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TURBOCHARGING OF IC ENGINES: AN OVERVIEW OF THE HISTORY, CURRENT STATE, AND THE DEVELOPMENT TRENDS

Dobrivoje Ninković[1]

UDC:621.431.73

ABSTRACT: Since the first patent granted in 1905 to the Swiss engineer A. Büchi for utilizing the IC engine exhaust gas energy for driving a turbine compounded with the engine, which is generally understood as the basis for the introduction of turbocharging, the latter has enjoyed a steady development first in the large Diesel engine segment, where it brought about efficiencies of 55%, becoming nowadays the indispensable technology for further progress in the field of IC engines in general.

Surveyed in the lecture is the development of the turbocharging concepts and realizations in the large engine field, where it matured and formed thus the basis for the introduction of the technology to the engines used to propel road vehicles. Although turbocharging is still an active R&D subject in the large engine segment, it is apparent that the main activities are now to be found in the field of small engines. The role of turbocharging as a means for further increasing the specific engine power, reducing both the production of CO2 and the emission of pollution gases detrimental to the environment while simultaneously improving driveability of the vehicles, is also discussed. Finally, an overview is presented of the current development trends in this area.

KEY WORDS: Turbocharging, IC engines, Downsiszing, Exhaust gases, Engine efficiency.

TURBOPUNJENJE MOTORA SUS: PREGLED ISTORIJE, TRENUTNOG STANJA I RAZVOJNIH TRENDOVA

REZIME: Još od 1905. godine, kada je švajcarskom inženjeru A. Büchi-ju bio odobren prvi patent za korišćenje energije izduvnih gasova u motorima SUS u svrhu pogona turbine spojene sa kolenastim vratilom motora, što se smatra osnovom za uvodjenje tehnologije turbpopunjenja, ta oblast se neprekidno razvijala prvo u oblasti velikih Dizel motora, gde je dovela do postizanja stepena korisnosti od 55%, postajući danas nezaobilazna tehnologija za dalji napredak na polju motora SUS uopšte.

U radu se prvo daje pregled razvoja koncepata turbopunjenja i njihove realizacije na polju velikh motora, gde je ta tehnologija sazrela i na taj način stvorila osnovu za transfer i primenu u motorima koji se koriste za pogon drumskih vozila. Iako turbopunjenje i danas predstavlja važnu razvojno-istraživačku temu u oblasti velikih motora, očigledno je da se sada težište istraživanja nalazi u segmentu malih motora. U radu se zato razmatra uloga turbopunjenja kao sredstva za dalje povećanje specifične snage motora, čime se smanjuje proizvodnja CO2 i emisija gasova štetnih za okolinu uz istovremeno poboljšanje voznih osbina vozila. Na kraju rada daje se prikaz aktuelnih razvojnih trendova u ovoj oblasti.

KLJUČNE REČI: Turbopunjenje, Motori SUS, Specifična snaga, Izduvni gasovi, Efikasnost motora

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UDC:621.431.73

INTRODUCTION

The advantages of filling the IC engine cylinders with air by using a compressing device of arbitrary design in comparison to their counterparts with air suction were recognized very early in the engine development. It is known that late in the 19th century both Gottlieb Daimler and Rudolf Diesel worked on concepts for forced cylinder induction by mechanically driven devices.

In 1905, Swiss engineer A. Büchi applied for a patent on a method for utilizing the exhaust energy of an IC engine for driving a turbine attached to the crankshaft of the same engine, creating thus the so-called compound engine. However, there was also a centrifugal compressor on the crankshaft, the purpose of which was to supply the engine with compressed air, increasing thus the engine specific power. This patent is generally seen as the generic one for the introduction of forced cylinder induction, i.e. charging with air, whereby the compressor is driven by a turbine that in its turn extracts the energy from the engine exhaust gases. For obvious reasons, this technology is referred to as turbocharging.

A similar method of forced cylinder induction whereby a centrifugal compressor is driven by mechanical means, such as e.g. the engine crankshaft, is commonly termed super-charging. This technology found wide¬spread use in the field of aircraft IC engines, especially in the period before and during the World War II. A great majority of aircraft engines of that time were equipped with sophisticated supercharging systems, usually driven by variable-speed drives.

However, turbocharging represents nowadays the prevalent method for forced cylinder induction practically in all IC engine sectors. It is seen as a prime means for increasing the power density of IC engines, which in turn results in reduced CO2 production. Although the former has always been the goal of turbocharging in the large engine sector, it is in conjunction with the intensifying application of this technology to road vehicle engines that a new term was coined, referred to as "downsizing". It is taken to mean improving the power and torque output of the engine while reducing its size, whereby the inevitable performance loss thus incurred is compen-sated by increased use of turbocharging.

Turbocharging is also instrumental in meeting the ever-stricter emission requirements posed to the engine manufacturers by the legislation bodies, since downsizing and emission reduc¬tion often represent conflicting requirements.

It is also mainly due to turbocharging that modern IC engines are to be found amongst the most efficient thermal machinery in general. For example, modern slow-running two-stroke Diesel engines attain efficiencies of 55%, and the newest generator sets driven by four-stroke gas engines with two-stage turbocharging reach an efficiency figure in excess of 46% referred to the electrical output [1].

The term two-stage turbocharging can refer both to series and parallel connections of two indivi¬dual turbochargers. The former is employed in order to obtain boost pressures

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higher than the ones possible with a single compressor, whereas the latter is used when a larger mass flow rate is needed at the same boost pressure. In this paper, this term will be used exclusively to refer to a series connection of two turbochargers.

MAIN REASONS FOR THE INCREASING SIGNIFICANCE OF TURBOCHARGING

Analyzing the history of turbocharging, it is possible to speak of two time periods in which the development and application have been located in two different IC engine segments, whereby the main driving forces were not quite the same. The beginning of the first one is located at the start of the past century, coinciding roughly with the first patents relating to turbo-charging. The field of application consisted mainly of large four-stroke Diesel engines, at first for ship propulsion, and later also for locomotives [2]. The reason for investing into the new techno-logy was that a turbocharged engine became by this means able to deliver more power than its naturally aspirated counterpart of the same size. This meant more room for the cargo, which was well understood by the ship owners, generating thus the demand for such engines, and driving the research and development. The production of turbochargers took place on the one-off basis, requiring close cooperation between the engine and turbocharger manufacturers [3].

The second period of intensive turbocharging technology development started some 25 years ago in the small IC engine sector. Customarily, the latter is taken to mean engines used on vehicles for road transportation, i.e. on trucks, buses, and passenger cars. The development pace has been very fast, resulting in practically all modern Diesel engines produced today being equipped with sophisticated turbocharging systems, whereas engines even with simple turbochargers were a minority only twenty years ago.

This trend has been brought about by the exhaust gas legislation in the developed countries, which started at the beginning of the seventies of the last century by limiting the carbon monoxide (CO) for Otto engines and smoke and sulphur oxide (SO2) emissions for Diesel engines, and culminating in the UNECE Convention of 1979 [4]. While the engine manufacturers were able to meet the ever stricter CO and HCx emission norms for Otto engines by a combination of catalytic exhaust gas converters, closed loop engine control, and engine design changes, it was the Kyoto Protocol of 1993 regarding the emissions of carbon dioxide (CO2) that posed a new challenge to the entire road transportation industry. Limiting the CO2 emissions means reducing the fuel consumption, and this can not be achieved by the exhaust gas treatment methods only.

Referring to Fig. 1 below, between 2005 and today CO2 emissions of newly produced vehicles were either reduced or requested to be reduced at annual rates between 2 and 4.8%. The technology that helped achieve most of these results has been the downsizing, as already mentioned in the Introduction.

According to the above diagram, the CO2 emission values of 2015 will have to be reduced by 50% in the next ten years, which in combination with the simultaneous reduction of other pollution gases constitutes a radical requirement that calls for further development of the downsizing, of which turbocharging is an inseparable part.

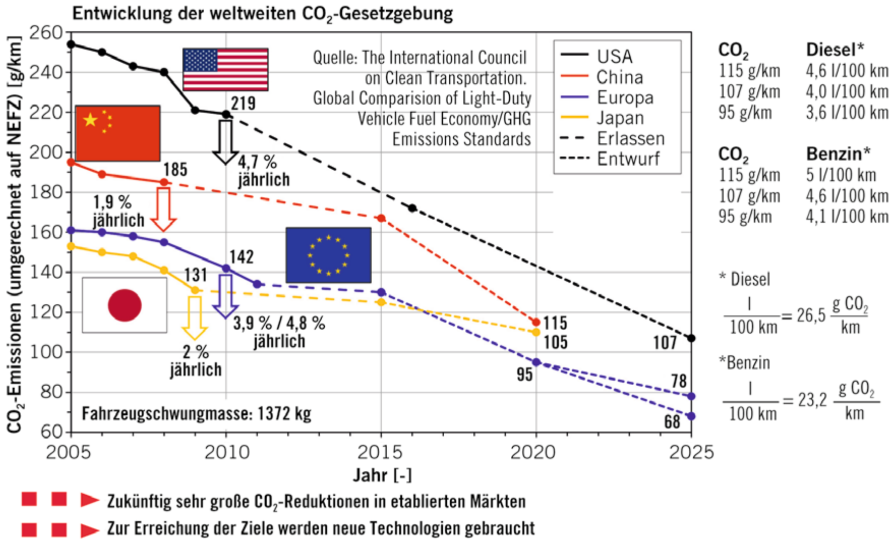

Figure 1. The CO2 emission legislation worldwide [5]

The above two periods in the development of the turbocharging technology are not sharply divided between the large and small engine sectors. For example, Swiss company Saurer (now Iveco) started series production of turbocharged Diesel engines for trucks and buses in 1938, the reason being power loss in naturally aspirated engines at high altitudes in Swiss Alps. A large number of turbochargers were built in the U.S.A. for aircraft engines in the Second World War, again in order to compensate for the power loss at high altitudes.

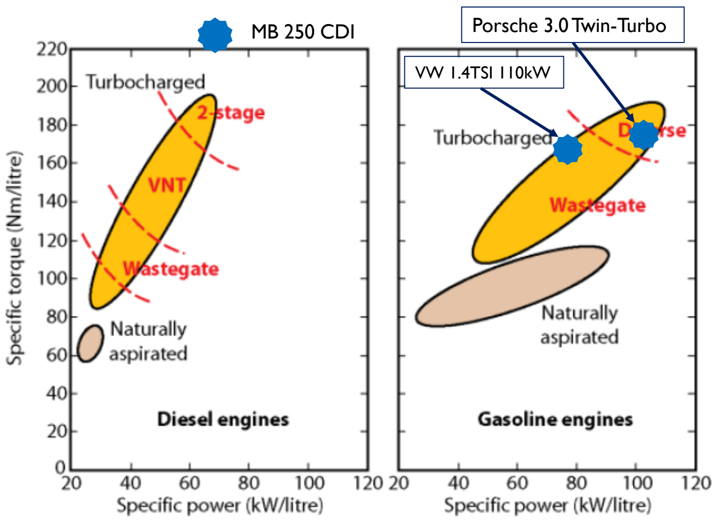

Figure 2. Comparison of naturally aspirated and turbocharged road transport engines [6]

Turbocharging modifies both the torque and power delivery characteristics of downsized engines. A comparison of naturally aspirated Diesel and gasoline engines with their turbocharged counterparts of Fig. 2 above shows a trend toward higher torque densities in the Diesel engine case and higher power densities in the gasoline engine case. The new data (MB 250CDI, VW 1.4TSI 110kW, and Porsche 3.0 Twin-Turbo) augmenting the ones published by Baines in 2013 [6], makes it apparent that these trends continue with modern engines.

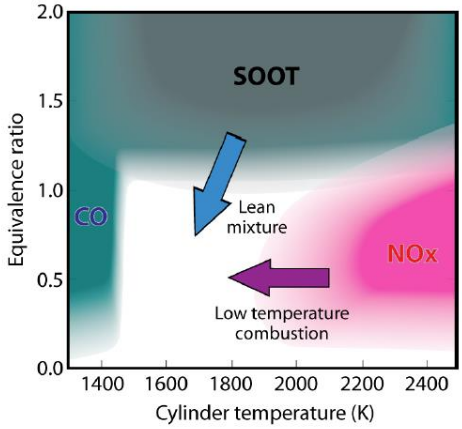

Figure 3. Clean combustion requirements [6]

Reduction of the emissions of polluting gases is the second area where turbocharging contributes in achieving the legislation targets. As the illustration in Fig. 3 above shows, in order to arrive at a clean combustion, the engine must operate with a lean mixture and at a low temperature, both of which tend to reduce the specific engine power. Turbocharging is the means for compensating the specific power loss thus incurred.

An important method for nitric oxide (NO_x) reduction in the exhaust gases is the Exhaust Gas Recirculation (EGR), which can be realized either as a low-pressure or a high-pressure variant, or a combination thereof. The high-pressure EGR can not be implemented without turbocharging; especially in the case of large engines because of the positive pressure difference across the engine. Referring to Fig. 4 below, it is clear that the cleaned exhaust gas, whose pressure is further reduced in the particle filter and/or catalyst, must be compressed before being introduced into the air receiver of the engine.

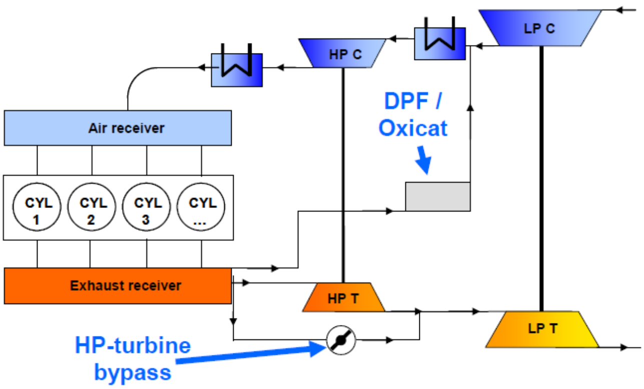

Figure 4. A high pressure EGR schema [7]

HISTORICAL REMARKS

As already mentioned, the first turbocharging patent is now almost 110 years old. The patent claim (see. Fig. 5 below) specified a turbine driven by the engine exhaust gases, and a centrifugal compressor supplying the engine with compressed air, all on the same shaft. Therefore, the invention specified both forced cylinder induction and a parallel connection of the turbine with the engine crankshaft, the latter being referred to as compound engine.

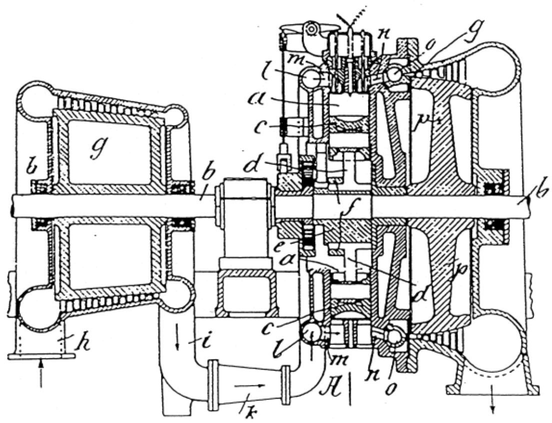

Figure 5. The Büchi compound engine and turbocharger of 1905

This patent was followed by another three, whereby the one of 1925 is customarily referred to as the key turbocharging patent. Referring to Fig. 6 below, the patent sketch shows an eight-cylinder engine with two turbochargers, whereby the turbines are connected to two suitably chosen cylinder groups. The criterion for the cylinder grouping was to minimize the interference between the individual cylinders in their respective exhaust phases, which could come about if an exhaust pulse from a cylinder would arrive at the exhaust valve of another one just at the moment of the latter's opening. In this way, the major part of the energy contained in the large-amplitude waves created by the exhaust pulses would be used for accelerating the turbine instead of being wasted in the wave reflections at the cylinder boundaries. This method is still in use today, augmented by special design features of the exhaust manifolds [8][9].

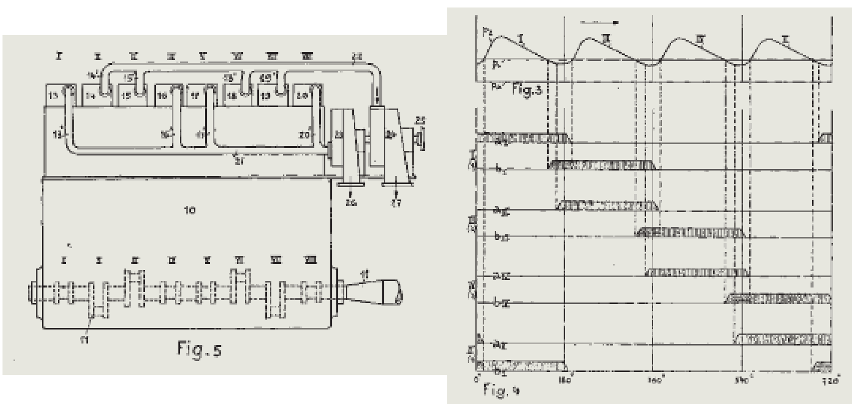

Figure 6. The Büchi pulse charging patent of 1925

The first turbocharged engine appeared in 1917. The turbocharger was designed by Prof. Rateau, and consisted of a tungsten steel turbine and a compressor with straight blades, rotating at max. 35000 rpm. It was mounted on a Renault aircraft engine, but due to a disappointing performance was later replaced by a mechanical supercharger [10].

This attempt was followed 1918 in the same application area, i.e. that of the military aircraft, by the design of S.A. Moss [10]. This one was a success, leading almost immediately to series production that extended throughout the entire World War II. Most of American WW II aircraft were equipped with turbochargers, whereas the European ones used almost exclusively mechanically driven superchargers.

The first commercially built turbocharger for a large engine was delivered in 1924 by the Swiss company Brown Boveri of Baden (BBC) to Swiss Locomotive & Machine Works (SLM) of Winterthur [3]. It was a single turbine, two-stage compressor design, delivering air at a pressure ratio of mere 1.35 to the scavenging blowers of a two-stroke Diesel engine running on a test stand.

The first two maritime vessels to become turbocharged Diesel engines as the main propulsion units were the sister ships *MS Preussen* and *MS Hansesstadt Danzig,* commissioned in 1927, and each fitted with two 10-cylinder, four-stroke Vulkan-MAN engines. The design and calculations of the turbocharging were done under the supervision of A. Büchi, and the engines employed Brown Boveri turbo-blowers.

The first two-stroke turbocharged Diesel engine had to wait until 1946 to be commissioned in the power station of Sulzer Works in Winterthur; and another six years passed before the first two-stroke, 6-cylinder unit made by Burmester & Wain, entered operation at high seas on board of a tanker *MS Dorthe Mærsk*. The engine was turbocharged by two Brown Boveri units that increased its power from 5530 to 8000 HP, which roughly corresponds to the charging pressure of 1.5 bar [2].

Although the advantages of turbocharging were apparent, the development proceeded rather slowly; it took 25 years after the first four-stroke marine Diesel engines were commissioned to see the first two-stroke Diesel engine on a ship. Generally, it was after the World War II that the development of turbocharging in the large engine sector accelerated, making it possible to replace the steam turbine as the main propulsion means at sea, and the steam engine on the rail.

The first turbocharged gasoline engines for passenger cars appeared in 1962, propelling the Oldsmobile Jetfire and Chevrolet Convair Monza models of GM Company. Due to reliability and performance issues, they did not have much success, and the sales almost stopped in 1965.

The first passenger car with a turbocharged Diesel engine was the Mercedes-Benz 300SD, released for sale only in the U.S.A. The car remained on the U.S. market until 1985, but the engine was also offered in Europe in other Daimler-Benz models, to be replaced by a more modern version in 1986.

Each of the above development areas will be dealt with in more detail in the Chapters that follow.

However, there is a subject common to all turbocharger application areas that must be briefly mentioned here. First turbochargers and their successors remained for quite a long time self-contained machines, practically custom-designed for a given engine. However, matching a set of two rotational machines connected by a common shaft to a positive-displacement machine, i.e. the engine, over a wide range of operating conditions represents an almost impossible task due to the widely differing characteristics of the turbocharger and the engine. While the matching task is somewhat simpler in the case of large engines because of the well-defined operational lines, achieving good matching over the operational areas of road vehicle engines cannot be made without some form of control over the characteristics of the turbocharger. In principle, the control function can be realized by varying the respective geometries of the turbine and the compressor and/or by manipulating the flow through the machine.

Since varying the compressor geometry is very difficult to accomplish, especially in the case of highly-optimized machines, and the turbine geometry is more amenable to being varied, many control devices have been developed for this task. Referring to Fig. 7 below, the most frequently used devices either vary the geometry of the turbine stator (commonly referred to as variable turbine geometry (VTG)), or change the cross-sectional area of the channels supplying the hot gas to the rotor.

Figure 7. Turbocharger turbine control elements [10]

The gas flow control concepts consist of placing valves of various design into the turbine and/or compressor gas conduits, varying thus the mass flow rate through the respective machines. Some of the traditional and new valve designs are shown in Fig. 8 below.

Modern turbocharging concepts involve sophisticated control schemes in order to achieve optimal engine characteristics over the entire operating area. Since the subject of turbocharger control is a rather special area, its treatment is beyond the scope of the present paper. However, it should be borne in mind that they nowadays represent an inseparable part of any turbocharged system.

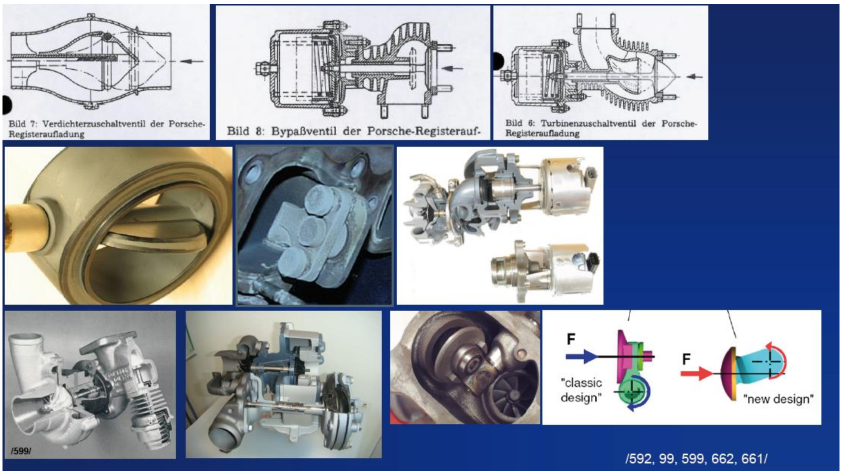

Figure 8. Gas flow control elements [10]

LARGE ENGINE TURBOCHARGING DEVELOPMENT

The large engine sector consists of Diesel and gas engines with a turbocharged power, i.e. power per turbocharger unit, of more than 500kW. The Diesel engines are furthermore divided into the two-stroke and four-stroke groups.

The two-stroke Diesel engines are almost invariably large, slow-running (n < 180 rpm) units, the chief application areas being ship main propulsion and stationary power generation plants. With thermodynamic efficiencies of the order of 55% they, together with combined-cycle gas turbines, represent the most efficient thermal machines available today. Due to their size, they are generally equipped with several turbochargers.

The four-stroke Diesel engines are used in a number of application areas, such as ship main propulsion, ship auxiliary services (e.g. electricity generation), stationary power plants, locomotives, and heavy off-road machinery. They are customarily divided into the low-speed (up to 300 rpm), medium-speed (300 – 1000 rpm), and high-speed (> 1000 rpm) groups.

Large gas engines represent an environmentally-friendly prime mover group, since the advantages of gaseous fuels in this regard are well known. However, sophisticated turbocharging and control systems are required in order for them to achieve performance figures comparable to large Diesel engines [1]. Their main application areas of large gas engines are gas transport (driving piston compressors recompressing the gas transported through pipelines), and the production of electrical energy.

In turbocharging, it is the combination of the boost pressure and mass flow rate of the turbocharger compressor that defines the engine performance. Referring to Fig. 9 below, where the development of the compressor pressure ratio of Brown Boveri (now ABB Turbo Systems) turbochargers with time is taken as an example, there has been a steady increase in the available boost pressure ever since the first units were manufactured, making it possible to achieve remarkable degrees of downsizing in the large engine sector [11].

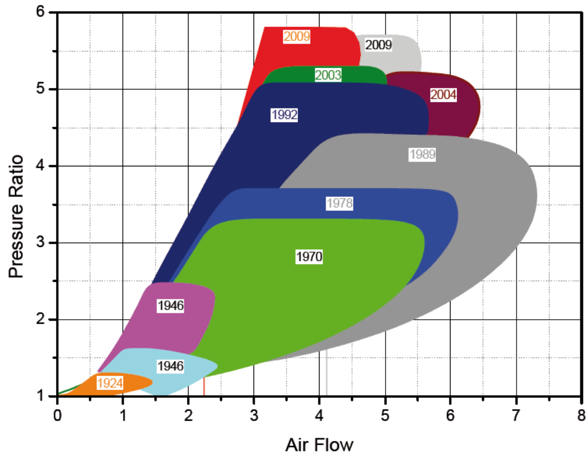

Figure 9. The development of the compressor pressure ratio of ABB turbochargers

Large engines typically operate along well-defined operation lines, such as e.g. the so-called propeller, or generator lines, which makes them amenable to a high degree of optimisation. As a consequence, the corresponding turbocharger compressor performance maps are rather narrow, extending to high charging pressures (see the LHS diagram in Fig. 10 below, showing the map of a modern ABB A140 turbocharger compressor for four-stroke engines). Road vehicles, on the other hand, must operate over a wide range of operating conditions, which in turn requires broad compressor maps. Since the delivery (boost) pressure and the mass flow rate are in opposition to each other when designing a centrifugal compressor stage, small engine turbocharging compressors generally have lower end pressures.

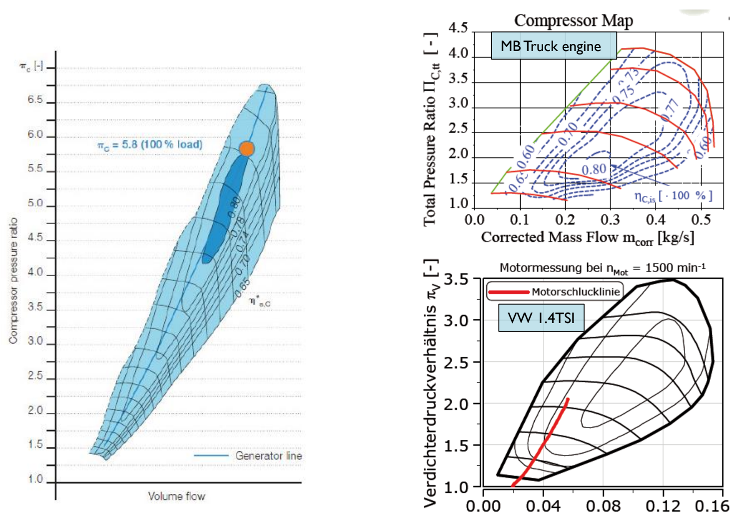

Figure 10. Compressor maps of large and small engine turbochargers

Although the high compressor pressure ratios of modern four-stroke large engine turbochargers make it possible to achieve high specific powers, meeting the current and future exhaust gas pollution norms without sacrificing the fuel economy calls for changes in the engine thermodynamics as well. One of the most capable ideas in this regard is the so-called Miller process, the application of which results in lower gas temperatures in the cylinder, reducing thus the production of nitric oxides in the combustion phase [8]. However, the Miller process needs much higher charging pressures than are possible even with the most modern single-stage compressors. Thus two-stage turbocharging has been developed, consisting of two turbochargers in series, capable of attaining charge pressure of the order of 10 bar [12]. The need for even higher boost pressures led to the development of the second generation of two-stage machines with pressure ratios increased by another 20% over the first generation ones [13].

Referring to Fig. 11 below, the diagram on the LHS shows a comparison of the ABB Turbo Systems two-stage turbocharging compressors of the first and second generations. The latter is characterized by a higher delivery pressure, but also by a broader map at higher pressure ratios.

The RHS diagram in Fig. 11 is more significant for judging the performance of the two-stage turbocharging, and especially of the 2nd generation. Plotted in the diagram is pressure difference across the engine as a function of the boost pressure ratio, with the turbocharger efficiency at two turbine inlet temperatures as parameter. It is seen that the turbocharger efficiency has been increased from 0.6 in the single-stage case to 0.75 in the 2nd generation of the two-stage one. At the same time, pressure difference across the engine has been considerably increased, which translates into fuel savings at lower NOx emissions. The interested reader is referred to the original article [13] for further details, especially as regards the trade-offs between the fuel consumption and exhaust emissions.

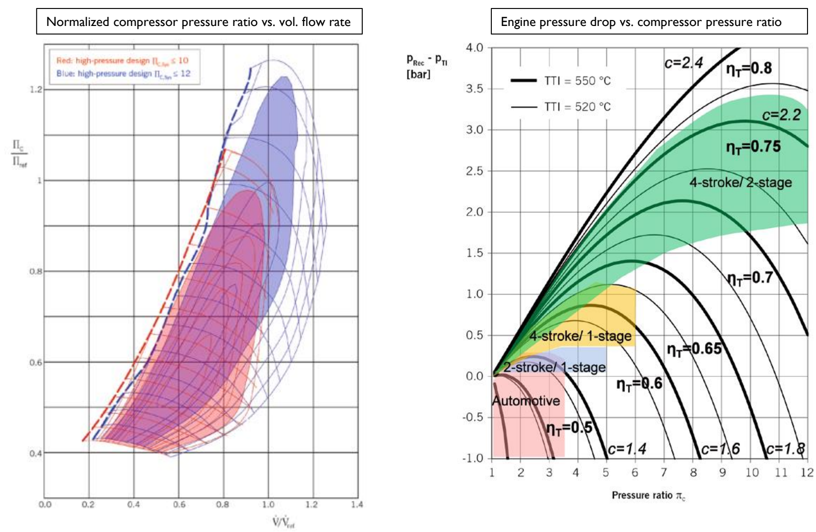

Figure 11. Performance maps of the 2nd generation of ABB two-stage turbochargers [13]

A good example of the potential already realized with two-stage turbocharging is the first large gas engine employing this technology. The engine is the 24-cylinder, 4.4 MW, GE-Jenbacher J624, driving a generator [1]. In comparison with the previous, single-stage turbocharged version with an electrical efficiency of 38%, the new one has an efficiency of 46.5%. The authors claim that by combining the engine with an excess heat recovery system, total efficiency of over 90% may be possible.

Increasing the BMEP (Brake Mean Effective Pressure) is also a means for increasing the engine power density. Newest research into the effects of raising the BMEP from 26 bar (routinely attainable with single-stage turbocharging) to 40 bar (peak cylinder pressure of 365 bar), carried out with a special single-cylinder experimental engine at the Hamburg University of Technology ([14][15]), revealed that meeting the IMO Tier II limits is possible with a simultaneous and remarkable fuel efficiency improvement. Major part of this success is due to two-stage turbocharging (gas exchange improvement) and Miller timing [15].

TURBOCHARGING OF COMMERCIAL VEHICLE DIESEL ENGINES

As already mentioned above, turbocharging of truck and bus Diesel engines commenced in 1938 in Switzerland with the production of the Saurer 8.55 litre BLD engine charged by a Brown Boveri turbocharger with a boost ratio of 1.4, increasing thus the engine power from 73 to 100 kW. As already mentioned, main reason for the introduction of turbocharging was to compensate for the power loss the naturally aspirated engines experienced when operating at high altitudes in Swiss Alps.

Referring to Fig. 12 below, the turbocharger can be seen in the upper right part of the engine. Clearly visible are the two exhaust pipes feeding the turbine, realizing thus the Büchi patent of 1925 as a two-pulse system. The latter is known to provide a faster acceleration of the turbocharger turbine by utilizing the energy of the large-amplitude waves being generated in the exhaust manifolds as a consequence of the cylinder discharge process. Fast turbine acceleration is instrumental in minimizing the so-called turbo hole, i.e. the delay in the engine power delivery in reaction to the demand signalled by the vehicle driver. Clearly, this is of special importance in the case of a heavy vehicle on a steep mountain road.

Apart from this engine and its derivatives, there were practically no further turbocharged engines of importance in the commercial road vehicle sector until the release of the MAN D1546 6-cylinder, 8.3 litre engine in 1951, and the Scania 8-cylinder, 11 litre engine in 1953, both with Brown Boveri turbochargers, operating in the pulse-charge mode. Mercedes-Benz also did some work on turbocharged truck engines at that time, but the application was limited to special vehicles, remaining so for another 30 years [16].

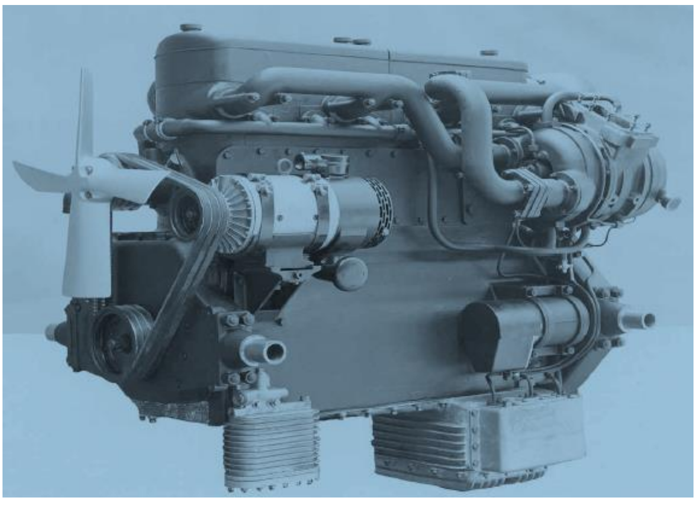

Figure 12. Saurer BLD engine with a Brown Boveri Turbocharger (1938)

Apparently, it was the exhaust gas legislation that gave rise to the intensification of the development and application of turbocharging to commercial vehicle engines, as witnessed by the Mercedes-Benz timeline in this area [16]. Referring to Fig. 13 below, there is a clear trend towards turbocharged engines from about 1980 onwards, i.e. from the time when the first exhaust gas legislation came into effect.

The accelerated development of turbocharging in this area has resulted in modern engines meeting all the relevant emission control norms. State of the art nowadays is single-stage, regulated turbocharging, although there is also a two-stage turbocharged MAN D2676 engine that meets the Euro VI and IMO Tier IV exhaust emission norms.

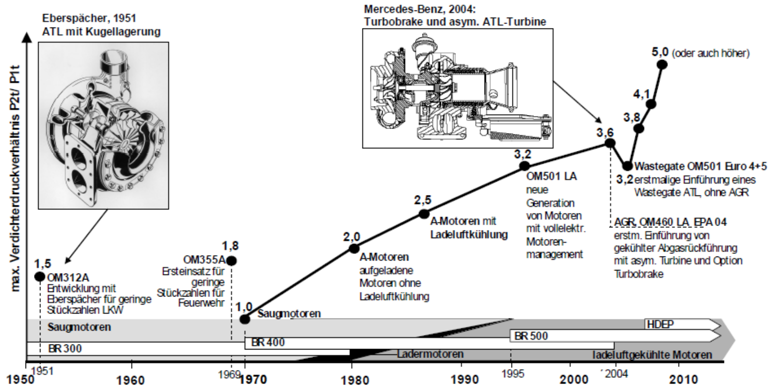

Figure 13. Mercedes-Benz commercial vehicle Diesel engine timeline [16]

However, in conjunction with the turbocharging of commercial vehicle engines there are two developments worth of being specially mentioned. The first one relates to the first patent application of Büchi in 1905 (the compound engine), whereas the second one deals with using the turbocharger as a vehicle braking device.

Compound Diesel engines have been designed and used in the past, but always in rather specialized fields, such as aircraft propulsion, e.g. Napier Nomad I and II [18], or the Zvezda M503 engine for military marine applications [17]. However, since the introduction and series-production of the Scania 11L turbo compounded engine in 1991, all major Diesel engine manufacturers have such systems in the production programs.

The Scania system (see Fig. 14 below) consists of a power turbine connected to the engine crankshaft by means of a hydraulic coupling and a gear reductor. Reportedly, such systems bring about BSFC efficiency increases of 5% on the average, but have also drawbacks in being complicated, bulky, and expensive. The new research seems to favour turbo generators as a more efficient, and easier to apply and control, solution for future designs [20][21].

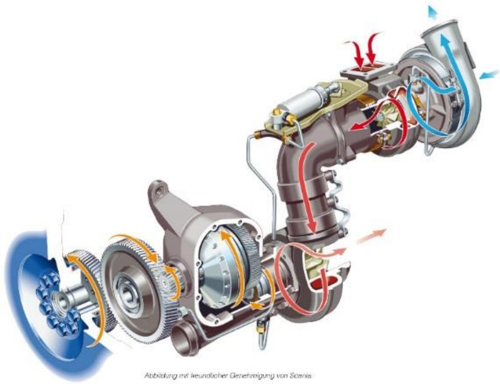

Figure 14. The Scania 11L engine turbo compounding system [19]

Turbo-braking is an interesting extension of the turbocharging in that the turbocharger is used as a brake by manipulating the gas flow cross-sectional area of the turbine. The need to develop this kind of a non-friction brake arose from the deficiencies of the standard Diesel engine retarder, such as temperature limits, fading action, weight, costs, etc. [16]. This solution disposes with the standard retarder throttle downstream of the turbine, replacing it with a redesigned, variable cross-sectional area, twin-scroll turbocharger turbine. The large mass flow rate available to the turbine at braking drives the compressor, which in turn charges the engine at high pressure, transforming the latter to a charged piston compressor. Braking power values of the order of 50 kW/litre of engine displacement have been realized with this system, which make it possible to increase the mean truck velocities in the long-haul road transport without having to invest into more powerful friction brakes [16].

PASSENGER CAR DIESEL ENGINE TURBOCHARGING

While the Mercedes-Benz 300SD of 1974 was the first series-produced turbocharged passenger vehicle, it remained the only one of this kind for almost 15 years. The turbocharged Diesel breakthrough in the passenger car market began with the introduction of the Audi 5-cylinder, 2.5 litre engine in 1989. Although preceded by the launch of the Fiat Croma 3.0 TD i.d. (*iniezione diretta*) in 1986, due to the latter being restricted for sales at the Italian market only, it was the Audi engine that created the necessary impact for the breakthrough to happen.

Passenger cars driven by naturally aspirated Diesel engines have been available long before the turbocharged ones came to the market, but the strengthened exhaust emission norms made a new concept necessary. The solution found was based on a combination of direct injection and turbocharging.

An exemplary overview of the Diesel engine development for passenger cars can be seen in the Mercedes-Benz timeline of Fig. 15 (the 300SD data added by the present author). It can be seen that the 300SD was better in terms of the specific power and torque than the 1983 2.5 litre, naturally aspirated engine of more modern design. The first common rail, direct injection, turbocharged engine was introduced in 1997 as a precursor of a very successful engine series, with the most recent one attaining the average fuel consumption of 4.1 l/100km. The engine version of 2014 and its predecessor of 2011 are equipped with a sophisticated two-stage turbocharging that includes gas path control actuators.

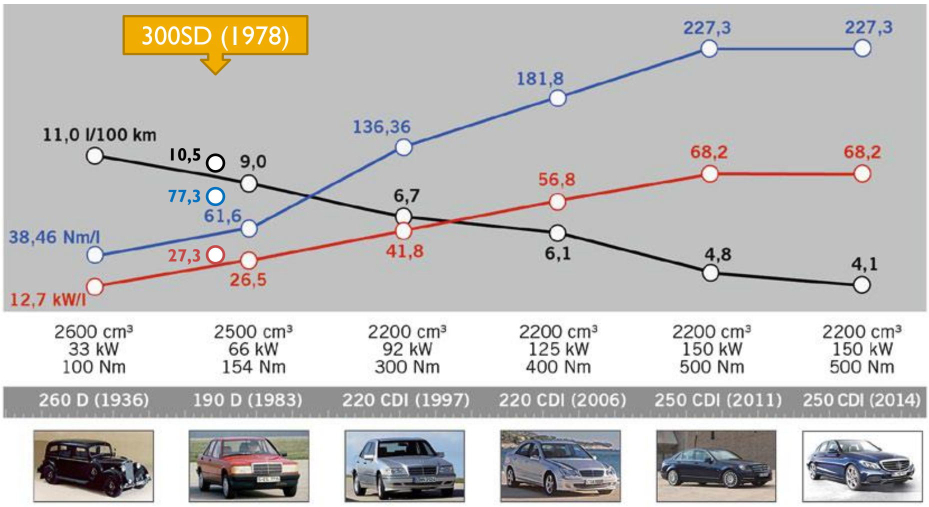

Figure 15. The Mercedes-Benz passenger car Diesel engine timeline [22]

An example of a modern two-stage turbocharged engine is presented schematically in Fig. 16 below. The engine in question is the VW/Audi 2.0 TDI, turbocharged by the Borg Warner R2S turbocharger with a single air cooler. Both turbochargers have fixed geometries, the control being effected by means of three valves in the air and exhaust manifolds. In addition, there is also a high-pressure EGR system, operating on the positive pressure difference between the exhaust and air manifolds (unlike the large engines, where the opposite is the case). With variations in the turbocharging tract, the engine is used in both passenger cars and light commercial vehicles.

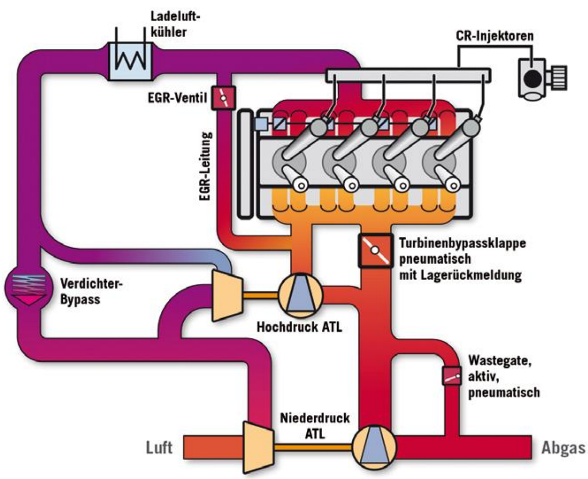

Figure 16. The VW/Audi 2.0 TDI engine [23]

The turbocharging control areas of the engine are shown in Fig. 17 below [8]. The engine map is shown in terms of torque vs. engine speed, with the shaded areas designating constant BSFC values. At low and high engine speeds, the turbocharger is running in the two-stage and single-stage configurations, respectively, whereas both machines are at various individual degrees of part-load in the middle region of the map. The map also shows that full torque is available at very low engine speeds, which is characteristic for all modern Diesel engines.

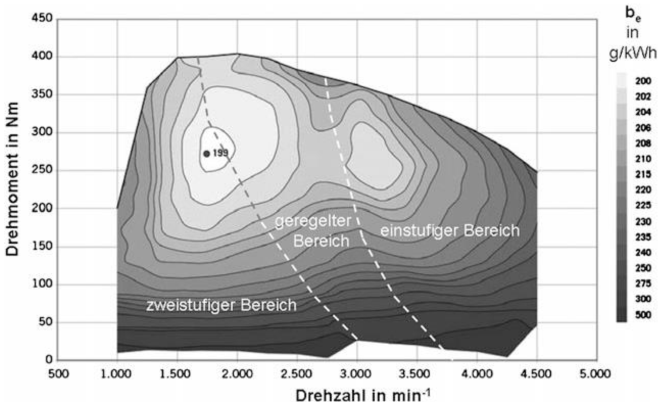

Figure 17. The control areas of the VW/Audi 2.0 TDI engine [8]

Two-stage, controlled turbocharging represents the state-of-the-art with the passenger car Diesel engines of today. According to Borg Warner [24], this concept is adequate at power densities of up to approx. 75 kW/litre. Three-stage turbocharging has already been applied, as in e.g. BMW 6-cylinder, 3.0 litre engine, bringing about a BSFC improvement of at least 8% [24].

PASSENGER CAR GASOLINE (OTTO) ENGINE TURBOCHARGING

After the limited success with the turbocharging of the two GM models in 1962, and considering the general tendency towards knock when the gasoline/air mixture is compressed to high pressures, turbocharged gasoline engines were not mass-produced until recently. Porsche seems to be the only manufacturer to have had turbocharged gasoline engines in the production program from 1974 onwards. But, as the Fig. 18 below clearly shows, the reason was not increasing the fuel efficiency, but rather improving "the pleasure of driving".

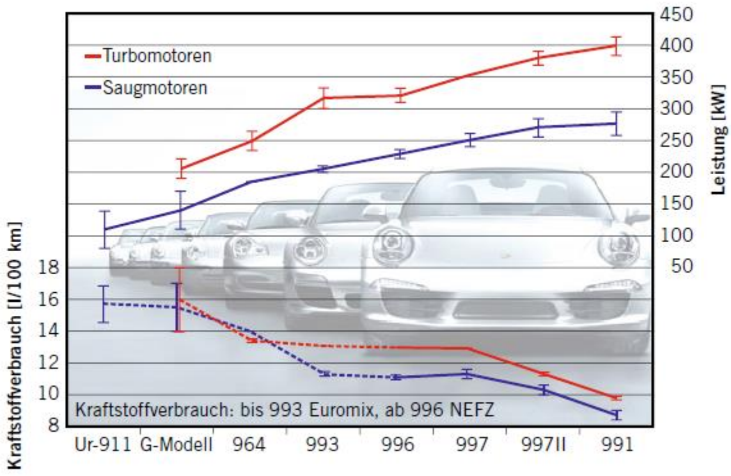

Figure 18. The Porsche 911 gasoline engines from 1974 until today [25]

Downsizing gasoline engines with fuel injection into the inlet channels was not successful, and the need thus arose for a different concept, namely direct gasoline injection into the engine cylinders. Bearing in mind that most WW II aircraft engines were equipped with this fuel supply system and were also supercharged, this concept was not entirely new. But apart from the Mercedes 300SL of 1953, which had a naturally aspirated engine with direct gasoline injection, no other engine manufacturer seems to have had such an engine until only recently, most probably on cost grounds.

However, inspecting the Mercedes-Benz gasoline engine timeline of Fig. 19 below, it is clearly visible that the 300SL engine was "downsized" to 50 kW/litre, a value that Mercedes only improved some 30 years later. The breakthrough with this engine type came with the combination of turbocharging and direct gasoline injection; and in order to meet the CO_2 legislation, downsizing of 50% was seen to be necessary. In addition, the norms regarding the emission of the pollution gases, especially of NO_x, can only be met in conjunction with other means, such as EGR, variable valve timing, etc.

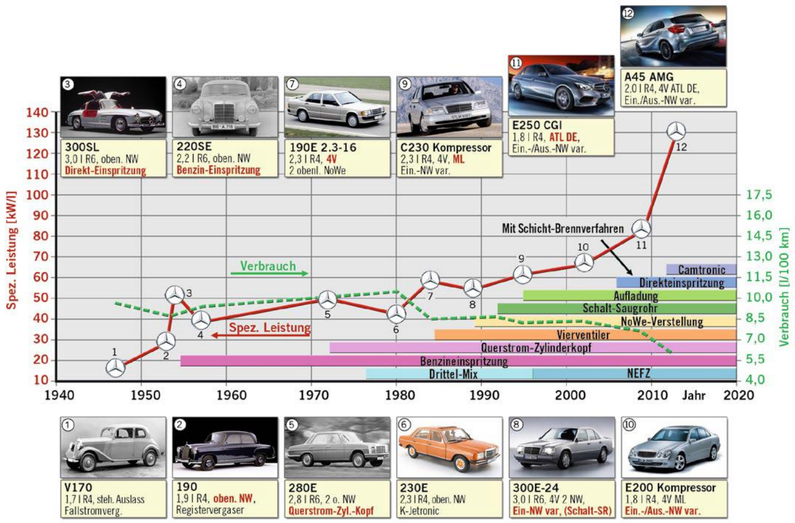

Figure 19. The Mercedes-Benz gasoline engine timeline [22]

The first downsized gasoline engines with direct fuel injection appeared some ten years ago. A characteristic example of the new engine generation is the VW 1.4 TSI, 132 kW engine, which in the first version of 2005 had a turbocharger and a Roots compressor. The torque curve was shifted towards low engine speeds, similar to the modern Diesel engines. The development of the new technology has been very fast, such that the internal efficiency of a GDI engine is now close to the values characteristic of modern Diesel engines [22].

The chart in Fig. 20 below illustrates the relationship between the engine power and specific power for various values of the displacement volume (the engine examples have been entered by the present author). The diagram also shows the limits of single stage turbocharging as of 2010 [26]; and it is also seen that two example engines from the current production are still below the downsizing limit possible with single-stage turbocharging. However, further development of single-stage turbocharging with a segmented (twin-scroll) turbine on a four-cylinder Mercedes-Benz engine clearly demonstrates further downsizing potential, reaching a specific power of 115 kW/litre simultaneously with a fuel consumption reduction by almost 20 g/kWh [33]; and one current GDI engine for sport cars (Mercedes

AMG 265 kW with a single-stage, twin-scroll turbine) demonstrates the downsizing potential of single-stage turbocharging taken to the extreme (see also engine No. 12 in Fig. 19 above).

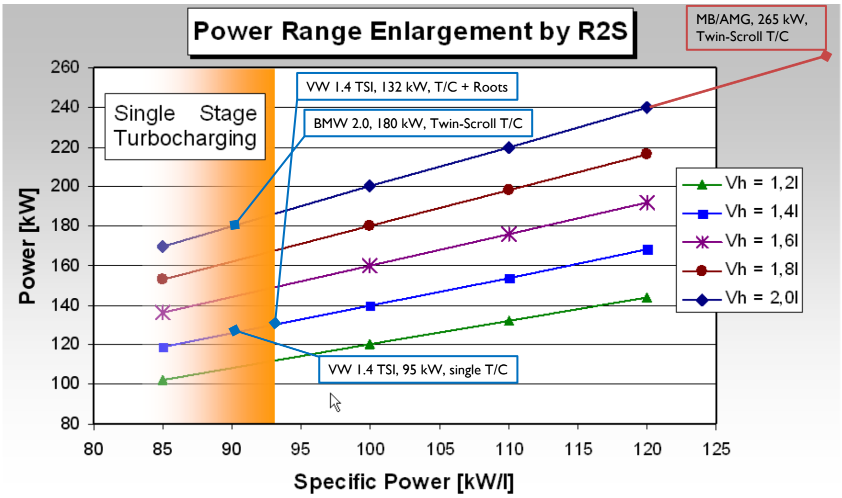

Figure 20. Downsizing possibilities and current engine examples [26]

Note that that on account of its firing order, the four-cylinder engine is the most difficult one to turbocharge; and in order to minimize the interference between the neighbouring cylinders, modern four-cylinder GDI engines are usually equipped with exhaust manifolds divided into two groups, which also provides for a full torque at very low engine speeds. An example of this is the Mercedes MB270 1.6 litre, 115 kW engine, which attains the full torque of 245 Nm already at the speed of 1250 rpm [8].

The turbocharged GDI engine of today is an established product, but there is still a need for further downsizing efforts, and other improvements. With regard to turbocharging, the three-cylinder engine seems to be an ideal basis platform for further development on account of its size and the ideal three-pulse exhaust pressure pattern [8]. Examples of this approach are the Ford EcoBoost 1.0 litre engine with the turbocharger integrated with the cylinder head [10], and the new PSA 1.2 litre engine [27].

One area where there is a clear need for improvement is the engine acceleration in response to the driver's demand for more torque. While most modern turbocharged GDI engines do have the full rated torque at low engine speeds under steady-state conditions, they are still behind their naturally aspirated counterparts in terms of the time needed to reach these values after a step demand (the so-called Time-to-Torque, or TTT). For example, a study by the French car part manufacturer Valeo [32] shows that a naturally aspirated engine has the specific TTT of the order of 100 $Nm/s/dm^3$, whereas a turbocharged one reaches 45 $Nm/s/dm^3$ (with a twin-scroll turbine). Referring to Fig. 21 below, it can be seen that the naturally aspirated engine needs one second to reach the torque of 240 Nm, whereas this time is about 2.7 seconds in the case of the turbocharged one. Note that the same engine equipped with a simpler turbocharger is capable of producing only 25 $Nm/s/dm^3$.

The solution proposed by Valeo consists of a small, electrically-driven auxiliary compressor, mounted on the turbocharger shaft, which is only taken into operation when the engine is about to accelerate. The total TTT (turbocharger + the auxiliary compressor) value thus achieved is 140 $Nm/s/dm^3$, i.e. higher than the one offered by the naturally aspirated engine. In this way, the main drawback of turbocharged GDI engines vs. the naturally aspirated ones as regards the vehicle acceleration can be eliminated, but there are a number of issues to be dealt with before this or similar solutions can enter the series production. These include the need for augmenting the electrical system of the vehicle with regard to voltage, battery capacity, improved charging in order to provide enough energy for frequent accelerations, etc. [35]. Examples of highly optimized engines with electro-assisted turbochargers can be found in the Formula-1 vehicles built in accordance with the 2014 regulations, e.g. [34].

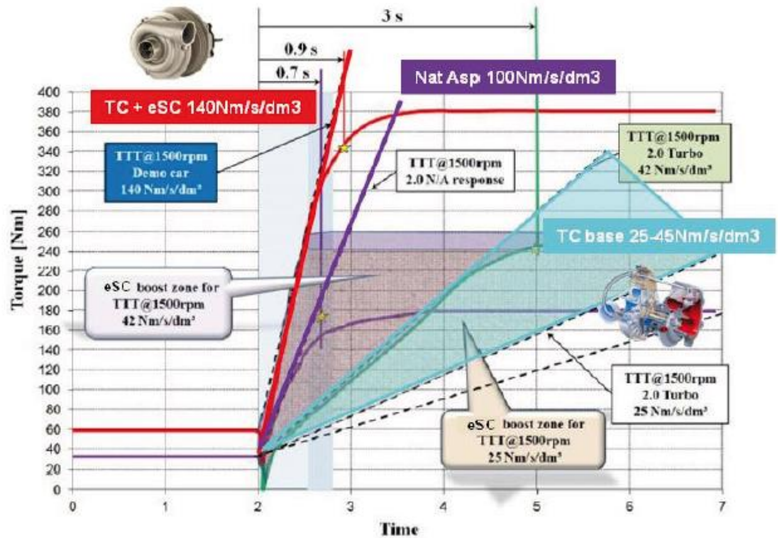

Figure 21. Torque build-up curves of various GDI engine versions [32]

Although the turbocharged GDI engine has already reached a high performance level through downsizing, there is still a considerable potential for further development. According to A. Schamel of Ford [28] and Prof. Geringer of TU Wien [36], there are possibilities for further CO_2 reduction in the two-figure percent range through downsizing; and there is an intensive activity aimed at the development and/or optimization of turbocharging concepts that should make this possible. Among the possibilities are the two-stage turbocharging in the series and parallel (sequential) connections, [29] and [30], respectively, and/or electrically-assisted turbochargers, such as e.g. the solutions of BorgWarner [31], Pankl [37], and already mentioned Valeo [32]. However, a fully-electrical turbocharging system does not seem to represent an attractive possibility at the moment [38].

The perennial subject of air management is certainly going to be actively pursued in the future; and from the standpoint of turbocharging, it primarily relates to improving the efficiency of the system components, i.e. turbine, compressor, and air path components [39].

CONCLUSIONS

- Turbocharging is one of the most important technologies for reducing the fuel consumption and exhaust emissions
- Large four-stroke Diesel and gas engines: two-stage turbocharging is a proven technology for the current and future engines

- Large two-stroke engines: advantages of two-stage turbocharging demonstrated at test stands, but the slow-steaming trend in the ship cargo transport deters investments in new technologies
- Commercial vehicle Diesel engines:
  - Two-stage turbocharging has been developed, but is not yet widely used
  - It is seen as a means for attaining the Euro VI and Tier 4 emission norms, as is e.g. the case with the MAN 2676 engine
  - Turbo-compounding and turbo-braking are established technologies
  - Passenger car Diesel engines:
  - Two-stage, regulated turbocharging represents the state-of-the-art today
  - Three-stage turbocharging has also been applied
  - In order to further improve the low-end torque, optimized compressors and turbines with variable geometry will be necessary
- Passenger car Otto engines:
  - Single-stage, regulated turbocharging represents the state-of-the-art today
  - Two-stage turbocharging has demonstrated potential for compensating the trend towards torque diminishing at higher downsizing values
  - However, some manufacturers (e.g. Daimler-Benz) see highly optimized exhaust channels, twin-scroll turbines, and/or electrically assisted turbocharges as a solution
- And a final quote: "We are still far away from the end of engineering possibilities [40].

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A CONTRIBUTION TO RESEARCH OF SOME PHYSICAL CHARACTERISTICS OF DISC BRAKES IN LABORATORY CONDITIONS

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UDC:629.017

ABSTRACT: It has been proved in literature that there is no generally accepted method for the evaluation of characteristics and output parameters of the disc brake mechanism. This paper presents experimental research of disk brake subsystem with brake fluid pressure in the brake cylinder and disc speed as input quantities and brake torque and brake noise (expressed as sound pressure level) as output quantities. The work of disc brakes during the braking process is defined with physical output parameters: activation force, brake torque, brake noise, thermal load, etc. The aim of this paper was to determine the minimum number of mutually independent physical parameters that represent, with reasonable accuracy, the braking characteristics of disc brakes.

KEY WORDS: disc brakes, physical characteristics, experimental research, data processing

DOPRINOS ISTRAŽIVANjU NEKIH FIZIČKIH KARAKTERISTIKA DISK KOČNICA U LABORATORIJSKIM USLOVIMA

REZIME: Može se videti u literaturi da ne postoji opšte prihvaćena metoda za procenu karakteristika i izlaznih parametara mehanizma disk kočnice. Ovaj rad predstavlja eksperimentalno istraživanje podsistema disk kočnice sa pritiskom kočne tečnosti u kočnom cilindru i brzinom diska kao ulaznim veličinama i kočnim momentom i bukom kočnice (izraženom kao nivo zvučnog pritiska) kao izlaznim veličinama. Rad disk kočnica tokom procesa kočenja je definisan fizičkim izlaznim parametrima: silom aktiviranja, kočnim momentom, bukom kočnice, termičkim opterećenjem, itd. Cilj ovog rada je bio da se utvrdi minimalan broj međusobno nezavisnih fizičkih parametara koji predstavljaju, sa prihvatljivom preciznošću, kočne karakteristike disk kočnica.

KLJUČNE REČI: disk kočnice, fizičke karakteristike, eksperimentalno istraživanje, obrada podataka

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A CONTRIBUTION TO RESEARCH OF SOME PHYSICAL CHARACTERISTICS OF DISC BRAKES IN LABORATORY CONDITIONS

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INTRODUCTION

The main task of the brake mechanisms is to achieve the necessary braking torque acting on the wheel of the vehicle, causing its deceleration, and thus braking of the vehicle. Therefore, the braking torque is a fundamental characteristic of every brake, the measuring tool of its functional properties or *performances*.

Calculation methods and testing procedures for disk brake mechanisms, presented in a large number of papers [1,2,3,4], have passed through several stages in their development. Calculation methods have experienced the most intense progress thanks to the development of computer techniques. In this way, numerical methods have become the basic calculation methods, without which, it cannot be possible to imagine the development of many products today in technologically advanced countries. These methods allow rapid analysis of a large number of different combinations and the selection of the optimal solution (optimization). Numerical methods [5,6] are approximate and they require reliable information based on which the boundary conditions are derived. Thus, there is their direct connection with the testing of brake mechanisms that provides information of high precision and quality, thanks to advancement of measurement and processing techniques and analysis of measurement results. In this way, the period needed for the development of new brake's elements is significantly reduced. The large part of the brake system testing is verification of compliance with the requirements of international and national regulations. Developmental researches are very diverse and necessary for the introduction of new product or improvement of the existing products.

A large number of physical parameters of the disc brake are recorded during the experimental research with special attention given to the brake fluid's pressure in the brake cylinder, the braking torque, disk speed and brake squeal expressed as sound pressure level (SPL). Over the years, many laboratory and road tests with a wide range of options and approaches have been developed. Modern dynamometers for testing the brakes have become

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the sophisticated test platforms for the identification of the brake characteristics and for the diagnostic of the noise problem

The work of disc brakes during the braking process is defined with physical output parameters: activation force, brake torque, brake noise, thermal load, etc. The aim of this paper was to determine the minimum number of mutually independent physical parameters which represent, with reasonable accuracy, braking characteristics of disc brakes. Thermal stress is an important parameter of brakes. However, it is not independent parameter because it can be calculated based on the braking torque and rotation angle of the brake disc. Bearing in mind that the aim of this paper is to analyse the correlation between the minimum numbers of independent brake parameters, it is justified not to pay attention to thermal load as a dependent parameter in this paper.

Performed analyses show that there are different approaches and there are still no generally accepted methods for evaluation of characteristics and output parameters of disk brake mechanisms [7]. One method for definition of criteria for evaluation of physical properties of disc brakes of motor vehicles that includes the phase of experimental measurements and the subsequent processing of the recorded data is presented in this paper.

EXPERIMENTAL RESEARCH

Description of measurement installation

Brake dynamometer for testing disc brake noise has been developed at the Laboratory for IC engines at the Faculty of Engineering, University of Kragujevac. Since the moment of inertia of the brake disc on rotating drive shaft is equivalent to a linear inertia of the vehicle, it is possible to include the influence of vehicle inertia and thus provide more reliable results in laboratory conditions. The installation consists of several functional units:

- Test bench with the electromotor, power transmission and disc,
- Electric power installation,
- Installation for activation of the disk brake i.e. for generation of the braking torque,
- Measuring equipment.

Electromotor is mounted on the stand that is rigidly connected. The drive block consists of: asynchronous electric motor fed by frequency regulator, friction belt transmission with ratio equal to 1, inertial mass and brake disc. The speed sensor is mounted at the free end of the electromotor's shaft. The inertial mass on the input shaft is made in the form of the disc with diameter of 0.35 m, width of 0.045 m and mass of 35 kg. Mass moment of inertia has value of 0.54 kgm2 and it corresponds to the kinetic energy of the test vehicle, at low initial speeds, which are critical in terms of the frequent appearance of brake squeal [8].

The components of the measuring chain formed to record the brake fluid pressure in the brake cylinder, p, brake torque, Mk, disk speed, n, and the sound pressure level, SPL, during the investigation of the brakes in the laboratory conditions, as well as the connections of individual components are shown in Figure 1 and in the block diagram in Figure 2.

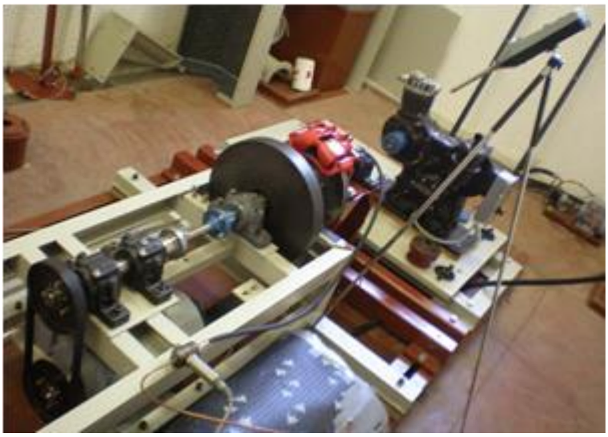


Figure 1: Photo of designed measuring installation for research of the brake noise [8]

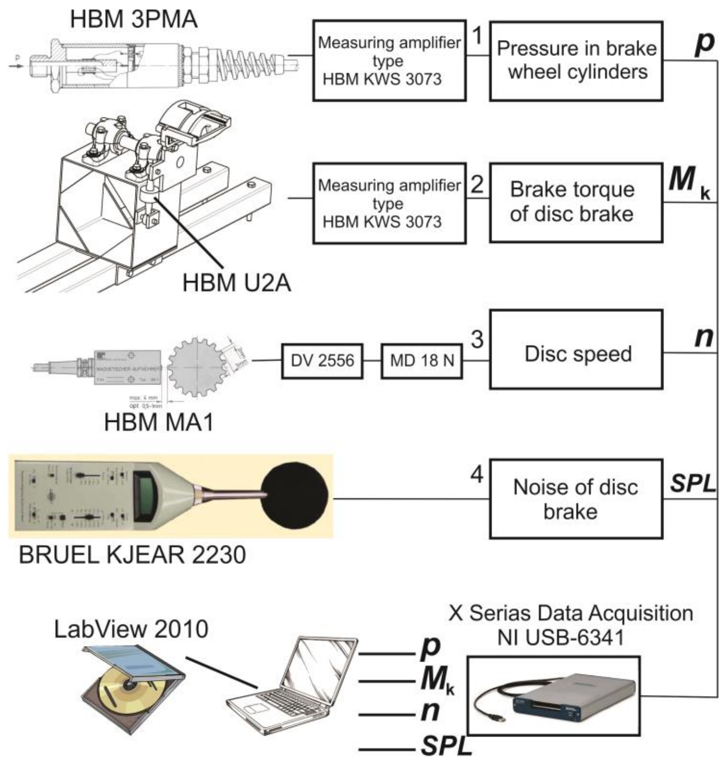


Figure 2: Measuring chain for analysis of disc brake parameters in laboratory conditions [8]

During the experimental research, the two categories of tests were performed: with constant disc rotation speed and for different braking pressures in the disc brake cylinder and with constant brake pressure and varying speeds. Performed tests correspond to braking of the vehicle with disengaged clutch i.e. with interrupted power transmission.

Test regimes:

Constant disc speed and different brake pressures:

- Constant disc speed in range from 250 to 1000 min^{-1} is applied.
- For every disc speed value, different pressures in range from 0.5 to 3 MPa with increment of Δp = 0.5 MPa are applied.

Constant brake pressure and different disc speeds:

- Constant maximum pressure in the range from 0.5 to 3 MPa is applied.
- For every pressure value, the different disc speeds in the range from 250 to 1000 min^{-1} with increment of Δn = 50 min^{-1} are applied.

The range of disc speeds at the beginning of the braking process and the corresponding speed of the real vehicle for the tested disk brake with ventilated disc with a diameter of 266 mm and the tire with dynamic radius of 260 mm in are shown in Table 1.

Table 1. Comparative overview of vehicle's speed and the corresponding disk speeds

N o	*Vehicle speed, km/h*	*Brake disc speed, min^{-1}*
1.	25	255
2.	30	306
3.	35	357
4.	40	408
5.	45	459
6.	50	510
7.	55	561
8.	60	612
9.	65	663
10.	70	714
11.	75	765
12.	80	816

In some braking regimes (e.g. low brake pressure and high initial speed of electromotor), only partial braking exists i.e. reduction of disc speed from the initial speed v1 to final speed v2, while in some other cases braking until stopping of the brake disc occurs.

The measurement of the temperature at the midpoint of the brake disc surface is important in order to avoid the overheating. If the system is cold, the working temperature should be raised to 80 - 90 °C. This is achieved by braking with lower pressure and with engaged electric motor.

Measuring signals are then led from sensors to the data acquisition system NI USB-6341 produced by National Instruments that, in interface with the software LabVIEW 2010, acquires, analyses and presents real-time data and stores the measurement results. The sound pressure level is the measuring signal with the highest frequency, so it sets the sampling rate during experimental research. Digitization of data is achieved in the 131072 points, with step of 2□10-5 sec, which enabled the reliability of data processing in the area from 0.381 Hz to 25000 Hz (this includes a relevant area of frequency of vibration and noise in motor vehicles [6,7,9,10,11]).

Results of research in time domain

For illustration, measured parameters that define the braking process - brake pressure, p, disc speed, n, braking torque, Mk and the sound pressure level, SPL, are displayed in Figures 3 to 6. Although the measurements in the case of maximum braking pressures from 0.5 MPa to 3.0 with step Δp=0.5 MPa are performed, only regimes related to limit values of pressure variation interval and to initial disc speed corresponding to the vehicle initial speed of 75 km/h are shown. Diagrams of measured parameters in other regimes are similar to the described limit ones and will not be shown here.

Since a new friction couple (disc - brake pads) is used in research, a low friction on the friction surface in the brake assembly is present, even when the brake is not activated. This can be noticed in the diagram that shows the variation of the braking torque where braking torque values of 5÷7 Nm exist before and after the braking process (braking pressure greater than zero), independently of the brake pressure and disc speed value applied.

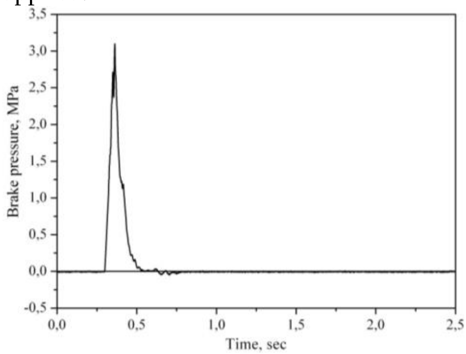


Figure 3: Brake pressure at a maximum pressure of 3.0 MPa and initial disc speed of 765 min-1 (v=75 km/h)

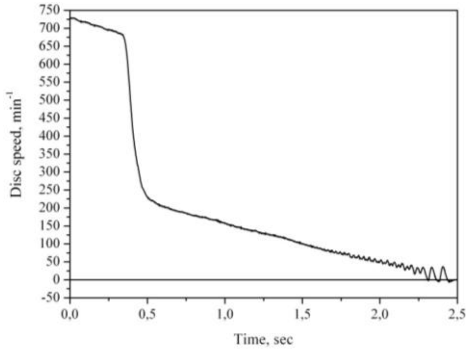


Figure 4: Brake disc speed at maximum pressure of 3.0 MPa and initial disc speed of 765 min-1 (v=75 km/h)

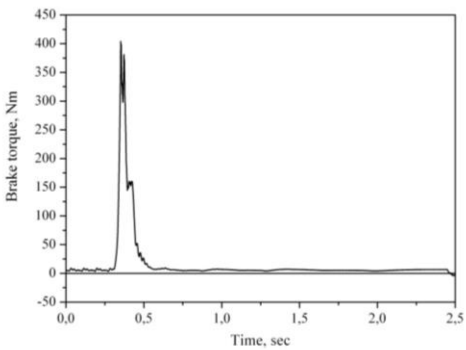


Figure 5: Braking torque at a maximum pressure of 3.0 MPa and initial disc speed of 765min-1 (v=75 km/h)

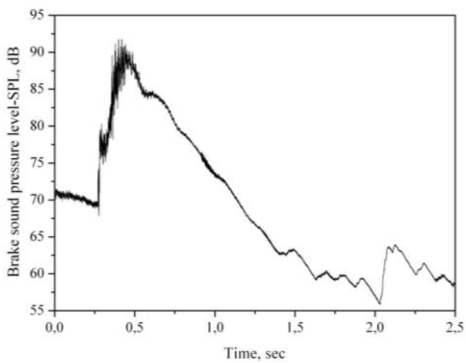


Figure 6: Sound pressure level (SPL) at a maximum pressure of 3.0 MPa and initial disc speed of 765 min-1 (v=75 km/h)

The diagrams in the mentioned figures are related to braking at the maximum pressure of 3.0 MPa, and the disc speed of 765 min^{-1} at the beginning of the measurement. Since this is a regime of large brake pressure, after the period of 0.5 sec when the process of braking is complete, the braking pressure drops to 0, while the disk speed, due to inertia, drops to 0 after 2.5 sec. Diagrams correspond to the braking regime until complete stopping of the brake disc. At initial disc speed, power transmission from electric motor is disengaged (the case of stopping the vehicle without brakes), so at the moment when the brake is activated (t=0.3 sec), the disc speed is reduced to 690 min^{-1}.

It is justified to show that there is a time delay between recorded signals. Therefore, Figure 7 illustrates the response time of disc brake hydraulic installation obtained by monitoring changes in the brake pressure and torque for the initial disc speed of 765 min^{-1} (75 km/h) and a maximum pressure of 0.5 MPa. The time delay between the observed signals is 0.02 sec. It is present throughout the whole braking process and caused by brake hoses deformation and overcoming the gaps in the disc brake assembly. The time delay has similar values for all investigated braking regimes.

It was important to show a comparative display of the acquired signals for different test conditions. The influence of different regimes of maximum brake pressure on changes of this parameter during the braking process is shown in Figure 8. The change in shape of the curves due to pneumatic actuation of the brake master cylinder may be seen. At the lowest brake pressures, the trapezoidal shape of actual parameters of the braking process and maintenance of the maximum constant braking pressure over a long period of time are observed. For higher values of maximum pressure, changes of the pressure have a triangular form, as stipulated by noise squeal matrix dynamometer test procedure SAE J2521 [12].

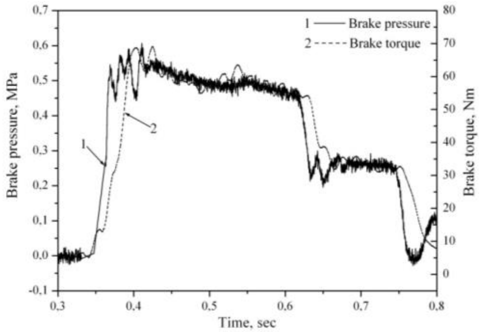


Figure 7: Time delay between the braking torque signal and the brake pressure signal at a maximum pressure of 0.5 MPa and the initial disc speed of 765 min-1 (v=75 km/h)

The influence of different regimes of maximum brake pressure on the braking torque variation during braking is shown in Figure 9. It is clear that the braking torque follows to a large degree the form of changes in the brake pressure, and when maximum pressure is low (0.5 and 1.0 MPa) the residual braking torque is shown (partially activated brakes).

The sound pressure level, SPL, is the most important quantity for estimation of the occurrence of brake squeal. With the variation of the maximum braking torque, the expected increase in noise level is present and it exceeds the threshold of 70 dB in all regimes (Figure 10). In the regime of the highest test pressure of 3.0 MPa, the intensive variations which correspond to squeal noise are observed. This is better evaluated in the frequency domain by analysis of the power spectra of the sound pressure levels.

The calculated kinetic coefficients of friction for different regimes of maximal brake pressure are shown in Figure 11. The analysis of the obtained curves shows that the friction coefficient is in the expected values of between 0.2 and 0.4 in large part of the brake interval. Range of the coefficient of friction is the widest for the highest values of brake pressure.

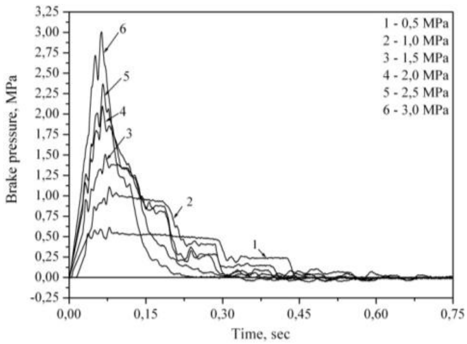


Figure 8: Variation of the brake pressure for the initial disc speed of 765 min-1 (v=75 km/h)

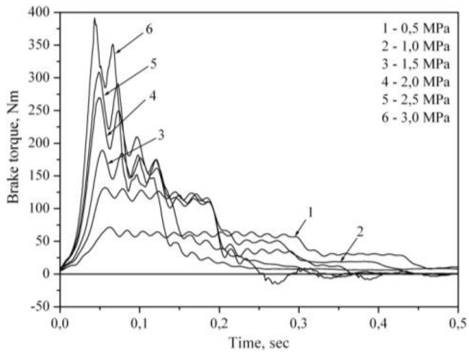


Figure 9: Variation of the braking torque for the initial disc speed of 765 min-1 (v=75 km/h)

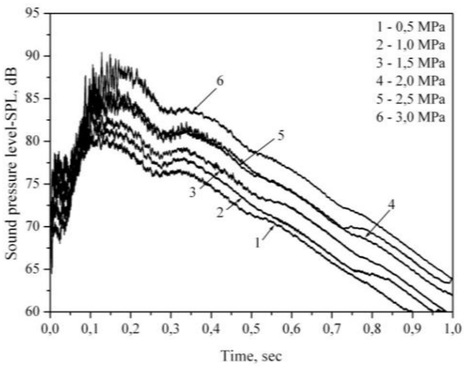


Figure 10: Variations of the sound pressure levels for the initial disc speed of 765 min-1 (v=75 km/h)

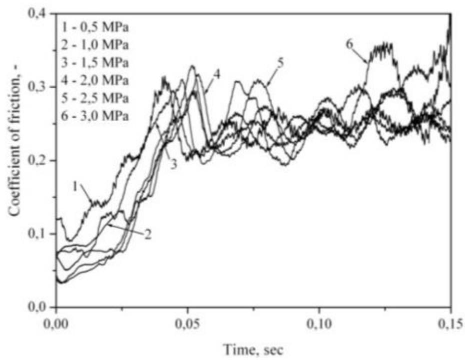


Figure 11: Variations of the coefficient of friction for the initial disc speed of 765 min-1 (v=75 km/h)

DATA PROCESSING AND ANALYSIS

In order to evaluate the validity of the selected number of points and the time increment during signal digitization, it was necessary to check it not only in terms of the minimum and maximum frequency, but also in terms of errors that are made when calculating the spectrum and autospectrum of signals. In this sense, calculated "bias" errors of the auto-spectrum and cross-spectrum have values of -0.012, while the relative random error of auto-spectrum has value of 0,1 and corresponding value for cross-spectrum is 0.118, which is acceptable [9,10,11]. Based on this, it can be stated that the selection of parameters of the digitization is correctly executed.

During the data processing, the computer program “ANALSIGDEM” [13] was used, based on theory from [9,10,11].

The boundary conditions of the conducted research

As it has been already mentioned, the tests were performed using a large number of combinations of initial disc speeds and maximum pressures in the brake installation, but a preliminary analysis have shown that it is justifiable to perform the analysis for two boundary conditions of the mentioned parameters (minimum and maximum values of the braking pressure in the installation).

In this regard, the following parameters of disc brakes were analyzed for extreme conditions:

a) the same initial disc speed (around 750 min-1), but for different pressures in the brake installation and

b) the same maximum pressure (3 MPa), but for different initial disc speeds.

In order to determine the validity of performing tests with different initial conditions, it is necessary to analyse the existence of the correlation between the identical registered values in various tests, because if there is a high correlation, then it is not necessary to perform a large number of tests, and vice versa. In order to analyse the influence of test conditions on the physical characteristics of the disc brakes, the cross-correlation coefficients functions [9,10,11] were calculated for test a) and test b).

The interactions between the following parameters are observed during the test a) and the test b):

- disc speed,
- braking torque,
- pressure in the brake installation and
- brake noise.

The following couples of registered parameters are observed:

- disc speed - brake torque,
- disc speed - brake noise,
- brake pressure in the installation - braking torque,
- brake pressure in the installation - brake noise and
- disc speed - brake pressure in the installation.

Cross-correlations coefficient in the case of the same initial disc speeds

Using the previously mentioned program, the cross-correlation coefficients are calculated and shown in Figures 12 and 13.

Analysis of the data in Figure 12 that refers to the observed parameters for the constant initial speeds shows that the cross-correlation coefficient decreases with the increase of time delay. This can be explained by the fact that there is a weak correlation between the observed parameters. It should be noted that there is the strongest correlation related to the brake noise, and that slightly smaller for disc speed.

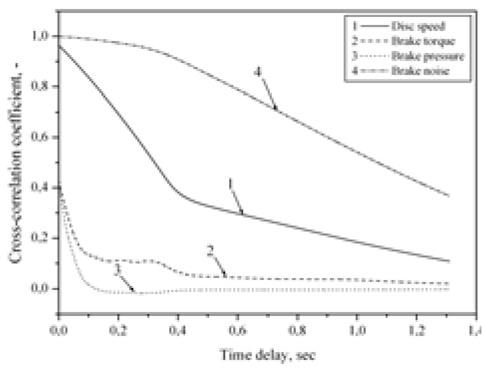


Figure 12: Cross-correlation coefficients for the same initial disc speed of 750 min-1

Cross-correlation coefficient for the same maximum pressure

Analysis of the data in Figure 13 shows that the cross-correlation coefficient also decreases in the case of the same maximum pressures for the observed values of disc speed and sound pressure level. A slight increase of the observed coefficient at around 0.1 sec is noticed, and then a quick drop in case of the braking torque and pressure. All this confirms the fact that there is no strong correlation between the observed data couples, and this leads to the conclusion that the tests must be carried out for a large number of combinations of initial conditions. In this case, there is a slightly stronger correlation that is related to the brake noise, too.

It is now reasonable to look for a coupling between the observed parameters that are not acquired for the same initial conditions (0.5 MPa, 765 min-1; 3 MPa, 714 min-1). The obtained results are shown in Figure 14. Based on an analysis of the data from this figure, it can be concluded that the forms of diagrams are not significantly different compared to those obtained for the same disc speeds and the same brake pressures in the installation.

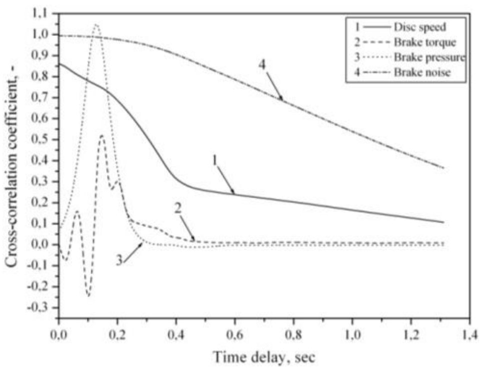


Figure 13: Cross-correlation coefficient for the same maximum pressure of 3 MPa

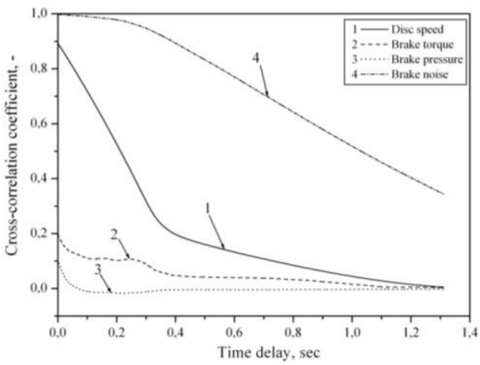


Figure 14: Cross-correlation coefficient for 0,5 MPa, 765 min-1 and 3 MPa, 714 min-1

Based on the previously mentioned, it may be concluded that the initial test conditions have an influence on the realized and acquired physical parameters of disc brakes and that this must be taken into account during the evaluation of their characteristics. Therefore, in further data analysis, we should use the results obtained in the most rigorous testing conditions.

Data analysis for the most rigorous testing regime

In order to determine whether it is necessary to record all relevant parameters of disc brakes during the brake disc testing in laboratory conditions, within the same initial testing conditions, their mutual correlation will be studied. In this case, the pressure in the installation of 3 MPa and disc speed of about 750 min-1 are observed. Taking into account [9,10,11], the following values were calculated:

- mutual probability density function,
- cross-correlation coefficient and
- cross-spectrum.

Thereby, the following couples of registered parameters are observed in the most rigorous regime of disk brake testing:

- disc speed - brake torque,
- disc speed - brake noise,

- brake pressure in the installation - braking torque,
- brake pressure in the installation - brake noise and
- disc speed - brake pressure in the installation.

Mutual probability density functions

Densities of mutual probability functions are partially shown in Figures 15 and 16. The analysis of all data shows that a mutual probability density has a value of 2.09% for data relating to the disc speed - torque, 4.53% for the pressure - torque. Based on the analysis of the maximum value of the mutual probability density functions, it can be concluded that they are in the range from 0 to 4.53%, which shows that there is a very low correlation between the observed values.

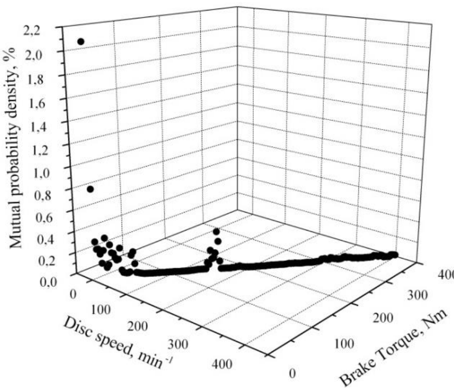


Figure 15: Mutual probability density (Disc speed -Brake torque)

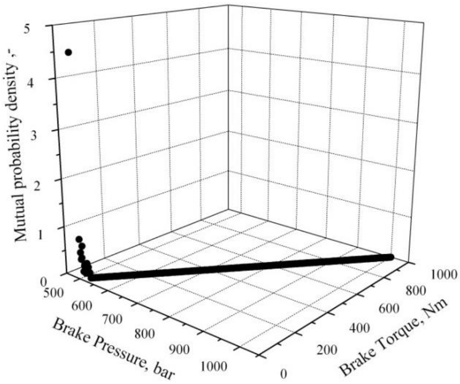


Figure 16: Mutual probability density (Brake pressure - Brake torque)

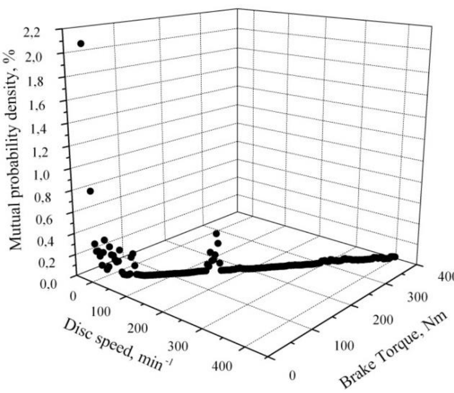


Figure 15: Mutual probability density (Disc speed -Brake torque)

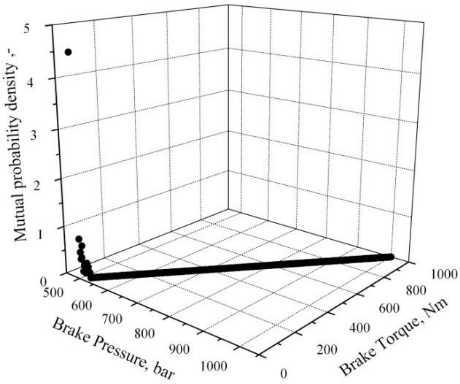


Figure 16: Mutual probability density (Brake pressure - Brake torque)

Cross-correlation coefficients

Cross-correlation coefficients are calculated and shown in Figures 17 and 18.

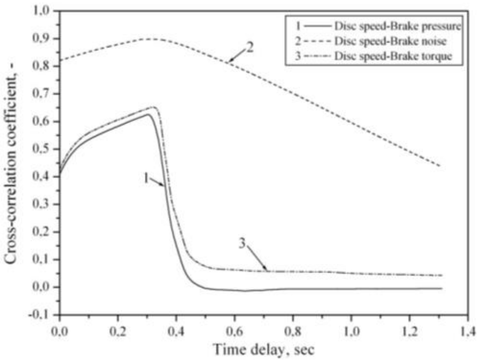


Figure 17: Cross-correlation coefficient
———— *Disc speed - Brake pressure*
- - - - - - - - *Disc speed - Brake noise*
-·-·-·-·- *Disc speed - Brake torque*

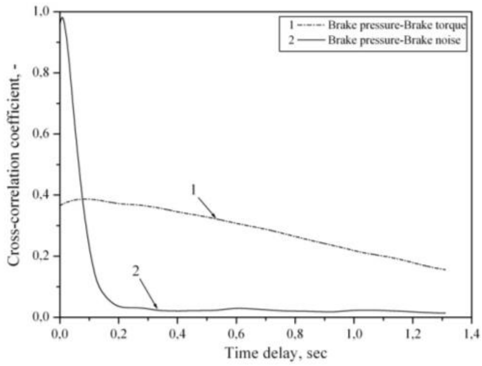


Figure 18: Cross-correlation coefficient
———— *Brake pressure - Brake torque*
-·-·-·-·- *Brake pressure - Brake noise*

Based on an analysis of the data from these figures, the maximum values of the cross-correlation coefficients have been determined and given in Table 2. From the figures, it can be noted that, after achieving the maximum value, the mentioned coefficients rapidly decrease to zero for all couples of parameters.

Based on the analysis of registered time series, it is noted that the change of the observed parameters occurs after about 0.2 to 0.3 sec from the start registration, which means that, before this interval, changes occur due to the fact that the brake friction couple was new, and that they are at the level of statistical errors. Having this in mind and the fact that the cross-correlation coefficients decrease after their maximum value, it can be stated that there is no significant correlation between the observed couples of parameters.

Table 2: The maximum values of the cross-correlation coefficients

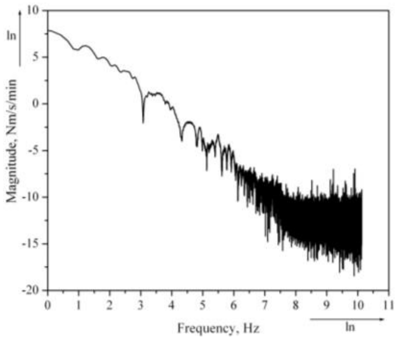
Observed couples of parameters	Time delay, sec	Cross-correlation coefficient,-
Disc speed - Brake torque	0,319	0,651
Disc speed – Brake noise	0,308	0,897
Pressure - Brake torque	0,008	0,981
Pressure - Brake noise	0,090	0,386
Disc speed - Pressure	0,302	0,625

Please, note that it is particularly interesting that this refers also to the change of the observed coefficient for the couple “Disc speed - Brake pressure in the installation”, and it shows that the initial conditions are mutually independent in the field of interest for the brake noise.

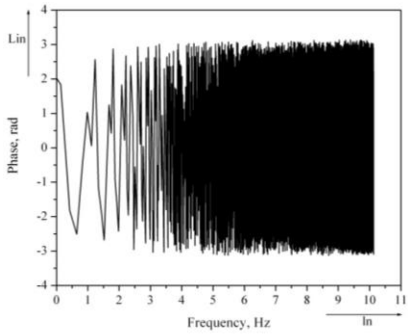
Cross-spectrum

In addition to the preceding calculations, the cross-spectra of the analyzed couples of time series are calculated. As the values of the cross-spectra are complex numbers [9,10,11], they are presented in the Euler's form (magnitude and phase).

Preliminary analyses show that there is a huge difference between the minimum and maximum values of the magnitude, which was not the case with phase angle. Therefore, the magnitudes are shown in diagrams on the natural logarithm scale and the phase angles on linear scale. As the frequency was in the range of 0.381 to 24 999 Hz, the abscissa is shown on the natural logarithm scale too, and the results are partially, for illustration, shown in Figures 19 and 20.



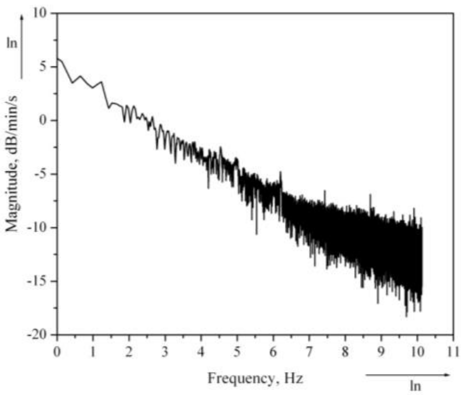
a)



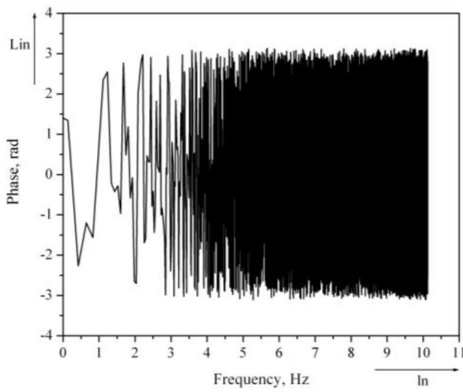
b)

Figure 19: Cross-spectrum of disc speed – brake torque

Based on the analysis of all data, it can be seen that the magnitude of the cross-spectrum is the largest at the frequency of 0.381 Hz, and it drops significantly with the increase of frequency, especially above frequency value of about 1000 Hz. This phenomenon occurs in all observed couples of time series, whereby the frequency interval around 1000 Hz is very important for low-frequency brake noise. For illustration, the magnitudes of power spectra for the minimum and maximum frequency for all the observed couples of time series are shown in Table 3. It can also be concluded that the phase angle is highly dependent on the frequency, but also on the observed couple of measured parameters, so the range interval will be given in the same table.



a)



b)

Figure 20: Cross-spectrum of disc speed – brake noise

Table 3: Some parameters of the cross-spectra

	Magnitude (ln)	Magnitude (ln)	Phase (linear-Lin)	Phase (linear-Lin)
Observed couples of parameters	f_{min}	f_{max}	f_{min}	f_{max}
Disc speed - brake torque	8,26	-13	-3,05	3,05
Disc speed - brake noise	7,78	-12	-2,59	2,47
Pressure - brake torque	3,51	-19	-2,67	2,44
Pressure - brake noise	3,03	-19,02	-2,94	2,25
Disc speed - pressure	5,82	-14,5	-2,99	2,83

Based on the analysis of all the diagrams and data from Table 3, it can be determined that the values of the magnitudes in the frequency interval important for noise are negligible. This indicates that there is no correlation between the observed couples of time series, which is consistent with the analysed cross-correlation coefficients and mutual probability density. Thereby, the higher values of magnitude of the cross-spectrum are not of interest for the brake noise, and will not be separately analysed.

It should be emphasized that this applies also to variation of magnitude of cross-spectrum for a couple “disc speed - pressure”, indicating that the initial conditions are independent in frequency interval interesting for brake noise.

Based on the previous analyses, it can be stated that all used data that characterize the physical behaviour of disk brakes must be considered as functions of initial values of disc speed and brake pressure in the installation as independent parameters. With that in mind, it is obvious that statistical values of all registered parameters of brakes should be considered separately.

Probability density, auto-correlation function and auto-spectrum

The following statistical parameters for the chosen, the most rigorous regime (one test) are calculated:

- probability density,
- auto-correlation functions and
- auto-spectra.

These parameters were calculated for the most rigorous test regime, while the following registered parameters were observed:

- disc speed,
- braking torque,
- brake pressure in the installation and
- brake noise.

Probability density

The calculated values of the probability density are, for partial illustration, shown in Figures 21 and 22. Based on analysis of all data, intervals and percentage of occurrence of the observed parameters by levels were determined and given in Table 4.

Table 4: Probability density

	Interval of occurrence	Percentage of occurrence
Disc speed, min^{-1}	-2,31 ÷ 813	2,18
Brake torque, Nm	-50 ÷ 405	47,1
Brake pressure, MPa	-1 ÷ 31	64,3
Brake noise, dB	-55 ÷ 91	19,9

Based on Table 4, it can be determined that the maximum value of the occurrence refers to pressure, and the lowest to disc speed, in the areas around the mean values of the signal. This is understandable due to the fact that the disk was always decelerating, and the pressure in the installation was varying around the set value during the test.

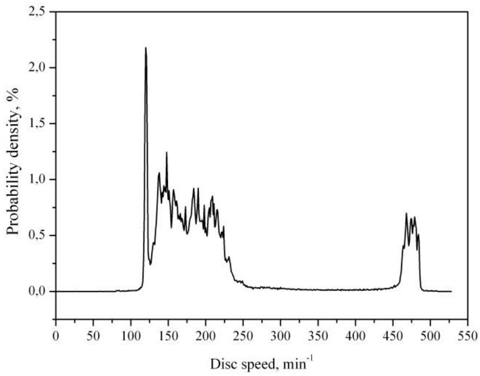


Figure 21: Probability density - Disc speed

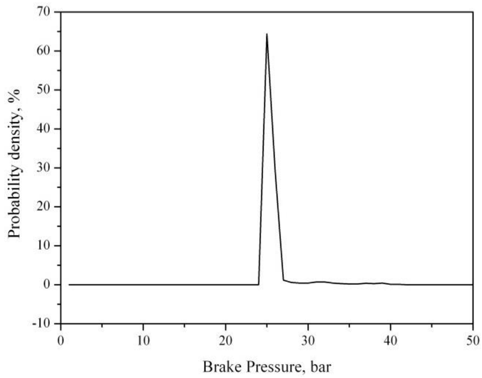


Figure 22: Probability density - Brake pressure

Auto-correlation function

The calculated values of autocorrelation functions are partially shown in Figures 23 and 24, for illustration. Based on the analysis of all data, it can be determined that the autocorrelation function has a maximum for a time delay that is approximately equal to zero, while the maximum values depend upon analysed signal and their range is between 14.589 bar^2s^{-1} (pressure) to 49541.5 $min^{-2}s^{-1}$ (disc speed). All the observed autocorrelation functions rapidly decrease with increase in time delay. This is understandable, since these are short, quasi-impact (transition) processes, which tend to zero after the braking process is finished.

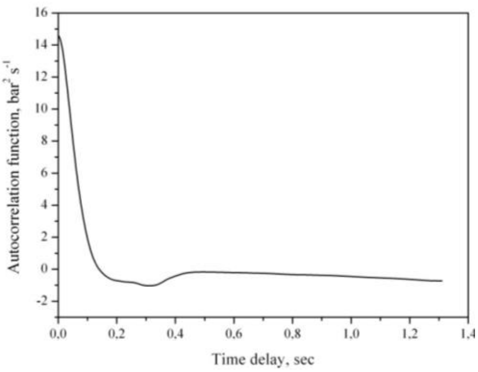


Figure 23: Auto-correlation function for brake pressure

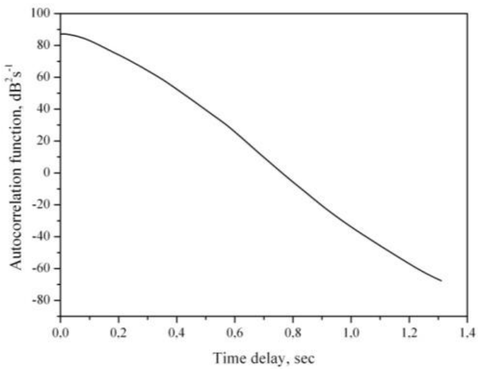
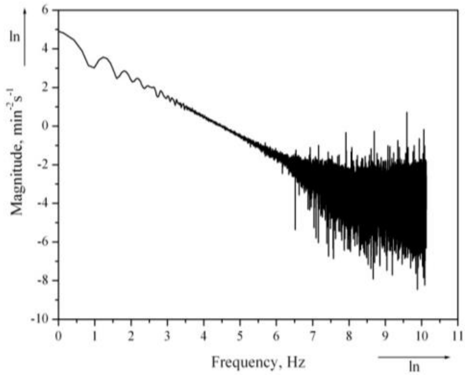


Figure 24: Autocorrelation function for brake noise

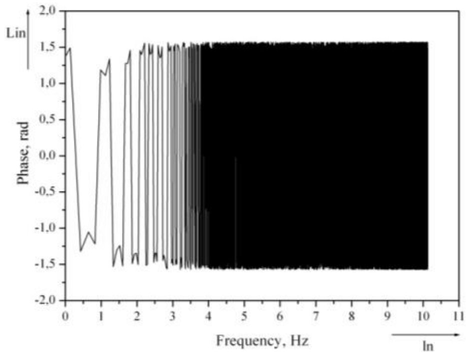
Auto-spectrum

As the auto-spectrum values are complex numbers, they are presented in the Euler's form in Figures 25 and 26.

Based on the analysis of all data, it can be determined that auto-spectra depend on the analyzed signals and frequencies. The highest values are calculated in the area of low frequencies, but some values appear at higher frequencies, which is interesting for investigation of the disc brake noise. It is observed that the largest values of amplitude spectra are found for disc speed at low frequencies, and the lowest refer to noise, which is very understandable when the physical values of the observed parameters are taken into account. The same trend is present when it comes to high frequency.

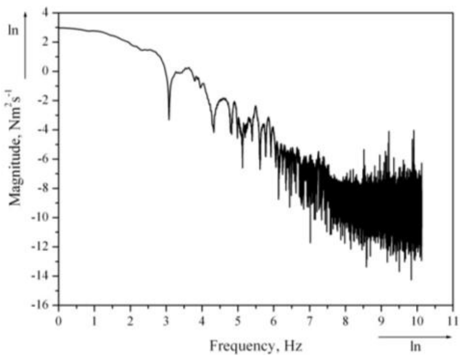


a)

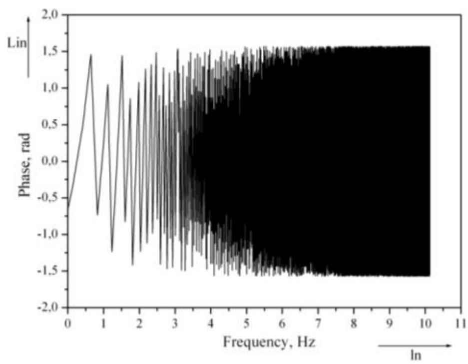


b)

Figure 25: Auto-spectrum for disc speed



a)



b)

Figure 26: Auto-spectrum for brake torque

It was also found that the phase angles depend upon analyzed signal's frequency and that their limit values are ±1.5 rad.

CONCLUSIONS

Based on the conducted research, the following conclusions can be made:

1. The developed test bench may be successfully and reliably used for research of the physical characteristics of disk brakes (high frequency brake noise and brake torque) in the laboratory conditions, because it allows observation of influence of initial disc speed (equivalent to the vehicle's speed in real braking conditions) and brake pressure in the installation (equivalent to the force acting on the brake pedal of the vehicle).
2. Testing of disc brakes at different initial conditions does not lead to identical values of measured physical properties of brakes.
3. Physical characteristics of the brakes should be measured at the most rigorous regimes (high disc speed and high pressure in the installation) and
4. Since no significant correlations between the registered parameters in the same test were established, clearly there is a need to observe and analyze each of the registered parameters separately.

ACKNOWLEDGMENTS

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THE CAR IN THE YEAR 2014

Josip Vlahović[1]

UDC:629.113

ABSTRACT: The passenger vehicle today is modern and highly equipped. Already in the basic version, it has a servo steering, power brakes, Antilock Braking System, Electronic Stability Control, air bag, etc. It has crash safe and dynamically robust body. Body corrosion resistance is significantly better. Engines consume less, and yet (and nevertheless) the emission is nearly 100 times cleaner than 30 years ago.

In any case, the passenger cars today are much better, safer, more reliable and environmentally friendly. They became larger and heavier, but they are fulfilling all requirements. All this is possible through the application of new methods of body construction, by application of the know-how in new production technologies and through application of the new metallurgical classes of Steel and Aluminium. There are also new plastics and composites, too.

Great progress has been made in electronic monitoring and control of vehicle systems. Today, on the market, there are vehicles with automatic control. Future cars will soon not need a driver. On the other hand, new limitations on the road, like speed limits, are coming more and more. Congestions on roads would lead toward more speed limits.

Because of these, it is quite difficult to say how would passenger cars look like in the future? It could happen that private cars would be controlled from some control centre.

KEY WORDS: Chassis-Body concept, self-supporting body concept, Space Frame body concept, self-supporting monocoque body concept, new materials, new technologies, Hydroforming, Heatforming, Superplastic Forming, Tailored- /blanks, tubes, strips; welding, brazing, joining, rivets, Clinch-spots, Flow Drill Screws, Tack impact, FSW - Friction Stir Welding

AUTOMOBIL 2014. GODINE

REZIME: Putnički automobil danas je moderan i veoma opremljen. Već u osnovnoj verziji, ima servo upravljač, električne kočnice, sistem protiv blokiranja točkova, elektronska kontrola stabilnosti, vazdušni jastuk, itd. Vozilo ima karoseriju koja je robusna i bezbedna na udar. Otpornost karoserije na koroziju je značajno bolja. Motori troše manje, a opet (a ipak) emisija je gotovo 100 puta čistija nego pre 30 godina.

U svakom slučaju, putnički automobili danas su puno bolji, sigurniji, pouzdaniji i ekološki prihvatljiv. Postali su veći i teži, ali oni ispunjavaju sve uslove. Sve to je moguće kroz primenu novih metoda izgradnje karoserije, primjenom znanja u novim proizvodnim tehnologijama i primenom novih metalurških klase od čelika i aluminijuma. Takođe tu su i nove plastike i kompoziti.

Veliki je napredak postignut u elektronskom praćenju i kontroli sistema vozila. Danas, na tržištu, postoje vozila s automatskom kontrolom. Budućim automobilima uskoro neće biti potrebni vozači. S druge strane, nova ograničenja na putu, kao i ograničenja brzine, postaju sve stroža. Zagušenja na drumovima direktno vode ograničenju brzine kretanja.

[1] *Received: September 2014, Accepted October 2014, Available on line December 2014*

Zbog toga, vrlo je teško reći kako će putnički automobili izgledati u budućnosti? Moglo bi se dogoditi da se putnički automobili kontrolišei iz nekog kontrolnog centra.

KLJUČNE REČI: Komcept šasija-karoserija, koncept samonoseće karoserije, prostorni koncept rama, koncept samonoseć jednodelne karoserije, novi materijali, neove tehnologije, oblikovanje mlazom, oblikovanje toplotom, superelastično deformisanje, oblikovanje cevi, trake, zavarivanje, lemljenje, spajanje, zakivanje, pertlovanje, zavarivanje trenjem

THE CAR IN THE YEAR 2014

Josip Vlahović[1] PhD

UDC:629.113

INTRODUCTION

he car, the most wanted and favourite object of a modern family is almost 128 years in the production. Officially, the first four-wheeler with the engine running on fossil fuel was a Mercedes-Benz from 1886. However, even before that, there were attempts and prototypes, which are worthy of attention because of their completeness. For instance, Siegfried Marcus from Vienna made his first car with two-stroke gasoline engine and wooden car body in 1864.

After the appearance of the first car, the development of the car has gained importance and the development of new models has begun in several countries in Europe and America. Thus, there were several car manufacturers already at the beginning of the 20th century. The development had so advanced, that, already in 1903, a car has appeared for which the aluminium superstructure was alternatively offered in sale, since, at that time, there could not have been any talk about car body. This revolutionary step was a consequence of the development of a „miracle“ material that does not rust and does not absorb water, which, back then, was of great importance, because the superstructures were made from poor iron and wood. It was aluminium, the production of which has also started in 1886. The beginning was marked by production of aluminium castings, while the first metal sheets were found in mass production in 1897. Of course, in the same year, the first sheets were already used in practice, only not for car production, but for the covering the roof of the church of St. San Gioacchino in Rome, which still exists today.

MILESTONES

Which cars have marked these 128 years of the production of passenger cars by their particularly successful concept and long-term production? One of them is certainly Daimler-Benz, the first patented passenger car from July, 1886. It was documented as a symbol of the beginning of a new era, the era of a new technical wonders. Almost two decades have passed since the appearance of the first car, when a famous Ford model had set the first records in production and technical solutions. It was the Ford Model T "Tin Lizzy" from 1908, which had been produced until 1927 and was made in more than 16.5 million specimens. It is particularly important for this model that it was the first car assembled on the assembly line in 1913 and thus it had achieved a daily production of several thousand units, which was magnificent at that time. The next car which especially marked the post-war era of personal motorization is known to all as VW "Beetle", which was based on a simple vehicle concept and was considered as a very reliable family car. Until 2003, it had been produced in more than 21.5 million specimens. In the meantime, its successor with more modern concept and design has been searched for. Thus, in 1974, the first generation VW Golf was produced and it is still produced in its seventh generation. More than 29.3 million specimens were produced in total with the sixth generation. Since we are in Kragujevac, “Zastava 600” must be mentioned, a famous and simple, small "Fica", which is a symbol of an era of development and motorization of the post-war generations in our area.

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1 IDr. Josip Vlahovic, Germany, Freiberg a/N

This small car have been produced from 1955 to 1985 and 923,487 specimens has been made (Figure 1).

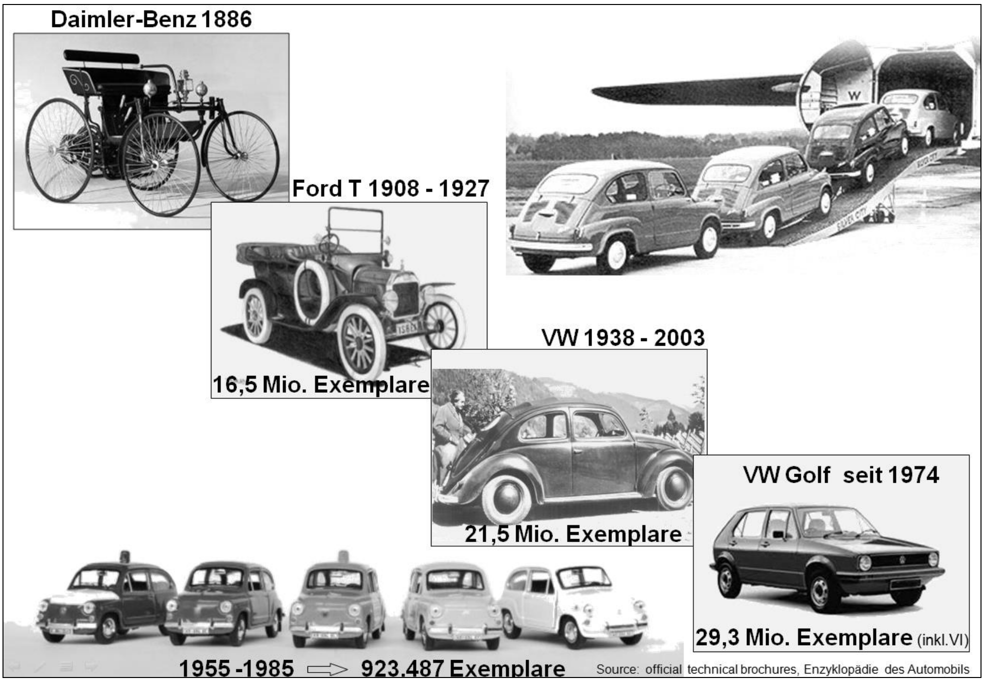

Figure 1 Milestones of motor cars

THE FIRST VEHICLE CONCEPTS

Production of the first cars has been based on a chassis - body concept, which generally represents a drive frame with the engine and the running gear and superstructure (car body) only lightly attached to the frame. Of course, this concept represented a rational, reliable and profitable production solution at the time, and it held for a long time, practically until after the World War II. Developments in technology and higher requirements, particularly in passive safety, have gradually pushed the original concept out of production. The first attempt at making a better, compact and integral carrying structure, which combines the chassis and the car body, now known as self-supporting body, was made with the model of Lancia Lambda in 1922. Of course, it was left just for the history, because the first official self-supporting body was patented by Joseph Ledwinke in 1931. The first officially produced serial car with a self-supported body was Citroen 7A from 1934. Right after that, there was Opel Kapitän, and then a large series of Opel Olympia (Figure 2). This marked the era of production of modern passenger cars with self-supporting/monocoque body, which takes and spreads the collision energy more efficiently, thus partially absorbing it and transferring it with small accelerations to the passenger compartment.

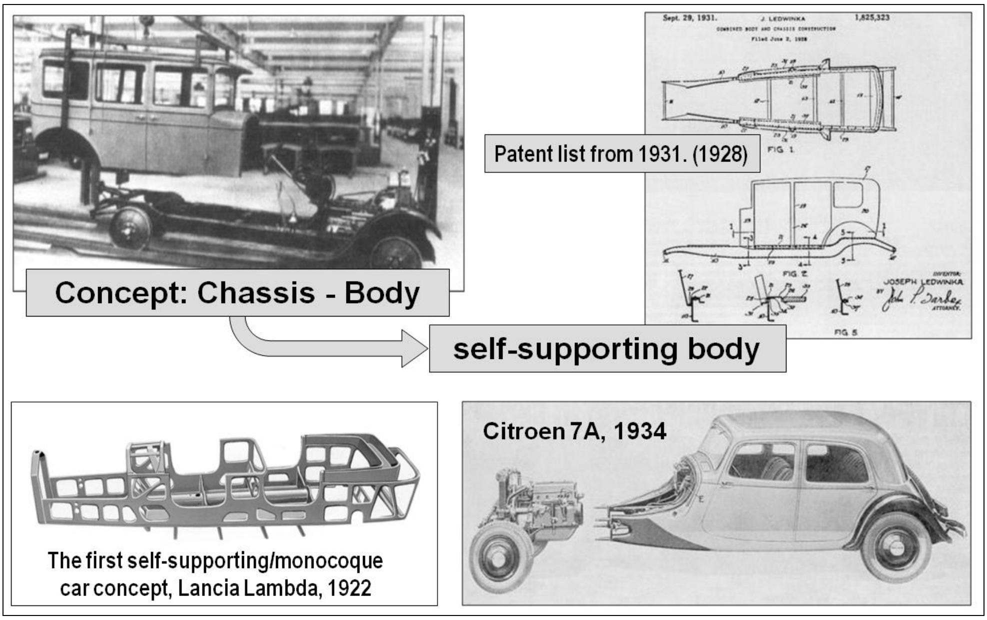

Figure 2 Car concept developments [3,4]

It is interesting to note that, already in 1953, the first car with a self-supporting body had appeared which was not made out of steel sheets, but aluminium sheets. It was the Panhard Dyna Z1, which was made conventionally, using the spot welding, without any chemical preparation of sheets before welding and without any subsequent corrosion protection. Regardless of that, an analysis from several years ago, made on one of those preserved cars that happened to be driven for a long time, have proved aluminium to be an excellent material for the car body, which opened further investments for the wider use of this material in serial production of car bodies. That car was in production until 1956 and was made in 39,460 specimens.

NEW CONCEPTS

Premises under which new concepts for passenger cars have been developed were, in general, strictly in accordance with the standards and definitions of lightweight design in the developmental stage. Figure 4 shows an Audi study of future, which is in accordance with one of the several published definitions of lightweight design.

Of course, creativity of our ancestors could not remain without visible results, so, with the help of new technologies, first in aluminium applications, a new way of designing the car body was developed, called SpaceFrame, which enabled the design of lighter, more compact, and more "crashworthy" carrying structures. These are the supporting structures, which consist of closed profiles (mainly made of aluminium), with special cross sections that are interconnected at nods through corresponding, spatially defined closed elements (mainly aluminium castings).

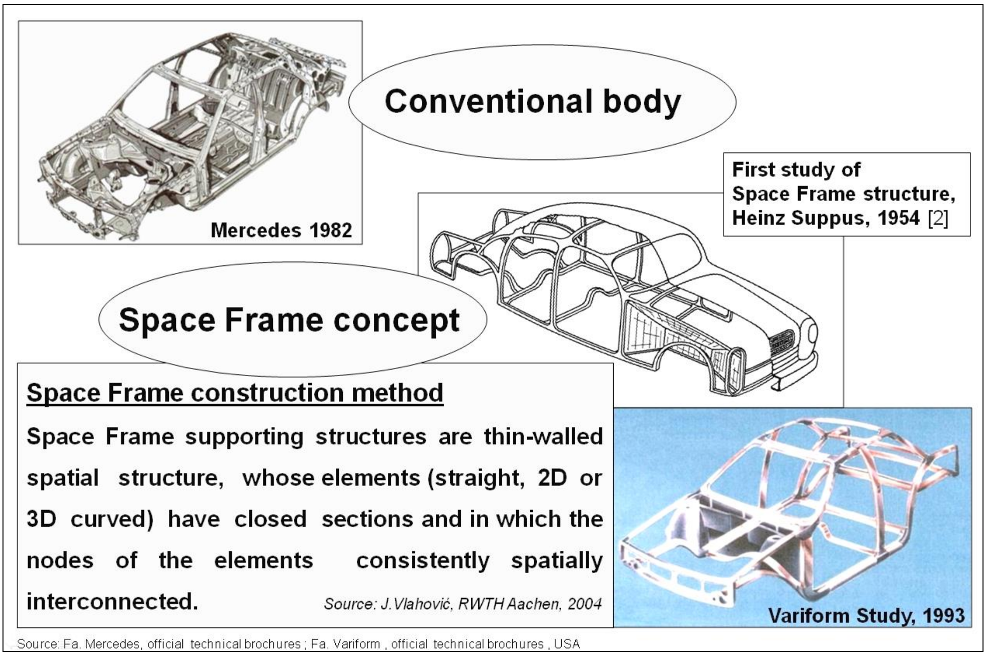

Figure 3 Lightweight design definitions

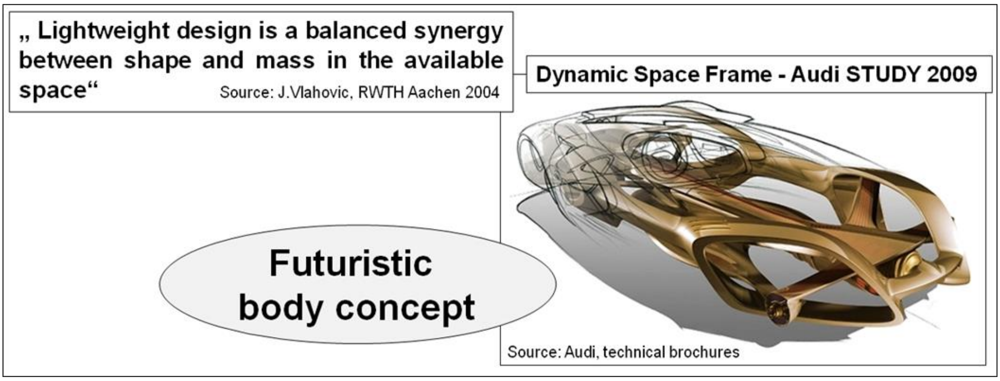

Figure 4 Different construction methods of supporting structures

The first steps in this direction can be seen already in 1954 (Figure 3) in the study of Heinz Suppusa [2], and the first car was made in several specimens by Treser (D) in 1987. It was more than surprising step in the car design - a car body which is not composed of sheet metal, but consists of a specially profiled aluminium profiles and moulded aluminium nodes. In these cars, we can no longer be talking about the car body, but of the supporting structure that takes on all static and dynamic loads and of panelling which closes it all, makes partitions and provides the optical appearance of the car body. Today, there are automobiles that have the aluminium car bodies that are made using these principles, and, officially, the first car made using this technology is the Audi A8 from 1994, Figure 5.

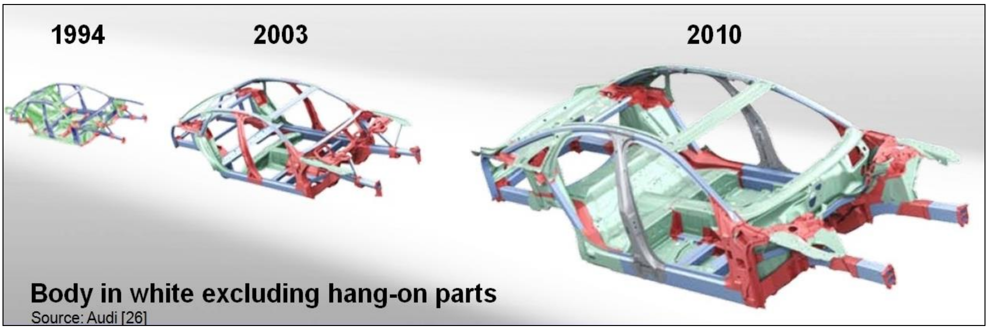

Figure 5 AUDI A8 development history

It is important to note, that, since 2008, there have been about thirty different types of cars with aluminium car body in production. As such construction (SpaceFrame) of car body of steel is concerned, it can be said that there were a few attempts to apply this type of construction of steel. Studies and prototypes were done (ARMCO study, USA, 1995, a prototype ULSAB Ultra-Light Steel Auto Body from 1996, and the prototype ULSAB AVC from 2002). Currently, this principle of construction in steel is applied only partially in serial production for certain parts of the self-supporting car bodies. Namely, the fact should be considered that certain principles of car body building, for example SpaceFrame, are more suitable for materials such as aluminium alloys. Copying of what is done with aluminium to steel will certainly not give fully effective results. For steel car body structures, there are another concepts and methods of construction today, which are elaborated in detail by Dr. Adam (ThyssenKrupp Drauz, 2009) and others.

THE LATEST DEVELOPMENT

What do we have today in large series production? Of course, mostly, highly profitable products, which implement the most optimal forms of modern techniques, but not superior technology and superior concepts of design. It may be found only in small series and exclusive cars. Today, a large series passenger car is modern in form, has reliable concept of building and is equipped with systems of active and passive safety to the maximum. Even in the basic version, modern passenger car has power brakes and power steering system, ABS (Antilock Braking System), ESC (Electronic Stability Control), multiple air bags and other previously unknown and expensive systems. Those include today both safe and lightweight car bodies, as well as economical and "clean" engines. If a modern engine is compared with an engine of thirty years ago, it can be stated that an engine manufactured at that time has produced the exhaust emission which corresponds to the value of the exhaust emission produced by at least a hundred engines today (Prof. Dr. Gruden).

ENVIRONMENTAL ASPECTS – THE CURRENT STATE

Why is it important that new cars are lightweight, with low fuel consumption and that the total exhaust emissions meet the latest relevant EU regulations? Since 1886 until today, about 2.5 billion passenger cars have been produced. About 28% of them are still drivable and in traffic. Every year, there comes a couple of millions of new cars (67.8 million in 2013), so we are approaching a mass of one billion passenger cars. If the total number of all cars in the world is considered, then we have to add about 307 million of trucks (in 2010) and about 360 million motorcycles (in 2013) to this number. It is clear that these technical pets represent a huge thermal and chemical load on nature and they affect the remaining energy resources on earth. Figure 6 shows the annual production and the total number of the passenger cars in the world.

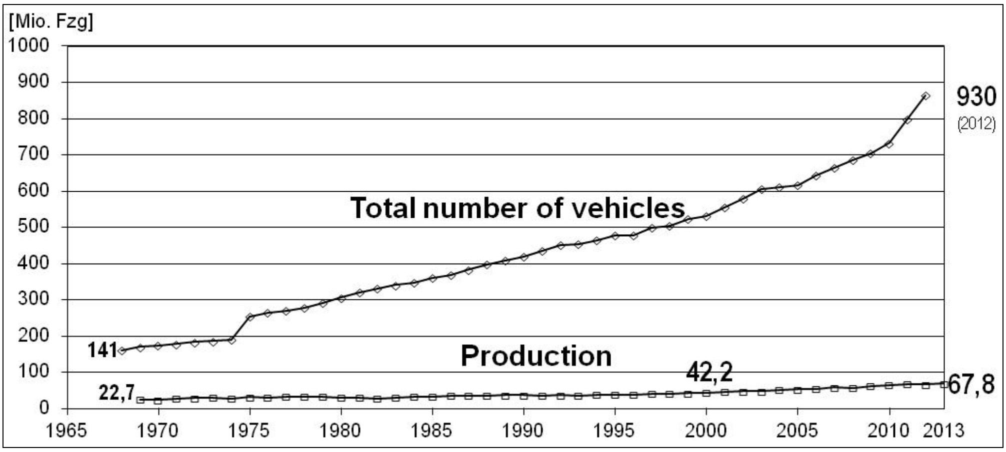

Figure 6 Passenger cars – Production and total number [1]

NEW TECHNOLOGIES IN PRODUCTION

Every day we witness the introduction of new technologies into mass-production. It's not easy, because new technologies are initially very expensive and not sufficiently mature for mass-production and dozens of years pass, from first experiments and test series, before a new technology or an invention is finally introduced into mass production. This is supported by two examples. The first one is from engine technology: two camshafts in the cylinder head and four valves per cylinder are known for decades, but just now they are an essential concept in the mass volume production. Another example is from the field of car body building: SpaceFrame construction concept is already known for a quarter of a century, but, due to the cost and complexity of tri-dimensional cross sections and welds, it would not be a dominating building technology for much longer.

Let's look at the list of currently present technologies and methods applicable in automotive production.

Method of construction (structure - bodywork)

As it may be seen from historical review, since 1886 until today, the way of building of passenger cars and their supporting structures has been significantly changed and improved. This is particularly evident in the last 25 years. Through the development of steel alloys, aluminium alloys and new plastic materials, the new principles of building self-supporting structures of the car were enabled, regardless of whether it is a self-supporting car body, "SpaceFrame" structure or self-supporting "monocoque" structure. The principles of building of supporting structures or complete car bodies in relation to the applied material are just listed here:

- Steel: mainly conventional self-supporting car bodies and seldom SpaceFrame car bodies
- Aluminium: seldom conventional self-supporting car bodies and mainly SpaceFrame car bodies
- Plastics and composites: mainly monocoque structure

- Multi-material concept, Steel/Aluminium/Plastic and composites in conventional self-supporting, SpaceFrame or in combination with monocoque bodies, too.

Parts production

Production of automotive structural parts has considerably developed in the last two decades, through completely new forms of design, but also through improvement of long known technologies. This is the case with "Hydroforming" technology, which has been known for more than 100 years, but only now is complemented with technological procedures to enable the development of stable and lightweight parts of the supporting structures. Current and new technologies in the production of parts are listed here:

- Conventional (stamping parts - mainly all steel cars)
- Hydroforming (Tubes, sheets), (A-Pillar BMW, B-Pillar Ford, rollover bar many cars, etc.)
- Heatforming (Tubes) (Al-structural part at Ferrari 458 Italia)
- Superplastic Forming (Aston Martin Vanquish, Morgan Aero 8, Mercedes SLS, etc.)
- Tailored blanks, Tailored tubes, Patchwork Blanks (many steel cars)
- Al - extruded profiles (many Al-cars)
- St - roll profiles (in development, Project ATLAS 2002)
- Tailored Strips Technologies (in development, ThyssenKrupp 2012)
- Electromagnetic pulse Technology (for suspension, in development in body).

Joining technology and connection methods

Spot welding at steel car bodies has dominated for the last hundred years. It has even been used for the three-year period (from 1953 to 1956) in the production of the first mass production car made of aluminium sheets (Panhard Dyna Z1). Main technologies and methods used for joining the parts of a supporting structure or car body are listed here:

- Spot welding (conventional method - at many steel cars)
- Laser welding (different systems at many steel and aluminium cars)
- Laser brazing (many steel cars)
- Adhesive joining (many steel and aluminium cars)
- Self piercing rivets (many aluminium cars)
- Clinch-spots /Clinching (many aluminium cars)
- FDS - Flow Drill Screws (many aluminium cars - Audi, Mercedes, etc)
- Tack impact (Mercedes SL)
- FSW - Friction Stir Welding (Ford GT40 and Mercedes SL)

Available materials

It is well known how the car bodies of large series cars were made about two decades ago. The car body was generally built from sheets having single quality of steel and different thickness. Today, in a modern car body or supporting structure, a large number of different materials and different wall thicknesses (sheet, profile or cast) is implemented, so, at the loaded locations, there is as much material as is necessary, and, at unloaded locations, material is reduced. In this way, the basic principle of lightweight design is fulfilled. All this enables a large number of new materials and countless number alloys to be used in modern cars (Figure 7). New kinds of plastics and composites must also be accounted for here (a lot of new steel and aluminium alloys and new plastics and composites). Despite all this, it must be recognized, that steel, which has been improved and optimized thanks to the research projects of large steel producers, still dominates the large-scale production (Figure 8). Thus, they have contributed to a high quality, efficiently used mass (weight) and safety of new steel supporting structures or car bodies.

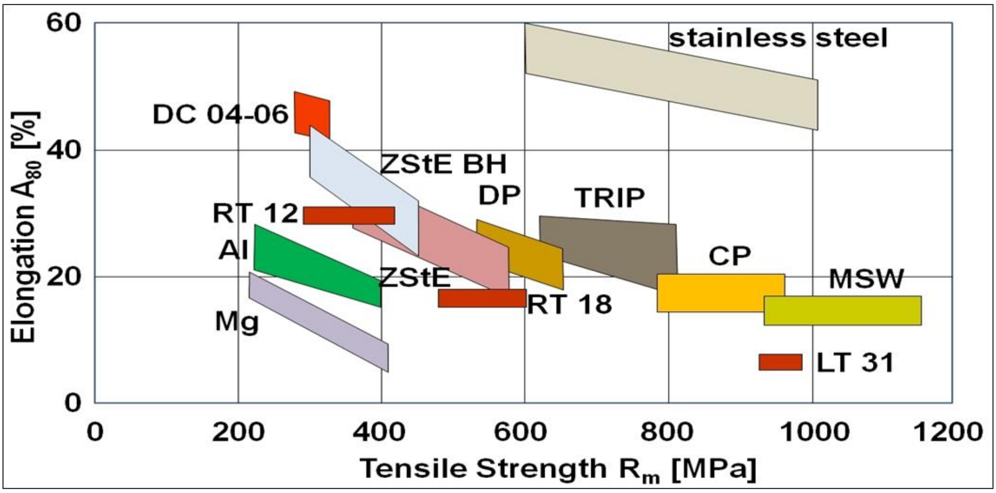

Figure 7 The properties of metallurgical classes of steel, aluminium and magnesium [8]

MAJOR DEVELOPMENT PROJECTS OF STEEL BODYWORK

Major development project of steel bodywork are:

- ULSAB - Ultra Light Steel Auto Body 1996 (Consortium, 35 steel producers, 18 States)
- ULSAB AVC 2002
- Project „ATLAS“ (Salzgitter/Karmann) 2002
- ArcelorMittal „ABC“ project 2004
- Benteler „PG-structure“ 2008
- ThyssenKrupp „InCar project“ 2009
- ArcelorMittal „S-in motion“ project 2010

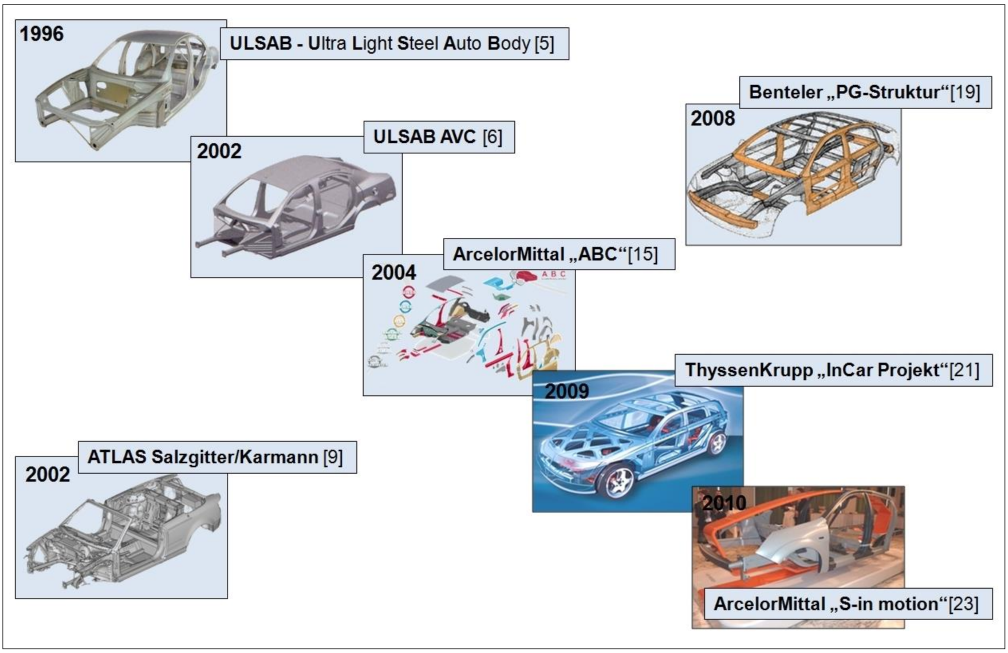

Figure 8 Major development projects of steel bodywork

CONCLUSIONS - NEW VIEWPOINTS FOR THE FUTURE

How did we actually come to the modern car? For years, the car has been developed without respect to man and nature and our intentions were for cars to be stronger, faster, larger and more booted up every day. Thus, a time period may be noticed, during which, with these project goals, the cars had gained big engines and had become not only faster, but also very heavy and large consumers of fuel. Unfortunately, it must be noted, that these cars had not met almost any of today’s crash-standards. This period of automotive history has finally stopped and, since July 1992, by introduction of exhaust emissions control (EU emission regulations) and regulations limiting fuel consumption and CO2 emissions per kilometre, has finally been directed. This has led to today's serial production cars whose engines already meet EURO 6 standard. Soon after, standards on passive safety (safety standards) had occurred. It's nice to say that the car bodies of new cars meet these current standards, important for passenger safety. Through a lot of competition in the market, unwritten rules started to apply, on equipping the car with all the known and available serial elements of active safety during steering or manoeuvring the vehicle, as well as elements of passive safety. All this has contributed to the fact that today we drive secure, very efficient (lightweight) as far as weight is concerned, equipped with very "clean" engines and technically functional cars.

It is clear that the premises, on which the cars of the future will be conceived, will be different from the former, and partly present premises and ecological demands and requirements on conservation of energy resources will get priority, which will greatly affect the concept, look and manner of use of future cars. Cars of the future will be strong enough, fast enough and sufficiently dynamic, which means that everything will be in accordance with the electronically strictly controlled roads and highways. In Europe, only in Germany, there are several sections of highways where you can drive without a speed limit. Speed limits are conditioned by the current number of vehicles at any given time on such sections without speed limits. Given that the world's current number of the passenger cars amounts to about one billion, that there are about 307 million of trucks and about 360 million of motorcycles, and all with a tendency of further growth, it is clear that many things will change in terms of traffic and its regulation. All this indicates that it is more likely that today's way of driving will no longer be possible in the future. Soon, there will be no roads without speed limits and cars will not be able to be controlled individually, but they will have to be harmonized with other participants in traffic, and that would be very tedious for the everyday driver. Considering the existing differences in control skills regarding the car (age differences and speed of reaction), it is clear that electronically driven vehicles will have supremacy in the future, which will significantly increase the traffic safety. The one charm of free driving well known by older generations and longed for by the young, will be unattainable.

We should bear in mind the fact, that the designers of future cars will face the problems that have been neglected so far and those are the types of fuel and total autonomy of movement. The future cars, the ones that will still move on the ground, will lose their specific and individual appearance - it will be cars having functional and unified rational forms, absolutely electronically controlled and managed. The driver will have very little of creative possibilities available when driving.

These considerations can very easily lead to the conclusion that, in the future, there would probably be private category cars, but the possibility of creating the pace and the dynamics of movement and road behavior will not exist. Driving will give the impression that there is a private programmed tram.

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THE EFFECT OF INTERMODAL TRANSPORT ON THE REDUCTION OF CO_2 EMISSION

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UDC:629.014.1

ABSTRACT: In October 2013 at the demand of Germany, The European Union postponed the adoption of regulations that stipulate that the average carbon dioxide emission of newly-manufactured passenger vehicles would be reduced from 130 gCO_2/km in 2015 to 95 gCO_2/km until 2020, which does not understate the importance of the need to reduce the emission of pollutants in road transport. European Automobile Manufacturers Association (ACEA) considers that crediting is necessary and they remind that credits should be used as incentives to technological innovations concerning reduction of carbon dioxide emission. In freight transport more and more carriers deliver monthly reports about carbon dioxide emission and inform their clients about the emission of pollutants and the savings generated in transport. The aims of EU-28 are directed towards the reduction of existing CO_2 emission by 20% until 2020 and by 80-95% by 2050 in comparison to 2010. One of the possibilities for CO_2 emission reduction is the application of intermodal transport which would meet all basic goals of relevant national and international treaties. Many studies have shown that the application of intermodal transport can reduce CO_2 emission in road transport by 13% or more, especially on distances over 250 km long. This paper shows possible reductions of CO_2 emission depending on the vehicle load and distance according to various intermodal technologies.

KEY WORDS: intermodal transport, CO_2 emission, emission factors.

EFEKTI INTERMODALNOG TRANSPORTA NA EMANJENJE EMISIJE CO_2

REZIME: U oktobru 2013. godine na zahtev Nemačke, Evropska unija odložila je donošenje propisa koji propisuju da je prosečna emisija ugljen dioksida novo-proizvedenih putničkih vozila bude smanjena sa 130 g CO_2/km u 2015. na 95 gCO_2/km do 2020. godine, što je važan činilac koji se ne sme potceniti zbog potrebe za smanjenjem emisije polutanata u drumskom saobraćaju. Evropsko udruženje proizvođača automobila (ACEA) smatra da je kreditiranje je neophodno i da oni treba da posluže za podsticaj tehnološkog razvoja za smanjenje emisije uljen-dioksida. U teretnom transportu sve više i više prevoznika dostavlja mesečne izvještaje o emisiji ugljen dioksida i obaveštavaju svoje klijente o emisiji polutanata i ostvarenim uštedama u transportu. Ciljevi EU-28 su usmerena na smanjenje postojeće emisije CO_2 za 20% do 2020. godine i za 80-95% do 2050. godine u odnosu na 2010. Jedna od mogućnosti za smanjenje emisije CO_2 je primena intermodalnog transporta koji će zadovoljiti svi osnovne ciljeve relevantnih nacionalnih i međunarodnih ugovora. Mnoge studije su pokazale da je primena intermodalnog transporta može smanjiti emisiju CO_2 u drumskom saobraćaju za 13% ili više, pogotovo na udaljenostima većim od 250 km. Ovaj rad prikazuje moguće smanjenje emisije CO_2, zavisno od opterećenja vozila i udaljenosti prema različitim vrstama intermodalnih tehnologija.

KLJUČNE REČI: intermodalni transport, emisija CO_2, emisioni faktori

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THE EFFECT OF INTERMODAL TRANSPORT ON THE REDUCTION OF CO_2 EMISSION

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INTRODUCTION

Oxides of carbon CO_2 and CO, methane CO_4 and volatile organic compounds represent serious atmosphere pollutants by creating the so-called “greenhouse efect”. Over the last 250 years there has been a constant increase of total concentration of carbon dioxide from various sources in the world.

Table 1 Overview of CO_2 concentration and mass in the period 1750-2100

Year	CO_2 imission (ppmv)			Mass of fractions (%) CO_2 (ppm)	Mass of CO_2 in the atmosphere (t)
1750	A		278	422.2	2.173E+12
1960	B		310	470.8	2.423E+12
2014 (March)	C		399	581,7	2.993E+12
2100	D	(i)	541	821.7	4.228E+12
		(ii)	970	1473.2	7.581E+12

Source: CO2 Home, “ppm” stands for "parts per million", "ppmv/w" stands for "ppm by volume/water" ppmv = (mg/m3)(273.15 + °C) / (12.187)(MW)

The period from 1750 represents pre-industrial era characterized by low concentration of CO_2; from 1750 to 1860 the concentration had increased by 250 000 million tons of CO_2, that is, 1190,48 million tons per year. From 1960 to 2007, new 570 000 million tons of CO_2 were emitted in the atmosphere, that is, 12 127,7 milllion tons per year, which is almost ten times more than in the previous two centuries (Table 1).

According to US EPA, the total emissioin in 2012 amounted to around 6 526 million of metric tons of CO_2, 28 % of which is transport. In EU-27 average emissions by sectors point to the fact that transport has a share in emissions 19,6 % - 21 % varying from one state to another; 6% of this number falls on freight vehicles due to the impact of different factors in road transport: roads (traffic density, maximum vehicle weight, climate conditions, topography, driving style, etc.), Figure 1.

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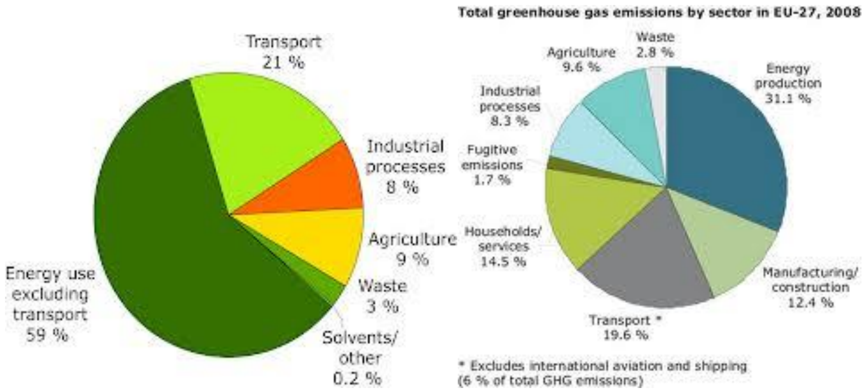
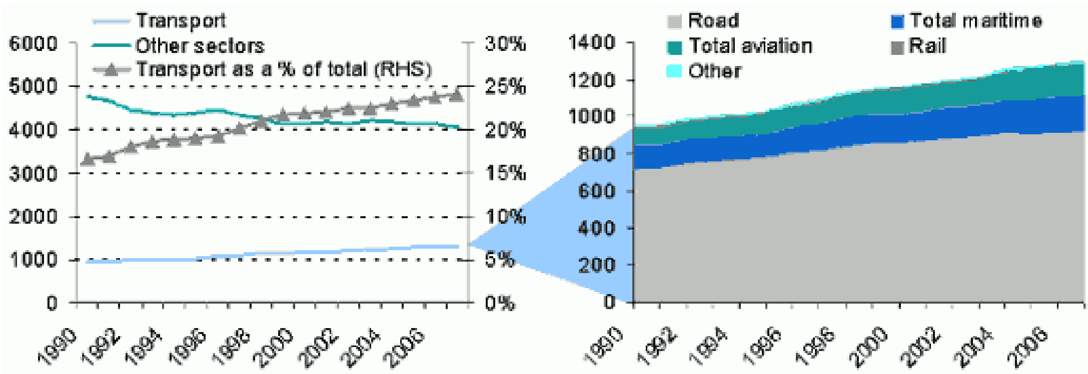


Figure 1 Total emission by sector in EU-27, 2007/08

Considering the types of transport, the biggest emission comes from air transport, followed by road transport, Figure 2. The intensity of the emission in railway transport is affected by: percentage of electrified railways, freight and free rails, density of terminals within the network, etc. In internal waterways the following factors have the biggest impact on emission: weight and age of the vessel, draft depth, density of terminals within the network, weather conditions, etc.



Source: EEA

Figure 2 Emission by types of transport in EU, 1990-2006

In order to reduce the emission, it is necessary to apply new technologies, redirect technological operations to “greener” types of transport that emit the lowest amount of CO_2 per tkm or other unit of measurement. It is essential to connect and optimize transport chains, improve the planning process, increase the exploitable carrying capacity, minimize the movement of freight vehicles without load, etc. On the other hand, it is very important to increase the efficiency of fuel combustion and reduce the share of carbon in fuel.

CURRENT RESEARCH OF CO2 EMISSION IN INTERMODAL TRANSPORT

A significant number of studies, projects and articles published in the world show applied methodologies and main results from different aspects of the effect of intermodal technologies on the air quality and the possibility of reducing the negative effect of CO2.

According to [2], the study (IFEU and SGKV, 2002) compares the energy consumption and CO_2 emissions between road and road-rail combined transport on 19 different European routes in container transportation. The study shows that 3 of 19 routes demand higher energy consumption than in road-rail combined transport; in eight cases the demand for fuel in combined transport is 20 % lower than that of road transport, in six cases it is 20 % - 40 % lower, and in two cases it is lower by more than 40 % which resulted in lower gas emission by 20-50%. The differences occur when using different ways of organizing rail transport (types of trains, lengths of trains where 700 m long trains consume 60 % less energy), PPH distances of heavy goods vehicles (HGV), age of the vehicle and fuel consumption, conditions on the roads, possible return trips, etc. The study is limited to comparing technologies and performance of adopted networks and operative features: length of the distance of main intermodal transport at around 450 km, trains have 21 – 28 cars (with maximum train length of 400-550 m), barges are Neo Kompenaar (32 TEU) and European barge (208 TEU). The study finds a quite low fuel consumption, 29 litres per 100 km, of tugboats (IRU/BGL: 34 liters/100 km). Road vehicles transport goods in both directions. "Transport en Logistiek Nederland" have come to the conclusion that space occupation in intermodal rail door-to-door transport is three times bigger than in road transport, table 2.

Table 2 Ecological performance of intermodal and unimodal transport

Technology Performance	Rail-maritime transport	Rail-road transport	Barge-maritime transport	Barge-road transport
Energy need	Favourable	Non-favourable	Non-favourable	Non-favourable
CO_2	Favourable	Non-favourable	Non-favourable	Non-favourable
NOx	Favourable	Slightly favourable	Non-favourable	Non-favourable
SO_2 and suspended particles	Non-favourable	Non-favourable	Non-favourable	Non-favourable
Land occupation	Non-favourable	Non-favourable	-	-

The paper, "Comparison of external costs of rail and truck freight transportation" (Forkenbrock, 2001), demonstrates that if external costs were included in total transport costs, the costs of freight delivery would be increased by around 13% (0,86 \$/tkm +8,42 \$/tkm) in road transport, and 9,3%-22,6% in rail transport.

Table 3 External costs for goods transport via road vehicle and railway (\$/tkm)

External costs Means of transport	Accidents	Air pollution	Greenhouse effect	Noise	Total
Road vehicle	0,59	0,08	0,15	0,04	0,86
Heavy unit train	0,17	0,01	0,02	0,04	0,24
Mixed freight train	0,17	0,01	0,02	0,04	0,24
Intermodal train	0,17	0,02	0,02	0,04	0,25
Double-stack train	0,17	0,01	0,02	0,04	0,24

The study concludes that based on the transport operation (tkm) transport via road vehicle has higher external costs that go up to three times higher than any other means of transport in railway traffic, table 3.

In paper "Combined transport is environmentally friendly; fiction or reality?” (De Leijer i Ruijgrok, 1990) the authors note that external costs of transport (pollutants emission and fuel consumption) should be compared based on technical characteristics of technologies (loading capacity, degree of vehicle exploitation, type of vehicle in use, distance of pre- and post-haulage PPH, geographical and topographical conditions of the routes, congestion on the road, etc.) by tons transported per km. However the problem is that this is not always suitable grounds for comparison. For instance, in algebra, 1000 tons transported over 10 kilometres is the same as 10 tons transported over 1000 kilometres. Even so, the selection of operations, vehicles and operative costs of transport-manipulative means would be different in both cases.

In paper “Transport by rail, comparison between the emissions of unimodal road transport and combined road-rail transport” (Van Binsbergen and Shoemaker,1993), the authors compare emissions of combined container rail-road transport depending on a load per distance (table 4). It is shown that variation of vehicle load led only to the increase of SO_2 emission while the emissions of other polluters decreased.

Table 4 *Emissions of intermodal rail-road transport compared to unimodal road transport*

Trip with load in both directions	Depart trip with load and return wihtout load	Depart trip without load and return trip with load
NO_x, aerosols, C_xH_y up to 80% reduction	NO_x, aerosols, C_xH_y up to 81% reduction	NO_x, aerosols, C_xH_y up to 83% reduction
CO_2, CO reductions between 36% - 52%.	CO_2, CO reduction between 38% - 53%.	CO_2, CO reduction between 39% - 58%.
SO_2: increase of 52%	SO_2: increase of 47%	SO_2: increase of 46%

In the paper “Emissions of combined transport” (Walstra et al., 1995), the authors compare the emissions and energy consumption between unimodal road transport and combined transport, from the theoretical aspect. The results clearly showed that combined transport is more favourable to the environment, despite the facts that its pre- and post-haulage produces a lot of emissions. The study, also, shows that the differences in emissions between road-barge and road-rail transport are very small. CO_2 emissions are always lower in barge transport than rail transport, while the NOx, CO, CxHy and aerosol emissions are always higher. In rail freight transport on electrified lines the emissions of CO2 are practically insignificant (0,001%). If the electrical energy is “green”, i.e., produced by water plants or nuclear plants than the emissions can be disregarded.

There are other projects from various countries supported by EU (Recordit, Externe, QUITS, PETS etc.), from which we can see that the analyses were done for specific routes and modalities with specific technical characteristics of vehicles and the origin of primary energy with different effects of emissions. Additionally it can be concluded that:

- countries have different ambitions when it comes to reduction of CO2 according to modalities,
- the countries have policy instruments in the transport sector, these policies are not intended to directly reduce CO2 emissions, however they have indirect influence on potential reduction of carbon dioxide emissions (safety measures),
- countries make different decisions in the selection of instruments of the policy of the reduction of carbon dioxide emissions,
- some countries involve target groups in the process of defining the policy and in decision-making process,

- the effects of non-transport policy with the effects of transport emissions are usually not considered in the policy of reduction of carbon dioxide emissions,
- CO2 emissons produced by transport have increased and they vary from one country to another,
- the differences in emission trends seem more connected to specific circumstances in countries than the policy of the reduction of carbon dioxide emissions,
- comparison of national policies in their attempt to reduce the effect of transport regarding CO2 is becoming a complex matter in most countries,
- the emissions produced by fuels without CO2 have not been analysed.

The analyses show that there are no significant differences in emissions (from -2 % to +3 % according to the average value) between EU countries on annual basis, and that other European countries have adopted the policy of reduction of transport impacts on the environment.

NECESSARY DATA AND EMISSION FACTORS IN INTERMODAL TECHNOLOGIES

The analysis of the effect of intermodal transport on the reduction of carbon dioxide emission consists of 5 steps:

- Step no. 1 Defining the goals of the research
- Step no. 2 Choosing assessment approach and defining research limits
- Step no. 3 Collecting data and choosing emission factors
- Step no. 4 Emission calculation
- Step no. 5 Verification and presentation

Basically, there are three levels of specification: (1) the lowest level which determines the emission based on average carrying capacity, type of engine, type of fuel and ballast factors for all the vehicles used for transportation, (2) medium level which analyzes homogenous vehicle types and defines average engine type, fuel type, and ballast factor for each type of vehicle, and (3) the highest level which analyzes more details and determines CO2 emission for each vehicle. Total emission of CO2 is expressed in different units of measurement (g/veh, g-kg-t/km, g-kg-t/tkm, kg/TEU, kgCO2/kg diesel etc.) and in the unit of measurement that expresses capacity in a specific chain and defined time period (day, month, year). In road freight transport, emission depends on several factors: type of the vehicle, type of the road, fuel consumption, steering techniques, fuel quality, velocity, etc. The calculation of warm emission of CO2 for a road vehicle in an undefined time period can be calculated through the following equation (1).

$$CO_2\ \text{Emission} = Nvoz \cdot Ef \cdot lpr \cdot .Fm\ \ (g/veh) \qquad (1)$$

where: Nvoz is the number of trips in a certain period of time; Ef – emission factor,(g/km); lpr – mileage per trip, (km); Fm is ballast level, (%)

Models of the emission factors are based on a unique factor for specific modalities, types of vehicles and transport technology which perform transport process in specific driving conditions. Emission factors are determined as a mean value of total emission's repeated measurements through the driving cycle that is usually defined as a mass of pollutant produced per distance unit or any other unit used for defining emission.

In the calculation of emissions, emission factor (Ef) can be used as 'standard' that is in accordance with 2006 IPCC principles or through Life Cycle Assessmet (LCA), which takes into consideration whole life cycle of an energy carrier. In transport, it is expressed by different units of measurement and the unit which defines total emission is used. CO2 emission is proportional to fuel consumption in general, so 1 l of consumed diesel fuel normally produces approximately 2,64 kg of CO_2 for diesel fuel mass density 0.84 kg/l and it applies under the condition that the amount of carbon in the fuel completely oxidizes into CO_2. Conversion basis for emission calculation 1 g/km CO2 for diesel fuel =0,043103448275862 l/100 km, conversion base: 1l/100 km = 23,2 g CO2/km. Value of the emission factor in road transport varies depending on the exploitation degree of vehicles' carrying capacity and empty vehicle mileage and ranges from 0.780 kg/km to 0.884 kg/km on average.

If total emission in road transport is evaluated in g/tkm, then standard emission factor is 62 gCO_2/tkm (61.9) (table 5). This value is formed based on the 80 % usage of 40 t - 44 t vehicle's carrying capacity of 40 t - 44 t and 25 % mileage of vehicle when without load. With the increase of load weight inside a vehicle and the decrease of mileage without load, significant decrease of CO2 emission occurs which can be characterized as negative exponential distribution.

Table 5 Emission factor changes depending on load weight and covered road structure in gCO_2/tkm

Load weight (t)	Percentage of covered distance without load in total covered distance (%)										
	0%	5%	10%	15%	20%	25%	30%	35%	40%	45%	50%
10	81,0	84,7	88,8	93,4	98,5	104,4	111,1	118,8	127,8	138,4	151,1
11	74,8	78,2	81,9	86,1	90,8	96,1	102,1	109,1	117,3	127,0	138,6
12	69,7	72,8	76,2	80,0	84,3	89,2	94,7	101,1	108,6	117,5	128,1
13	65,4	68,2	71,4	74,9	78,9	83,4	88,5	94,4	101,3	109,5	119,3
14	61,7	64,4	67,3	70,6	74,2	78,4	83,2	88,7	95,1	102,7	111,8
15	58,6	61,0	63,8	66,8	70,3	74,2	78,6	83,7	89,7	86,8	105,3
16	55,9	58,2	60,7	63,6	66,8	70,5	74,6	79,5	85,1	91,7	99,7
17	53,5	55,7	58,1	60,8	63,8	67,2	71,2	75,7	81,0	87,2	94,7
18	51,4	53,5	55,8	58,3	61,2	64,4	68,1	72,4	77,4	83,3	90,4
19	49,6	51,5	53,7	56,1	58,8	**61,9**	65,4	69,5	74,2	79,8	86,5
20	48,0	49,8	51,9	54,2	56,8	59,7	63,0	66,9	71,4	76,7	83,0
21	46,6	48,3	50,3	52,5	54,9	57,7	60,9	64,5	68,8	73,9	80,0
22	45,3	47,0	48,8	50,9	53,3	55,9	59,0	62,5	66,5	71,4	77,2
23	44,2	45,8	47,6	49,6	51,8	54,3	57,2	60,6	64,5	59,1	74,7
24	43,2	44,7	46,4	48,3	50,5	52,9	55,7	58,9	62,7	67,1	72,4
25	42,3	43,8	45,4	47,3	49,3	51,7	54,3	57,4	61,0	65,2	70,3
26	41,5	42,9	44,5	46,3	48,3	50,5	53,1	56,0	59,5	63,6	68,5
27	40,8	42,2	43,7	45,4	47,3	49,5	52,0	54,8	58,1	62,1	66,8
28	40,2	41,5	43,0	44,6	46,5	48,6	51,0	53,7	56,9	60,7	65,3
29	39,7	41,0	41,4	44,0	45,7	47,8	50,1	52,7	55,8	59,5	63,9

Ballast level (Fm) depends on vehicle's carrying capacity (t), weight of loaded cargo (t), and weight of internal cargo within certain company, and in road transport its value is approximately 0.75 with high frequency transportation and 0.50 with a single trip. If carrying capacity of a vehicle is 10 t and load weight 8 t, then the usage of nominal carrying is 0.80. In groupage transport, it can occur that out of 8 t of loaded cargo, 5 t belongs to another sender, so the usage is 0.50, and by dividing 0.5/0.8=0.63 we get correction factor. If the emission factor is 0.78, mileage is 100 km, number of rides is 10, with the correction factor which is 0.63, then the total emission is 487.5 kg of CO_2. If the cargo belongs to one sender, then correction factor is equal to static vehicle utilization coefficient. The higher vehicle utilization factor is, total emission is lower, table 6.

Table 6 *CO_2 emission of road vehicle depending on ballast percentage*

Existing exploitation of vehicle's carrying capacity	Increase of carrying capacity %	Saving in CO_2 emission in %
40%	50	16
	60	27
	70	34
50%	60	13
	70	22
	80	29
60%	70	10
	80	18
	90	24

In road traffic, total emission can be assessed based on the equation:

$$CO2\ Emission = Eg \cdot lpr \cdot Ef\ (kg) \qquad (2)$$

where: Eg – fuel consumption for specific vehicle ballast, (l/km); lpr – total mileage per vehicle, (km); Ef – emission factor of the fuel, (kgCO2 / l fuel)

Theoretical fuel consumption (Eg) in road transport (HGV>3.5 t) is defined by the manufacturer and it has following values: MAN trucks, from manufacturer OAF & Steir, using Euro 3, or older Euro 2 diesel fuel have the consumption from 33 to 37 l/ 100 km (loaded) and 29 to 32 l/ 100 km (empty). Renault vehicles using Euro 3 have the consumption of diesel between 29.3 and 33.4 l/ 100km (loaded). Scania vehicles have the consumption of Euro 3 from 31to 32.6 l / 100 km (fully loaded) and from 22 to 23 l/ 100 km (empty). Volvo vehicles which use Euro 3 have the consumption from 29 to 32 l/ 100 km (loaded) and 18 to 20 l /100 km (empty). Because of the age and state of the vehicles, analyses should use average consumption of 39 l/100 km for a loaded vehicle and 29 l/100 km for an empty vehicle. According to (7), actual average fuel consumption of heavy goods vehicle with carrying capacity of 16 to 32 tones is from 210 to 251 g/km and for the vehicles over 32 tones is from 251 to 297 g/km.

In freight road transport emission is defined with the following relation:

$$CO_2\ emission = U \cdot Ef\ \ (g) \qquad (3)$$

where: U – cargo amount transported,(tkm); Ef - emission factor (g/tkm).

Approximate emission values of subcategorized HGV are: <7,5 t (452), 7,5 t-14 t (294), 14-20 t (294), >20 t (218), road train <20 t (161), 20-28 t (133), 28-32 t (128), >32 t (128) semitrailer truck <32 t (114) i >32 t (111), acerage of 147 gCO2/tkm.

In railroad transport, total emission for engines that are diesel or electric-powered can be estimated based on the equation:

$$TE = Q_c \cdot l \cdot \frac{EF_{CO_2}}{1 \cdot 10^6} \cdot \frac{153.07 \cdot f_t \cdot Q_v^{-0.5}}{FO} \quad TE = \sum_z Q_c \cdot l_z \cdot \frac{EF_{Z,CO_2}}{1 \cdot 10^6} \cdot \frac{675 \cdot f_t \cdot Q_v^{-0.5}}{FO \cdot (1 - TI)} \tag{4}$$

where:TE – total emission; (t CO2); Qc – cargo weight, (t); l – transport distance, (km); Efco2 – emission factor for diesel-powered engines, (kgCO2/kg diesel); Ef z-CO2 – emission factor for electric-powered engines, (kg CO2/KWh); ft – terrain factor (for the flat terrain, it should be decreased by 20%, for the hilly terrain there is no alteration and for mountainous terrains, it should be increased by 20 %); Qv – total train weight, (t); FO – ballast factor (0,72 for bulk cargo, 0,58 for general cargo and 0,44 for high volume cargo); TI – percentage of energy loss due to transportation losses; z – country.

Energy consumption and therefore CO2 emission in railroad transport depends on: type of engine power (diesel or electric) and the length and gross weight of a train (train with the gross weight of 1500 t consumes 17.6 (Wh/brtkm), of 1000 t consumes 21.0 (Wh/brtkm), of 600 t consumes 26.7 (Wh/brtkm), (NTM Rail, 2008), type of the engine and consumption, type of cargo, spatial position of the railroad, valid transport limitations, number and condition of frigo cars (with power between 10 and 2 kW). Additional emission occurs when the interior of a car and/or container is cleaned with the use of steem (for the consumption of 189 kWh emission is 38 kg CO2 per car or container).

It was estimated that for cargo transportation by international fully loaded container train consumption is 30 kWh/veh.-km, and 20kWh/veh.-km for the empty containers transport. It is presumed that electric-powered engines cover 75% of railways, and diesel-powered engines 25% in most European countries. If there are no data about share (in percentage) of individual systems within diesel-electrical power matter, then for the rough emission of electrical-powered engines we can use emission of diesel-power multiplied by 0.25 and then add calculated emission based on the equation of electric power multiplied by 0.75. In railroad transport, the value of the emission factor based on realized gross tkm ranges from 1.8 g CO2/tkm to 19 g CO2/tkm for electric and from 21 kg CO2 /tkm to 55 kg CO2/tkm for diesel power. Since it is very hard to determine the value of the emission factor in the calculations of rough emission, its average value of 22 g CO2 /tkm can be used.

In water transport total emission can be calculated with following equation:

$$TE = PG \cdot l \cdot EFCO2 \tag{5}$$

where: TE – total emission of CO2, (kgCO2); PG - fuel consumption, (t/km); l – transportation distance, (km); EFCO2 – emission factor (kg CO2 per ton of fuel).

Recommended average value of the emission factor in national water transport of barges in container transportation is 31 g CO2/tkm or 0.367 kg/km, although there are differences depending on whether the transport is upstream or downstream, or on the canal, and depending on the size of barges (small 90 TEU, medium sized 208 TEU and big 500 TEU). The bigger the barge, and if it is going downstream, the factor is lower (for example, small barges going upstream have the emission factor of 63.4 g CO2/tkm or the big barge going downstream has the emission factor value of 10.2 gCO2/tkm. In maritime transport, depending on the size of a ship, the emission factor value is 13.5 g CO2/tkm for small ships (up to 2500 t), for those of medium size, it is 11.5 g CO2/tkm, and for the big ships it is 8.4 g CO2/tkm. The emission factors vary depending on the ships, (table 7).

Table 7 Estimation of emission factors for cargo ships in gCO2/tkm

Ship type	Carrying capcity	Emission factor
Tankers	0 - 60000 + dwt	5,7 – 45,0
Chemical tankers	0 - 20000 + dwt	8,4 - 22,2
LPG carriers	0 - 49,999 + m^3	9 – 43,5
LNG carriers	0 - 200.000 m^3	11,9 – 14,5
General cargo	0 - 9.999+ dwt + TEU	11,9 – 19,8
Reefer ship	Svi	12,9
Container ship	8000 + TEU	12,5
Container ship	5000 - 7999 TEU	16,6
Container ship	3000 - 4999 TEU	16,6
Container ship	2000 - 2999 TEU	20,0
Container ship	1000 - 1999 TEU	32,1
Container ship	0 – 999 TEU	36,3
Ro-Ro ship	2000 + Im	49,5
Ro-Ro ship	0 – 1999 Im	60,3

If in water transport the emission factor is expressed based on covered distance, then its average value is around 0,357 kg/km. Correction factor in water transport has average value of 0.80 with direct transportation, 0.50 with shuttle transportation and 0.80 with air freight transport.

EMISSION ESTIMATION ACCORDING TO TRANSPORTATION CHAINS

Application of intermodal technologies represents one of the organizational measures which give significant results in reduction of emissions. Without plunging too deep into detailed analysis of possible variations of intermodal technologies we should identify transportation chains and within them applied technologies by organizational structure and types of transported commodities. If we take into consideration only land technologies 'vehicle-vehicle' there are 4 types of road-rail (Version A, B, C and Bi-modal), while at land-water, and entirely water more types exist. Version B (unaccompanied), represents transportation of road semitrailers by special rail vehicles, and along with the version C (unaccompained, shipping containers) represents the most commonly used land combined technology. In order to make simpler estimation, approximate values of total distance of 500 km and the amount of the load (1000000 t) are used.

Also, it is necessary to take into consideration the emission caused by the transhipment in the starting and finishing point of operation. In most of the land terminals load, as well as shipping containers, are transhiped (loading, unloading and/or transhipment) by container crane electrically powered by 100kW to 250kW whose average emission is around 0.002t CO2 / operation or in case of shipping containers diesel-powered cranes (RTGC, SCU/UC et.al) are used as container handlers. Their emission is 0.007t CO2 /operation. In the work with shipping pallets emissions are lower since lifting trucks with low power engines 20kW to 100kW (diesel, electrical energy or gas) are used. In determining the emission factor for means of mechanisation, apart from the installed power, the number of effective working hours per day/shift (6-12-18h) should be considered as well as average load of the crane hook in precentages (50%). Average emission factor of different types of cranes (100kW to 600kW) used in transportation chain has an approximate value of 567 gCO2 /kWh of work. In one hour of operating, cranes can make 24-180 operations in a shift, depending on technological demands. In foreign literature, emissions at starting and finishing points of operations, that is, at terminals are separately shown.

EXAMPLE 1. Unimodal chain

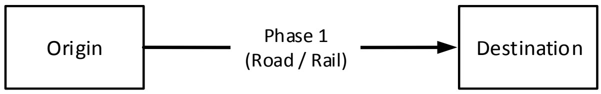


Figure 3 Direct unimodal land chain

In the case of a direct transportation of state-of-the-art freight units by unimodal transportation, transportation chain is simple and it practicaly represents an emission of a single type of transportation in which there are two tranships with their emissions (Figure 3). Estimated emission in road transport was performed based on formula (3) and previously mentioned emission factors (table 5). The difference in emissions indicates that the same amount of cargo weight at the same distance produces less emission by rail transport than by road transport.

Table 8 The estimation of CO2 emission in unimodal transport per ton-kilometer

Modality/technology	Load weight (t)	Distance (km)	Cargo amount transported (tkm in 10^3)	Em.factor (gCO_2/tkm)	Emission CO_2 (t)
Road	100000	500	50000	62	3100
Rail	100000	500	50000	22	1100

In case of a more detailed estimation of the total emission in one chain in which more subcategories of HGV (heavy goods vehicle) circulate, it is necessary to conduct the calculation of emisions per each subcategory by lenght of the transportation and add their emissions in observed time period.

Furthermore, it is necesarry to know exact statistical data on fuel (mass or volume), consumption by subcategories (Conventional, EURO I to EURO IV and EUROV-VI), number of the vehicles, average distance covered, amount of carbon in conventional fuel and fuel enhanced with additives, structure of the road covered, age of the vehicle, etc., by which the total emission could be estimated more precisely. With transhipping, given emission is related to one t-operation. In case of cargo unitization it is necessary to turn the amount of cargo into a number of containers or manipulation pallets.

EXAMPLE 2. BIMODAL CHAIN

When the emissions in bimodal land chain are estimated, the procedure is the same except for the fact that the single emissions of both types which participate in transportation are estimated by the same dimension and then added to each other. In intermodal land technologies, road transportation is used for the purposes of transporting the cargo to and from certain destination, and rail is used for the long distance transportation, Figure 4.

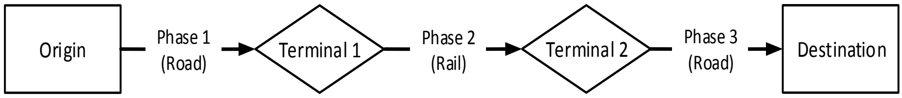


Figure 4 Bimodal land transport chain

As the value of the emission factor in road transportation depends on the exploitation of useful carrying capacity of the vehicle and length of the transportation without load, the value of emission factor in land bimodal chain depends on the length of road vehicle transportation in chain, table 9. Based on the total mileage covered in bimodal technology, in many European countries road traffic has a share of approximately 10% in total traffic.

Table 9 *Emission factor of rail vehicles in land bimodal chain*

	Means of transport	Distance in road transport in % according to the toal distance of the transportation			
		5%	**10%**	15%	20%
		gCO2/tkm			
Combined technology Road-Rail-Road	Rail (average)	24.0	**26.0**	28.0	30.0
	Electrical power (average EU)	21.2	**23.3**	25.2	27.6
	Diesel power	25.9	**27.8**	29.7	31.6
	Ro-Ro – Rail	38,3	**39,5**	40,8	42,0

The change of distance from 5% to 20% of the total planned length of the transportation chain often occurs in road transportation and it influences the value of the emission factor. It is recommended to use the deviations of 10% of total planned length of the transportion by road vehicles, which changes the value of the emission factor, table 10. If shipping conatainers are transported, the procedure is the same except it is necessary to know the weight of the specific shipping container, type of the load that is transported, whether the load is palleted or not, which type of palletes are used in a shipping container, etc. Characteristics of bimodal chain are: loading into a road vehicle at sender's location, two transhiping from road vehicle to rail car in both terminals and one unloading at receiver's location. Should there be more containers, then three loadings and unloadings should be considered.

Table 10 Estimation of CO_2 emission CO_2 in bimodal land chain

Modality/Technology	Load weight (t)	Distance (Km)	Cargo amount transported (tkm)	Em.factor (gCO_2/tkm)	Emission CO_2 (t)
Road	100000	2 · 25	5000000	53,7	2685
Rail	100000	450	45000000	23,3	9786

<u>EXAMPLE 3</u>. TRIMODAL CHAIN

Trimodal chains (road, water and rail) are much more complex in their structure and length of the distances, kinds and types of means of transport and mechanization, types of the load which is transported, etc., Figure 5. For every specific transportation chain it is important to know the technical characteristics of the means which are part of the chain. With water technologies there are other transport-manipulative diesel-powered means, whose functions in terminals are: locotractors which pull classic or LUF semitrailers, container handlers, different types of stakers, pneumatic and/or band transporters, air compressors, pumps (10kW to 70 kW), etc.

Figure 5 Trimodal transportation chain

As with bimodal chain, it is necessary to take into consideration the distance which road vehicle crosses in the total length of the transport. Based on the total distance covered with trimodal technology, in many countries of the world, road traffic takes aproximately 5% of the total traffic, table 11.

Table 11 Emission factor according to trimodal technologies

Intermodal technology	Means of transport	Distance in road transportation in % according to the total length of the transport			
		5%	10%	15%	20%
		gCO_2/tkm			
Road-Barge		**32,6**	34.1	35.7	37.2
Road-maritime short-distance technologies	RoRo-Ship	**49,7**	50.3	51.0	51.6
	RoRo-Rail	**38.3**	39.5	40.8	42.0
	Small tanker ships (up to 844 t)	**22.1**	24.2	26.3	28.4
	Big tanker ships (approximately 18371 t)	**7.9**	10.7	13.6	16.4
	Small bulk carrier (approximately 1720t)	**13.6**	16.1	18.7	21.2
	Big bulk carriers (14201 t)	**9.8**	12.5	15.3	18.0
	Small container carriers (approximately 2500 t)	**15.9**	18.4	20.8	23.2
	Big container carriers (20000 t)	**14.0**	16.6	19.1	21.6
	Other short-distance carriers	**18.3**	20.6	22.9	25.2

Table 12. gives total values of carbon dioxide emissions for the trimodal transportation chain given in Figure 5.

Table 12 *CO_2 emission estimation in trimodal land chain, road-water-rail-road*

Modality/Technology	Load weight (t)	Distance (km)	Cargo amount transported (tkm)	Em.factor (gCO_2/tkm)	Emission CO_2 (t)
Road	100000	$2 \cdot 25$	5000000	53,7	2685
Rail	100000	200	20000000	23,3	4660
Water	100000	250	25000000	18,3	4575

Depending on applied technology, using intermodal transport can make savings of 13% of CO_2 emission in comparison to all-road transport. Significant savings can be observed on the example of unaccompanied transport (versions B and/or C). In one closed block-train 32 trailers are transported on average (Version B), if there are two pairs of trains, 8 in total (4 at the departure and 4 at the arrival) per week, then the weekly capacity is 256 semi-trailers. One hauler pulls a semi-trailer (23800 kg) and spends 0.34 l/km on average. While burning one liter of diesel fuel it produces around 2.9 kg of CO_2, which shows that at every 100 km a hauler produces more than 91,8 kg of CO_2, 0,24 kg of NOx, around 0,003 kg PM etc. With this transport technology emission is around 40% of the total emission of the road transport (100%), which, in turn, creates a significant saving of 60%. If we would use accompanied transport (Version A) emission could be around 77% of the total emission of road transport, with which it would be possible to make a saving of 23%, Figure 6 [1]. If during 45 weeks of the year 11600 trailors are transported, 1 290 000 of trees are needed for the absorption of CO_2 emission, which is equivalent to 3 225 ha of forest, which would be enough for the recovery of the emission caused by road vehicles (one tree absorbes around 22 kg of CO_2, which means that there should be around 400 trees on one ha) which is practicaly impossible.

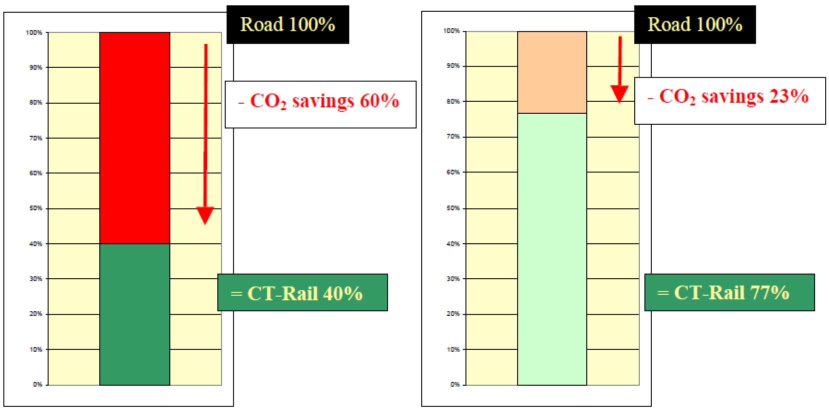


Figure 6 *Comparison of savings in CO_2 emissions with accompanied and unaccompanied transport*

There are various softwares for the estimation of CO_2 emission: ADMS Road, CORSIM based on Trafficware-Synchro 4 (Traffic Signal Coordination Software), VISUM/GIS, SoFi offered by PE INTERNATIONAL's, E coPorts self-diagnosis Method, EcoTransit calculator etc. which, along with the world's best quality databases on sustainability, provide intereseted parties with data on emission, financial effects of the sustainability, suggestions on how to improve sustainability of the operations, supply chains and/or the quality of the product. For the estimation of the emission from passenger vehicles 'electronic calculators' (Myclimate, Alpha Vehicle, WRI et al.) can be used, which include information on distance of the trip and fuel consumption per 100km.

CONCLUSION

The aim to increase security, protect the environment, improve and stimulate all types of clean technologies in traffic represent the basis of state policies, which also applies to stimulating the use of "clean" technologies. Guidelines for this kind of development are defined at the level of EU transport system (ECMT Council, Prague 2000) and based on reduction of harmful effects of transport on the environment, using the "3i" principle, application of modern information technologies and rational use of available capacities.

Carbon dioxide emissions and the lack of fossil fuels represent key guidelines for automotive development in the near future. Regarding road vehicles, in recent years the end goal is Zero Emission Vehicle (ZEV), even though it is known there is no human product that has zero effect on the environment, since the man alone produces around 4 tCO_2 per year. Only recently, new goals were introduced regarding emissions form vehicles: *NZEV* – Near *ZEV* i.e. near minimum emission and *EZEV* – equivalent zero emission vehicle.

There are different ways of affecting high emission from transport: improvement of technical chracateristics of vehicles by modalities, development of sustainable fuel with reduction of carbon intensity and the intensity of propulsion systems, rational use of vehicles and infrastructure through improved traffic management by using information systems (e.g. ITS, SESAR, ERTMS, SafeSeaNet, RIS), selecting the most favourable transport modality that produces less CO_2 per ton-km or crossed km, optimal SCM management (harmonization of requirements, quality contracts, optimal route planning, making less mistakes in the delivery on transport networks etc.), by increasing the carrying capacity of vehicles, limiting vehicle speed, minimising load-free rides etc.

Several methodologies for estimating CO_2 emissions in intermodal transport were presented in a range of international projects: ARTEMIS (Appendix 4, CO_2 Emission Estimation Methodology for Road Transport, 2008), STREAM (Study on the TRansport Emissions of All Modes), EcoTransIT, GHG Protokol, NTM (Road methodology 2007, NTMWater 2008 i NTM NTMRail 2008 etc.), and all estimates of the emissions of CO_2 were based on standards ISO 14064:2006, ISO 14067:2013 and EN 16258:2011 Methodology For Calculation And Declaration On Energy Consumptions And Ghg Emissions In Transport Services (goods and passengers). In our country, in accordance with requirements defined by The European Monitoring and Evaluation Programme (EMEP) under *The Convention on Long-range Transboundary Air Pollution* and European Environment Agency (EEA), the model COPERT 4 version 10, based on MS Windows [7] is used for calculation of emissions in road transport for heavy goods vehicles (HGV, diesel 16-32 t, Euro I-1991 to Euro IV-2005, diesel >32 t, Euro V-2008 and Euro VI-2012). The project COFRET (Carbon Footprint of Freight Transport) has a particular significance in the calculation of emissions in freight transport and it was used for development of harmonized methodology of emissions calculation.

It is concluded that there is a high correlation between emissions in intermodal modalities depending on the length of road vehicle trip in overal distance and the degree of vehicle load. Further research should be focused on characteristics of diesel fuel in HGV according to the structure of transport chains, with classification of vehicles by types, technologies, logistics providers, etc. This research should be supported by a detailed database about the technical characteristics of technological elements, variations in the values of variable emission factors, which are used in calculation, with special attention paid to comparison of emissions according to technologies, modalities and directions. Such approach would lead to sustainable and safe transport.

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APPLICATION OF HIGH STRENGTH STEELS TO RESPONSIBLE WELDED STRUCTURES ON MOTOR VEHICLES

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UDC:629.021

ABSTRACT: There is constant tendency of engineers to decrease a weight of vehicles and to increase their capacity and mobility. Parallel with development of high strength steels, which starts in 90's years of last century, starts their application in industry for producing motor vehicles with special purpose (construction mechanization vehicles, fire and military vehicles, etc.). With increase of strength of used materials there are conditions for using lower thickness of cross sections what have direct influence to the weight reduction. With respect to that, one of the most used method for producing that kind of structures is welding, in this paper is analysed the weldability of used HS steel and it is proposed the optimal welding technology for welding HSS class S690QL. The assemblies and parts on one military vehicle are made of that steel. The optimal welding technology should preserve good mechanical properties in weld metal, transition zone and in HAZ as the most critical zone of the welded joint.

KEY WORDS: motor vehicles, high strength steel, S690QL, mechanical properties, weldability

PRIMENA ČELIKA POVIŠENE ČVRSTOĆE KOD ZAVARENIH STRUKTURA MOTORNA VOZILA

REZIME: Postoji stalna težnja inženjera da se masa vozila smanji i da se povećaju kapaciteti i mobilnost. Paralelno sa razvojem čelika visoke čvrstoće, čija primena 90-tih je godina prošlog veka, počinje njegova primjena u industriji motornih vozila s posebne namene (građevinska mehanizacija vozila, vatrogasna vozila i vojnih vozila, itd). S povećanjem krutosti korišćenih materijala postavljaju se uslovi za primenu elemenata manjih poprečnih preseka što imaju direktan uticaj na smanjenje težine vozila. S obzirom na to, jedan od najčešće korišćenih metoda za proizvodnju takve konstrukcije je zavarivanje, u ovom radu je analizirana zavarljivost HS čelika i predlažu se optimalne tehnologije zavarivanja za zavarivanje HSS klase S690QL. Sklopovi i delovi na jednom vojnom vozilu su izrađeni od ove vrste čelika. Optimalna tehnologija zavarivanja treba da sačuva dobre mehaničke osobine šava metala, prelazne zone i u HAZ kao najkritičnije zone zavarenog spoja.

KLJUČNE REČI: motorna vozila, čelik visoke čvrstoće, S690QL, mehanička svojstva, zavarivanje

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INTRODUCTION

For considered vehicle several very responsible assemblies are made, and the aim of this paper is to propose welding technology which will ensure obtaining reliable welds. In paper is estimated the weldability of high strength steel on which number of factors have influence. Some of them are chemical composition of the base metal (BM), type of filler material (FM) and welding method, amount of diffusible hydrogen from weld metal into base metal, thickness, type and positions of welds, heat input, type of applied heat treatment, sequence of welding, etc. The optimal welding parameters are based on the results obtained from the mechanical tests performed at room as well as at elevated temperatures, visual inspection of joint and measured hardness and metallographic examinations of some zones of welded joints, and special with regards on results obtained for impact toughness.

WELDABILITY OF THE BASE METAL

The S690QL class steel belongs into a group of special thermo-mechanical (TMO) low alloyed steels. The producer provides declaration of chemical composition on delivery [1-3]. The carbon content is limited to 0.20%, so the steel should possess good weldability. Microalloying elements cause improvement of mechanical properties of those steels~~;~~. Especially effective are niobium and boron, which are deoxidizing the steels and cause the fragmentation of metal grains. There are three different modifications of the S690 steels: S690Q, S690QL and S690L1, which only differ with regard to guaranteed impact toughness: S690Q – KV = 27 *J* at -20°C; S690QL – KV = 69 *J* at -40°C, S690QL1 – KV = 27 *J* at -60°C [2, 3]. Mass application of the high strength steel of this class occurred due to exceptional mechanical characteristics (tensile strength and yield stress) as well as favorable impact toughness. Basic data provided by the steel manufacturer can be found in corresponding references [1-3, 5].

It should be emphasized that the commercial mark of this steel is WELDOX 700 (SSAB Sweden).

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It should be emphasized that application of these steels is limited for the working conditions when the temperature does not exceed 580°C, since above this limit the mechanical properties become worse [2, 3].

Weldability of high strength steels can be determined by calculating according to the equivalent carbon (CE) and proneness of the steel towards the cold cracks. Depending on formula for calculating the CE and thickness of the welded parts, the limiting values of CE vary, Table 1.

Table 1 Values of the chemically equivalent carbon [1, 2, 4]

Mark	Thickness, *mm*	chemically equivalent carbon, CE, %	
		$CEV = C + \frac{Mn}{6} + \frac{Cr+Mo+V}{5} + \frac{Ni+Cu}{15}, \%$	$CET = C + \frac{Mn+Mo}{10} + \frac{Cr+Cu}{20} + \frac{Ni}{40}, \%$
S690QL	20	0.43 - 0.55	0.29 - 0.36
	30	0.46 - 0.55	0.31 - 0.36
	60	0.57 - 0.55	0.35 - 0.36

Based on results from Table 1, steel manufacturers recommend the preheating temperature that enables hydrogen diffusion from the joint zone and extending of the HAZ cooling time, for the purpose of obtaining the softer structure [1, 2, 5]. Besides the chemically equivalent carbon, one can theoretically estimate risk of appearance of cold, hot, lamellar and annealing cracks [1, 5]. According to formulas of different authors, the considered steel is highly prone to appearance of cold cracks. Risk of hot cracks is not so prominent, but risk of lamellar and annealing cracks is [1, 2]. Thus, the manner and procedure of welding should be so chosen that the reliable welded joint can be realized, which during exploitation would not be prone to appearance of any cracks that can cause brittle fracture of the welded structure.

For the S690QL steel the preheating temperature is recommended to be within range 150 – 200°C, and maximum interpass temperature should be $T_{interpass}$ = 250°C in order to prevent porosity in the weld metal caused by air turbulence, as well as worsening of the steel's mechanical properties realized by the thermo-mechanical treatment of steel

SELECTION OF THE OPTIMAL WELDING TECHNOLOGY

Based on manufacturer's recommendation and experience of other users, it was decided, for responsible joints, to apply filler metals of austenitic structure of the smaller strength than the base metal for the root weld layers, while for the rest of the weld layers (filling and cover ones) to apply the filler metals of the strength similar to that of the BM.

Thus, the proposed welding technology assumes deposition of the root welds by the MMAW electrode E 18 8 Mn B 22 – diameter Ø 3.25 mm; deposition of the filling layers is done by the GMAW electrode wire Mn3Ni1CrMo – diameter Ø1.2 mm (Figure 1). For deposition of the covering layers the GMAW procedure was selected due to better productivity with respect to the MMAW [1, 2]. The welded plate dimensions were 400×200×15 mm. After deposition of the root pass 1 it was subsequently partially grooved by the graphite electrode arc-air procedure and the new root pass was deposited in the complete argon protected atmosphere by the austenite electrode 8.

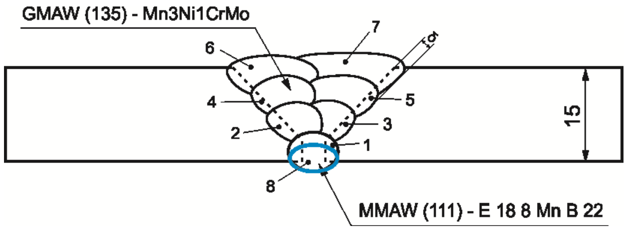


Figure 1 Combined MMAW/GMAW deposition of weld layers

Table 2 Chemical composition and mechanical properties of the filler metals [1, 2].

Electrode type	Chemical composition, %						Mechanical properties of weld metal			
	C	Si	Mn	Cr	Ni	Mo	R_m, MPa	R_p, MPa	A_5, %	KV, J
E 18 8 Mn B 22 *	0.12	0.8	7	19	9	-	590 - 690	> 350	> 40	>80 (+20ºC)
Mn3Ni1CrMo **	0.6	0.6	1.7	0.25	1.5	0.5	770 - 940	> 690	> 17	>47 (-40ºC)

Table 3 Welding parameters [1]

Parameters	I, A	U, V	v_z, cm/s	v_t, m/min	q_l, J/cm	δ, mm	Shielding gas	The flow of shielding gas, l/min
Root weld (MMAW)	120	24.5	0.2	-	12000	1.7	-	-
Filler welds (GMAW)	240	25	0.35	8	14885	2	Ar/18% CO_2	14

EXPERIMENATAL INVESTIGATION OF THE WELDED JOINTS

Tensile testing

Experimental tests of the S690QL steel specimen consisted of both mechanical tensile test and impact toughness test. The tensile test was performed on specimens prepared from the 15 *mm* thick plate; 4 specimens were prepared for the Base Metal (BM) tests and 4 specimens were aimed for testing the welded joint material (Figures 2 and 3) [5, 6]. Tests were performed according to standard SRPS EN 10002-1 [8].

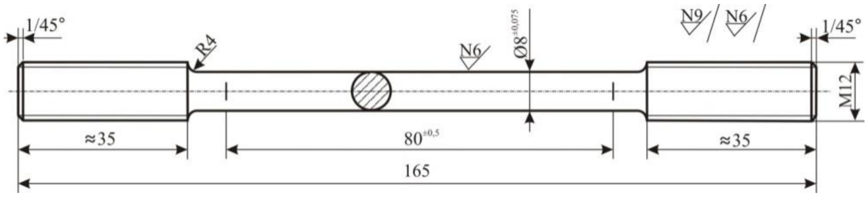
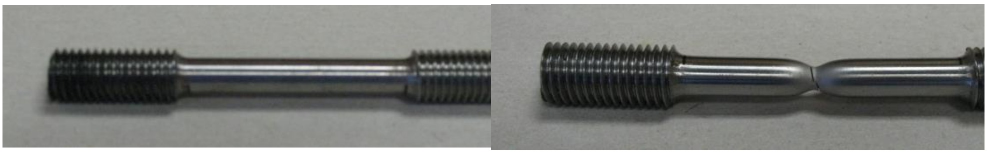


Figure 2 *Specimen for tensile testing*



Specimen before testing - BM *Specimen afrer testing - BM*

Specimen before testing - WJ *Specimen after testing - WJ*

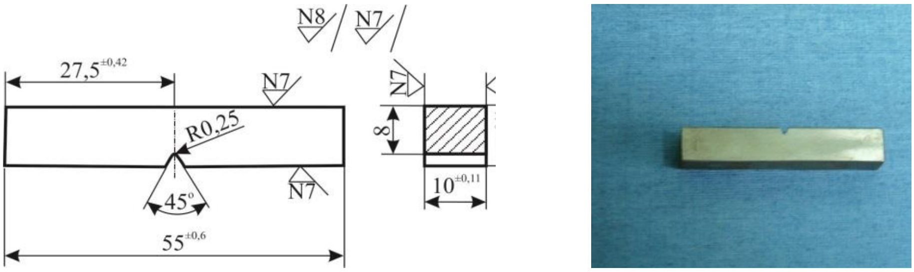
Figure 3 Specimen appearance before and after testing

Table 4 Experimental results of the tensile test [1-5]

Sample No	L_0, *mm*	S_0, *mm*^{2}	$R_{p0.2}$, *MPa*	R_m, *MPa*	$A_{11.3}$, %
Base material – S690QL					
1	89.28	50.27	781.94	797.81	14.19
2	89.28	50.27	809.40	839.92	11.30
3	88.42	50.01	800.41	835.52	9.98
4	88.29	50.27	811.95	842.45	10.92
Welded joint – plate 1 (REL/MAG)					
1	89.28	50.27	809	840	11.30
2	88.42	50.27	764	831	9.77
3	86.96	49.39	760	812	5.49
4	86.96	49.39	740	804	5.38

IMPACT TOUGHNESS TESTING

Specimens were prepared for the impact toughness tests (Figure 4), according to the similar procedure as for the tensile test specimens; six aimed for testing the BM and three for testing the impact toughness of the weld metal, root metal and HAZ. Tests were done on the Charpy pendulum, both at room and lower temperatures, according to standard EN 10045-1 [9]. Table 5 gives results of the impact tests.



a) b)

Figure 4 Appearance of specimen for impact toughness testing: drawing (a) and photo of the specimen (b)

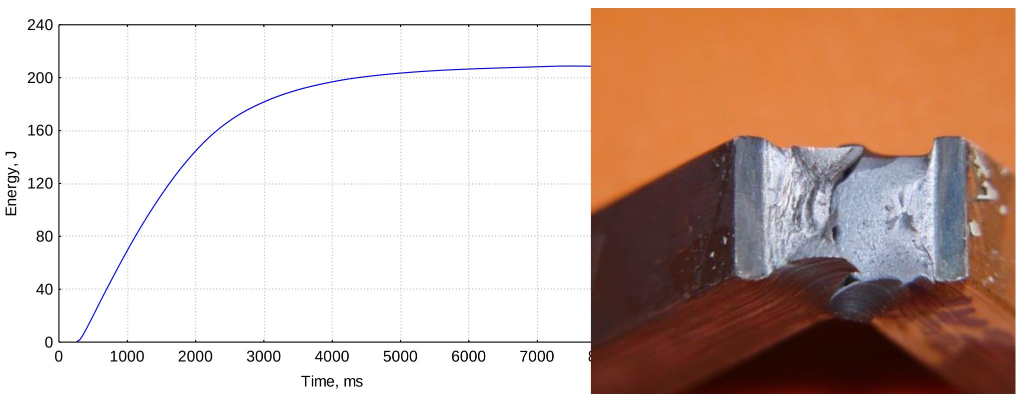
Table 5 Impact energy absorbed at room and lower temperatures [2, 3]

Steel mark	Temperature, ºC	Impact energy, J			
		Base material	Weld metal	Root weld metal	HAZ
S690QL	+20	235.2; 222.4; 234.7	24.2; 45.5; 34.7	85.8; 89.5; 54.1	189.2; 172.8; 209.7*
	-40	219.6; 19.8; 206.1	-	-	-

*Marked value is shown on diagram in Figure 5.

In figure 5a is shown the impact energy vs. time diagram and in figure 5b fracture surfaces appearance. Figure 5 is related to one representative specimen.

Besides the presented results, in favor of selected welding technology, were deciding also the plastic properties of the executed welded joints, estimated according to the share of the plastic fracture in the total fracture surface (Figure 2b), which was, in all the zones of the executed joint, within range 92.41 – 99.81%. That represents exceptional results from the aspect of the welded joints plasticity.



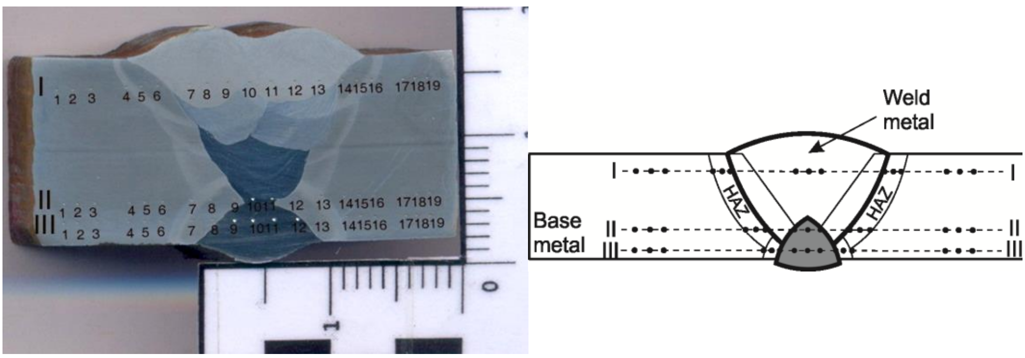
a) *b)*

Figure 5 Representative specimen – HAZ – sample 3: a) impact energy vs. time diagram; b) fracture surfaces appearance

HARDNESS MEASUREMENT

For hardness measurements of individual welded joints' zones, special metallographic samples were prepared, 2 per each plate (Figure 6), on which the microstructure of characteristic zones was red-off as well. Hardness was measured by the Vickers method, indentation force was 100 N, according to standard SRPS EN 1043-1:2007 [2-4].

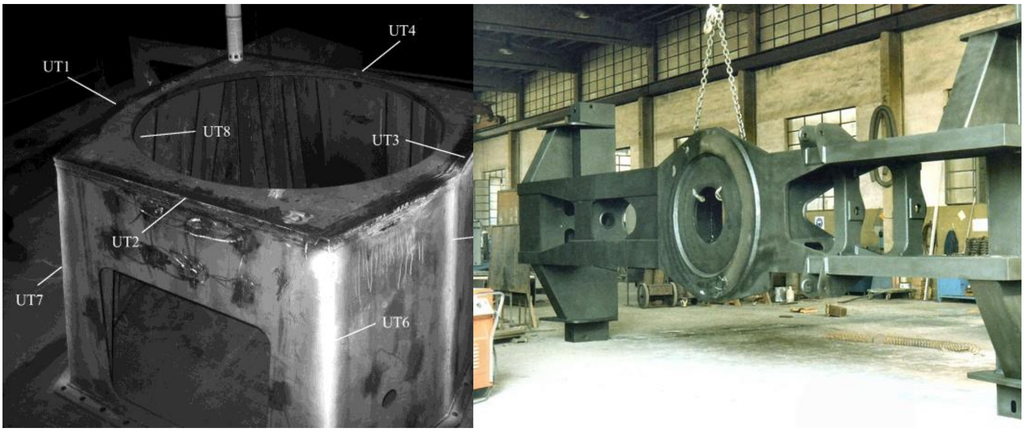
Hardness was measured of the base metal (BM), in the HAZ and weld metal (WM) along the straight lines perpendicular to the welded joint, Figure 16. Along a single line hardness was to be measured at least at three points for each of the characteristic zones, WM, HAZ (both sides) and BM (both sides). The first indent in HAZ ought to be as close as possible to the melting zone (border WM – HAZ). This also applied for the root. Obtained results show slight deviations of values for the homogeneous zones (BM, WM), but those deviations are somewhat larger for the HAZ, as well as for the melting zone.



a) *b)*

Figure 6 Metallographic sample for hardness measurement and microstructure estimate (a) and hardness measurement directions (b)

In Figure 7 are shown some examples of welded construction made according to proposed welding technology and also some details from production process.



a) *b)*

Figure 7 Details from motor vehicle production process: a) ultrasonic defectoscopy and b) rear axle of a truck

CONCLUSION

After detail analysis of the most important properties of base material and estimation of its weldability, choosing the optimal combination of filler materials, welding method and technology, and extensive model investigation, the optimal welding technology was established which has been applied on real construction. That welded construction is than installed on vehicle and was subjected to rigorous tests and it fulfilled all the requirements necessary for field work, where it turned as a very reliable one.

In establishing the optimal technology experimental results of the sample welds tensile and impact tests were used as indicators. During the tensile tests samples' fractures occurred outside the welded joint zone, what means that strength of the welded joint was higher than the base metal strength. Impact toughness was within limits required by the appropriate standards, especially in the HAZ as the most critical zone of the joint. The recorded metallographic structure revealed that appearance of the brittle martensitic structure of the welded metal was avoided.

Due to all quoted in paper, the required properties of welded joint, similar to properties of the base material, it is recommended to use this particular steel for production very responsible welded constructions.

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