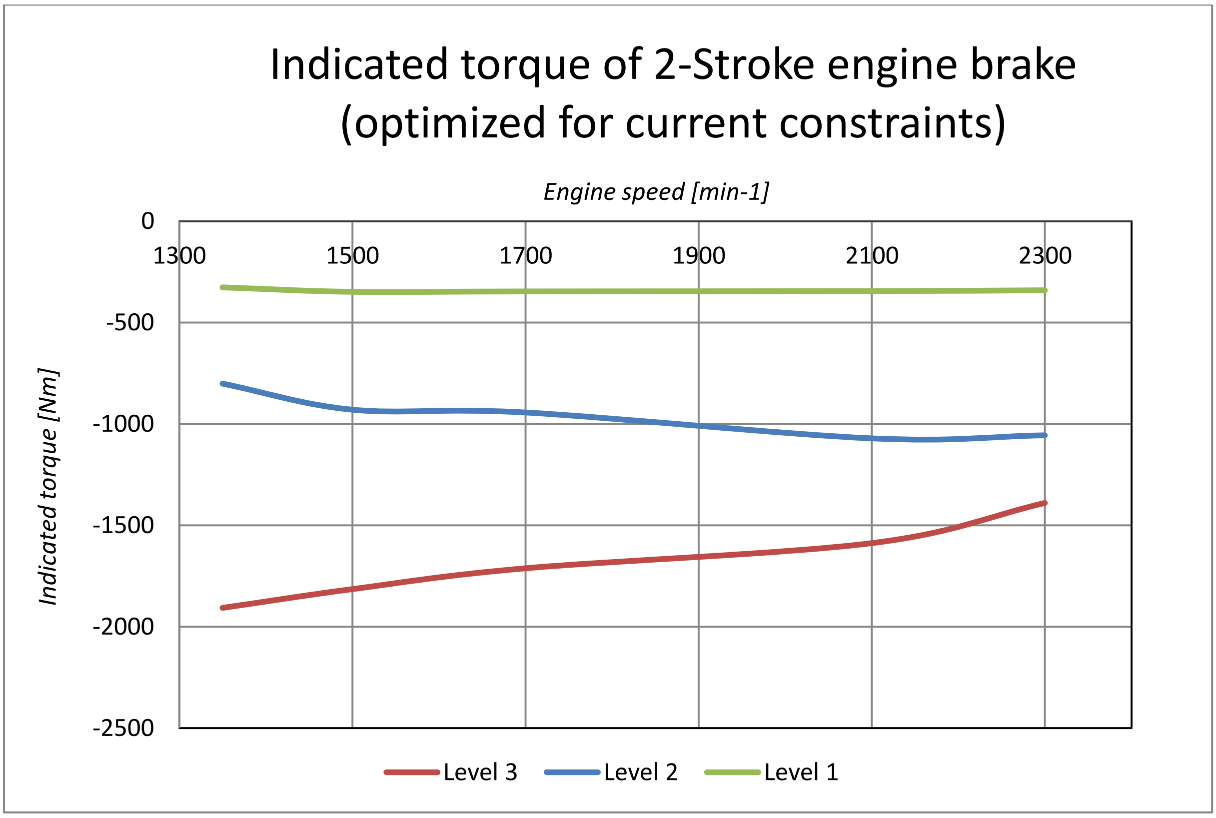


Chart 16 - Brake torques of Two-Stroke engine brake optimized for current constraint for different levels

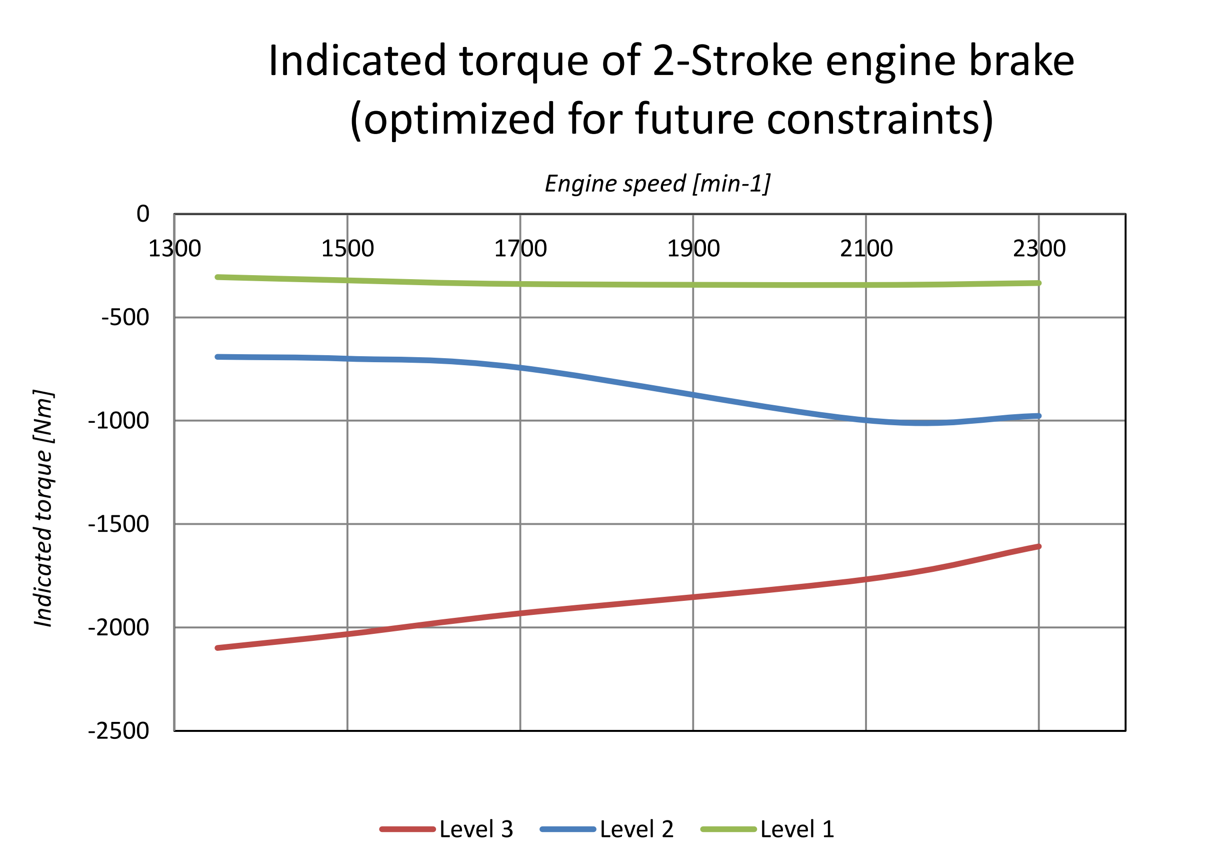


Chart 17 - Brake torques of Two-Stroke engine brake optimized for future constraint for different levels

6.2 Drivetrain retarder control

Drivetrain retarder torque is controlled by brake pedal. Based on the results of sensitivity testing, the best solution *(Fig. 60)* is described below:

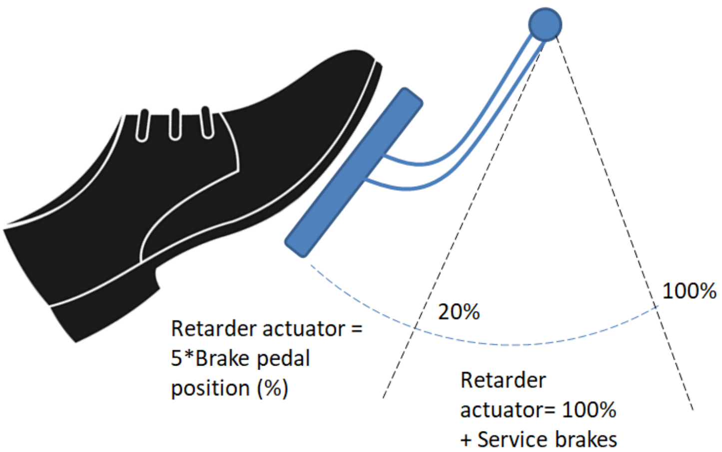


Fig. 59 - Retarder actuator control

- When the brake pedal is lower than 20%, the service brakes are off and retarder actuator position is multiplied by 5 times of brake pedal position.
- When the brake pedal is higher than 20%, the service brakes are controlled by position of brake pedal, and the retarder has a full power.

7.2 Braking strategy (Brake control unit)

Braking strategy *(Fig. 61)* is set to minimise use of service brakes. It is realised by an event manager template like a time sequence events. Monitored parameters are:

- difference between current and target speed *($\Delta v = v_{current} - v_{target}$)*
- vehicle acceleration *($acc = \Delta v / \Delta t$)*

Every event has a *Event exit criterion*, which must be met to move on to the next event. Then the software evaluates the *decision conditions*. For example Event *"Engine brake level 1"* - when one of the Event exit criterions are met (*$\Delta v>2$ or $\Delta v<-2$) then go to decision conditions:*

- *Is vehicle acceleration lower or equal than $0{,}1 m \cdot s^{-2}$?*
    - *Go to intermediate step - "Keep Level 1" until one of the following criterions will be met:*
        - *$\Delta v<-5$ - Go to "Engine brake deactivation"*
        - *$\Delta v>0$ - Go to "Engine brake Level 2".*

- *Is vehicle acceleration higher than $0{,}1 m \cdot s^{-2}$?*
    - *Go to - "Engine brake Level 2".*

For detailed strategy please see the Annex 4.

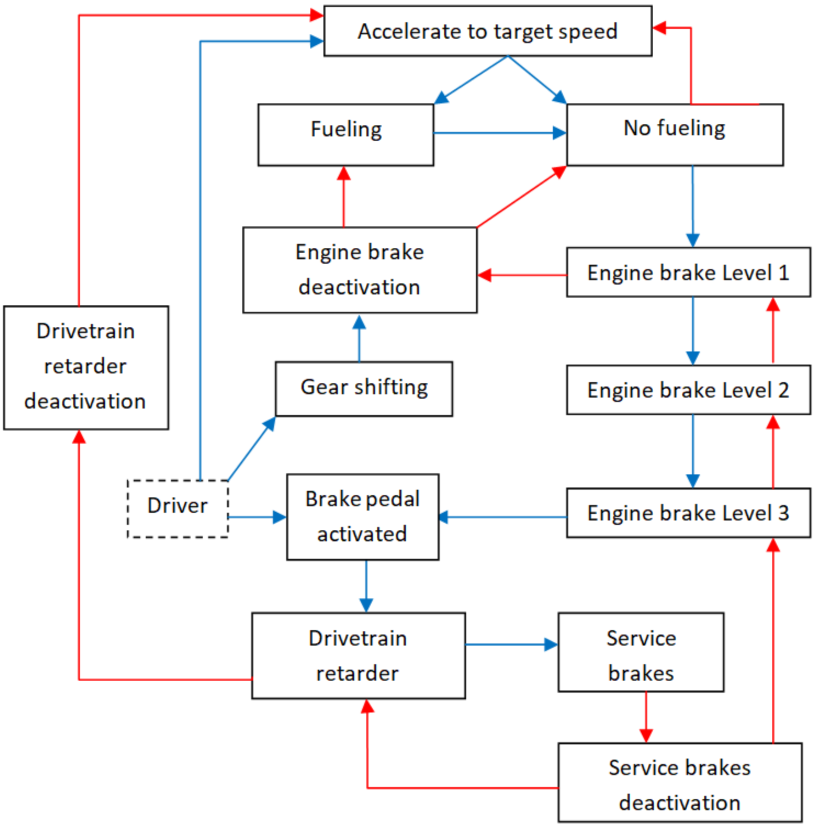


Fig. 60 - Braking strategy

8.2 Consequences of reducing driving resistances

This part of thesis discusses consequences of lower driving resistances on truck and is divided to static and dynamic calculations.

8.2.1 Static calculations - Comparison of engine brakes

Static calculations are quick tool for determination how much braking power is needed for cruising constant downhill grade at constant vehicle speed and comparison, what type of engine brake is sufficient for the case. It's simple, when the required braking power on crankshaft is lower than power of engine brake, vehicle will be accelerating without using any other brake system. For calculations of required brake power on crankshaft are used parameters from the table X and the gear ratios are same like in GT-SUITE model. In the future, some of the parameters what are described in teoretical part will be probably not that radically changed in reality, but I consider in calculations the case of the limit of reduction in driving resistances.

To compare most used types of engine brakes, are used data from *Jacobs vehicle systems*, because the brake power is in relationship to engine displacement. The data are approximated by third degree polynom to have braking power like a function for every engine speed.

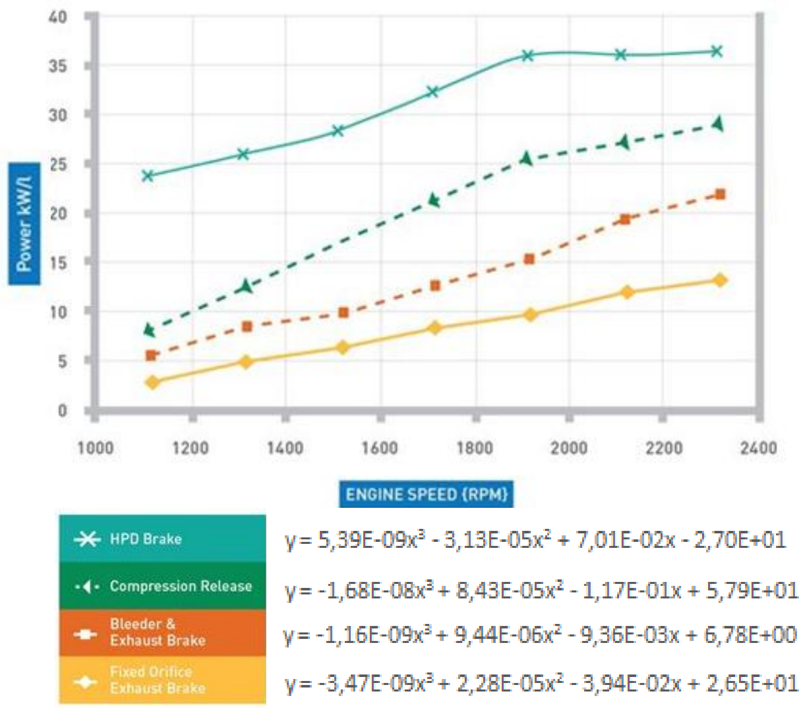


Fig. 61 - Jacobs engine brakes characteristics [29]

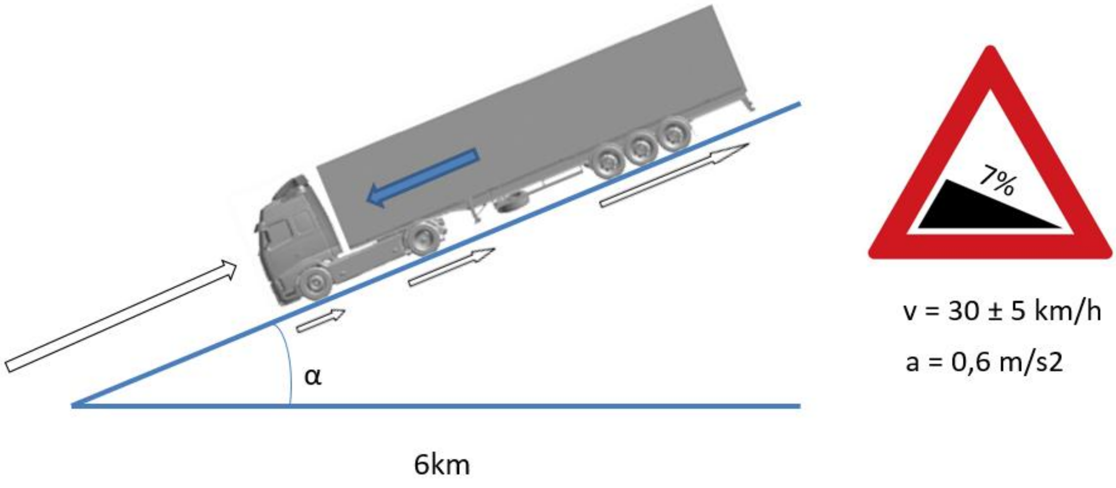
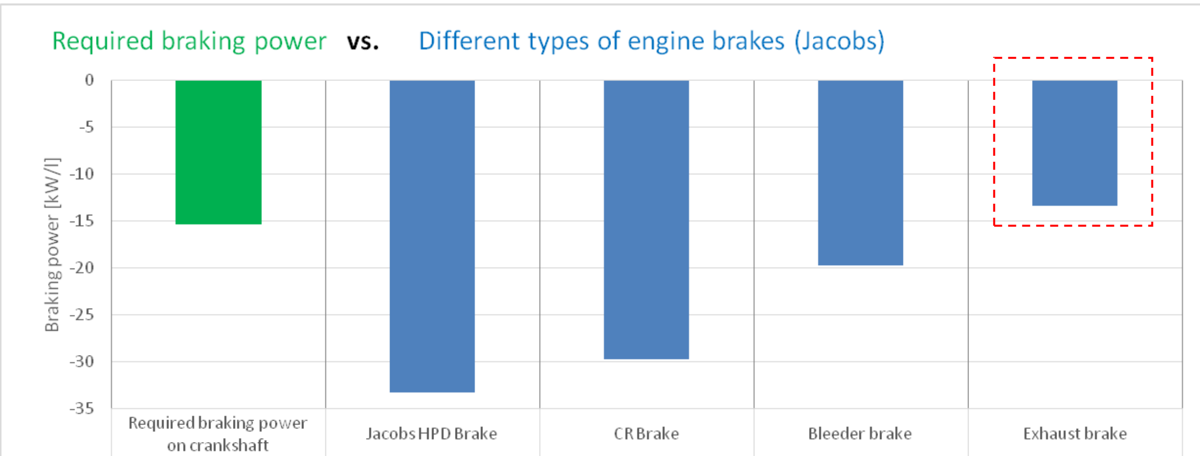


Fig. 62 - Test Type II.A

Case 1 and 2 *(Chart 18 and 19)* are representing legislation test type II.A. In first case is shown that natural retarding power of engine is insufficient, because the simple exhaust brake is already insufficient. This assumption is also fulfilled in the **chapter X**. The difference, when the driving resistances are lower in second case, the Bleeder brake would be insufficient as well. It follows that, nowadays trucks with engine displacement lower than 10.8l, have to be equipped at least with Bleeder brake in combination with Exhaust brake.

Case 3 and 4 *(Chart 20 and 21)* are representing typical demands from truck customers. In Case 4 is shown that if the assumptions on driving resistances will be met, none of the engine brake from Jacobs vehicle systems would be sufficient to keep constant speed at downhill cruising and the vehicle becomes dangerous without using drivetrain retarder or friction service brakes.

CASE 1: Current truck on 7% grade, by cruising speed of 30km/h, engine speed 2212min^{-1}



	Required braking power on crankshaft	Jacobs HPD Brake	CR Brake	Bleeder brake	Exhaust brake
[kW/l]	-15,22059113	-33,2524838	-29,74452469	-19,71506826	-13,35320262

Chart 18 - Static calculations - CASE1

CASE 2: Future truck on 7% grade, by cruising speed of 30km/h, engine speed 2212 min^{-1}

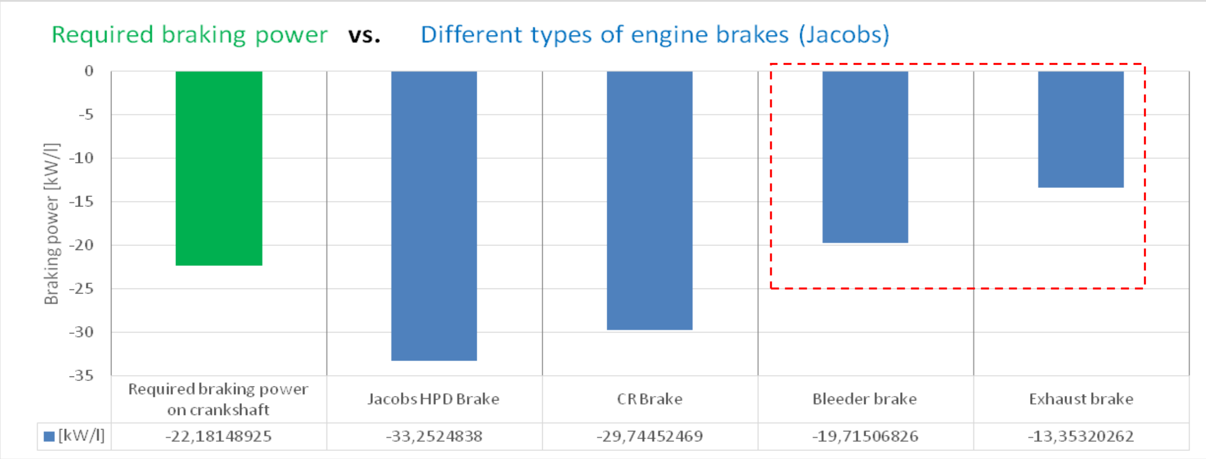


Chart 19 - Static calculations - CASE2

CASE 3: Current truck on 8% grade, by cruising speed of 50km/h, engine speed 2286 min^{-1}

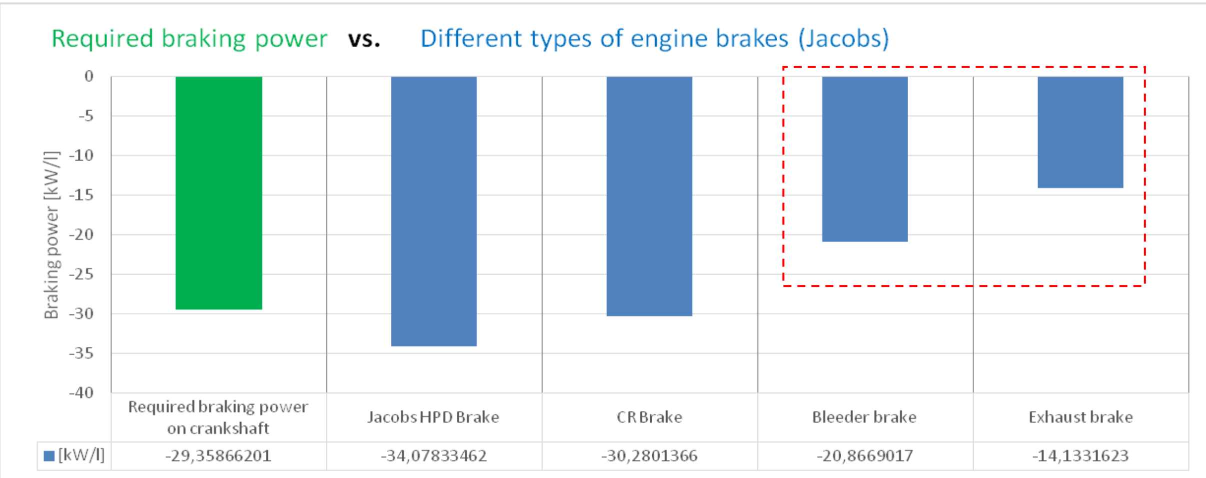


Chart 20 - Static calculations - CASE3

CASE 4: Future truck on 8% grade, by cruising speed of 50km/h, engine speed 2286min^{-1}

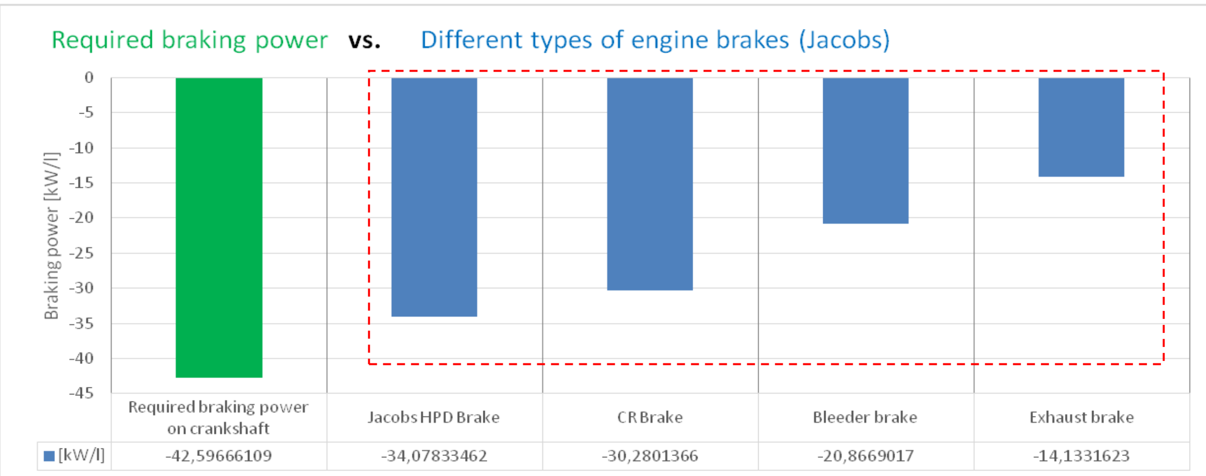


Chart 21 - Static calculations - CASE4

8.2.2 Dynamic calculations

Results of static calculations shows, that in the future, would be met legislation type II.A *(Chart 19)* only by Two-stroke engine brake (HPD) and Compression release engine brake, so dynamic calculations are focused only for these two types of engine brakes.

8.2.2.1 Stopping distance

Test of stopping distance (*Chart 22*) is done from vehicle initial speed 96km/h to total stop. Compared are high resistances (Current) vehicle and low resistances (Future truck) vehicle. For engine brakes is this event too fast, so engine is disconnected just a few seconds after the emergency braking started. Testing with engine brake shows that the engine always stalls below the idle speed so only possible auxiliary brake for this test is a drivetrain retarder. For retarder is this event fast too, because has a long activation time and is off when the vehicle speed is low (*Chart 23*), but certainly helps a bit with braking.

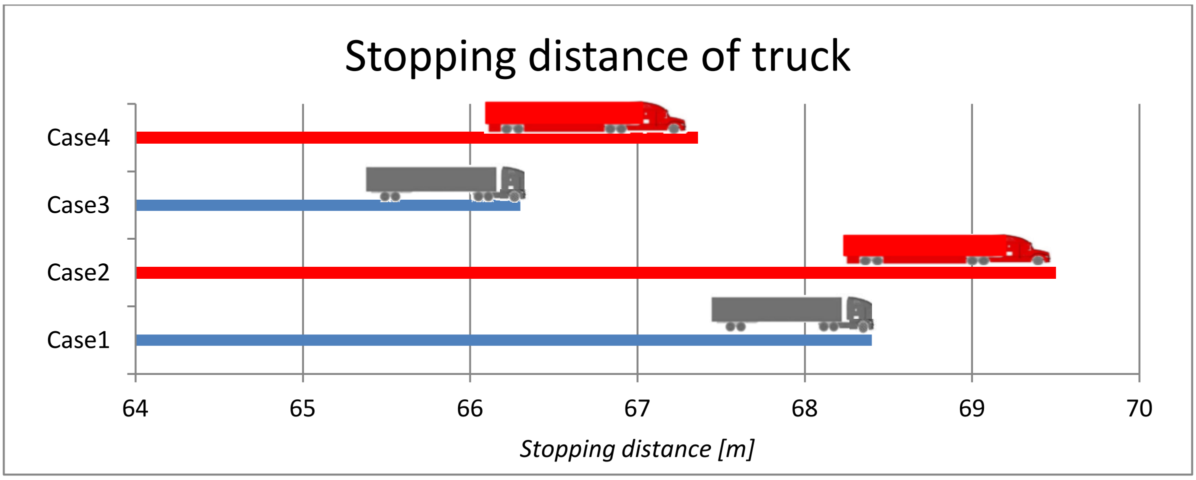


Chart 22 - Consequence of lower driving resistances to stopping distance

Case 1 -High vehicle resistances without auxiliary braking systems – 68.4m
Case 2 - Low vehicle resistances without auxiliary braking systems – 69.5m
Case 3 - High vehicle resistances with retarder – 66.3m
Case 4- Low vehicle resistances with retarder – 67.4m

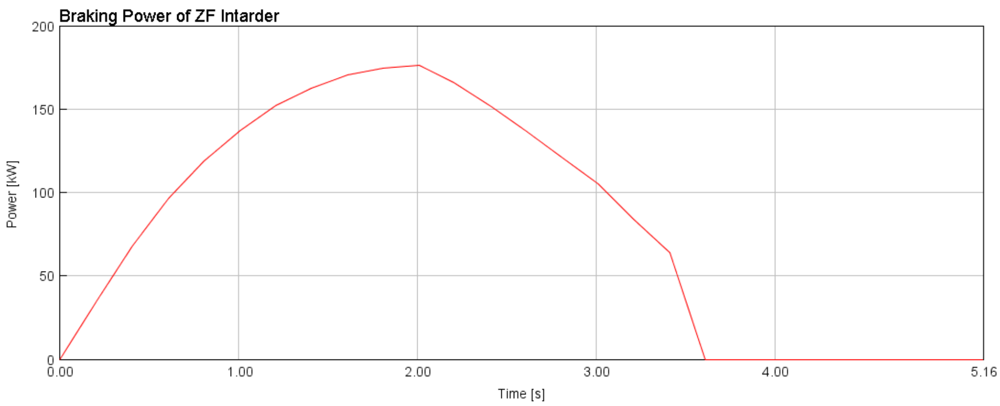


Chart 23 - Reaction of ZF Intarder during the stopping distance test

8.2.2.2 Dynamic grade test

Dynamic grade test shows brake systems reactions on truck model with braking strategy from chapterX on a predefined road with dynamic change of road grade *(Chart 24)* to find out the amount of energy that was dissipated through friction brakes *(Chart 25)* on different configurations of vehicle and brake systems. Target speed for test is 60km/h.

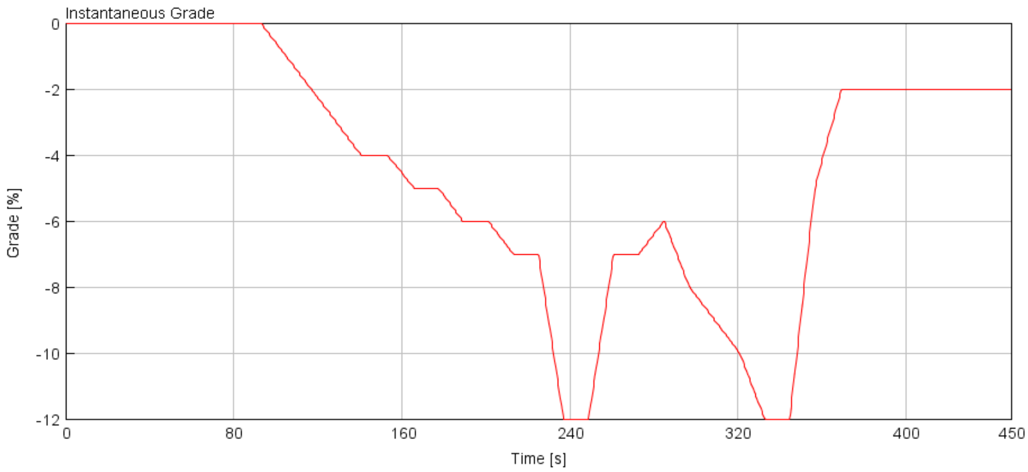


Chart 24 - Predescribed road on dynamic grade test

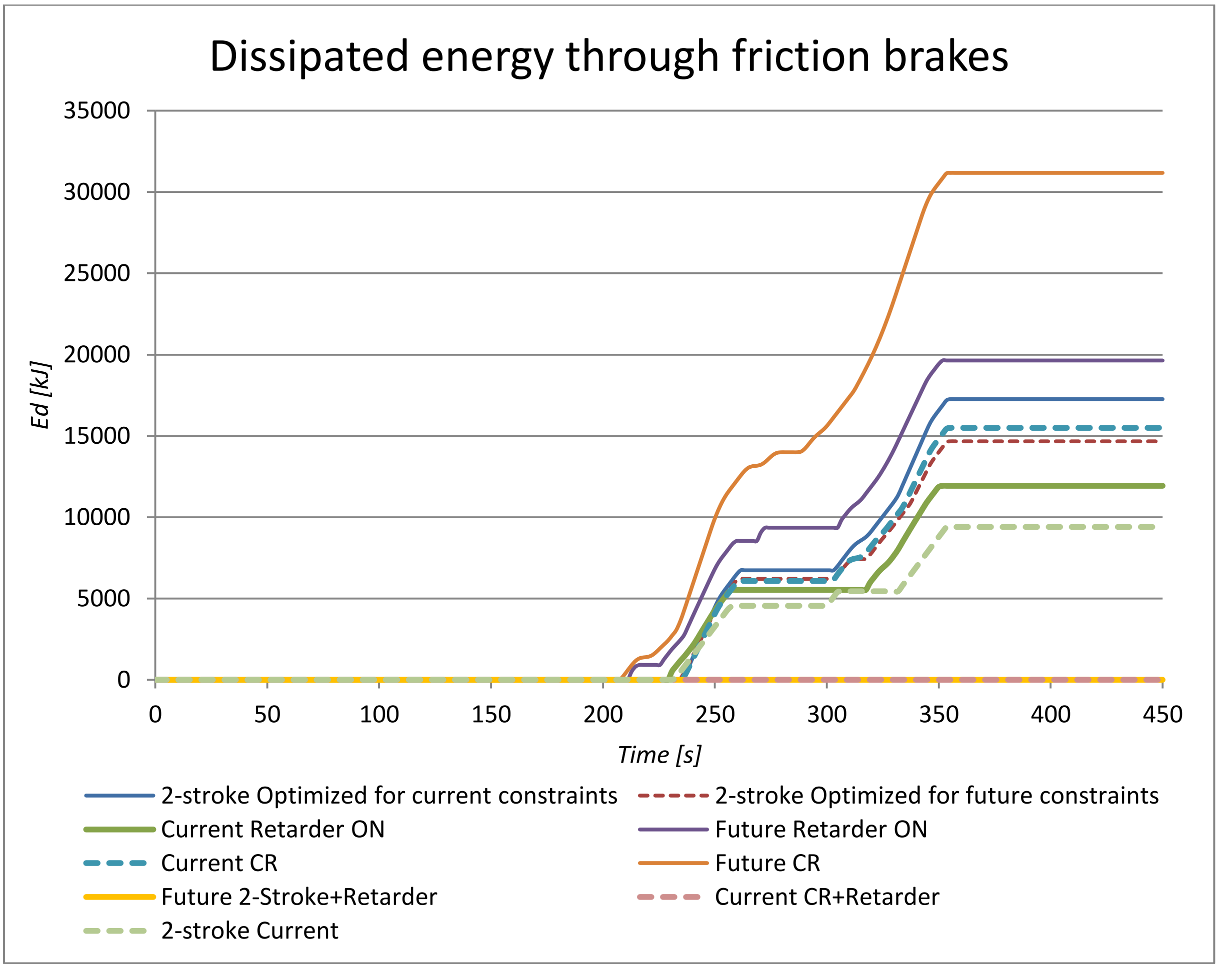


Chart 25 - Results of Dynamic grade test

Table 10 - Results of Dynamic grade test

Type of auxiliary brake	Brake distance with using friction brakes [m]	Estimated Brake dust emmisions depends on brake distance [mg]	Dissipated energy through friction brakes [kJ]
CR+EX *Current truck*	874.7462	43.73 - 69.97	15502.104
ZF Intarder *Current truck*	676.365	33.81- 54.10	11925.644
ZF Intarder with CR+EX *Current truck*	0	0	0
2-Stroke(10.8l engine) *Current truck*	715.6122	35.78 - 57.24	**9394.058**
CR+EX *Future truck*	1427.5294	71.37 - 114.20	31176.67
Two-Stroke (optimized for current constraints) *Future truck*	998.8032	49.94 - 79.90	17267.7
Two-Stroke (optimized for future constraints) *Future truck*	876.6565	43.83 - 70.13	**14658.555**
ZF Intarder (8.4l engine) *Future truck*	952.4585	47.62 - 76.19	19632.39
ZF Intarder with Two-Stroke (optimized for future constraints) *Future truck*	0	0	0

Current truck

For current truck *(Table 10)* with 10.8l engine and high driving resistances is the amount of dissipated energy with CR higher than using ZF Intarder for this road. Combination of use of both systems shows, that the service brakes weren't used, so no emmisions were dissipated to the air. If Two-Stroke engine brake is used on current truck, the amount of the dissipated energy would be decreased and lowest from all of the auxiliary brake systems. But the brake dust emissions are higher because total brake distance on service brakes is higher than with ZF Intarder. However, if brake dust emissions were calculated from amount of dissipated energy, the real emissions would be probably lower.

Future truck

For future truck *(Table 10)* with 8.4l engine and low driving resistances, the amount of dissipated energy would be approximately twofold higher with using CR, so CR would be inappropriate. Two-Stroke with current limitations would be adequate replacement for CR in current truck, but dissipated energy would be a little higher. But if engine limitations were improved, the amount of dissipated energy would be lower as well as brake dust emissions. For using only one auxiliary brake system would be the best choice Two-Stroke, because the amount of dissipated energy is the lowest from all of the systems. If both systems were used (Engine brake and Drivetrain retarder) in future truck, to compensate consequence of lower driving resistnaces on higher dissipated energy using ZF Intarder and insufficient CR, the Two-Stroke would be used like an ideal substitute, because still no energy would be dissipated.

8.2.2.3 Performance limit test

To find out the maximum performance limit of using auxiliary brake system without using of service brakes is done by performance limit test. The test is done on road *(Chart 26)* with progressive decreasing of road grade with vehicle displacement. Target speed is 50km/h during the whole test. The time and then road grade limit is measured, when the friction brakes power begin to rapidly increase for a long time *(Chart 27)* – on this value of road grade decrease, vehicle becomes dangerous without using service friction brakes – what is the performance limit for auxiliary brake.

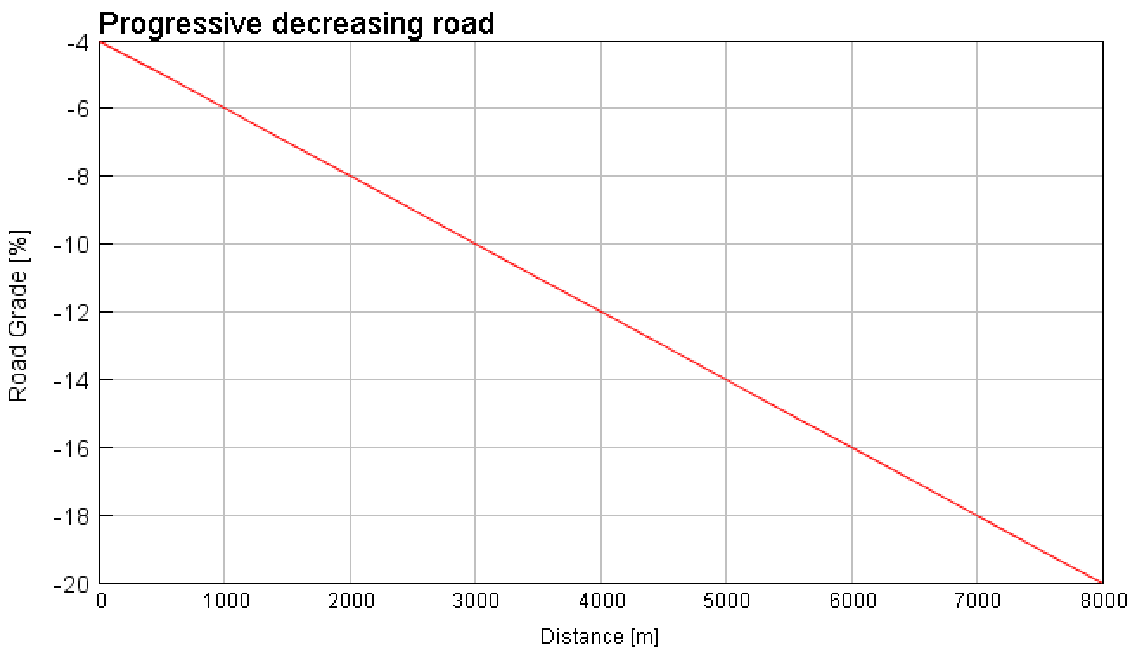


Chart 26 - Predesribed road for Perforance limit test

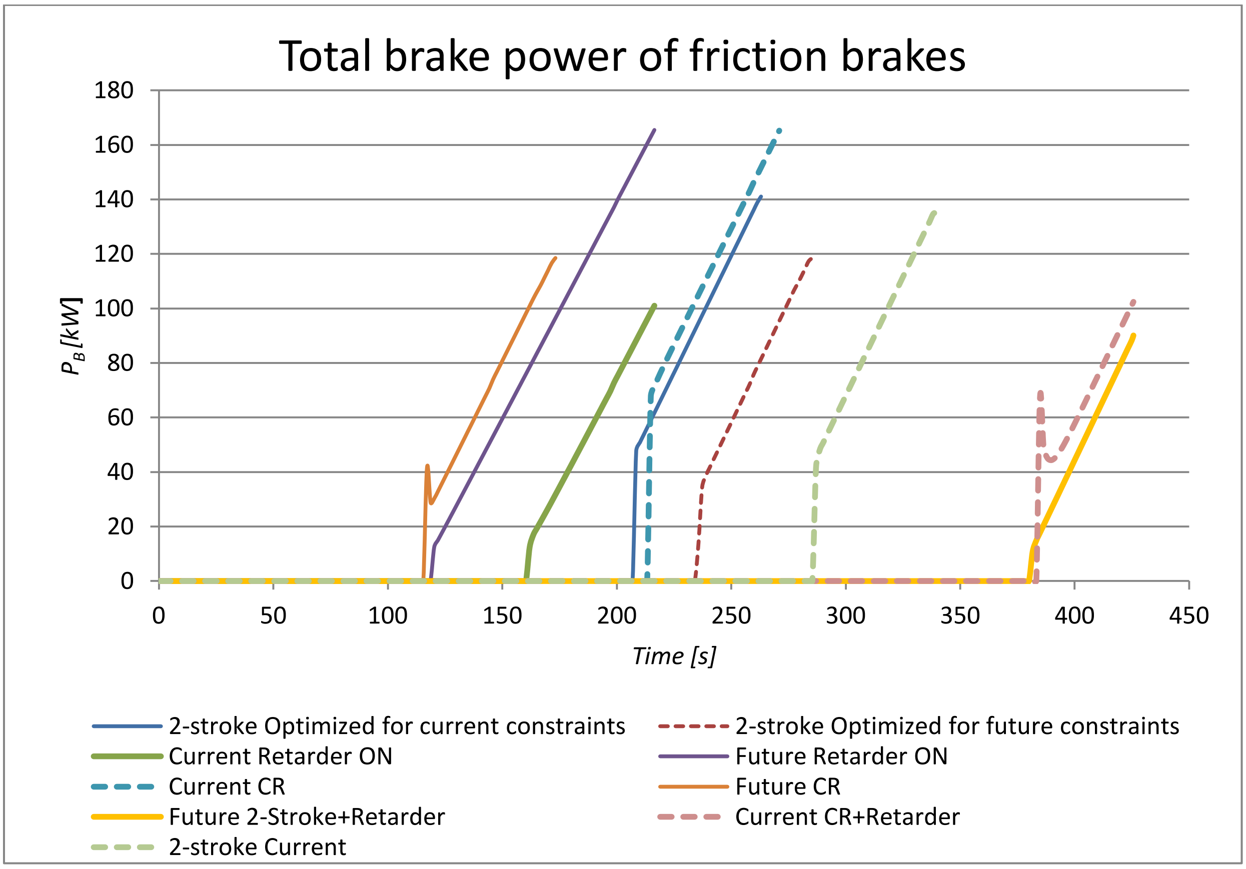
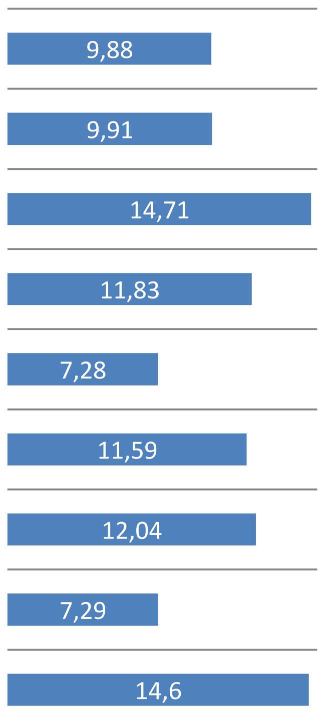


Chart 27 - Results of Performance limit test

Table 11 - Results of Performance limit test

Type of auxiliary brake	Time [s]	Grade [%]
CR+EX *Current truck*	213.24	-9.88
ZF Intarder *Current truck*	210.43	-9.91
ZF Intarder with CR+EX *Current truck*	383.18	-14.71
2-Stroke(10.8l engine) *Current truck*	285.38	-11.83
CR+EX *Future truck*	118.65	-7.28
2-Stroke ***(optimized for current constraints)*** *Future truck*	276.36	-11.59
2-Stroke ***(optimized for future constraints)*** *Future truck*	290.39	-12.04
ZF Intarder *(8.4l engine)* *Future truck*	117.05	-7.29
ZF Intarder with 2-Stroke ***(optimized for future constraints)*** *Future truck*	379.97	-14.6



Comparison of current and future truck *(Table 11)* shows that lower drive resistances of future truck would substantly reduce maximum possible value of road grade cruising without using friction brakes. Result of test shows that the limit of grade for CR and ZF Intarder is approximately same because ZF Intarder power is small in low vehicle speeds. Using Two-Stroke in combination with ZF Intarder confirms assumption mentioned in Dynamic grade test that Two-Stroke would be ideal replacement for CR in the future truck because the maximum road grade limit is similar like in case of CR in combination with ZF Intarder on Current truck.

8.2.2.4 Dynamic Type-IIA test

Legislation test Type II.A (Chapter Legislation) is used for endurance braking performance of auxiliary brake systems and service friction brakes after end of the test have to keep cold. For testing were chosen only ZF Intarder and Compression release because Two-Stroke is much higher above the limit of required brake power for this test. Legislation permits tolerance in case of engine braking for vehicle speed 30±5 km/h.

DAF MX Engine brake (CR+EX)

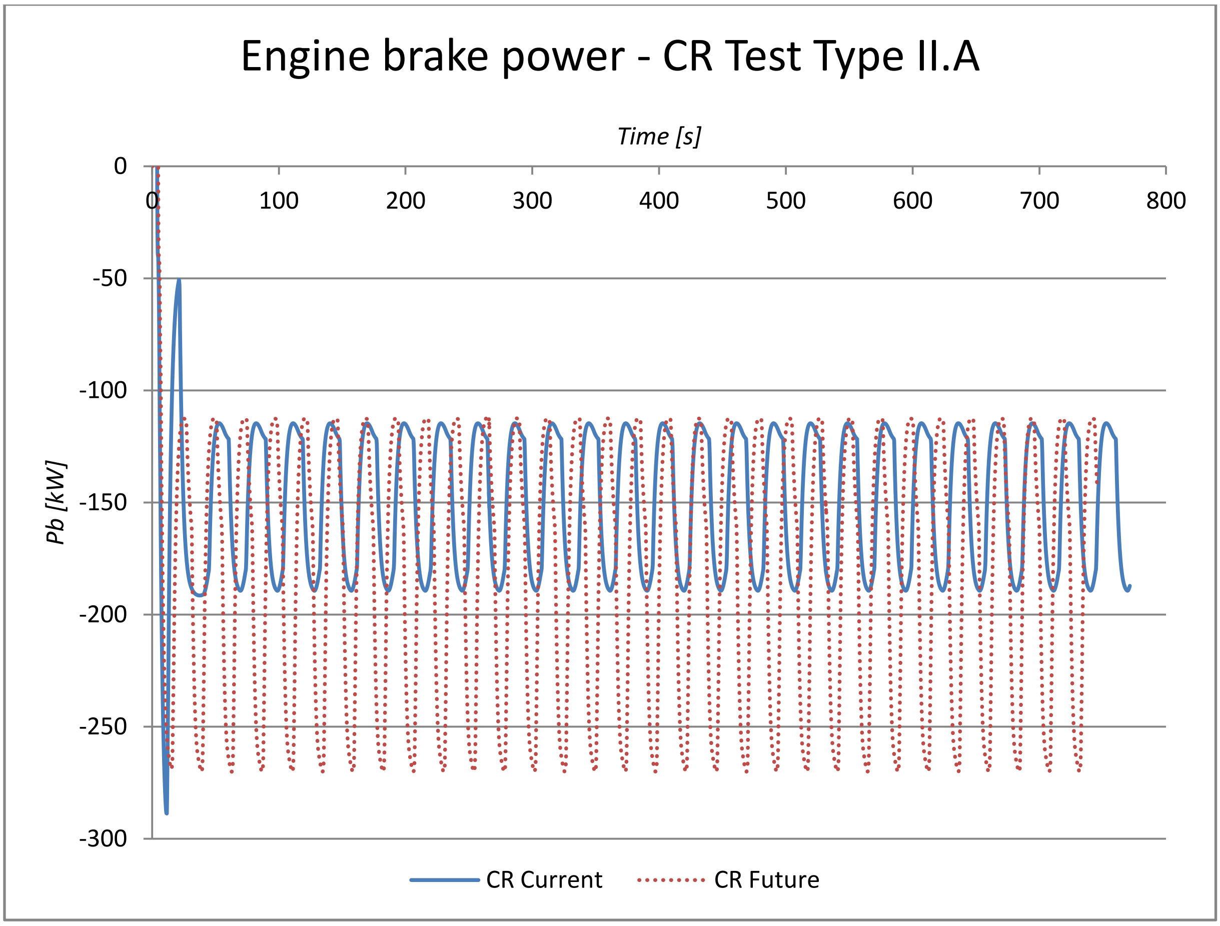


Chart 28 - Result of test type II.A with CR brake for brake power

CR Current: τ=29.21s 27x **CR Future:** τ=23.6s 31x

Results of testing CR+EX with designed braking strategy *(Chart 28)* shows that for legislation requirements would be CR+EX still sufficient in the future. During the test no service brakes would be used.

Current CR

In case of Current CR, is the maximum brake power of engine brake (Level 3) used only in the start of the test. After, the brake control unit is switching between Level 1 and Level 2 because no more brake power is needed for oscilating around target speed.

Future CR

In case of Future CR, when considered assumptions about driving resistances would be met, the consequence for future truck would be more frequent using of engine brake, because vehicle is accelerating faster with lower drive resistances (*Chart 29*) and amount of power metering is higher. In comparison with Current CR, the brake control unit have to use engine brake Level 3 for oscilating around target speed.

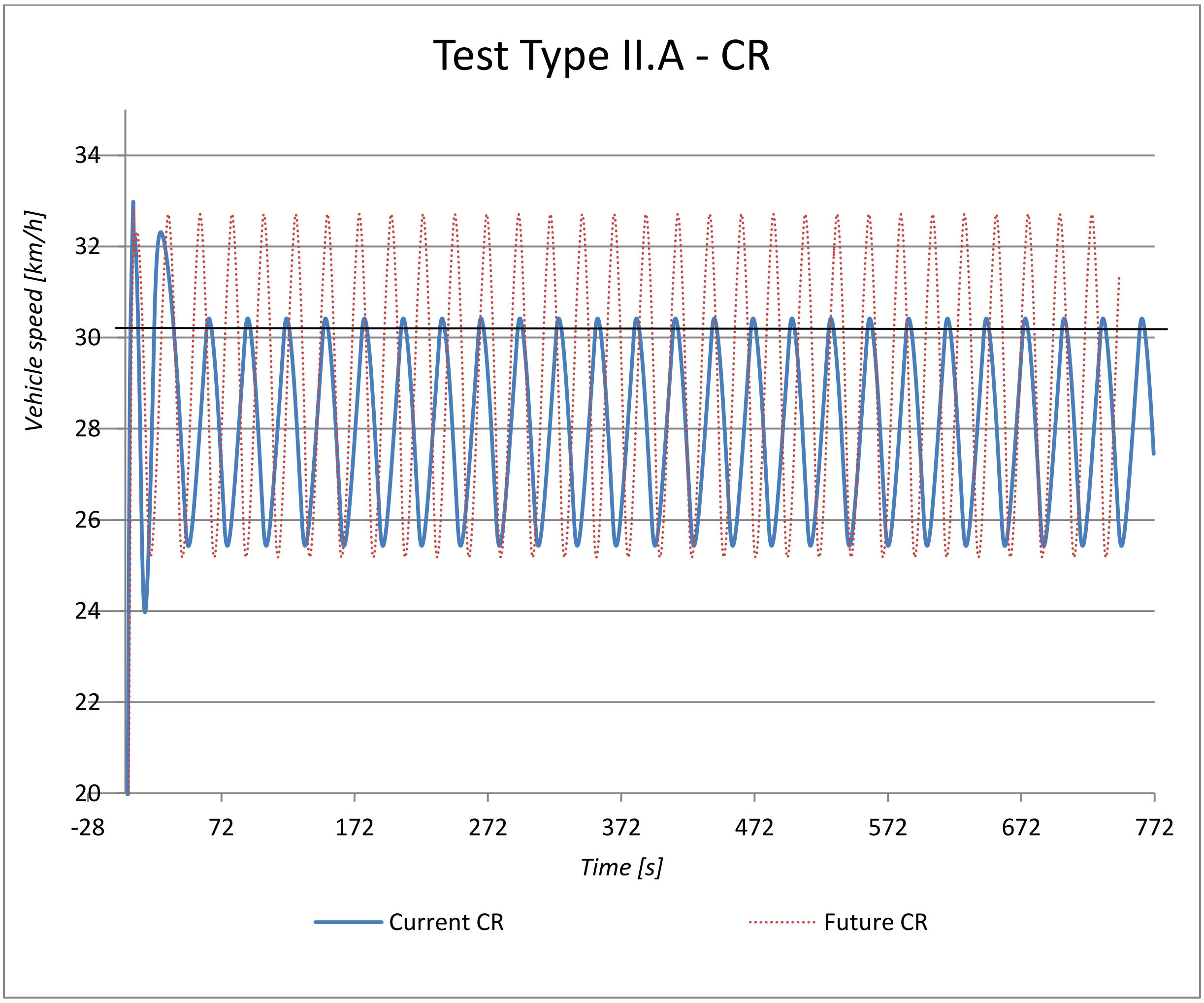


Chart 29 - Result of test type II.A with CR brake for vehicle speed

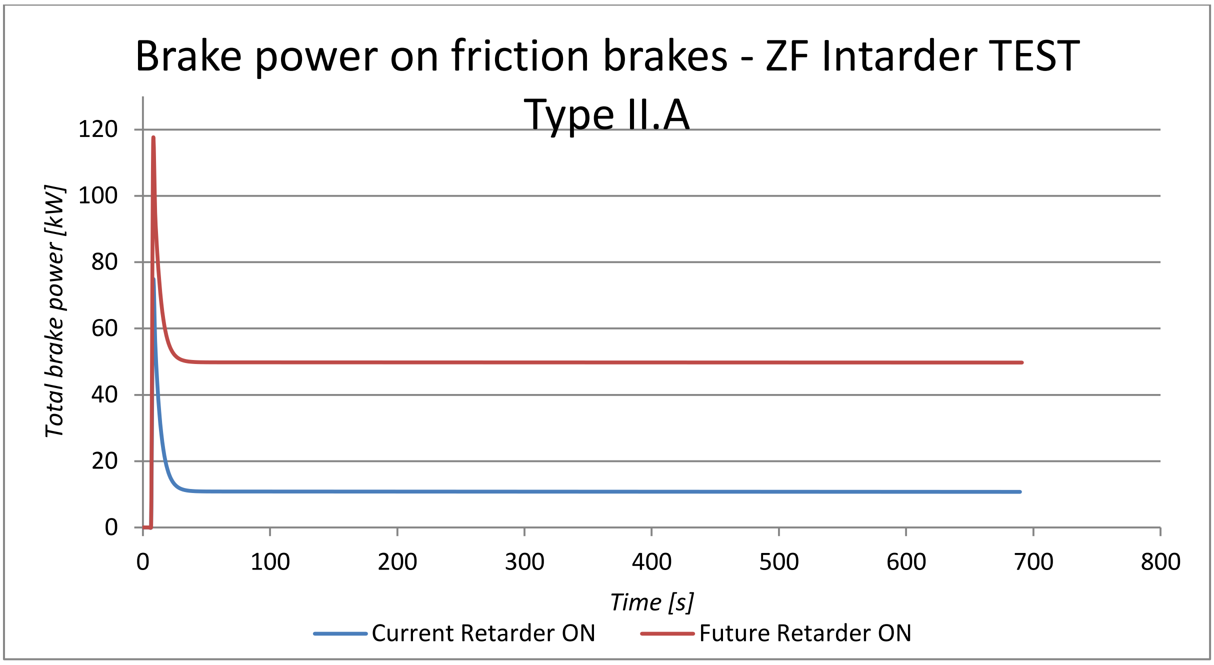


Chart 30 - Result of test type II.A with ZF Intarder for brake power

Testing of ZF Intarder confirm that the power of Intarder is small at low vehicle speed. Chart 26 shows, that Intarder is insufficient for legislation test in both cases. In current truck, the vehicle speed is steady on 31km/h, but maintaining the vehicle at this speed requires about 10kW brake power on friction brakes. Brake power of ZF Intarder is on the limit and is affected by differential gear ratio and by dynamic radius of tire, so in reality would be ZF Intarder probably on the edge of legislation limit. But in future truck would be ZF Intarder certainly insufficient for legislation requirements. In comparison with CR, where is vehicle speed with CR (Chart 25) oscilating arround the target speed and in case of ZF Intarder is regulation smoother because brake torque is regulated by amount of oil passing through Intarder.

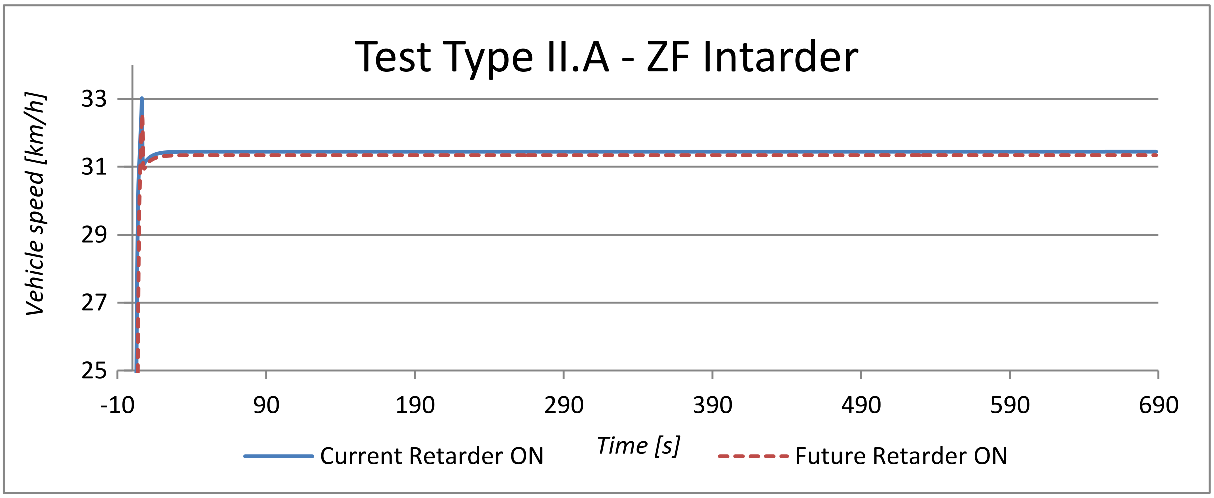


Chart 31 - Result of test type II.A with ZF Intarder for vehicle speed

Conclusion

To conclude, if all of the assumptions mentioned in this thesis were met, the following findings would occure.

Large engine downsizing has a substantial impact on brake power of Two-Stroke engine brake. Sensitivity analysis has shown that increasing of engine limitations would reduce the impact on downsizing the way that brake power of downsized engine would be just a little smaller than the original one.

Regulation of the engine brake could be more sensitive by adding more sophisticated regulation of Variable geometry turbine if the model of the engine brake would be connected to parametrical model of the vehicle. For purposes of this thesis was this engine brake regulation sufficient. Connection of the engine brake model with parametrical model of the vehicle could be the subject for next thesis with detailed emphasis on brake dust emissions and to find more exact responses of engine brake to vehicle.

By static calculations has been found that not only Conventional exhaust brake but also Bleeder brake would be insufficient to meet requirements of legislation type II.A.

Dynamic grade test shows that the currently most used type of engine brake - Compression release would be insufficient because high amount of energy would be dissipated through friction brakes. Two-Stroke engine brake would be in the future an ideal replacement for insufficient Compression release engine brake.

Performance limit test confirmed that the Two-Stroke engine brake would be an ideal replacement in the future for Compression release engine brake, because in case of use combination of both systems – engine brake and drivetrain retarder, the performance limit of future truck would be similar like in case of current truck.

Legislation test type II.A shows that ZF Intarder would be insufficient in terms of legislation. In the future, the brake power ot the ZF Intarder in low vehicle speeds should be increased or the ZF Intarder would has to be used in combination with any auxiliary brake.

To summarise all dynamic test confirmed that the Two-Stroke engine brake would be the best auxiliary brake for truck in low and mid vehicle speeds range.

References

[1] In: *Wikipedia: the free encyclopedia* [online]. San Francisco (CA): Wikimedia Foundation, 2001- [cit. 2019-07-10].

[2] CEDR report 2017/05 – Conditions for efficient road transport in Europe: Appendices ISBN: 979-10-93321-33-2

[3] *DUFFY, Owen C. a Gus WRIGHT. Fundamentals of medium/heavy duty commercial vehicle systems. Burlington, MA: Jones & Bartlett Learning, [2016]. ISBN 9781284041163.*

[4] LIMPERT, Rudolf. *Brake design and safety*. Warrendale, PA: Society of Automotive Engineers, c1992. ISBN 1560912618.

[5] *Hydraulic Drum Brake System* [online]. [cit. 2019-07-10]. Available from: https://www.quora.com/What-are-differences-between-Disc-Brake-and-Drum-Brake

[6] GSCHEIDLE, Rolf. *Příručka pro automechanika*. 3., přeprac. vyd. Přeložil Iva MICHŇOVÁ, přeložil Zdeněk MICHŇA, přeložil Jiří HANDLÍŘ. Praha: Europa-Sobotáles, 2007. ISBN 9788086706177.

[7] *Disc Brake Operation (floating calliper single piston)* [online]. [cit. 2019-07-10]. Available from: https://www.aa1car.com/library/brake_calipers.htm

[8] *With Braking Faster to Your Destination. Retarder [online]. In: . [cit. 2019-07-10]. Available from: https://voith.com/ita-en/brochure_1028_e_cr272_en_retarder_2014-12.pdf*

[9] *Driving resistances of lightduty vehicles in europe: present situation, trends, and scenarios for 2025* [online]. ,43 In: KÜHLWEIN, Jörg. Berlin, 2016 [cit. 2019-07-10]. https:/theicct.org/sites/default/files/publications/ICCT_LDV-Driving-%09Resistances-EU_121516.pdf

[10] *FIGURE 5-1 Energy "loss" range of vehicle attributes as impacted by duty cycle, on a level road.* [online]. [cit. 2019-07-10]. Available from: https://www.nap.edu/read/21784/chapter/7

[11] *Setting the Stage for Regulation of Heavy-Duty Vehicle Fuel Economy & GHG Emissions: Issues and Opportunities* [online]. , 69 [cit. 2019-07-09]. Available from: https://theicct.org/sites/default/files/publications/MJB_HDFEissues_091.pdf

[12] *Truck Powertrain* [online]. [cit. 2019-07-10]. Available from: http://www.coastpowertrain.com/graphics/flexslider/slide1.jpg

[13] *Volvo Trucks - Introducing the SuperTruck Concept Vehicle* [online]. 2016. [cit. 2019-07-10]. Available from: https://www.youtube.com/watch?v=KndAcoPLZQY

[14] ŠILAR, J. *Truck aerodynamic parameters.* Brno: Brno university of technology, faculty of mechanical engineering, 2013. 37p . Supervisor Ing. Petr Hejtmánek

[15] *Deturbulator* [online]. [cit. 2019-07-10]. Available from: http://www.sinhatech.com/Sinhadeturb/deturbulator.asphttps://www.quora.com/What-are-differences-between-Disc-Brake-and-Drum-Brake

[16] *DISCOVER HOW AERODYNAMICS CONTRIBUTE TO SAVING FUEL*[online]. [cit. 2019-07-10]. Available from: https://wattsupwiththat.files.wordpress.com/2009/11/semi-truck-savings.png

[17] *Wabco Boat Tail Deploys Automatically* [online]. 2016 [cit. 2019-07-10]. Available from:https://fleetimages.bobitstudios.com/upload/truckinginfo/content/product/_migrated/WABCO-OptiFlow-AutoTail-27Feb2017-1.jpg

[18] *Teardrop trailer* [online]. [cit. 2019-07-10]. Available from:*http://www.donbur.co.uk/gb-en/products/aerodynamic-teardrop-trailer.php#3d*

[19] *Correct tire pressure* [online]. [cit. 2019-07-10]. Available from: https://www.pneumatiky.cz/maji-vase-pneumatiky-spravny-tlak-t4

[20] *M.J.Bradley &Associates LLC: Summary of Available Technologies to Reduce Aerodynamic Drag & Rolling Resistance from Heavy Trucks* [online]. , 33 [cit. 2019-07-09]. Available from: https://theicct.org/sites/default/files/publications/AERO_RR_Tech nologies_Whitepaper_FINAL_Oct2012.pdf

[21] *Analysis of Key Parameter Ranges for HD Engine Market, 2011-2018* [online]. [cit. 2019-07-10]. Available from: https://www.slideshare.net/sandeepkar/north-american-truck-engines-are-shrinking

[22] Downspeeding Engines Make It Easy For Drivers To Enjoy The Ride While Saving On Fuel. *Road King*. **2015**, 2.

[23] KALNICKÝ, J. *Provozní a odlehčovací brzdy užitkových vozidel.* Brno: Vysoké učení technické v Brně, Fakulta strojního inženýrství, 2015. 54 s. Vedoucí bakalářské práce Ing. Jan Fojtášek.

[24] *TELMA TECHNICAL SPECIFICATIONS Axial AC61-60 retarder*[online]. , 6 [cit. 2019-07-09]. Available from:https://www.telmausa.com/Downloads/OC442101.pdf

[25] *Web of company Voith* [online]. [cit. 2019-07-10]. Available from: http://voith.com/corp-en/braking-systems/retarders-trucks.html

[26] *WRIGHT, Gus. Fundamentals of medium/heavy duty diesel engines. Burlington, Massachusetts: Jones & Bartlett Learning, 2017. ISBN 9781284067057.*

[27] *XIN, Qianfan. Diesel engine system design. Cambridge: Woodhead Publishing, 2011. Woodhead Publishing in mechanical engineering. ISBN 978-1-84569-715-0.*

[28] *Web of company Jacobs Vehicle systems* [online]. [cit. 2019-07-10]. Available from:https://www.jacobsvehiclesystems.com/technologies/bleeder-brake

[29] *Web of company Jacobs Vehicle systems* [online]. [cit. 2019-07-10]. Available from:https://www.jacobsdrivesthefuture.com/jacobs-introduces-1-5-stroke-hpd-high- power-density-engine-brake/

[30] ERBACH, Gregor. *CO2 emission standards for heavy-duty vehicles* [online]. April 2019, 10 [cit. 2019-07-10]. Available from: http://www.europarl.europa.eu/RegData/etudes/BRIE/2018/628268/EPRS_BRI%282018%29628268_EN.pdf

[31] *ECE R.13: PROPOSAL FOR DRAFT AMENDMENTS TO REGULATION No. 13*. In: . ročník 2004. Available also from: https://www.unece.org/fileadmin/DAM/trans/doc/2004/wp29grrf/TRANS-WP29-GRRF-56-inf03e.pdf

[32] *The brake dust particle filter from MANN+HUMMEL* [online]. 2017 [cit. 2019-07-10]. Available from: https://www.mann-hummel.com/en/the-company/current-topics/press- releases/the-brake-dust-particle-filter-from-mannhummel/

[33] *Perspectives: Enviromental Newspaper for Voith Turbo*. 09/2008 [cit. 2019-07-10]

[34] BENDL, Jiří. To byste nevěřili, kolik emisí z otěrů pneumatik, asfaltu a brzd dýcháme. *Ekolist.cz*[online]. 7.2.2011 [cit. 2019-07-10]. Available from: https://ekolist.cz/cz/publicistika/nazory-a-komentare/jiri-bendl-to-byste-neverili-kolik-emisi-z-oteru-pneumatik-asfaltu-a-brzd-dychame

[35] *Web of company Eaton* [online]. [cit. 2019-07-10]. Available from: https://www.eaton.com/us/en-us/catalog/transmissions/rt-18-transmission.specifications.html

[36] *Bibliography of company DAF* [online]. [cit. 2019-07-10]. Available from: http://www.dafcomponents.com/~/media/files/daf%20trucks/trucks/new%20cf%20and%20xf/infosheets/mx11/69637_daf_paccar_mx-11_engines_en.pdf

[37] *20 Things You Didn't Know About Enzo Ferrari* [online]. 05/08/2014 [cit. 2019-07-10]. Available from: https://www.thrillist.com/cars/20-enzo-ferrari-facts-that-you-might-not-know

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