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EU ENERGY AND PROPULSION TRANSITIONS IN THE MOBILITY SECTOR OF GERMANY – A REALIZABLE STRATEGY OR EVEN RATHER IDEOLOGICAL ASTRAY?

*Ralph Pütz[1]**

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RESEARCH ARTICLE

ABSTRACT: The EU policy – with the intended exclusion of combustion engines and consequently the dictate to electric mobility in the on-road and off-road sectors on the one hand, and on the other hand with the exclusive focus of the Euro legal limit stages solely on driving operation (tank-to-wheel, TtW) while neglecting the influences of all relevant processes of vehicle production (cradle-to-gate, CtG), energy supply (well-to-tank, WtT) and recycling/disposal (end-of-life, EoL) – leads to a misguiding distortion of the ecological facts while at the same time ignoring the constraints of a free market economy. The necessity of a "propulsion transition" is questionable, and the feasibility of an "energy transition" with an exclusive focus on German domestic renewable energies and thus energy autarky seems to be underestimated by far. With comprehensive "system thinking" (vehicle load and range, charging times and charging spaces, energy generation, distribution and refuelling infrastructure as well as required power grids) the intended energy transition for Germany is more than questionable.

KEY WORDS: *EU energy transition, EU propulsion transition, New Green Deal, holistic ecological balancing, technology-neutrality*

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EU TRANZICIJE ENERGIJE I POGONA U SEKTORU MOBILNOSTI NEMAČKE – OSTVARIVA STRATEGIJA ILI ČAK IDEOLOŠKA ZABLUDA?

REZIME: Politika EU – sa nameravanim isključivanjem motora sa unutrašnjim sagorevanjem i posledično diktatom električne mobilnosti u drumskom i terenskom sektoru s jedne strane, a s druge strane, sa isključivim fokusom evropskih zakonskih ograničenja isključivo na vožnju („od rezervoara do točka“), dok zanemaruje uticaje svih relevantnih procesa proizvodnje vozila („od kolevke do kapije“), snabdevanja energijom („od izvora do rezervoara“) i reciklaže/odlaganja („kraj životnog veka“) – dovodi do pogrešnog iskrivljavanja ekoloških činjenica, a istovremeno ignoriše ograničenja slobodne tržišne ekonomije. Neophodnost za „pogonsku tranziciju“ je upitna, a izvodljivost „energetske tranzicije“ sa isključivim fokusom na nemačke domaće obnovljive izvore energije i samim tim energetsku autarkiju, čini se da je daleko potcenjena. Sa sveobuhvatnim „sistemskim razmišljanjem“ (opterećenje i domet vozila, vremena punjenja i prostori za punjenje, infrastruktura za proizvodnju, distribuciju i punjenje energije, kao i potrebne elektroenergetske mreže), planirana energetska tranzicija za Nemačku je više nego upitna.

KLJUČNE REČI: *Energetska tranzicija EU, pogonska tranzicija EU, Novi zeleni plan, holističko ekološko uravnoteženje, tehnološka neutralnost*

EU ENERGY AND PROPULSION TRANSITIONS IN THE MOBILITY SECTOR OF GERMANY – A REALIZABLE STRATEGY OR EVEN RATHER IDEOLOGICAL ASTRAY?

Ralph Pütz

INTRODUCTION: POLITICAL REQUIREMENTS IN THE EU WITH THE FOCUS ON THE TRANSPORT SECTOR ESP. COMMERCIAL VEHICLES

The political and social goal is to limit global warming to 1.5°C. On 14 July 2021, the European Union (EU) adopted the "European Green Deal" to reduce global emissions (greenhouse gas emissions, GHG) in the EU by 55% by 2030 compared to 1990 and to then no longer cause any net emissions of greenhouse gases or CO2 equivalents by 2050 [1]. As early as 2045, no more greenhouse gas emissions may be emitted in Germany. Every sector in the EU - thus also the transport sector - must achieve these ambitious goals, for which EU policy provides coupled political measures as so-called "transitions", which, in addition to the necessary "energy transition" to the exclusive use of renewable energies, also consider a "mobility transition" (traffic avoidance, shift of individual transport to public transport) and a "propulsion transition" (accelerated use of electromobility) to be necessary. However, it is questionable whether the absolutely indispensable energy transition in connection with an intended energy self-sufficiency with domestic energy can be realised at all - especially in Germany. In addition, the sense of a "propulsion transition" with the exclusion of locally highly clean combustion engines while using regenerative primary energies must be questioned in a holistic balance. So in the recent European elections, for example, the European People's Party (EPP/EVP) propagated the withdrawal of the decided exclusion of combustion engines as an election promise.

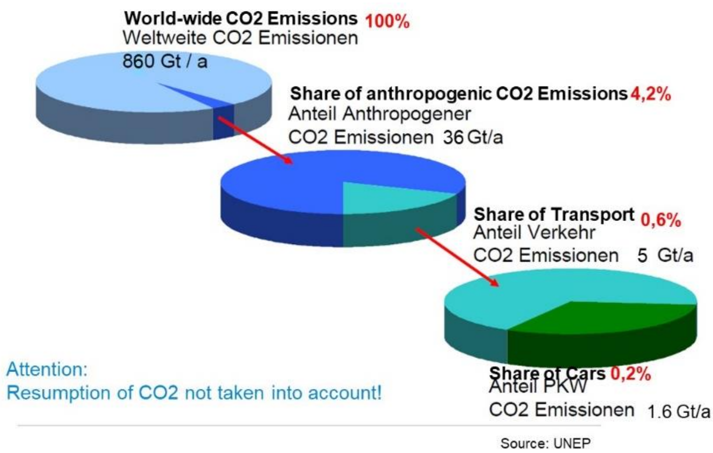


Figure 1 World-wide annual CO2 emissions 2020 – overall and anthropogenic share

Currently, global emissions from all transport modes (including ships and aircraft) in the EU account for around 25% of EU anthropogenic greenhouse gas emissions, of which road transport is responsible for four-fifths (20%) overall, which, however, corresponds to just under 2.4% of global anthropogenic CO2 emissions (road transport: 1.9%). The commercial vehicle sector (trucks including coaches and buses) emits 6% of CO2 anthropogenic emissions in the EU, which is only a neglecting 0.4% of global anthropogenic CO2 emissions. Taking into account that world-wide anthropogenic CO2 emissions only amount to 4,2% of the overall world-wide CO2 emissions (see Fig. 1), the EU road transport emissions amount to only 0,08% (sic!) of the overall world-wide CO2 emissions. If every CO2 molecule regardless of its origin – anthropogenic or natural – has the same influence, this should be classified accordingly and be a reason to consider. On the other hand, the reports of the International Panel on Climate Change (IPCC) leave no doubt about the urgent need to drastically reduce greenhouse gas emissions [2]. However, in order to achieve the required climate neutrality in the EU by 2050, enormous demands will be placed on the transport sector without, in all likelihood, having a measurable effect on climate change. In the fight against climate change, therefore, the EU's actual, limited "leverage arm" must be taken into account in its actions. The EU in total is responsible for about 9.5% of anthropogenic global CO2 emissions (or 0,4% (sic!) of the overall world-wide CO2 emissions), see Fig. 2. The largest CO2 emitter within the EU, Germany, is responsible for a total anthropogenic global contribution of 1.85% (0,078% of the overall world-wide CO2 emissions), followed by Italy with 0.93%. Against this background, the EU's pioneering role in combating climate change alone seems futile if the four main emitters - China, the USA, India and Russia, which together account for around 55.7% of anthropogenic CO2 emissions - do not live up to their responsibility.

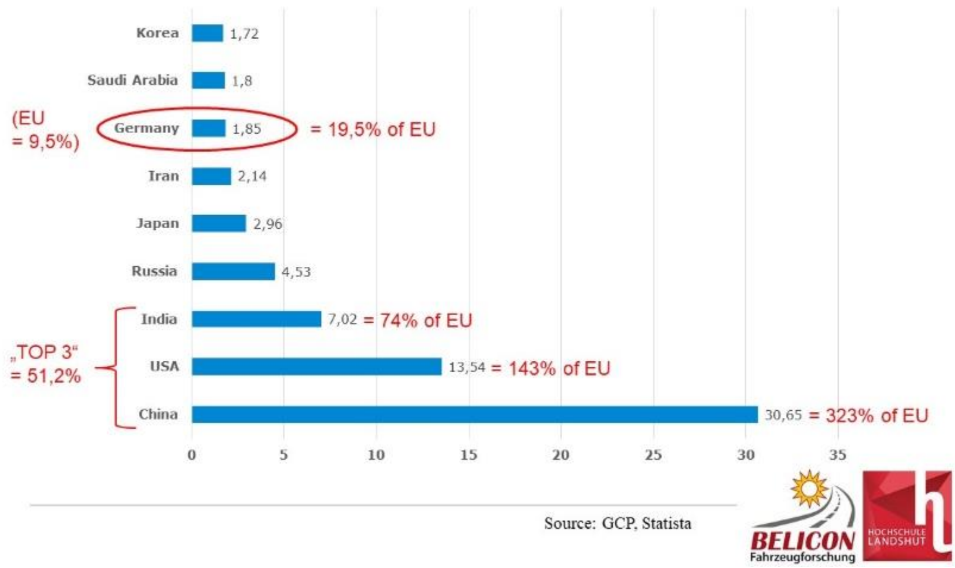


Figure 2 World´s largest anthropogenic CO2 emitters (share in %) and limited lever of EU measures

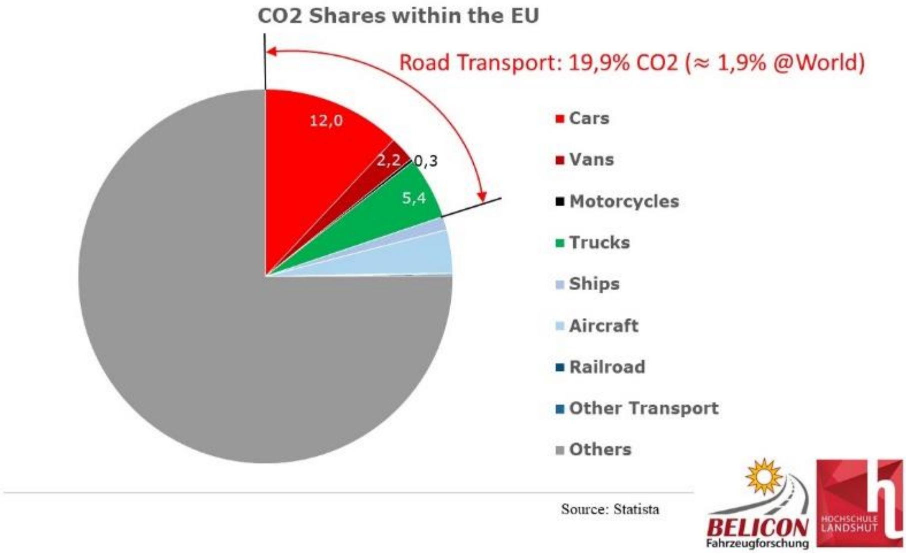


Figure 3 Contribution of the EU Transport Sector to EU global anthropogenic emissions in 2022

In the public transport (PT) bus sector, the so-called "EU Clean Vehicles Directive" (implemented in Germany since 2.8.2021) has prescribed fixed quotas for the procurement of new public buses with "clean" and "emission-free" drives, see Fig. 4 [3]. The EU policy focus in this directive only on local tailpipe emissions inadmissibly completely ignores the provision of raw materials and energies for vehicle production including drive systems (cradle-to-gate, CtG), the generation and distribution of energy in the fuel supply chain (well-to-tank, WtT), and subsequent recycling and disposal (end-of-life, EoL), but all these subsystems are highly relevant for a robust holistic ecological (and economic) balance.

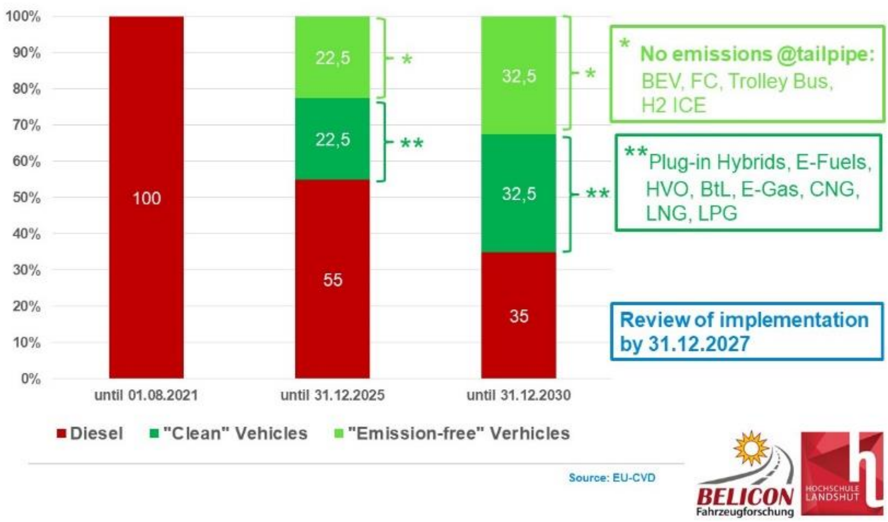


Figure 4 EU „Clean Vehicles Directive“: Fixed quotas for procurement of new PT buses

While the targeted elimination of pollutants such as particles (particulate mass PM and particle numbers PN) and nitrogen oxides (NO, NO2) from the tailpipe makes perfect sense, the elimination of CO2 from the tailpipe is completely irrelevant, as GHG emissions have a "global" effect and it is irrelevant whether the emissions of CO2 occur in the upstream chain (WtT), in driving cycles (TtW), in vehicle production (CtG) or recycling/disposal (EoL).

1 ANALYSIS OF FIELDS OF ACTION FOR LOCAL EMISSIONS

As far as particle emissions during driving are concerned, Fig. 5 shows for a Euro VI PT diesel bus with DPF that the intake air (in this case polluted ambient air) contains significantly more particles than the exhaust gas - which hence represents a partial air cleaning that is not possible with the electromobility options (battery BEV or fuel cell FC drives).

Fig. 6 shows the NOx emissions measured during the real operation of articulated buses in urban traffic with typical SORT 2 pattern ("stop-and-go operation") for a route with demanding topography. The emission behaviour of a Euro V articulated bus is shown on the left and that of a Euro VI articulated bus on the right. In the case of the Euro VI articulated bus, nitrogen oxide emissions are almost at ambient air level after only 10 minutes due to the effective SCR exhaust gas aftertreatment. Incidentally, the same also applies to diesel engines of Non-Road Mobile Machinery (NRMM) stages 4f and 5 in tractors in heavy agricultural use.

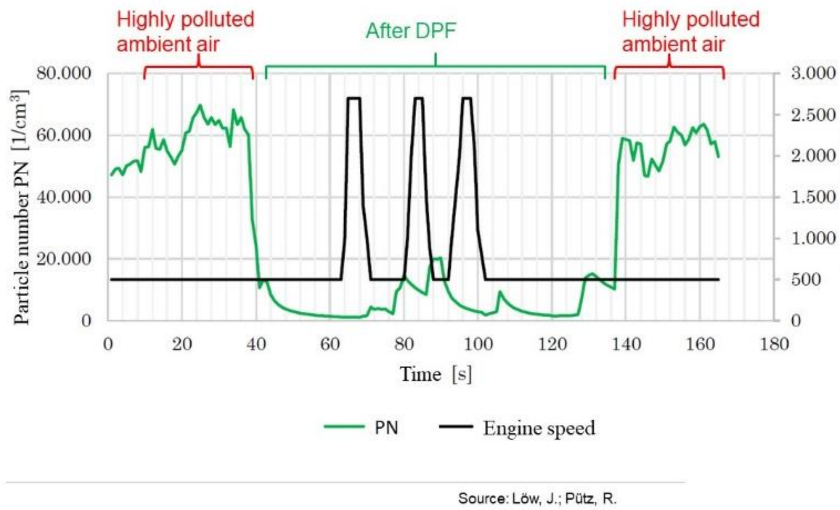


Figure 5 Particle emissions @tailpipe of a Euro VI bus with DPF: cleaning the ambiant air

In view of the near-zero emission level of internal combustion engines already achieved today, it is doubtful whether the Euro VII (or 7 in cars) limit values envisaged after 2025 can be proven in practice to improve air quality at all.

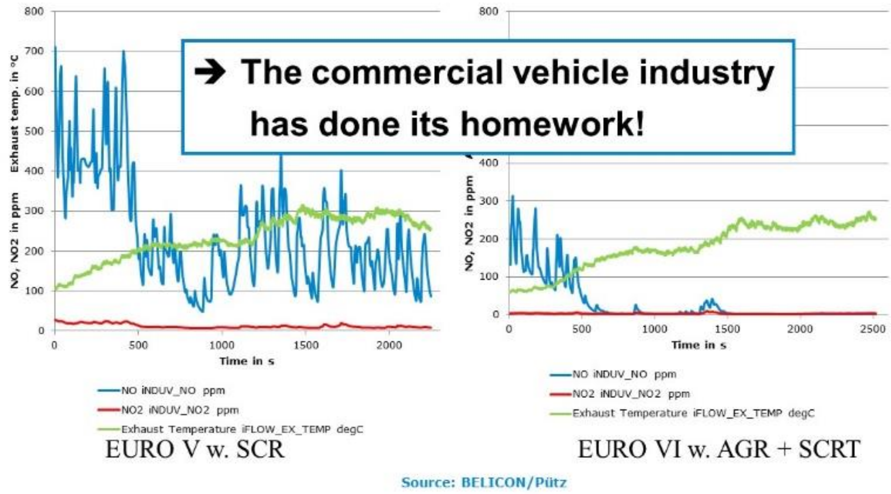


Figure 6 Comparison of nitrogen oxide emissions @tailpipe of Euro V and Euro VI articulated buses (SORT 2, demanding topography)

2 ANALYSIS OF THE FIELDS OF ACTION FOR GLOBAL EMISSIONS, ENERGY CONSUMPTION AND RANGE

Today, there is an undisputed need for action in the forced use of renewable energy sources, not only to significantly reduce globally effective emissions, but also to conserve fossil resources. In order to analyse the energy demand for the German „energy transition" in the transport sector, a commercial vehicle-specific scenario is examined below, in which initially only the energy demand for handling the heavy-duty trucks´s toll kilometres on German motorways with currently 359 motorway filling stations is considered. 500,000 domestic trucks and 360,000 foreign trucks, each with an average mileage of 93,000 km/year, travel a total of 41.6 billion toll kilometres on German motorways with an average energy consumption of 30 l Diesel fuel/100 km (2.97 kWh/km). This corresponds to a total of 12.49 billion litres of Diesel fuel per year for the completion of toll kilometres. It was assumed that, due to the company filling stations available at the haulage companies depots, only about half of the fuel (52%) is filled up at the motorway filling stations. The following promising drive/energy system options for heavy-duty trucks were to be investigated:

- •EURO-VI-Diesel with E-Fuel (Power-to-Liquids; PtL),
- •EURO-VI-CNG with E-Gas (Power-to-Gases; PtG),
- •Fuel Cell (FC or BZ) with „green" compressed gaseous hydrogen (CG H2) from wind power,
- •Battery electric vehicle (BEV) with wind power.

In each case, a range of 1,500 km with one tank filling/ charging process was assumed, as this represents a minimum requirement from the operator's point of view. A lower range with more frequent refuelling/charging processes would theoretically be possible, but from the operator's point of view it appears to be an unrelated restriction. The "In-motion charger" truck option (trolley truck with overhead catenary or induction rails and small batteries on board) is not considered here due to the lower operational flexibility and probably greatly reduced interoperability in international freight transport. In public transport, however, it could be sensibly implemented as a trolley hybrid bus due to the fixed bus lines [6].

While the lower energy consumption in driving mode (TtW) speaks in favour of the use of BEV and FC trucks, the BEV option is already ruled out de facto when the required volumetric and gravimetric tank capacity is taken into account, as there are significant losses in payload and load volume, see Fig. 7. The influence on the load density ρN,g in t/m^3 and the payload factors GN' in t/m and GN" in t/m^2 are shown in Fig. 8. It can be seen that to cope with the load that can be transported in diesel vehicles with E-fuel, BEV trucks would require an additional vehicle demand of 17.2%, with an equivalent increase in energy demand and also in driver cost, the latter which account for about 35% of km costs in long-distance transport (50% in short-distance transport). When considering energy consumption and load planning factors at the same time, only the two options "EURO VI diesel + E-fuel" and "FC + CG H2" appear to be target-oriented for heavy-duty long-haul trucks.

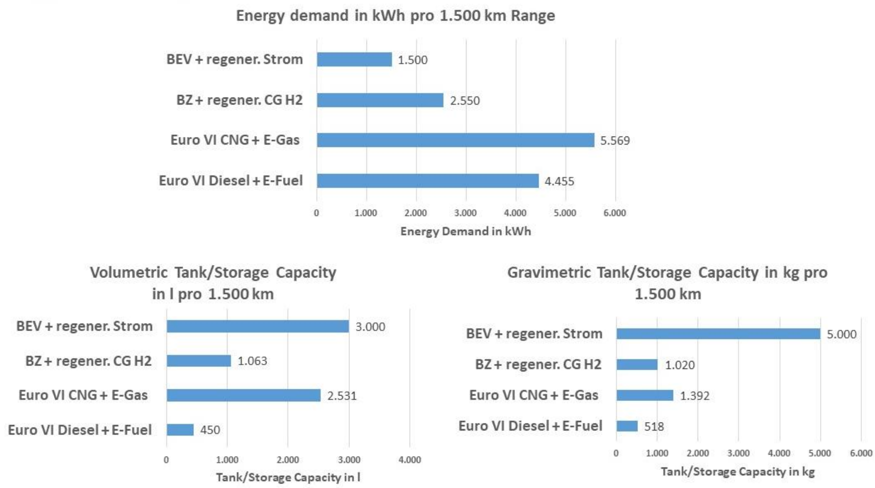
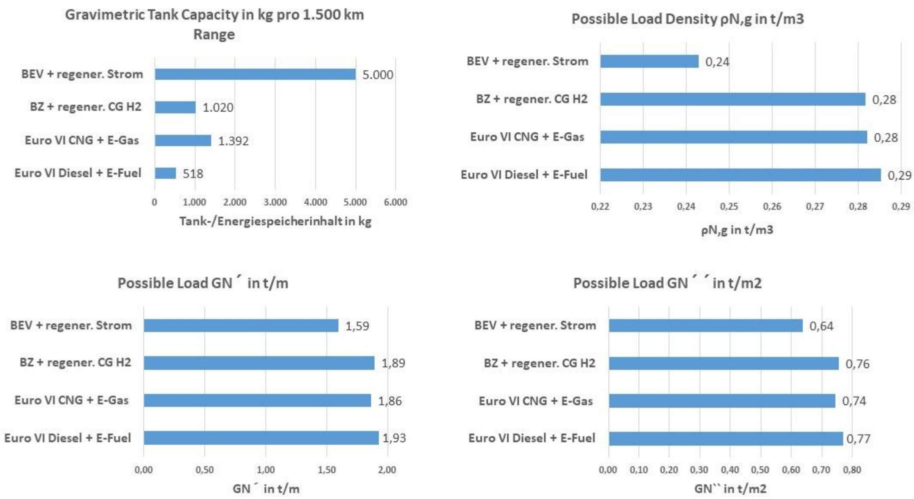


Figure 7 Energy requirements and tank contents for usual truck ranges of 1,500 km



BZ = FC

Figure 8 Impact of gravimetric energy content on truck payload

The refuelling times for the investigated options are shown in Fig. 9 for a refuelling or driving distance of 1,500 km on the one hand and on the other hand for the daily refuelling processes at a motorway filling station with around 50 truck refuellings per day. A fast charge with 250 kW was assumed on average for the BEV option because the power dissipation in the form of process heat with 500 kW or even MW charging is excessive! While the daily net refuelling times of all trucks at an average motorway filling station can easily be completed in 24 hours for the target options "EURO VI diesel + E-fuel" and "FC + CG H2", BEV trucks would require more than two weeks (17.3 days) - 48 times as long as the "EURO VI diesel + E-fuel" option - considering the additional vehicles mentioned, which would only ensure truck operation with an equivalent, hardly feasible increase in space at filling stations.

Due to the politically intended energy self-sufficiency with domestic renewable energy from preferably wind power and photovoltaics, the feasibility of this toll transport scenario within the "energy transition" in the transport sector was analysed on the basis of "10 GWh wind power plants" (also referred to as "Growian" = Große Windkraftanlage; large wind power plant) with an average yield of 10 GWh/a in Germany. To cover the energy demand for the trucks toll kilometres on German motorways, 2,023 of the "10 GWh wind turbines" would be required for the BEV truck option, see Fig. 10. For this purpose, the average number of full-load hours (VLS) of wind power of 1,500 VLS/a in Germany was taken into account. For the operation of the BEV truck option, however, the long-haul transport companies would have to accept highly significant, non-practical restrictions in the up to now flexible operation. For the practical option "FC + CG H2", however, 4,543 of the "10-GWh wind turbines" would be required in Germany, and for the likewise practical option "EURO-VI Diesel + E-fuel" even 9,202 of "10-GWh wind turbines" in Germany, which would make a regular production of E-fuels in Germany absurd. However, if the required volumes of E-fuels were produced in Patagonia, for example, with an average of 5,200 VLS/a, only 2,654 "10-GWh wind turbines" would be needed at typical German wind speeds. On the other hand, it must be taken into account that the wind turbines in sparsely populated Patagonia could well be larger and, due to significantly higher wind speeds, even significantly fewer than the number of wind turbines required for BEV trucks in Germany would be needed - because the kinetic energy flux density is proportional to the third power of the wind speed (v3) [7]. Consequently, a wind speed of just 1.5 times higher than in Germany increases the extractable wind power by almost 3.4 times, which means that comparatively 781 Growians (about 40 per cent of the number of wind turbines needed for BEV trucks) would only be required for E-fuel production in Patagonia. Comparably favourable conditions would also result from the use of photovoltaics for E-fuel production in North Africa. However, despite the highest energy efficiency in driving operation, the BEV truck option for the truck toll kilometre scenario would mean that statistically an average of 5.6 "Growian" wind turbines would be required per motorway filling station in Germany, and even 12.7 for the "FC + CG H2" option. The preconditions for this would be either on-site energy production at the filling stations (if sufficient wind power is available on site) or a sufficient electricity distribution network, which, however, could only be established in the long term.

In order to classify the above-mentioned regenerative energy demand for covering only the trucks toll kilometres, the previous and projected further development of wind energy development in Germany must be considered. Under the conservative assumption that the progression of the previous development of renewable energy expansion is continued, the scenario shown in Fig. 11 results. It was assumed that the energy yield from wind turbines (on- and off-shore wind energy) doubles from 2020 to 2030. It can be seen that with this development, already today's net electricity consumption (488 TWh in 2020 and 484 TWh in 2022) can only be covered by renewable energies in 2035 and consequently there is until then still no capacity available to provide the energy demand for E-mobility – not even only for trucks toll kilometers!

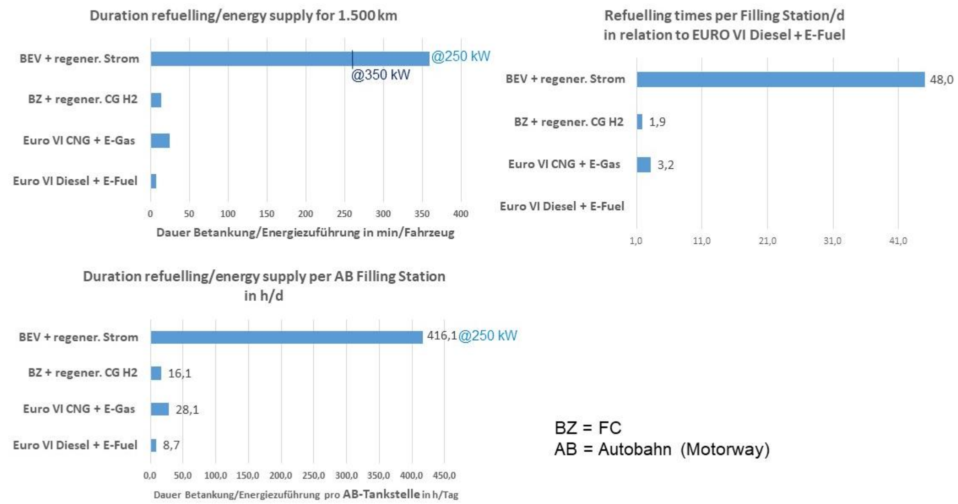


Figure 9 Comparison of heavy-duty trucks refuelling times

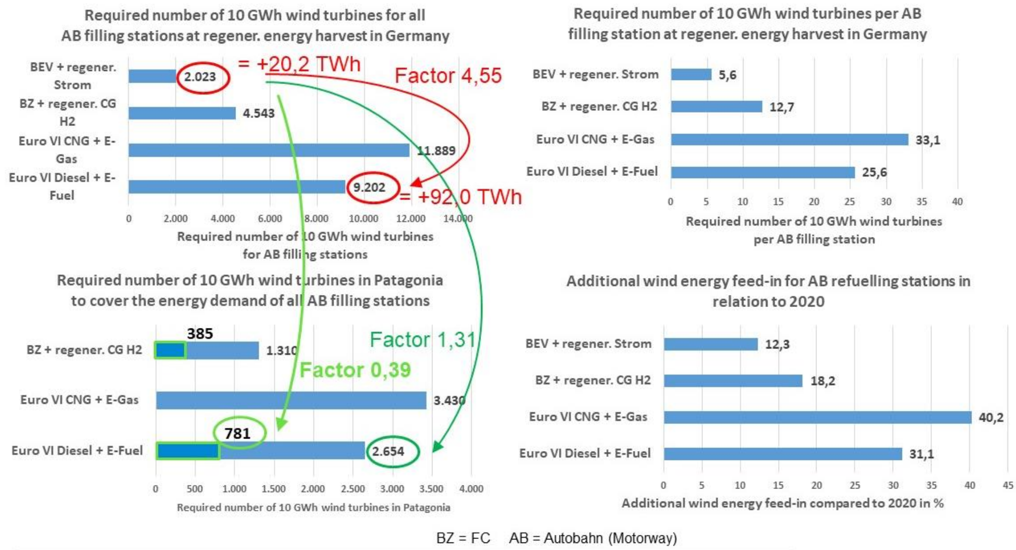


Figure 10 Required additional number of „10 GWh wind turbines" in Germany for the coverage of the trucks toll kilometres @motorway filling stations

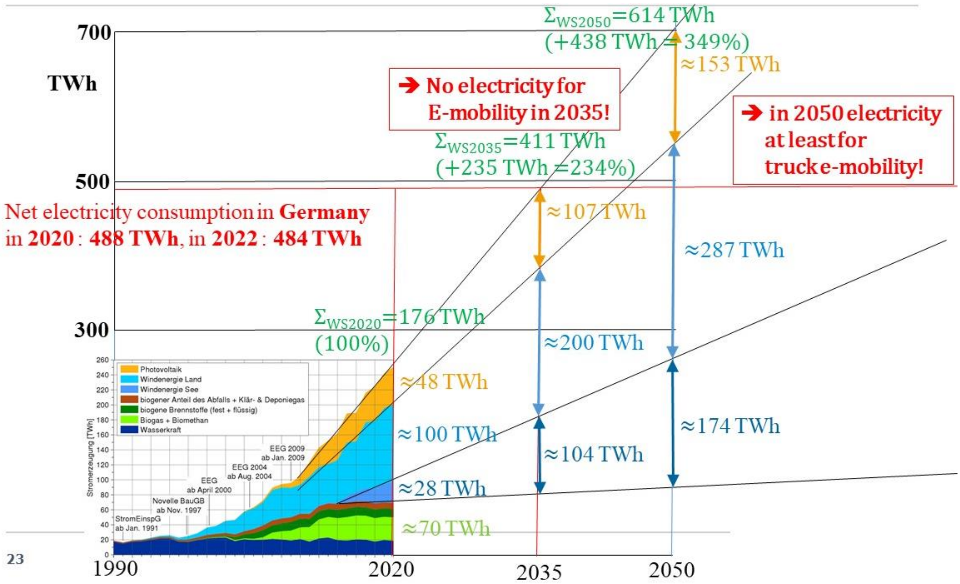


Figure 11 Development of renewable energies in Germany (conventional scenario)

Within the foreseeable future an insufficiently developed electricity distribution network and assuming that renewable energy would be sufficiently available on site, the BEV truck option would require an average area of 2.2 ha per motorway filling station, and even 5.1 ha for the "FC + CG H2" option, the latter which is significantly more practicable from an operational point of view. The total additional land requirement of 809 ha for the „BEV with German wind power“ option and 1,817 ha for the "FC + CG H2" option would be euphemistic and completely insignificant for Germany (see Fig. 12). Compared to "BEV with German wind power", the option "EURO VI Diesel + E-fuel" would only require a slightly larger area of 1,062 ha in Patagonia if E-fuel is generated in Patagonia, and even significantly less if higher wind speeds are taken into account – e.g. only 312 ha at 1.5 times the wind speed. The advantage of the latter option would also be the unchanged use of the existing infrastructure for energy distribution and refuelling in Germany and Europe, while the planned expansion of the German electricity grid according to the German Grid Development Plan for Electricity and the required construction of fast charging stations for trucks involve high, presumably by far underestimated cost dimensions.

As a further scenario for analysing the energy demand for the „energy transition“ in the transport sector, a passenger car-specific scenario was examined, in which only the energy demand for handling the entire domestic passenger car traffic with 45.8 million vehicles (31 million petrol and 14.8 million diesel passenger cars) at 14,460 filling stations throughout Germany was considered. It was assumed that 90% of the driving of domestic cars takes place in Germany. An average vehicle of the lower middle class from the German passenger car fleet was chosen as the average value. Furthermore, a range of 700 km was assumed, which is common for passenger cars today.

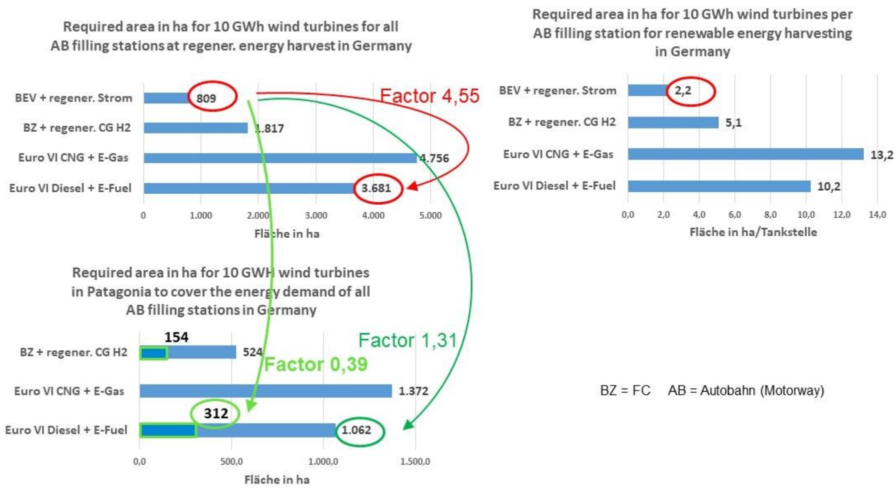


Figure 12 Required areas for "10 GWh wind Turbines" in Germany for heavy-duty trucks on German motorways

For the refuelling times for one „tank filling“, a fast charge with 50 kW and with 250 kW was alternatively investigated for the BEV car option. While the net "refuelling time" of a BEV car at 250 kW charging power takes only a little more than half an hour (34 min), taking into account the charging losses, the usual charging with 50 kW today would require almost three hours (168 min). Not only the latter would lead to an almost unrealisable increase in filling station space with today's battery technology, which is obviously also an underestimated item. To cover the required energy demand with domestic wind energy, 21,707 "10 GWh wind turbines" would be needed for the "BEV car" option, and 48,972 for the "FC car + CG H2" option. The required areas for wind turbines in Germany would - purely superficially considered - not be a problem either for the "BEV car" option with 12,161 football pitches or 0.2% of the grassland in Germany or for the "FC car + CG H2" option with 27,436 football pitches, if a sufficiently stable electricity distribution grid were available. The equivalent production of E-fuels for the internal combustion engine option would require only an area of 8,205 football pitches in Patagonia, due to the significantly higher wind full load hours (VLS) and assuming a 1.5-fold average wind speed compared to Germany. This would mean only 0.005% of the area of Patagonia. It follows that the conventional scenario shown in Fig. 11 for the further development of domestic renewable energy production in Germany is far from sufficient for the "BEV passenger car" and "FC passenger car + CG H2" options; instead, a forced scenario would be required as shown in Fig. 13. It further follows that for the exclusive regenerative energy supply of domestic cars, until the year 2050 around six "10 GWh wind turbines" for the "BEV cars" option and as many as ten of these wind turbines for the "FC cars + CG H2" option would have to be newly erected in Germany at every working day - with immediate effect! Considering the previously analysed trucks toll kilometres scenario with renewable energy, for which 52% of the energy consumption of heavy-duty trucks on German motorways was assumed, and additionally the energy supplied at the companies´ depot filling stations, the number of "10 GWh wind turbines" to be erected per working day for the "BEV" option increases to at least seven and for the "FC + CG H2" option to at least eleven of these wind turbines! However, this still does not take into account the high number of trucks under 7.5 t including the van class as well as the substitution of today's still fossil net electricity generation and as well as the additional electrical energy with the intended German exclusion of oil and gas heating systems etc., which would still highly significantly increase the requirements for the daily erection of "10 GWh wind turbines“. In this respect, the emphatically positive statements on BEVs in some recent studies and discussion papers are definitely to be questioned, although these are often used by politicians, while other recent studies and open letters to the resonsible EU representatives, signed by hundreds of scientists, are apparently completely ignored by politicians.

Since the generation and consumption of renewable energies do indeed not correlate in a resilient manner, energy storage systems are also indispensable, because energy management ("smart grids"; bidirectional charging) alone will hardly be effective without dispensing with unacceptable interventions such as limited electricity allocations.

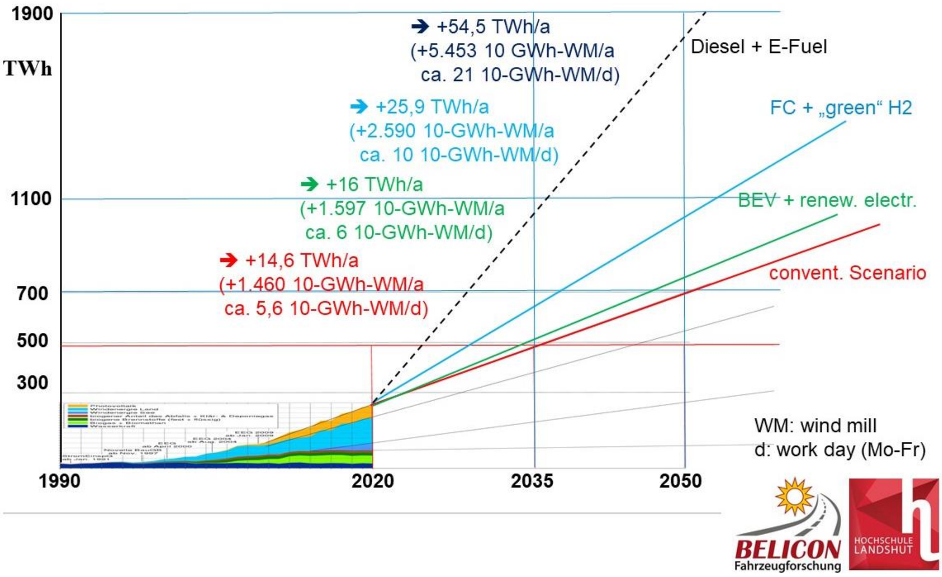


Figure 13 Required availability of renewable energy in Germany only for sustainble car mobility in 2050

3 ANALYSIS OF THE OVERALL ECOLOGICAL PROFILES OF DIFFERENT PROPULSION/FUEL OPTIONS AND CONCLUSION

For a comprehensive ecological and economic evaluation of vehicle fleets of different propulsion systems and fuel types, all stations of the life cycle of transport systems must be included in a comprehensive analysis, as mentioned at the beginning, since the isolated consideration of only the actual driving operation (TtW) can lead to completely wrong conclusions [8]. The following analysis is carried out using the example of a PT bus fleet.

The overall ecological assessment for the time horizon "today" (year 2020) shows that for comprehensive sustainability, the modern Euro VI Diesel bus fleet is already today sufficiently positioned with even fossil Diesel fuel, and with the German electricity mix, little to no improvements are achieved through the procurement of alternative electric drive options (BEV and FC), although according to the EU Clean Vehicles Directive, all electric vehicles - regardless of the electricity mix - are mistakenly declared as "emission-free". Even in the long term, there would therefore be de facto no ecological need to dispense with internal combustion engine technology, especially as further ecological potential can be tapped with E-fuels. Depending on the scenario, the World Energy Council (WEC) assesses the demand for PtX in 2050 to be between 10,000 TWh (low demand) and 41,000 TWh (high demand), according to which even the aforementioned high demand could be covered by the potentially available global generation volumes [9]. However, the calculation with the average German power plant mix, as optimistically also applied here, is an „ecological whitewash“, because the fluctuating residual load must be predominantly covered by controllable power plants. In concrete terms, this means that when the demand for charging electricity increases, the ecologically worst electricity mix is used. This significantly worsens the eco-balance of electromobility. With E-fuels, Re-Fuels (as HVO100) and E-gas, the same GHG emission level is already achievable today in combination with EURO VI engines as with the alternative drive variants of the electromobility spectrum with exclusively renewable energies.

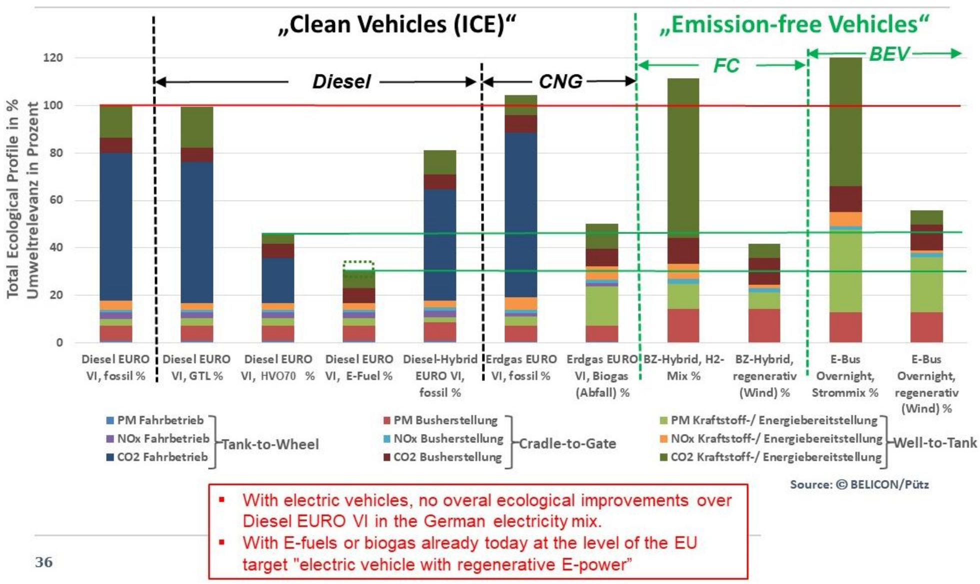


Figure 14 System-related overall environmental profile of a real PT bis fleet for the time horizon „today“

The implications of the range problem of BEV vehicles have already been discussed before in section 3 in the scenarios for heavy-duty trucks and passenger cars. For PT buses, this problem is less significant due to fixed routes. Nevertheless, it must be taken into account that the range of BEV buses drops significantly, especially at low temperatures, as the heating energy must be covered from the limited energy supply of the battery. Considering the additionally limited passenger capacity due to the mass of the required battery capacity, it can even be assumed in the best case that one Diesel bus must be replaced by at least by two battery buses in winter, which worsens the environmental balance of the BEV buses accordingly, see Fig. 15. The equivalent additional demand for drivers must also be taken into account. An operational evaluation is shown in Fig. 16.

By the way: The demand for fossil energy in Europe before the start of the Corona pandemic was around 17,100 TWh. In order to be able to replace this demand with renewable energies, more than 2.9 million (sic!) new wind turbines would have to be installed in Europe in addition to the 82,000 already existing. The calculation would be similar for photovoltaics: The current photovoltaic area of just under 2,100 km_2 would have to be increased to around 230,000 km^2 by a factor of 110 (11,000%!). The decision-makers in the EU should be well aware of these dimensions - which border on utopia - bearing also in mind the requirements of a completely new electricity grid design.

The well-intentioned climate protection goals of the EU may be honourable, but the quasi-dictated paths seem politically naive and possibly economically disastrous - if key industries are sacrificed and ultimately the rest of the world is not willing to follow the European lead.

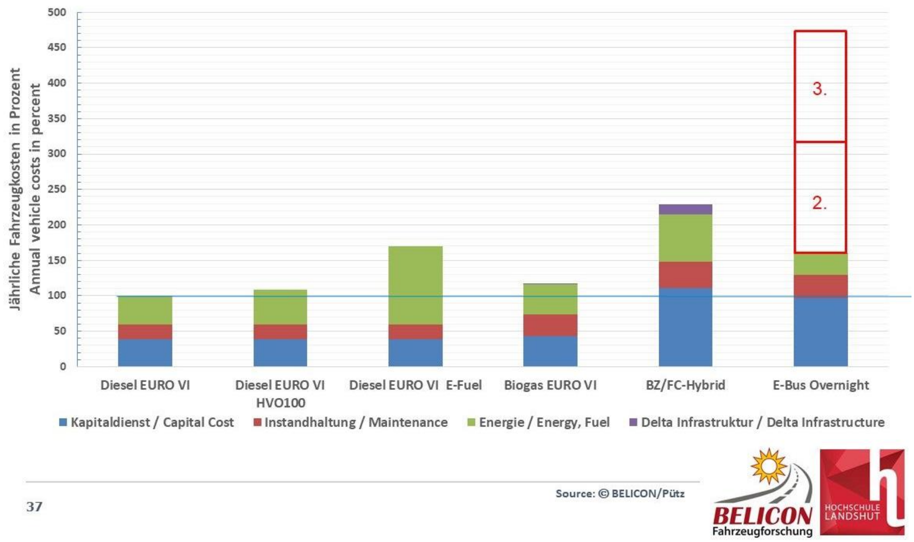


Figure 15 System-related overall Total Cost of Ownership (TCO) of a real PT bis fleet for the time horizon „today“

	Gravimetric Tank Capacity	Volumetric Tank Capacity	Payload Density	Refuelling Time	Operational Flexibility	Infrastructure Availability	Infrastructure Flexibility
EURO-VI-Diesel + E- or Re-Fuels	++	++	++	++	++	++	++
EURO-VI-CNG + E-Gas	+	+	+	++	++	+	+
FC + CG H2	+	+	+	++	++	-	+
BEV	- -	- -	- -	- -	-	- -	+

Figure 16 Evaluation of perational requirements for selected propulsion options with regenerative energy

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DETERMINING THE RELIABILITY OF BRAKE BOOSTERS IN LIGHT COMMERCIAL VEHICLES

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RESEARCH ARTICLE

ABSTRACT: The paper presents the results of research related to the analysis of the reliability of brake boosters in light commercial vehicles. To consider the reliability of mechanical systems, it is necessary to understand the structure, functioning, and potential failure modes that can lead to the loss of operational capability of the system under consideration. Based on operational data, statistical indicators of the random variable of time until the first failure of the brake boosters and the parameters of the approximate model were determined. Statistical data processing was performed using a computer program. For the approximation of the empirical distribution of the random variable, due to its generality and flexibility in adapting to the empirical distribution, the three-parameter Weibull distribution was used. The results of graphical and non-parametric testing confirm that the Weibull distribution can be used as an approximate model. The conclusion of the paper highlights the importance of considering and modeling the reliability of elements in technical systems, the possibilities of using the obtained results, and the directions for further research in this field.

KEY WORDS: *reliability, motor vehicles, braking system, brake booster, Weibull's distribution*

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UTVRĐIVANJE POUZDANOSTI POJAČIVAČA KOČNICA U LAKIM KOMERCIJALNIM VOZILIMA

REZIME: U radu su prikazani rezultati istraživanja vezanih za analizu pouzdanosti pojačivača kočnica u lakim komercijalnim vozilima. Da bi se razmotrila pouzdanost mehaničkih sistema, neophodno je razumeti strukturu, funkcionisanje i potencijalne načine otkaza koji mogu dovesti do gubitka operativne sposobnosti razmatranog sistema. Na osnovu operativnih podataka, određeni su statistički indikatori slučajne promenljive vremena do prvog otkaza pojačivača kočnica i parametri približnog modela. Statistička obrada podataka izvršena je pomoću računarskog programa. Za aproksimaciju empirijske raspodele slučajne promenljive, zbog njene opštosti i fleksibilnosti u prilagođavanju empirijskoj raspodeli, korišćena je Vajbulova raspodela sa tri parametra. Rezultati grafičkog i neparametarskog testiranja potvrđuju da se Vajbulova raspodela može koristiti kao približni model. U zaključku rada ističe se značaj razmatranja i modeliranja pouzdanosti elemenata u tehničkim sistemima, mogućnosti korišćenja dobijenih rezultata i pravci za dalja istraživanja u ovoj oblasti.

KLJUČNE REČI: *pouzdanost, motorna vozila, kočioni sistem, pojačivač kočnica, Vejbulova raspodela*

DETERMINING THE RELIABILITY OF BRAKE BOOSTERS IN LIGHT COMMERCIAL VEHICLES

Dobrivoje Ćatić, Vladimir Ćatić

INTRODUCTION

The braking system is one of the vital systems of the complex mechanical system of a motor vehicle and plays a crucial role in the safety of the vehicle and the people in traffic [9]. It is a typical example of systems in motor vehicles, whose structure is determined by the function of its purpose, as defined by current international and national regulations on vehicle safety in traffic. To increase the efficiency of the braking system in terms of deceleration and stopping the vehicle, and to reduce the force needed to activate the brake control, brake boosters are integrated into the transmission mechanisms of service brakes [5]. Researching the reliability of these devices is of great importance for improving vehicle safety in traffic.

The reliability modeling of brake boosters was carried out using the Weibull distribution. This distribution is, without a doubt, the most widely used in the field of reliability. This directly results from its parametric nature and the wide range of possibilities to interpret very different laws of random variables by selecting appropriate parameter values. Thanks to this, the Weibull distribution can be used as an approximate model for all three periods of the lifespan of mechanical elements. The determination of the parameters of the Weibull approximate distribution is most quickly and accurately conducted using a computer and appropriate software. The application of the least squares method for the analytical determination of the parameters of the three-parameter Weibull distribution, the explanation of the program algorithm, and its practical application are provided in [4].

To reduce the time and cost of testing for reliability evaluation, various accelerated testing procedures are applied. Accelerated testing plans enable the statistical acceleration of tests. By applying accelerated testing plans, the reli-ability of the drum brake cylinder [2] and the steering linkage joint of light commercial vehicles [3] were determined.

1 THE ROLE AND IMPORTANCE OF THE BRAKE BOOSTER IN THE HYDRAULIC TRANSMISSION MECHANISM

The transmission mechanisms of the service brakes in motor vehicle braking systems are designed in various ways [7]. Depending on the energy source used to activate the brakes, the transmission mechanisms can be: without servo assistance, with servo assistance, or with full servo action. In cases where braking cannot be achieved solely by the driver’s effort, certain amplification from an external energy source is introduced, meaning that brake boosters are incorporated into the transmission mechanism. These devices can differ not only in their construction but also in their operating principles. In passenger vehicles, the brake booster is most commonly directly connected to the master brake cylinder and is shared by both branches of the hydraulic system. In commercial vehicles, brake boosters are separate units, and typically two are installed (one for each branch). The simplest way to supply energy to the brake booster is by using the vacuum in the intake manifold of an internal combustion engine. If this is insufficient, for example in diesel engines where the vacuum in the intake manifold is relatively low and inadequate for this purpose, special vacuum pumps are used.

A schematic diagram of the hydraulic transmission mechanism (hydraulic system) of the service brake of light commercial vehicles is shown in Figure 1 [8]. This mechanism belongs to the group of transmission mechanisms with partial servo action. It is a so-called dual-branch mechanism, which allows the vehicle to brake via one branch if the other branch fails (for example, if oil leaks from one branch due to a hose damage). The connection of the brake cylinders is designed so that one branch comprises the brakes on the front wheels, and the other branch on the rear wheels.

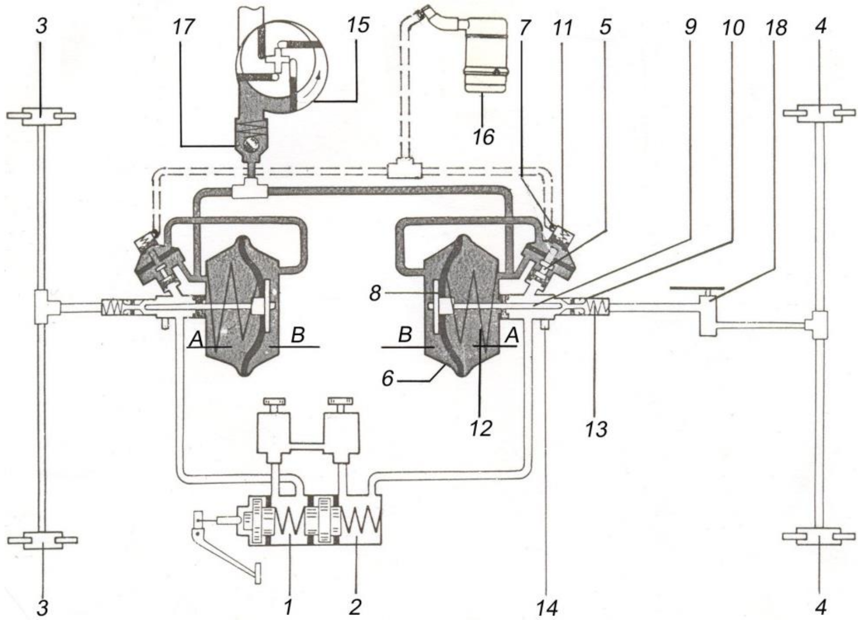


***Figure 1** Schematic diagram of the hydraulic transmission mechanism of the service brake*

The components of the hydraulic transmission mechanism include: the master brake cylinder, wheel brake cylinders, brake boosters, a depressor, a brake regulator, an oil reservoir, pipes, connectors, etc. In the study [1], along with the description of these components of the hydraulic transmission mechanism, explanations of their operation are provided. The operation of the brake booster and vacuum pump is explained alongside the operation of the hydraulic system.

The hydraulic system shown in Figure 1 operates as follows [6]. Pressing the brake pedal creates hydrostatic pressure in chambers (1) and (2) of the master brake cylinder for the dual-branch hydraulic system. From the master cylinder, the oil is carried through pipes to the brake boosters, separately for each branch of the system. The increased pressure in the hydraulic system activates the valve (5) in the brake booster, which interrupts the connection between chambers (A) and (B) of the pneumatic cylinder (6), in which a certain vacuum is always maintained while the engine is running. By interrupting communication between chambers (A) and (B), valve (5) simultaneously allows atmospheric pressure to enter chamber (B) through opening (7). The pressure difference between the chambers (vacuum on one side and atmospheric pressure on the other) causes the elastic diaphragm (8) to move, to which the piston (9) of the hydraulic cylinder (10) is firmly attached at its center. In this way, a force proportional to the pressure difference in chambers (A) and (B) is applied to the piston (9). The movement of the piston in the hydraulic cylinder significantly increases the pressure received from the master brake cylinder. This increased pressure is then carried to the brake cylinders of the front (3) and rear (4) wheels, and the brakes are activated.

When the action on the brake pedal ceases, the return spring (11) opens valve (5), thereby establishing a connection and equalizing the pressure in the chambers on the left and right sides of the elastic diaphragm of the pneumatic cylinder. The return springs of the pneumatic cylinder (12) and the hydraulic cylinder (13) of the brake booster return the diaphragm (8) and the piston (9) to their initial positions, releasing the brakes. The screw (14) is used to release air from the part of the hydraulic system extending from the master cylinder to the brake booster. The brake boosters are powered by a a vacuum pump (depressor). The depressor (15) is driven by an internal combustion engine and serves both brake boosters. For simpler power transmission, it is mounted on the engine block. The air that the depressor draws in at the beginning of operation, until a certain vacuum level is reached, is compressed into the engine sump. Therefore, it is important that the air, which enters the engine block via the brake booster and depressor, is purified. This is achieved by supplying air to chamber (B) of the brake booster through the engine's air filter (16). If the air were not purified, dust particles would lead to accelerated abrasive wear of engine parts and a significant reduction in the lifespan of this system. It is important to note here that if the pneumatic (air) hoses of the brake booster rupture, the depressor becomes a compressor. Since the outlet branch of the depressor is connected to the engine housing, it leads to the creation of high pressure in the housing and the expulsion of oil. A greater capacity of the depressor can be achieved by installing a vacuum reservoir. The check valve (17) allows air to pass only from the pneumatic cylinders to the depressor.

As shown in Figure 1, the brake boosters consist of: a housing, a diaphragm, a piston and cylinder, a return spring, and a valve. The housing is attached to the vehicle’s body. It is made of steel or aluminum. The flexible diaphragm is located inside the housing and divides the device into two chambers. The diaphragm converts the difference in air pressure in the chambers into a force that acts on the piston of the hydraulic cylinder, increasing the hydrostatic pressure in the hydraulic system. The valve controls the airflow in the chambers.

The loss of operational capability of the brake booster can occur due to the failure of its components or the failure of connecting elements or the vacuum device [10]. Pipes, flexible hoses, and connectors are used to connect the brake booster to the other components of the hydraulic system. Damage to the pipes and hoses leads to the leakage of brake fluid and to the complete or partial loss of the braking system’s operational capability. Damage to the air system results in the failure of the brake booster. Additionally, damage to the diaphragm prevents the servo action. A broken return spring slows down the complete release of the brakes. The complete or partial loss of the brake booster’s operational capability significantly affects the efficiency of the service brake and the safety of the vehicle in traffic.

2 STATISTICAL DATA PROCESSING AND ANALYSIS OF RESULTS

As part of the research on the reliability of the braking system of light commercial vehicles, operational tests of critical components of the braking system were conducted. Due to the importance of efficient and reliable operation of the braking system, data on the operating time until failure of the brake booster was collected. The brake booster belongs to the group of repairable items. After a failure, appropriate maintenance measures are taken to restore its operational capability. Therefore, it is important to emphasize that the collected data pertains to the operating time until the first failure. The operating time until failure was measured in kilometers traveled. The obtained dataset on the operating time until failure of the brake booster, arranged in ascending order, is shown in Table 1.

Table 1 Operating times until failure of the brake booster system of light commercial vehicles.

Ordinal numeral of failure	Distance travelled	Ordinal numeral of failure	Distance travelled	Ordinal numeral of failure	Distance travelled	Ordinal numeral of failure	Distance travelled
1	57,570	6	97,350	11	134,162	16	245,709
2	63,000	7	116,138	12	143,434	17	256,850
3	63,521	8	119,706	13	203,861	18	258,285
4	68,635	9	126,170	14	205,405	19	272,125
5	72,047	10	132,833	15	231,542	20	314,515

With the use of software to determine the theoretical model of empirical distribution, as described in detail in [1], numerical characteristics of the statistical series are obtained:

- mean value t_{sr} = 159,143 km,
- standard deviation σ = 81,921 km,
- median t_{50} = 133,497 km,
- coefficient of asymmetry K_a = 0.332 and
- coefficient of flatness K_e = 1.608.

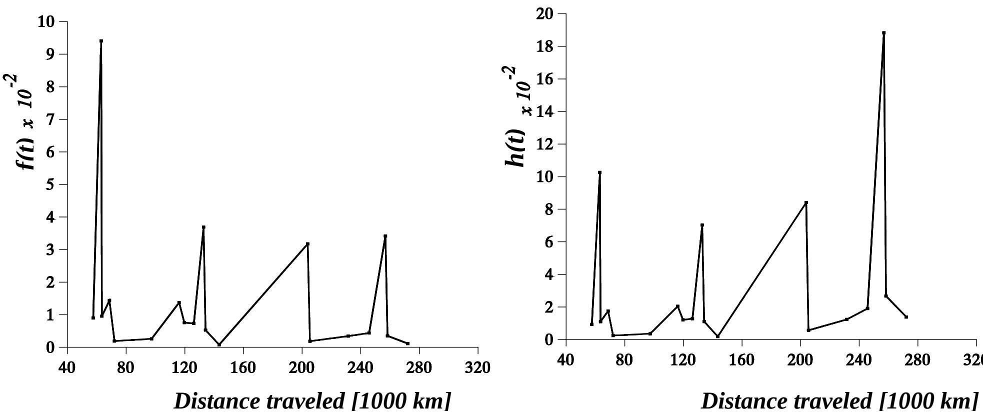
In continuation of the program, and based on procedures for assessing functional indicators of the distribution of a random variable for a small number of samples (n < 30), estimated values are obtained: unreliability $F(t)$, reliability $R(t)$, operational time density until failure $f(t)$ and failure intensity $h(t)$ of brake boosters. These values are calculated for the midpoints of time intervals and are presented in Table 2.

Table 2 Estimated values of functional indicators of the distribution of the random variable

i	t_i	$F(t_i)$	$R(t_i)$	$f(t_i)$	$h(t_i)$
1	57,570	0.03431	0.96569	0.90275E-02	0.93483E-02
2	63,000	0.08333	0.91667	0.94088E-01	0.10264E+00
3	63,521	0.13245	0.86765	0.95854E-02	0.11048E-01
4	68,635	0.18147	0.81863	0.14367E-01	0.17550E-01
5	72,047	0.23049	0.76961	0.19373E-02	0.25173E-02
6	97,350	0.27941	0.72059	0.26091E-02	0.36208E-02
7	116,138	0.32843	0.67157	0.13739E-01	0.20458E-01
8	119,706	0.37755	0.62255	0.75835E-02	0.12181E-01
9	126,170	0.42657	0.57353	0.73570E-02	0.12828E-01
10	132,833	0.47559	0.52451	0.36884E-01	0.70321E-01
11	134,162	0.52451	0.47549	0.52868E-02	0.11119E-01
12	143,434	0.57353	0.42647	0.81122E-03	0.19022E-02

13	203,861	0.62255	0.37745	0.31748E-01	0.84112E-01
14	205,405	0.67167	0.32843	0.18755E-02	0.57104E-02
15	231,542	0.72069	0.27941	0.34601E-02	0.12384E-01
16	245,709	0.76961	0.23039	0.43999E-02	0.19098E-01
17	256,850	0.81863	0.18137	0.34160E-01	0.18834E+00
18	258,285	0.86765	0.13235	0.35419E-02	0.26761E-01
19	272,125	0.91677	0.08333	0.11564E-02	0.13877E-01
20	314,515	0.96579	0.03431		

Illustrations of graph charts depicting the estimated values of operational time density until failure $f(t)$ and failure intensity $h(t)$ for brake boosters, presented as polygons, are shown in Figure 2. In preliminary analyses, these graph charts can be used to determine hypothetical distribution models.



a) b)

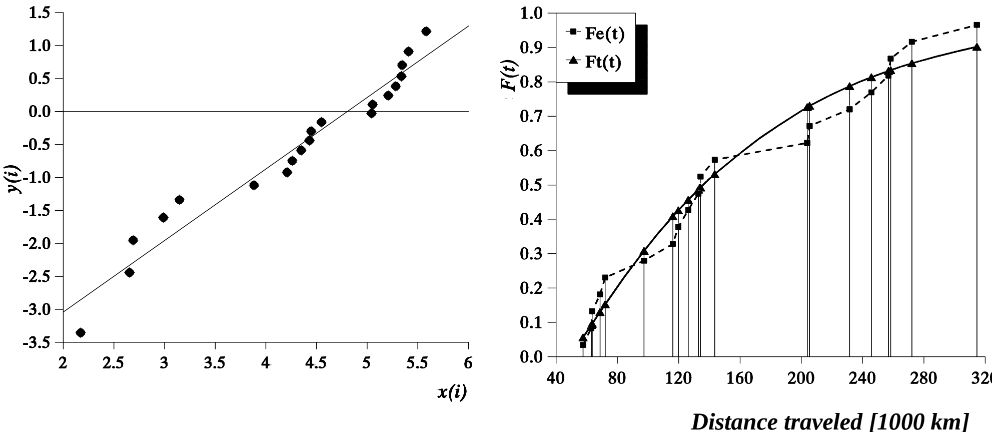
Figure 2 *Diagrams of estimated values for distribution of: a) density and b) intensity of brake boosters failure*

Analytical determination of Weibull distribution parameters using the least squares method is performed through computer-based approximation of the empirical distribution of operational time until failure of brake boosters. The algorithm used is detailed in the paper [4]. After 18 iterations of halving the interval of possible values of the location parameter of the Weibull distribution, and determining the sign of the second derivative of the approximate quadratic parabola, the following parameters are obtained: location parameter γ = 48,792 km, scale parameter η = 122,027 km, and shape parameter β = 1.084. Based on these parameters, the expression for the probability of faultless operation of brake boosters is formulated:

$$R(t) = e^{-\left(\frac{t-\gamma}{\eta}\right)^{\beta}} = e^{-\left(\frac{t-48.792}{122.027}\right)^{1,084}} \qquad (1)$$

During the determination of the probability of proper functioning for brake boosters and all other functional reliability indicators derived from expression (1), time t is expressed in kilometers of distance traveled.

To verify the validity of the approximation, graphical testing on Weibull probability paper and nonparametric testing using Kolmogorov and Mizes tests were conducted. Figure 3 a) displays the arrangement of points with transformed *x* and y coordinates for the specific value γ = 48,792 km on Weibull probability paper. The linear arrangement of the points in Figure 3 a) indicates that the approximate model meets the conditions for graphical testing.



a) b)

Figure 3 *a) Arrangement of points on probability paper for Weibull distribution, b) Graphical representation of deviations of Weibull approximate distribution from empirical distribution*

For testing of hypothetical distribution model, according to Kolmogorov test, it is necessary to determine the greatest absolute value of difference between theoretical model and estimated values of distribution functions of operation time until failure. Table 3 contains a segment of output list of a program that relates to this part. Figure 3 b) presents graphical representation of deviations of theoretical approximate model $F_t(t)$, from empirical distribution $F_e(t)$.

As it may be seen from Table 3, the largest deviation of theoretical model from empirical distribution is for the result No. 13 and amounts to 0.1040. For number of samples, n = 20 and given level of significance for Kolmogorov's test α = 0.20, λ_α = 1.07, permitted value of difference is:

$$D_n = \frac{\lambda_\alpha}{\sqrt{n}} = \frac{1.07}{\sqrt{20}} = 0.2393. \qquad (2)$$

Since the maximal deviation is less than permitted value of difference, Weibull approximate distribution satisfies the Kolmogorov's test for adopted level of significance.

For non-parametric testing of the Weibull approximate distribution, the Mises test was also used. The calculated comparative size of the deviation between the theoretical and empirical models for this test is $n\omega^2$ = 0.0548. With the adopted significance level α = 0.2, the allowable tabulated deviation value is $n\omega^2(\alpha)$=0.2412. Since the calculated value for the Mises test is lower than the tabulated value for the adopted significance level, it can be concluded that the Weibull approximate distribution satisfies this test.

Table 3 Deviations of Weibull approximate curve from estimated values of distribution function of operation time until failure

i	t_i	$F_e(t_i)$	$F_t(t_i)$	delta
1	57,570	0.0343	0.0560	0.0217
2	63,000	0.0833	0.0925	0.0092
3	63,521	0.1324	0.0960	0.0363
4	68,635	0.1814	0.1302	0.0512
5	72,047	0.2304	0.1527	0.0777
6	97,350	0.2794	0.3080	0.0286
7	116,138	0.3284	0.4084	0.0799
8	119,706	0.3775	0.4260	0.0485
9	126,170	0.4265	0.4567	0.0303
10	132,833	0.4755	0.4869	0.0114
11	134,162	0.5245	0.4928	0.0317
12	143,434	0.5735	0.5319	0.0416
13	203,861	0.6225	0.7266	0.1040
14	205,405	0.6716	0.7304	0.0588
15	231,542	0.7206	0.7877	0.0671
16	245,709	0.7696	0.8137	0.0441
17	256,850	0.8186	0.8320	0.0133
18	258,285	0.8676	0.8342	0.0335
19	272,125	0.9167	0.8543	0.0624
20	314,515	0.9657	0.9023	0.0634

Based on the graphs of the estimated values of the failure rate (Figure 2b)), it can be concluded that there is a constant failure rate. This means that the failures of the brake booster occur suddenly during normal operation. As a result of the program approximation of the statistical data set using the least squares method, a three-parameter Weibull approximate distribution with a shape parameter β = 1.084 was obtained. In the two-parameter Weibull distribution, if β = 1, an exponential distribution is obtained, whose parameter λ is the reciprocal of the scale parameter η. The obtained value of the shape parameter β might lead to the incorrect conclusion that an exponential distribution could be used to approximate the statistical data set. However, the operating time until the first failure at 57,570 km is not negligible in relation to the operating time until the failure of the last object. This means that there is a significant period of operation in which no failures occurred. The location parameter or the minimum value parameter of the random variable γ enabled the approximate model to be translationally shifted in the positive direction of the horizontal axis. This means that if a transformed coordinate system were adopted whose vertical axis is shifted to the right by the parameter γ, it could be asserted that an exponential distribution could be used for a hypothetical approximate model. In reality, it deals with a two-parameter exponential distribution, whose density graph is shifted to the right by the value of this second parameter, specifically in this case, by the value of γ.

CONCLUSIONS

By testing machine elements and systems for reliability assessment under operational conditions, a realistic picture of their behavior concerning the loss of operational capability is obtained. This approach takes into account all influential factors on lifespan, which are random in nature and difficult to assess and simulate for laboratory testing purposes. Determining the reliability distribution law represents the ultimate goal of any data analysis in the field of reliability. This process has significant implications for conclusions and decisions related to the practical application of the obtained results. Programmatic determination of the parameters of the three-parameter Weibull distribution, unlike graphical and graph-analytical methods, allows for achieving satisfactory accuracy of results with great speed. Due to this and the well-known properties of this distribution concerning the interpretation of different laws of random variables, the Weibull approximate distribution often emerges as the optimal solution compared to other theoretical models. Knowing the reliability distribution law of machine elements enables the determination of the reliability of the entire system based on the reliability of its components, the planning of maintenance measures, the planning of spare parts production, the determination of the optimal warranty period for components or the entire system, and so on. In this study, a small sample method was applied to determine the estimated values of reliability indicators, specifically through median rank estimation. In this regard, further research could involve determining the confidence interval of the distribution as a whole. Additionally, systematic collection of data on the failures of brake boosters during operation would allow for the planning and execution of accelerated tests for the reliability assessment of the considered object.

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MODELING AND VALIDATION OF TRUCK SUSPENSION SYSTEMS USING ADAMS SOFTWARE

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RESEARCH ARTICLE

ABSTRACT: This paper presents the modeling and validation of front and rear suspension systems in (4x4) trucks utilizing ADAMS software, focusing on leaf spring mechanisms essential for supporting light, medium, and heavy loads. The validation process compares the vertical stiffness of the multi-leaf spring in the front suspension and the dual-rate helper in the rear suspension which consists of main and auxiliary taper-type leaf springs. The stiffness values obtained from ADAMS simulations using the 'Parallel Wheel Travel' test are compared against those calculated analytically via beam theory and with reference values provided in engineering drawings. The comparative analysis indicates that the errors are within an acceptable range. Therefore, it is concluded that the suspension models are validated and can be used for full truck simulation.

KEY WORDS: *Leaf spring, ADAMS, Vertical stiffness, Truck, Beam theory*

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MODELIRANJE I VALIDACIJA SISTEMA VEŠANJA KAMIONA KORIŠĆENJEM ADAMS SOFTVERA

REZIME: UOvaj rad predstavlja modeliranje i validaciju sistema prednjeg i zadnjeg vešanja u kamionima (4x4) korišćenjem ADAMS softvera, fokusirajući se na mehanizme lisnatih opruga neophodnih za podupiranje lakih, srednjih i teških tereta. Proces validacije upoređuje vertikalnu krutost višelisnate opruge u prednjem vešanju i dvostruke pomoćne opruge u zadnjem vešanju, koja se sastoji od glavnih i pomoćnih lisnatih opruga konusnog tipa. Vrednosti krutosti dobijene iz ADAMS simulacija korišćenjem testa „Paralelni tok točka" upoređuju se sa onima izračunatim analitički pomoću teorije grede i sa referentnim vrednostima datim u inženjerskim crtežima. Uporedna analiza pokazuje da su greške u prihvatljivom opsegu. Stoga se zaključuje da su modeli vešanja validirani i da se mogu koristiti za kompletnu simulaciju kamiona.

KLJUČNE REČI: *Lisnata opruga, ADAMS, vertikalna krutost, kamion, teorija grede*

MODELING AND VALIDATION OF TRUCK SUSPENSION SYSTEMS USING ADAMS SOFTWARE

Abdeselem Benmeddah, Momir Drakulić, Aleksandar Đurić, Sreten Perić

INTRODUCTION

In the automotive sector, different types of springs are intended to absorb mechanical shocks and ensure good comfort for passengers. Among these are the leaf springs, widely used in utility vehicles, vans, and trucks and meant to support medium and large loads without compromising vehicle performance.

For many years, leaf springs have been used in vehicles, particularly commercial vehicles, due to their robustness, reliability, and cost-effectiveness [4], which are generally used in different positions within vehicles: in the rear axles of utility vehicles and vans or in the front and rear axles of trucks and trailers.

The leaf springs have sufficient rigidity to support the car body and transmit the load from the chassis to the axle. At the same time, long leaf springs can also contribute to maintaining longitudinal stability [8].

The leaf spring comprises several elements that interact with each other, making its structure complex compared to the other elastic elements [9]. The longest leaf at the top of the spring is known as the master leaf. It is curved at both ends to form spring eyes. The center bolt is the critical part located in the center of the leaf springs. Its task is to bring together all the leaves. Additionally, the U-bolt provides the necessary force to securely fasten the leaf spring and its associated components. The shackle ensures the connection between the leaf spring and the chassis, it can be located at the front or rear of the leaf spring. Rubber bushes offer a cushioning effect for the leaf springs, with steel enclosures for the front ones and an entirely rubber composition for the rear ones. Rebound clips maintain leaf spring alignment and prevent lateral movement while the vehicle is in motion. The bump stop is usually made of rubber. It protects the spring leaf against sudden impacts on the vehicle body and cushions metal-to-metal shocks during extreme compression. In addition, the leaf spring is characterized by its camber, which is the deflection of the leaf spring at its center, measured from the central line passing through the two eyes, which are the attachment points of the spring.

The Figure 1 illustrates all the details of the leaf spring. Each mentioned element affects the characteristics of the leaf spring, particularly its stiffness, meaning that any change in these elements results in a change in the spring's characteristics.

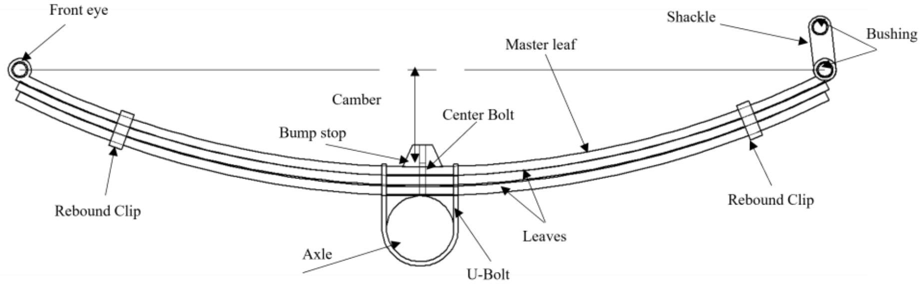


Figure 1 *Leaf spring components*

Leaf springs are classified based on several criteria. They can be classified according to the number of leaves they contain. Single-leaf or mono-leaf springs are commonly found in light vehicles such as passenger cars, vans, and some light trucks. Multi-leaf springs, made of several graduated leaves, are used across various vehicles including trucks, trailers, and some heavy-duty vehicles. In contrast to conventional leaf springs that maintain a uniform thickness throughout, tapered leaf springs are thicker in the center and gradually taper towards the ends. They are commonly used in various vehicles, including trucks, trailers, and some passenger cars.

Leaf springs also vary according to their structure such as semi-elliptical leaf springs, quarter elliptical leaf springs, three-quarter elliptical leaf springs, full elliptical leaf springs, transverse leaf springs, and cantilever-mounted semi-elliptic [3]. Additionally, they can have different rates, ranging from single-rate to variable-rate [6].

Various configurations exist to achieve a variable rate, including dual-rate helper springs, dual-rate extended leaf springs, progressive multi-leaf helper springs, progressive taper leaf helper springs, progressive dual-rate fixed cantilevers, and progressive dual-rate swing cantilevers [3].

Leaf springs present a challenge in design to achieve desired characteristics such as vertical stiffness, torsional stiffness …etc. Several methods have been developed to model the leaf springs and determine their characteristics. Among them: beam theory, the finite element method (FEM), discrete methods...etc.

Beam theory studies the leaf spring as a beam to determine its characteristics, where several beam theories have been used for this purpose such as Timoshenko's theory [7] or by applying the second theorem of Castigliano [1].

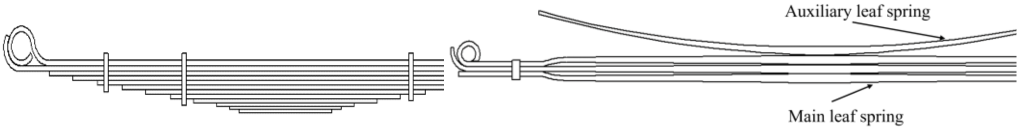
ADAMS Software employs the discrete method to determine the stiffnesses of the leaf spring. This method involves dividing the leaf spring into rigid elements, which are subsequently linked together using components such as torsion springs and dampers. The properties of these torsion springs and dampers are then refined until the force-displacement behavior of the modeled leaf spring matches or closely resembles, the actual force-displacement behavior of the physical leaf spring [4].

This paper aims to model and validate a truck's leaf springs, which are placed in the front and rear suspension in the ADAMS software. It aims to validate the model by comparing the vertical stiffness of the leaf springs obtained through simulation in ADAMS and analytical results using the beam theory with other results represented in the drawings considered reference values.

1 DESCRIPTION OF VEHICLE SUSPENSION

The Truck has a dependent suspension system in both the front and rear. The front and rear suspensions are structurally identical, each containing an axle connecting the two wheels, a shock absorber, and a leaf spring. The difference between them lies in the position of the shock absorbers relative to the axle and the type of leaf spring. The leaf spring used in the front suspension is a multi-leaf type with 12 graduated leaves, as shown in Figure 2 (a). The rear suspension comprises a dual-rate helper spring. It includes a main leaf spring of taper type with 03 leaves and an auxiliary leaf, a single taper leaf that comes into operation when the truck is subjected to a specific load, as shown in Figure 2 (b).

Detailed drawings provide all the details of the front and rear leaf springs. Based on these drawings, we can create a model of ADAMS\CAR's front and rear suspension.



(a) (b)

Figure 2 *(a) Front leaf spring; (b) Rear leaf spring*

2 MODELING OF LEAF SPRING IN ADAMS\CAR

Modeling a system in ADAMS\CAR follows a specific process that begins with creating models in the 'template', defining the major roles such as suspension, leaf spring, anti-roll bar, steering, etc. In these models, all details are provided, including hardpoints, element geometry, Attachment (joints and bushings), characteristics of shock absorbers and springs, and communicators (input and output) which have an important role in defining connections with other subsystems. Afterward, the creation of 'subsystems' based on previously created 'template', specifying minor roles such as front, rear, or any other role, where modifications can be made to the hardpoint, attachments, property files of the shock absorbers, springs, etc. After creating subsystems, they can be assembled into a "Full vehicle assembly" or "suspension assembly".

To assemble the rear truck suspension in ADAMS, three templates are created. The first template defines the axle with the shock absorber, specifying the 'suspension' as the major role. The second and the third templates represent the main and auxiliary leaf springs of taper type, with 'leaf spring' defined as the major role, with inputting all the characteristics of the leaf springs in the toolkit. Then, subsystems are created for each template, specifying 'rear' as the minor role.

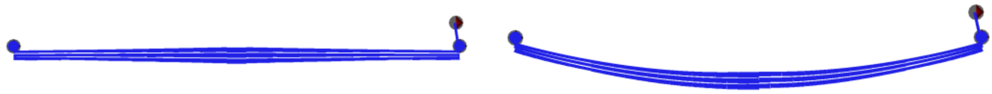
For the front suspension, two templates are generated, following the same procedure as the rear suspension modeling, and changing the type of leaf spring which is a multi-leaf spring. Additionally, the subsystems are created by defining the minor role for both subsystems (axle with shock absorber and leaf spring) as 'front'.

The next step is to create a 'Suspension Assembly' for each leaf spring to proceed with the simulation and determine the stiffness of each one.

To model a leaf spring in ADAMS\CAR, there is a toolbox where all characteristics of the leaf spring can be entered. These parameters are the coordinates (x, z) and the thickness of each leaf, the characteristics of the shackle and clips, and the positioning of the leaf about the axle, bushing, and more.

The leaf spring model is automatically generated with details such as shackles, clips, bushings, seals and communicators.

The details of the three leaf springs are represented in the drawings. However, extracting the coordinates directly from these drawings is impossible due to the deflection of the installed spring in the truck $(z_c = \delta)$, as shown in Figure 3(b), which is different from that represented in the drawing where there is no deflection ($z_c = 0$) as shown in Figure 3(a).



(a) (b)

Figure 3 *(a) Illustrated Configuration in the Drawing, (b) The Installed Configuration.*

To determine the coordinates (x, z) of each leaf, as installed in the truck, we conducted geometric modeling of the master leaf, as detailed in Figure 4. The coordinates of the other leaves are then deduced from those of the master leaf.

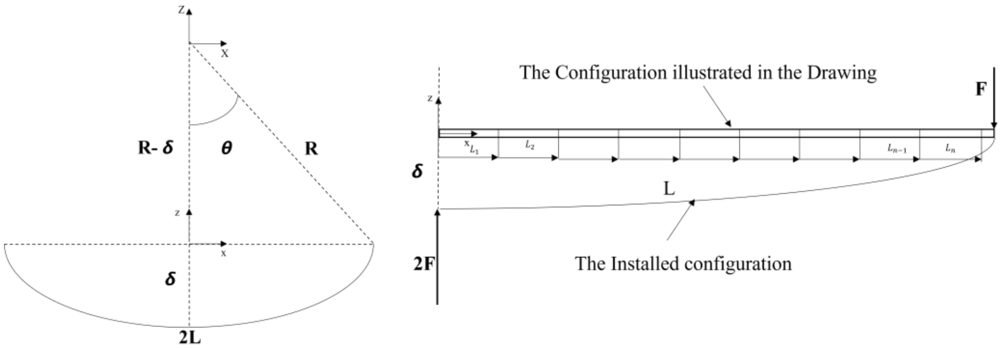


Figure 4 The geometric approach

Where δ represents the deflection, F is the force applied by the body on one side of the leaf spring, L_i are the lengths presented in the drawing, R and θ represent the radius and angle of the arc created when the master leaf is deflected, respectively, which are unknowns and $2L$ represents the total length of the master leaf as provided in the drawing.

According to the geometry presented in Figure 4, a system of equations *(1)* and (2) is extracted to describe the relationships between the parameters.

$$\frac{L}{2} = R\theta \tag{1}$$

$$\cos(\theta) = \frac{R-\delta}{R} \text{ and } \theta \in \left[0, \frac{\pi}{2}\right] \tag{2}$$

From equations (1) and (2), the function f(R) is defined and presented in the equation (*3*):

$$f(R) = \cos\left(\frac{L}{2R}\right) - \frac{R-\delta}{R} \tag{3}$$

By solving f(R) = 0, both the radius R and angle θ of the of curvature are determined and the coordinates (x, z) are found using the equations *(4)*,(5), and (6).

$$\theta_i = \frac{L_i}{2R} \tag{4}$$

$$x_i = R.\sin(\theta_i) \tag{5}$$

$$z_i = R(1-\cos(\theta_i)) - \delta \tag{6}$$

Numerical calculation based on the previously mentioned equations is necessary to find the coordinates (x, z). These coordinates are determined using the measured deflections for each leaf spring as installed in the truck, represented in Table 1.

Table 1 The measured deflection of the leaf springs

	Front leaf spring	Main rear leaf spring	Auxiliary rear leaf spring
Deflection $\delta\ (mm)$	5	130	85

The next step involves inputting the coordinates of each leaf and additional characteristics as detailed in Table 2 into the leaf spring toolkit within ADAMS \CAR.

Table 2 The characteristics of leaf springs

		Front leaf spring	Rear leaf spring	
			Main	Auxiliary
leaves Number		12	3	1
Leaf type		Multi-leaf	Taper leaf	
Shackle	Length (mm)	98		/
	location	Rear		/
Clips	Number	4	2	/
Axle mount type		Over slung		
Eye shape		Downturned	Upturned	/

The leaf spring will be generated automatically within the template and saved as 'leaf spring', as shown in Figure 5. It can then be assembled with the axle to create either the front or rear suspension.

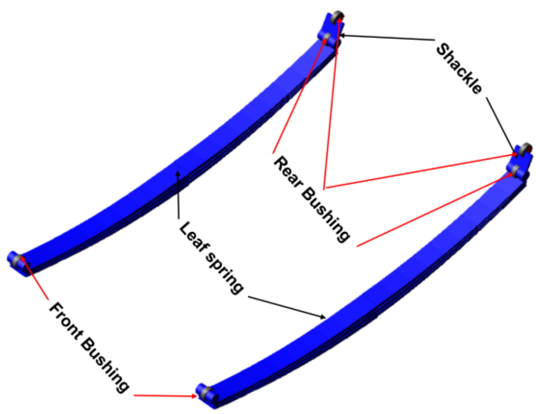


***Figure 5**: The created leaf spring in ADAMS*

3 RESULTS AND DISCUSSION

Generally, the process to determine the vertical stiffness of leaf springs is to apply a vertical force (F) through an actuator at the center of the leaf spring and translation joints are applied at the eyes of the leaf spring, to allow only horizontal translation, as shown in the Figure 6. The corresponding displacement of the center of the leaf (dz) is measured and the vertical stiffness (K) of the leaf spring is determined by plotting the curve $F(dz)$.

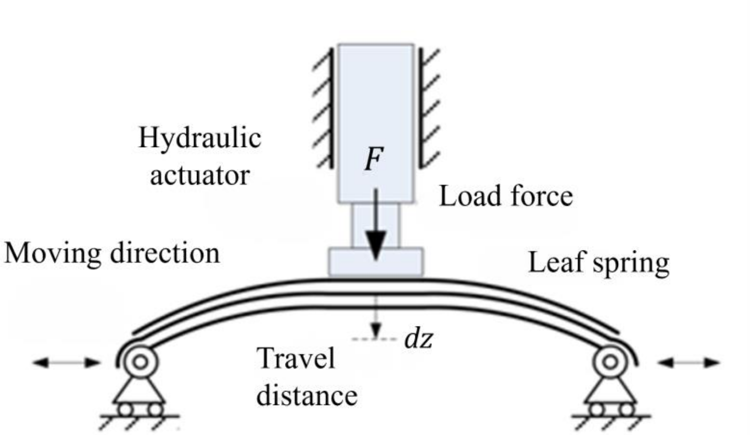


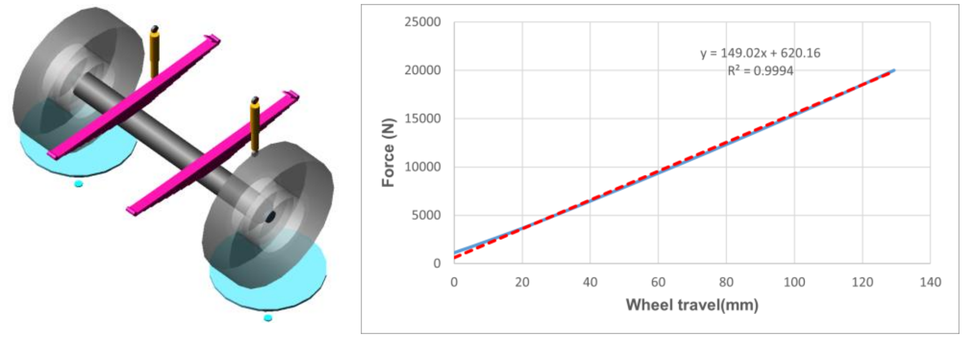
Figure 6 Procedure for determining vertical stiffness of the leaf spring *[5]*

3.1.1 ADAMS\CAR:

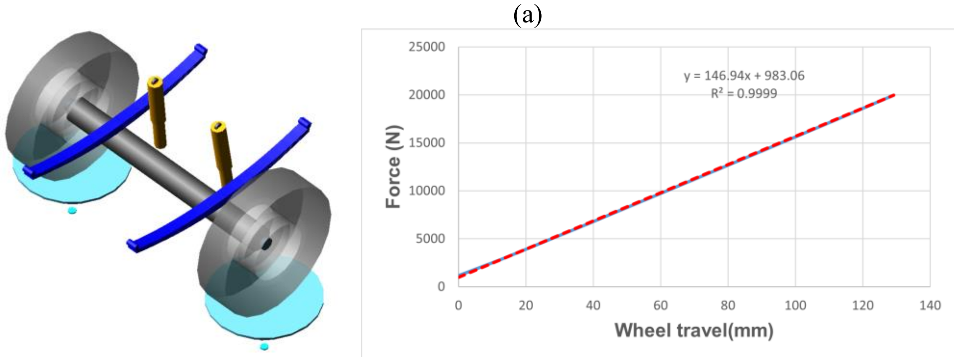
In ADAMS\Car, to realize the suspension assembly as shown in Figure 6, translation joints between the eyes of the leaf spring and the ground will be added. The automatically generated bushings during leaf spring modeling will be deactivated, and the shackle will be ignored. The next step involves creating a subsystem for each leaf spring and assembling them with the axle and shock absorber in the 'Suspension assembly'. At this stage, all three assemblies are ready for simulation.

Parallel Wheel Travel is conducted to determine the vertical stiffness of each leaf spring. In this simulation, vertical displacement is imposed by defining the upper and lower limits of wheel center displacement, known in the software as 'Bump travel' and 'Rebound travel'. When the truck wheels are in contact with the ground; the leaf springs only function under compression. However, when they lose contact with the ground, which is unfavorable, the leaf springs are subjected to traction, which is not taken into account. Therefore, the lower limit value is considered to be zero.

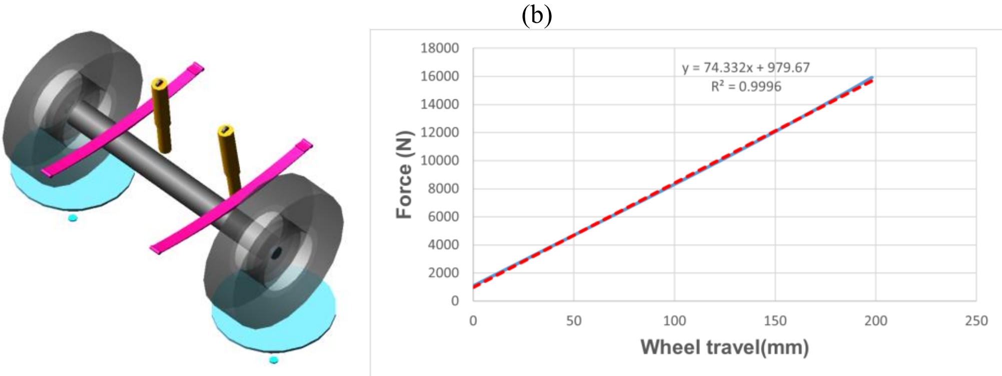
The assembles with the corresponding force-displacement curve for each leaf spring are represented in Figure 7.



(a)



(b)



(c)

Simulation curve   Linear approximation

Figure 7 *Assemblies and their corresponding Force-Displacement curves, (a) Front multi-leaf spring, (b) Rear main taper leaf, (c) Rear auxiliary Single taper leaf*

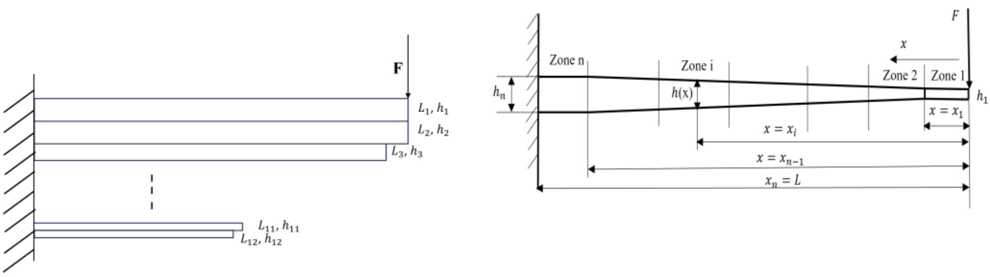
From the simulation results for each leaf spring, as shown in Figure 7, it can be observed that the force-displacement curves are linear ($R^2 \approx 1$). Comparing the three curves, it's evident that the curves of the taper leaf suspension (Figure 7 (b) and Figure 7 (c)) exhibit perfect linearity compared to the curves of the multiple-leaf suspension Figure 7 (a), which initially shows a linearity issue before becoming perfectly linear. This observation can be attributed to the behavior of the leaf spring: in the case of taper leaf springs, all the leaves react simultaneously when a force is applied.

Conversely, during the excitation of the multi-leaf spring containing graduated leaves, the leaves do not react simultaneously at the beginning of the excitation. However, when the applied force reaches a certain threshold, all the leaves react, resulting in a perfectly linear force-displacement curve. Therefore, we can conclude that the vertical rigidity coefficients remain practically constant during the excitation.

3.1.2 Beam theory:

The vertical stiffness of each leaf spring can be calculated using an analytical method commonly found in the literature based on beam theory. Based on the symmetry property of the leaf spring, beam theory is applied to the half-leaf spring to calculate its vertical stiffness. In this approach, one end of the half-leaf spring is fixed while a force is applied to the other, as presented in Figure 8. Then, the vertical stiffness is deduced by dividing the force by the deflection at the end of the beam.

Based on [10,11] and using Castigiliano's theorem [1], the vertical stiffness of each leaf is calculated.



(a) (b)

Figure 8 *The half model of the leaf spring (a) multi-leaf spring and (b) mono-taper leaf spring*

The analytical expressions for the vertical stiffness of a multi-leaf spring and a mono-taper leaf spring are given in the equations *(7)* and (8), respectively.

$$K = \frac{bE}{4\sum_{i=2}^{m}\frac{(L-L_i)^3-(L_1-L_{i-1})^3}{h_1^3+h_2^3+\ldots+h_{i-1}^3}+\frac{L_1^3-(L_1-L_m)^3}{h_1^3+h_2^3+\ldots+h_{i-1}^3+h_m^3}} \tag{7}$$

$$K = \frac{bE}{12\left[\frac{x_1^2}{3h_1^3}+\frac{x_n^2-x_{n-1}^2}{3h_n^3}+\sum_{2}^{n-1}\frac{1}{a_i^3}\left(Ln\left(\frac{h_i}{h_{i-1}}\right)+2b_i\left(\frac{1}{h_i}-\frac{1}{h_{i-1}}\right)-\frac{b_i^2}{2}\left(\frac{1}{h_i^2}-\frac{1}{h_{i-1}^2}\right)\right)\right]} \tag{8}$$

Where:

a_i and b_i are the coefficients that define the linear expression of the height in each zone for the taper leaf spring, as given in (*10*) :

$$h = \begin{cases} h_1 & \text{if} \quad 0 \le x < x_1 \\ a_i x + b_i & \quad x_1 \le x < x_{n-1} \\ h_n & \quad x_{n-1} \le x < x_n \end{cases} \tag{9}$$

- E: Elastic modulus of materials;
- b : Width of the leaves;
- m : Number of leaves;
- n : Number of zones in taper leaf.

To find the vertical stiffness of the main leaf, we either transform the leaf spring with three leaves into a mono-taper leaf spring where we calculate the equivalent thickness of the new leaf.

According to [10], the equivalent thickness can be found using the expression (*10*), this thickness is used in the equation (8) to calculate the vertical stiffness of the main taper leaf.

$$h_e(x) = \sqrt[3]{h_1^3(x) + h_2^3(x) + h_3^3(x)} \tag{10}$$

Through the comparison of the vertical stiffness of each leaf using the simulation and analytical methods with the reference value, as shown in Table 3, the error between the simulation and the reference values does not exceed $\mathbf{11}$%, which is in the acceptable range.

Table 3 Comparison between the simulation and the analytical with the reference results of vertical stiffnesses of the leaf springs

Emplacement	Type of spring	Reference stiffness (N/mm)	Analytical stiffness (N/mm)	Error (%)	Simulation stiffness (N/mm)	Error (%)
Front spring	Multi-leaf spring	141.95	154.436	$\mathbf{8.80}$	149.02	$\mathbf{4.98}$
Rear spring	Main taper leaf spring	148.59	157.60	$\mathbf{6.06}$	146.94	$\mathbf{1.11}$
	Auxiliary single taper leaf spring	67.25	65.30	$\mathbf{2.90}$	74.33	$\mathbf{10.52}$

4 CONCLUSIONS

This paper modeled the front and rear suspension of a (4x4) truck, each comprising an axle, a shock absorber, and a leaf spring. Due to the complexity of modeling the leaf spring in the ADAMS software, it was necessary to validate the model beforehand. The front suspension incorporated a multi-leaf spring consisting of 12 graduated leaves. In contrast, the rear suspension has two taper leaf springs: a main spring with 03 leaves and an auxiliary mono-leaf spring.

This validation process involved comparing the vertical stiffness values of the modeled leaf springs with those calculated using beam theory and the values provided in the engineering drawings which served as reference values.

A 'Parallel Wheel Travel' simulation was conducted for each type of leaf spring using ADAMS software to determine their vertical stiffness. The comparison indicated that the error between the simulated and analytically calculated vertical stiffness values with the reference values, were within acceptable limits. Consequently, the front and rear suspension models are validated, enabling us to use them in the complete truck model.

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APPLICATION OF HYDROSTATIC TRANSMISSION IN MOBILE MACHINES

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RESEARCH ARTICLE

ABSTRACT: Hydrostatic transmissions are widely used in the field of mobile machines in construction, mining, agriculture, forestry, both to start working bodies (multipliers) that perform technological operations, and for motion drives. In the first part of the paper, hydrostatic power transmissions and their classification are described, followed by an explanation of the working principles of hydraulic pump and hydraulic motor. In the second part of the paper, the application of the hydrostatic transmission in different mobile machines is presented. The third part represents the most crucial segment, where the calculation to obtain the key parameters for hydrostatic transmission design is presented. Finally, the main components of the hydrostatic transmission are selected.

KEY WORDS: *hydraulic pump, hydraulic motor, calculation, hydrostatic transmission*

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PRIMENA HIDROSTATIČKE TRANSMISIJE U MOBILNIM MAŠINAMA

REZIME: Hidrostatički prenosnici se široko koriste u oblasti mobilnih mašina u građevinarstvu, rudarstvu, poljoprivredi, šumarstvu, kako za pokretanje radnih tela (multiplikatora) koji obavljaju tehnološke operacije, tako i za pogone kretanja. U prvom delu rada opisani su hidrostatički prenosnici snage i njihova klasifikacija, nakon čega sledi objašnjenje principa rada hidraulične pumpe i hidrauličnog motora. U drugom delu rada predstavljena je primena hidrostatičkog prenosnika u različitim mobilnim mašinama. Treći deo predstavlja najvažniji segment, gde je predstavljen proračun za dobijanje ključnih parametara za projektovanje hidrostatičkog prenosnika. Na kraju, odabrane su glavne komponente hidrostatičkog prenosnika.

KLJUČNE REČI: *hidraulična pumpa, hidraulični motor, proračun, hidrostatički prenosnik*

APPLICATION OF HYDROSTATIC TRANSMISSION IN MOBILE MACHINE

Vanja Šušteršič, Vladimir Vukašinović, Dušan Gordić, Mladen Josijević

INTRODUCTION

The hydraulic power transmission is carried out by means of liquids, which are most often mineral hydraulic oils and non-flammable fluids for hydraulics. In the components of these transmissions, the mechanical energy of the working fluid is converted into its fluid energy and vice versa. Depending on the principle of operation of these hydraulic devices for energy conversion, hydrostatic and hydrodynamic power transmissions are distinguished. In hydrostatic transmissions, the energy converters are a volume pump at the input and a volume hydro-motor or hydraulic cylinder at the output [1].

Compared to others, hydraulic drives have a significantly lower mass and smaller aggregate dimensions, and therefore have a low inertia; the rotational mass of hydraulic motors of rotational action is several times smaller than the rotational mass of electric motors of the same power. They make it possible to achieve a stepless change of the output speed, convert rotary motion into translational and translational into rotary motion; constructively, it simply ensures the protection of hydro aggregates from overloading. Disadvantages include the high price of aggregates, the complexity of exploitation, and a relatively short service life [1].

In recent years, the progress has been made in the development and practical application of hydrostatic transmission and management in all branches of economy. Hydrostatic transmissions (HST) are particularly widely used in the field of mobile machines in civil engineering, mining, agriculture and forestry, wind turbines, etc. These hydrostatic systems are intended for movement of working organs (manipulators) which perform technological operations and also as driving power [2].

When we talk about the global hydrostatic transmission market, it was at 5.1 billion USD in 2023. and is predicted to reach 7.9 billion USD by 2032 (Figure 1). This sector has a strong presence in the North American market, especially the United States and Canada in the agricultural sector. Technological improvements and the deployment of modern machinery drive the market in this area even further.

When we talk about the application of hydrostatic transmisisons in the EU, Germany, France, the United Kingdom, Italy, and Spain are leading the way. The increase of building and mining operations, notably in Eastern European nations, is driving market growth in Europe. Adoption of hydrostatic transmissions is further aided by stringent rules encouraging energy efficiency and environmental practices [3].

Configuration of hydrostatic transmission

There are four types of hydrostatic transmission, two with open circuit and two with closed circuit. In open-loop circuit transmission (Figure 2, a), the working fluid enters the regulating pump through the reservoir, then passes through the hydraulic motor and finally reaches the reservoir, or rather, returns to it. In a closed-loop circuit, (Figure 2, b) the path of fluid movement is continuous, so the fluid flows along a constant path from the output of the control pump to the input of the control motor and vice versa [2]. An open circuit is not used in vehicles, because it cannot be reversed; and lacks braking. It is used for conveyor belts where load is resistant and where rotation in one direction is possible.

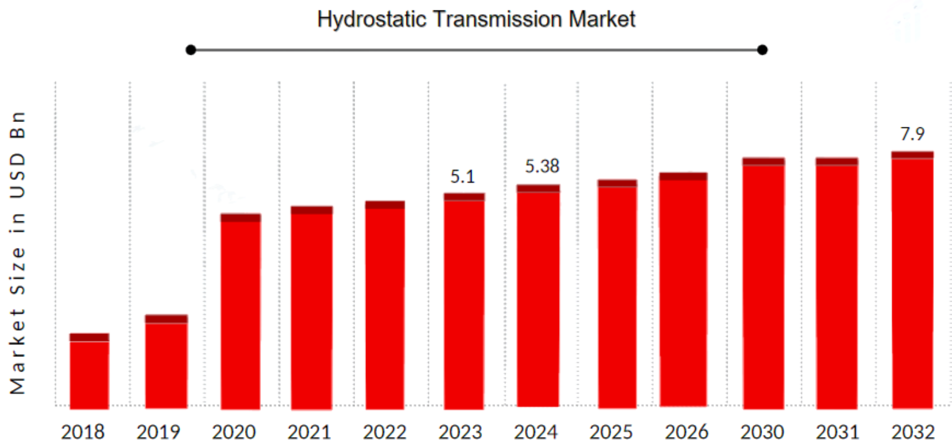
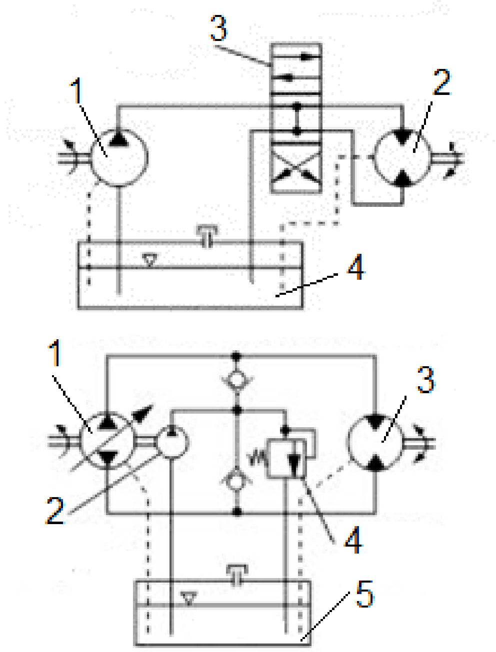


***Figure 1** The global hydrostatic transmission market [3]*



a) b)

***Figure 2** Scheme of the hydrostatic transmission open-loop hydraulic circuit, 1 - hydraulic pump, 2 - hydraulic motor, 3 – directional valve, 4 – oil tank closed-loop hydraulic circuit, 1 - hydraulic pump, 2 - feed pump, 3 - hydraulic motor, 4 - pressure regulator, 5 – oil tank [4]*

The configuration of the hydrostatic transmission can be with a pump with a constant (or variabile volume), and with a hydraulic motor with a constant (Figure 2, a) or variable working volume. HST with constant displacement pump and motor is cheap, but its application is limited, mainly because other types of power transmission are inefficient. The most widespread configuration of hydrostatic transmission consists of a regulating pump (Figure 2, b) and a regulating motor with a variable working volume. Theoretically speaking, this arrangement provides an infinite number of torque and speed ratios. With the throttle motor at maximum operating volume, the variable output volume of the throttle pump directly changes the speed and power output, while the torque remains constant. By reducing the working volume of the engine at the full working volume of the control pump, the speed of the control motor increases to its maximum value, the torque changes inversely with the speed, and the power remains constant.

Figure 3 shows the working characteristics of the hydrostatic transmission. Range 1 covers changing the angular speed of the motor by varying only of displacement of the hydraulic pump from zero to maximum values, while the working volume of the engine is fixed at its maximum value. Range 2 starts when the pump reaches its maximum move. To obtain higher values for the angular velocity, engine displacement must be reduced. Within this range, the system ensures constant power and flow, while hydraulic torque is proportional to engine volume move.

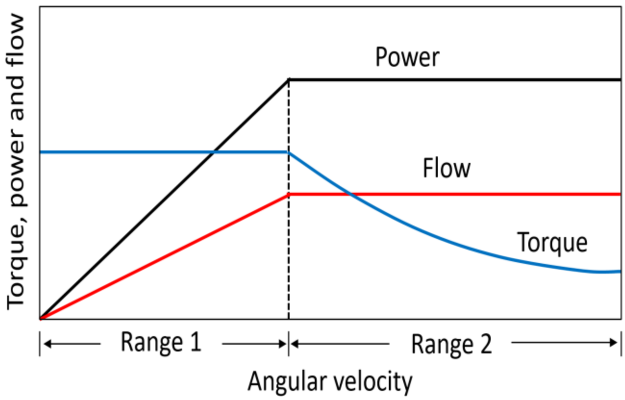


Figure 3 *Operational characteristics of the hydrostatic transmission [5]*

1 APPLICATION OF HYDROSTATIC TRANSMISSION

Hydrostatic transmission is widely used in agriculture. It is used in tractors, all types of self-propelled machines such as combine harvesters for sugar cane, sugar beet harvesters, silage harvesters, sprayers, telescopic manipulators, silo mixers, etc. Hydrostatic transmission is the most modern, but also the most expensive transmission and is used mainly in special agricultural machines. In order to achieve cost reduction, manufacturers of agricultural machinery apply hydro-mechanical transmissions with less or greater use of reducers and mechanical drive bridges [4]. At the beginning of the 2000's, the most commonly applied type of transmission was with rear-wheel drive only, with one variable-flow piston-axial pump and one two-flow piston-axial motor connected to a mechanical drive bridge (Figure 4).

Figure 4 *Mechanical drive bridge with attached hydraulic motor and reducers [6]*

Figure 5 shows the hydrostatic drive solution with a speed reducer between the hydraulic motor and the wheel. This a hydrostatic circuit consists of a piston-axial pump, a two-flow piston-axial hydraulic motor with an inclined axis and a speed reducer. The most common transmission ratio in such reducers is i = 35 - 45. An integrated parking brake or an active disc brake are installed in the reducer. The big advantage of this type of hydrostatic system is its price. Two-flow piston-axial engines have a price of about 60% lower compared to high-torque piston-radial low-speed engines, while the reducers are from mass production, slightly modified to meet the needs of the machine manufacturer [7].

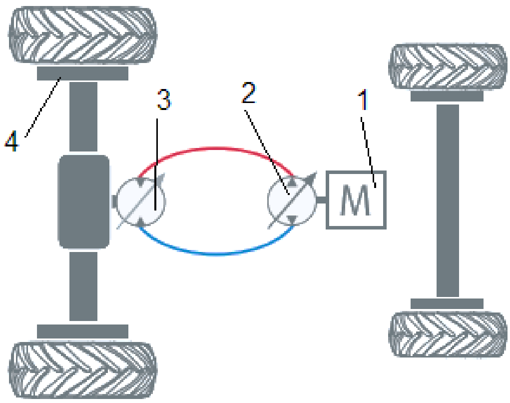


Figure 5 *Classic hydrostatic drive system with a speed reducer (1 - power unit, 2 - piston-axial pump, 3 - piston-axial hydraulic motor, 4 - reducer) [7]*

One example of the use of hydrostatic transmission in a combine harvester is shown in Figure 6. This harvester uses a diesel engine, a variable-displacement hydraulic pump with mechanical-hydraulic control, and a constant displacement hydro-motor connected to a three-speed gearbox. Gear ratios are adjusted for different conditions of use such as harvesting, moving around the field, as well as the road. In the cabin, there is a mechanical-hydraulic controller, with which the operator of the machine regulates the operation of the hydro-pump.

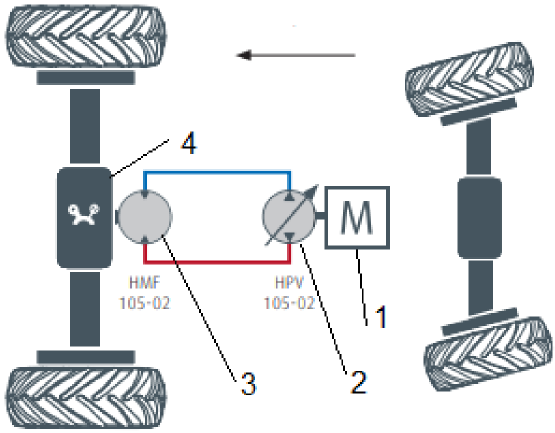


Figure 6 *Hydrostatic transmission for combine harvesters (1 - diesel engine, 2 - hydraulic pump, 3 – hydro-motor, 4 - gearbox) [7]*

Hydrostatic transmission is also used to drive sugarcane harvester. In Figure 7, the concept with two hydraulic circuits is shown and works without a transfer mechanism with a distributor. The power unit drives two variable displacement hydraulic pumps, which transmit hydraulic power, each separately, to variable displacement hydraulic motors that drive the tracks.

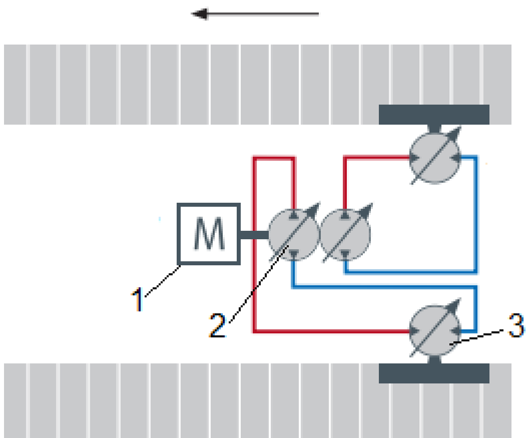
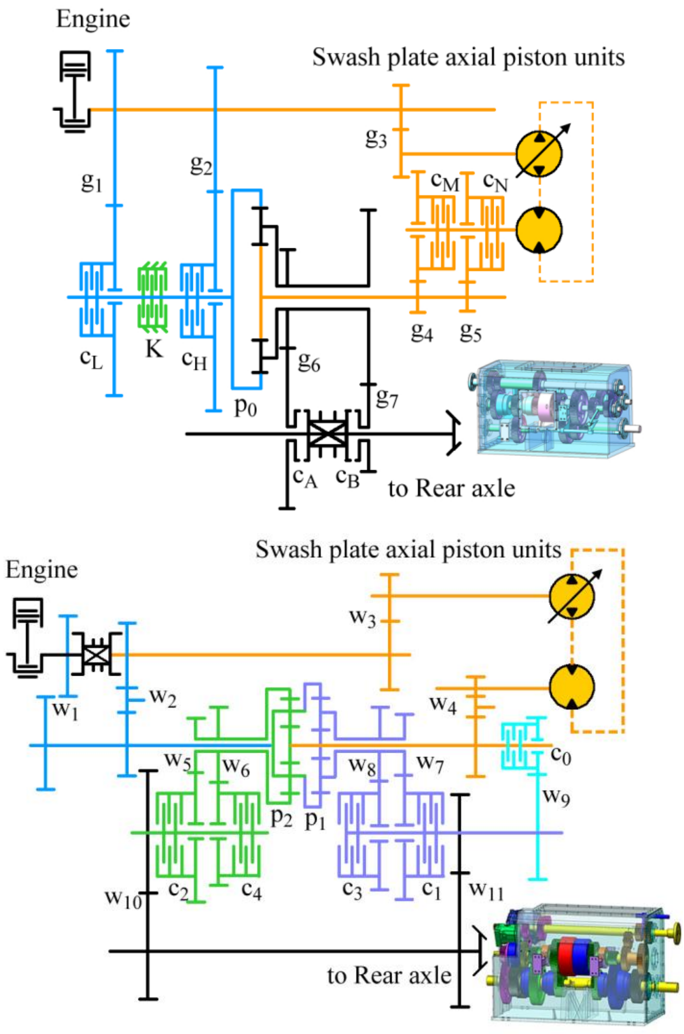


Figure 7 *Hydrostatic transmission at sugarcane harvester (1 – diesel engine, 2 - hydraulic pump, 3 - hydraulic motor) [7]*

When we talk about tractors, manual transmission, hydrostatic (HST) and hydro-mechanical transmission (HMT) are used today. HST and HMT transmission are continuously variable transmissions and can operate the engine with high thermal efficiency independently of the vehicle speed in the transmission range, thereby reducing fuel consumption and exhaust gases. In addition, being able to automatically control the gear ratio increases the driver's work efficiency, and finally, they are environmentally friendly and highly efficient [8]. In order to improve the comfort of driving a tractor with HST, this transmission is often combined with a planetary gearbox (Figure 8) [9].



a) b)

***Figure** 8 Powertrain of hydrostatic power-split transmission (a) Standard HMT, (b) Simpson HMT [9]*

2 CALCULATION OF HYDROSTATIC TRANSMISSION

The most important assumption for setting up a satisfactory solution to the problem is the previous system procedure of planning and execution of the hydrostatic system. When designing the hydrostatic system, the requirements and parameters of the function of the members of the kinematic chain of the machine that the actuators of the hydrostatic system should strengthen are first analyzed in detail. For the correct selection and definition of hydrostatic components, it is necessary to know their characteristics: principles, methods and conditions of operation, basic parameters and transmission functions, methods of installation and maintenance, prices and methods of delivery.

In this part, the calculation of hydrostatic transmissions is described, which is used to drive wheeled tractors and crawler tractors, and based on which the choice of hydro-pump and hydro-motor is then made. First, it is necessary to calculate the parameter values of individual components that are within this transmission. Some of the quantities are hydraulic power, maximum pump flow, specific flow, etc. The general scheme of the hydrostatic transmission used in the calculation is shown in Figure 9.

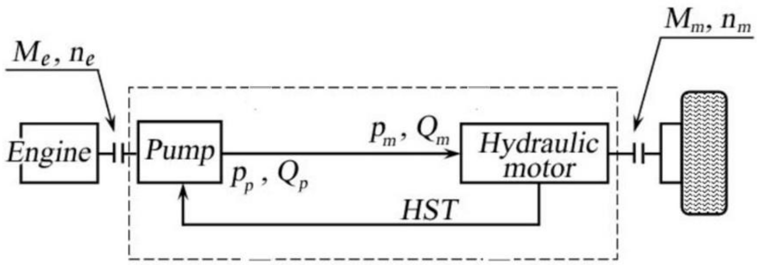


***Figure 9** Model of hydrostatic transmission [10] M_e, n_e - torque and motor shaft speed; M_m, n_m - torque and rotational speed of the hydraulic motor shaft; p_p, Q_p - pump pressure and flow rate; p_m, Q_m - pressure and supply to the hydraulic motor*

For wheeled tractor, the circulation of the working fluid is in a closed circuit. This type of hydraulic drive consists of a power unit, specifically a diesel engine, a hydraulic pump with a variable working volume, which drives a hydraulic motor with a constant working volume, and which is connected via a two-stage mechanical gearbox to the drive bridge, which drives the wheels and transmits power to the ground.

The basic equations for torque and flow of hydraulic pump and hydraulic motor are given by equations 1-4.

$$M_p = \frac{\Delta p \cdot q_p}{2 \cdot \pi \cdot n_p} \quad (1)$$

$$M_m = \frac{\Delta p \cdot q_m \cdot \eta_m}{2 \cdot \pi} \quad (2)$$

$$Q_p = q_p \cdot n_p \cdot \eta_{vp} \quad (3)$$

$$Q_m = \frac{q_m \cdot n_m}{\eta_{vm}} \quad (4)$$

where are: $\boldsymbol{\Delta p}$ - pressure difference across hydraulic motor/pump (Pa); $\boldsymbol{Q_p}, \boldsymbol{Q_m}$- flow rate of hydraulic pump/motor (l/min); $\boldsymbol{M_p}, \boldsymbol{M_m}$- torque of hydraulic pump/motor (Nm); $\boldsymbol{n_p}, \boldsymbol{n_m}$– number of revolutions of pump/motor (rpm), $\boldsymbol{q_p}, \boldsymbol{q_m}$– specific flow rate of hydraulic pump/motor - displacement (cm^3/rev); $\boldsymbol{\eta_{vp}}, \boldsymbol{\eta_{vm}}$- volumetric efficiency.

Table 1 gives the initial data for the calculation of hydrostatic transmission for wheeled and crawler tractors, and Figure 10 shows the calculation flow.

Table 1 Basic data for calculation

Wheeled Tractor		Crawler tractor	
Vehicle weight [kN]	42.8	Mass of tractor [t]	7.5
Wheel radius [mm]	590	Engine power [kW]	120
Nominal speed of pump [min-1]	2100	Maximum flow of pump [l/min]	250
Maximum vehicle speed [km/h]	25	Maximum working pressure [MPa]	40
Type of road	macadam	Maximum vehicle speed [km/h]	10
Maximum ascent [o]	15	Maximum required traction force of one track [N]	2590

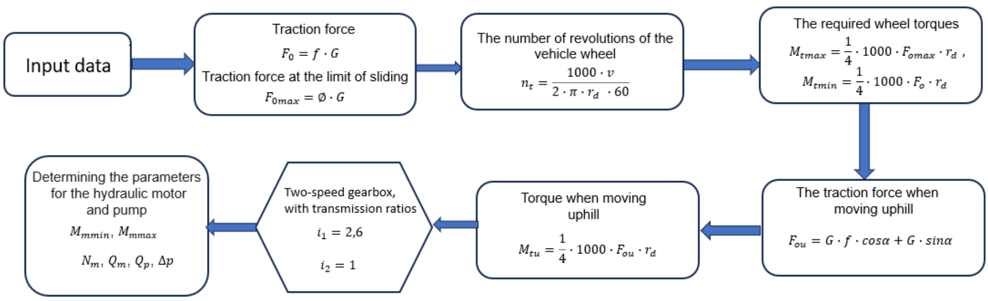


***Figure 10** Calculation flow for wheeled tractor*

After obtaining the necessary parameters, the pump and motor are selected. The hydraulic motor and hydraulic pump, based on the data obtained from the calculation, were adopted from the catalog of the manufacturer "Bosh Rexroth", model MCR-F, and the hydraulic pump AA4VG Series 32 (Table 2).

For the crawler tractor used a hydrostatic transmission system with two open circuits (Figure 11), consisting of: diesel engine, elastic coupling, toothed distributor, hydraulic pumps, distributors, modular drive transmission with an integrally connected hydraulic motor and planetary reducer to which the caterpillar's drive sprocket is attached. The calculation is based on the maximum required torque on the output shaft of the drive reducer, i.e. the maximum required torque on the drive sprocket of one caterpillar. It is necessary to first calculate the maximum required traction force of one caterpillar.

Table 2 Technical data on the hydraulic pump and motor from the manufacturer's catalog [11]

Radial piston hydraulic motor „*Bosh Rexroth MCR-F*“	
Displacement	565 cm^3/rev
Maximum torque	4047 Nm
Maximum pressure	450 bar
Maximum number of revolutions	385 min^{-1}

Axial piston pump „Bosh Rexroth AA4VG Series 32“	
Displacement	40 cm^3/rev
Nominal pressure	400 bar
Maximum pump flow	160 l/min
Maximum pressure	450 bar

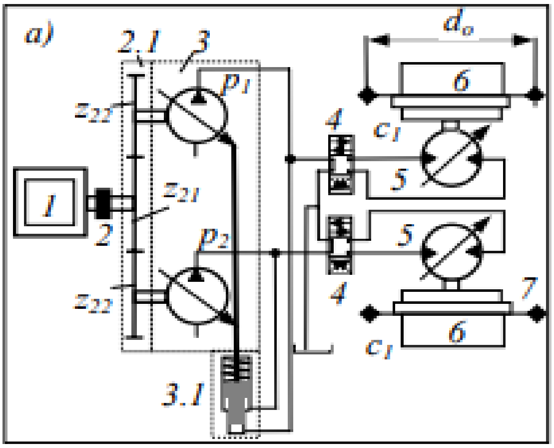


Figure 11 *Functional scheme of transmission with two open hydrostatic circuits [12]*

(1 – diesel engine, 2 – elastic coupling, 2.1 – gear distributor, 3 – hydraulic pump, 3.1 – collective distributor, 4 – flow regulator, 5 – hydromotor, 6 – reducer, 7 – sprocket)

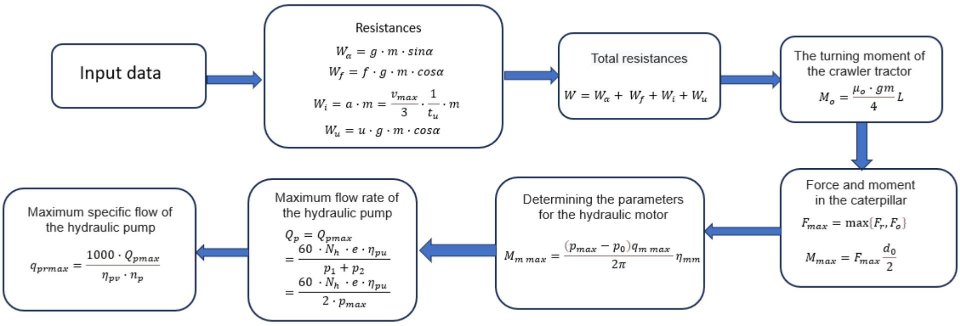


Figure 11 *Calculation flow for crawler tractor*

Based on the maximum specific flow determined by calculation, according to the manufacturer's catalog, the closest available value of the specific flow of the pump is adopted. If there is a significant difference between the magnitude of the required and available specific flow of the hydraulic pump, while maintaining the desired maximum flow of the hydraulic pump, the required gear ratio is determined from the power distributor. After that, the maximum number of revolutions of the hydraulic pump is determined through the ratio and checked in the manufacturer's catalog. The hydraulic motor and hydraulic pump were adopted, based on the data obtained from the calculation (Table 3):

- two piston-axial pumps from the catalog of the manufacturer "Danfoss", series 45, with frame E, and
- two hydraulic pump two hydraulic motors model A10VE manufactured by "Bosh Rexroth"

Table 3 Technical data on the hydraulic pump and motor from the manufacturer's catalog [11, 13]

Radial piston hydraulic motor "Bosh Rexroth A10VE"	
Displacement	45 cm^3/rev
Maximum torque	250 Nm
Maximum pressure	350 bar
Maximum number of revolutions	5400 min^{-1}
	

Piston - axial pump "Danfoss"	
Displacement	130 cm^3/rev
Nominal pressure	310 bar
Maximum pump flow	250 l/min
Maximum pressure	400bar
	

3 CONCLUSIONS

The paper presents a mathematical calculation that enables a preliminary calculation of the working volumes of hydraulic machines with further specification in accordance with standard values. The pressure produced by the pump is determined according to the load during the movement of the transport machine. The power of the drive motor should take into account both the movement of the vehicle and the possibility of creating special systems with a hydraulic drive.

The main disadvantages of the mechanical transmission are the sudden change in the transmission ratio due to the gearbox that works on the principle of toothed transmission, a small ratio of power per unit of mass, poor flexibility and the inability to regulate. On the other hand, the use of hydrostatic transmission in vehicles enables the achievement of large forces and moments with devices of small dimensions. Continuously variable transmission is also achieved within the entire working area, giving the best transfer measures between the drive motor and the wheels, which increases dynamic performance and reduces fuel consumption.

Further development of hydrostatic transmission components (primarily piston-axial hydraulic pumps and piston-radial hydraulic motors) as well as the integration of electronics and computers into these systems will dictate further directions of development of these systems. Some types of hydrostatic systems are the most acceptable with their price and will be used as basic systems for driving mobile machines for a long time. On the other hand, other types of hydrostatic systems presented in this paper are dominant in special agricultural machines (corn pickers, sugar beet harvesters), while the most modern hydrostatic systems are used in the most complex self-propelled agricultural machines (vegetable harvesters).

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QUANTITATIVE ANALYSIS OF NONCONFORMING PRODUCTS: A CASE STUDY IN THE AUTOMOTIVE INDUSTRY

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RESEARCH ARTICLE

ABSTRACT: This paper presents an analysis of nonconforming products within a case study conducted in an automotive industry company. Utilizing monthly data over one year, the study examines trends in nonconformance and associated costs. Descriptive statistics reveal significant monthly variations in nonconforming products, while correlation analysis indicates a strong relationship between the frequency of nonconforming products and the incurred costs. The findings suggest that addressing the most frequent types of nonconforming products could lead to substantial cost reductions. Key recommendations include implementing targeted preventive measures, such as enhancing employee training, adhering to operational procedures, and optimizing machinery and equipment settings. Establishing a robust quality control system is essential for minimizing nonconformance.

KEY WORDS: *nonconforming products, automotive industry, costs, statistics*

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KVANTITATIVNA ANALIZA NEUSAGLAŠENIH PROIZVODA: STUDIJA SLUČAJA U AUTOMOBILSKOJ INDUSTRIJI

REZIME: Ovaj rad predstavlja analizu neusaglašenih proizvoda u okviru studije slučaja sprovedene u kompaniji automobilske industrije. Koristeći mesečne podatke tokom jedne godine, studija ispituje trendove u neusaglašenosti i povezanim troškovima. Deskriptivna statistika otkriva značajne mesečne varijacije u neusaglašenim proizvodima, dok korelaciona analiza ukazuje na jaku vezu između učestalosti neusaglašenih proizvoda i nastalih troškova. Rezultati sugerišu da bi rešavanje najčešćih tipova neusaglašenih proizvoda moglo dovesti do značajnog smanjenja troškova. Ključne preporuke uključuju sprovođenje ciljanih preventivnih mera, kao što su poboljšanje obuke zaposlenih, pridržavanje operativnih procedura i optimizacija podešavanja mašina i opreme. Uspostavljanje robusnog sistema kontrole kvaliteta je neophodno za minimiziranje neusaglašenosti.

KLJUČNE REČI: *neusaglašeni proizvodi, automobilska industrija, troškovi, statistika*

QUANTITATIVE ANALYSIS OF NONCONFORMING PRODUCTS: A CASE STUDY IN THE AUTOMOTIVE INDUSTRY

Nikola Komatina, Danijela Tadić, Marko Djapan

1 INTRODUCTION

One of the significant challenges faced by industrial enterprises is the occurrence of nonconforming products. The presence of nonconforming products can result in a lower output of conforming, or good and acceptable products, compared to what was planned in the production schedule. A nonconforming product is one that fails to meet the criteria for delivery to customers and, therefore, cannot be considered an acceptable outcome of the manufacturing process. Even if a company effectively plans and directs its production capacities according to forecasted demand, the occurrence of nonconforming products can significantly impact the execution of plans and the achievement of production goals.

Nonconformities in the manufacturing process occur when a process or product does not meet defined requirements (such as standards, customer requirements, or defined plans and goals) [10]. Generally, nonconformities can result in various consequences, including inadequate product quality, loss of customer trust, and negative impacts on the company's reputation. To effectively manage nonconformities, it is essential to establish procedures and mechanisms for their identification, analysis, and resolution.

In this study, the occurrence of nonconforming products was analysed using descriptive statistics, interval ratings, and correlation analysis. The aim of this paper is to identify and analyse the causes of nonconforming products, as well as to evaluate their impact on the effectiveness of the manufacturing process. The analysis was further enriched with graphical representations and examples of applying specific statistical functions in Microsoft Excel.

The case study presented in this paper is based on data from a company in the automotive industry. The company belongs to the category of small and medium-sized enterprises. Its product range consists of aluminium products processed on hydraulic presses. The company's primary product is radiator housings for electric vehicles, but its product range also includes other products obtained through similar processing methods. The company is certified according to the IATF 16949:2016 standard, which represents a quality management system tailored to the needs of the automotive industry.

2 TYPES OF NONCONFORMITIES IN THE MANUFACTURING PROCESS

In the manufacturing process, both major and minor nonconformities may arise. The difference between major and minor nonconformities is reflected in their consequences and the hierarchical level at which they occur [10].

Major nonconformities typically occur more frequently and can significantly impact productivity. They can directly lead to customer dissatisfaction and negatively affect the manufacturing process, resulting in increased costs, waste, and potential financial and other penalties for failing to meet specific standards. Minor nonconformities, on the other hand, are not as frequent and, even when they occur, can generally be easily detected and quickly resolved by production workers [10].

Major nonconformities may include issues such as [16]: documentation problems, lack of management control, an absence of internal audits, inadequate employee training, and lack of customer feedback monitoring. These are systemic nonconformities that, starting from top management, affect all business processes, including the manufacturing process.

Examples of minor nonconformities in an industrial setting might include missing training records or invoicing errors, while in the manufacturing process, minor nonconformities might manifest as unnoticed nonconforming products or improperly calibrated devices. If a minor nonconformity recurs too frequently, it is then considered a major nonconformity. For example, if a nonconforming product is not identified and removed over a long period, it is treated as a minor nonconformity. Conversely, if this issue happens daily, it indicates a more serious problem and is regarded as a major nonconformity [16].

This paper focuses on nonconforming products. It is considered essential for every company to establish and regularly update procedures for product control to ensure that products meet defined requirements. Nonconformity of products can result from defects arising in the manufacturing process, non-compliance with standard requirements, and issues with material supply. Additionally, product nonconformity may involve physical damage, dimensional irregularities, non-compliance with technical specifications, or lack of functionality. Such products do not meet the expected standards and are unacceptable to customers.

Product nonconformity can negatively impact a company's reputation and may lead to customer loss or, at best, merely result in fines. Therefore, it is crucial to identify and address nonconformities promptly to ensure product quality and customer satisfaction. Implementing appropriate control measures and procedures can help reduce the number of nonconformities and enhance manufacturing process efficiency.

Establishing a procedure for managing nonconforming products is crucial for both the manufacturing process and overall business operations. This procedure allows for effective and prompt responses to nonconformities in the manufacturing process and prevents them from reaching end customers. The process for handling nonconforming products is illustrated in Figure 1.

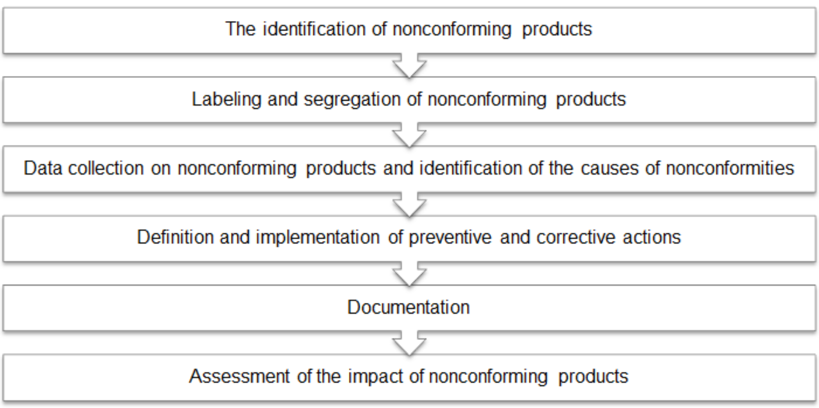

Figure 1 *Procedure for managing nonconforming products [13]*

When nonconforming products are identified, it is necessary to implement a system for labelling and segregating these products to prevent their mixing with conforming (correct) products. The labelling of nonconforming products should be clear and visible to avoid any confusion. The collection of data on nonconforming products involves gathering and documenting information about the nonconformities, including details about the product, the time and place of detection, as well as other relevant information. This data is essential for the successful identification of the causes of nonconformities and the implementation of preventive and corrective actions. In the end, all relevant information about nonconformities, identified causes, and applied measures should be properly and thoroughly documented. This allows for monitoring improvements, evaluating the effectiveness of the implemented measures, and assessing the damage potentially caused by the nonconformities [13].

3 OVERVIEW OF NONCONFORMANCE MANAGEMENT PRACTICES

The issue of nonconforming products has been extensively discussed in the relevant literature. Authors have approached this problem from various perspectives. For instance, papers [1, 17] have focused on the identification and analysis of nonconforming products. Additionally, a very common issue addressed in the literature is the prevention and management of nonconforming products [2, 11]. To approach the analysis of nonconforming products, some authors believe that it is essential to apply a specific methodology for their classification [4, 5].

In the automotive industry, the problem of nonconforming products has been primarily examined from the perspective of applying methods and tools for quality improvement [3, 7, 12]. Some authors have looked at this problem through the analysis of quality control efficiency [15], while a number of authors have approached the issue by proposing quality improvement strategies in the automotive industry [6, 14].

In the context of addressing nonconforming products, one method that has proven to be highly useful is Failure Modes and Effects Analysis (FMEA). FMEA enables the systematic identification of potential nonconformities in the manufacturing process and the evaluation of their potential effects on product quality. This method helps prioritize nonconformities based on their severity, occurrence, and detectability, allowing for a focus on the most critical areas and the development of effective strategies to address them. In the automotive industry, FMEA is used to enhance the reliability and efficiency of manufacturing processes, significantly reducing the number of nonconforming products and improving overall customer satisfaction. Relevant literature includes studies where FMEA has been applied in the automotive industry to address nonconformity issues in the production process [8, 9].

4 CASE STUDY

The subject of analysis in this paper is a company whose production facility is located in the vicinity of Kragujevac. The company’s headquarters is actually in Italy, while the production facility in the Republic of Serbia was established in 2013. The examined company is a supplier in the automotive industry's supply chain. In the Republic of Serbia, the company currently employs 52 workers, of which 42 are production operators. Thus, based on size criteria, this company can be classified as a small to medium-sized enterprise. Additionally, based on connectivity criteria, the company is part of a global supply chain. The company engages in mass production and operates at a mechanized technological level. Due to the confidentiality of the data and information presented, the name of the company under consideration is not disclosed.

The company's product range consists of 25 types of parts, predominantly aluminium, which are further incorporated into various types and purposes of vehicles. The main product the company supplies to its customers is a radiator grille for electric vehicles. The standard dimensions of the grille are 1400×1200 mm. Most of the products are manufactured using hydraulic presses, with additional manual and machine processing. In its production facility, the company currently operates 8 hydraulic presses, each with a maximum pressing force of 500 tons.

The production process involves several different job positions:

- Team Leader
- Quality Controller
- Forklift Operator
- Press Operator
- Maintenance Operator
- Warehouse Operator
- Other Staff
-

Production in this company is carried out in two shifts. Typically, the first shift operates at full capacity, utilizing the production equipment to its maximum and engaging approximately 2/3 of the production workforce. The second shift usually operates at reduced capacity, but, if necessary and in accordance with customer requirements, job reorganization is frequently carried out.

4.1 Analysis of the Occurrence of Nonconforming Products

The first issue addressed in this case study is the impact of nonconforming products on the effectiveness of the production process. The company produces 25 different types of products. The Quality Manager maintains records of the number of nonconforming products on a monthly basis for each product type. Figure 2 shows the number of nonconforming products over a one-year period.

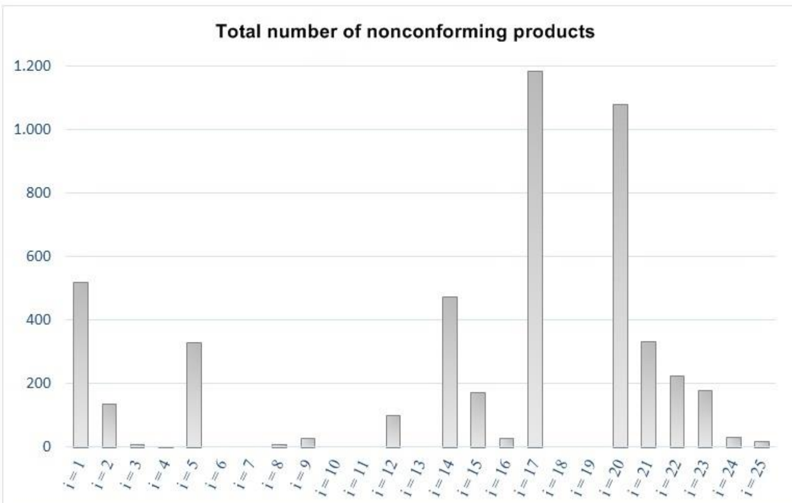

Figure 2 *Number of nonconforming products on an annual basis for each product type considered (Source: internal company documentation)*

Figure 2 shows that the highest number of nonconforming products is associated with product types $i = 17$ and $i = 20$. This information is particularly important for the Quality

Manager. Analysing the presented data can provide insight into problematic segments of the production process. In other words, based on the displayed data, the Quality Manager can identify critical points in the manufacturing process. Subsequently, it becomes possible to identify the causes leading to the occurrence of nonconforming products, which could result in specific changes to the production process or modifications to quality control procedures.

The occurrence of nonconformities in these two product types is most commonly due to:

- Incorrect press adjustments,
- Improper material placement, and
- Inadequate initial quality control.

In this case, it is crucial to conduct regular and proper quality control, as well as to train operators to recognize and promptly report any irregularities in the product manufacturing process. Additionally, proper maintenance and adjustment of presses and auxiliary tools are essential.

For the Procurement and Logistics Manager, the number of nonconforming products on a monthly basis is also important, especially for the aforementioned two product types. The number of nonconforming products on a monthly basis for product types $i = 17$ and $i = 20$ is shown in Figure 3. The availability of this information is crucial for procuring raw materials and optimizing the logistics process. This enables the Logistics and Procurement Manager to manage inventory more efficiently, which further leads to a reduction in operational costs.

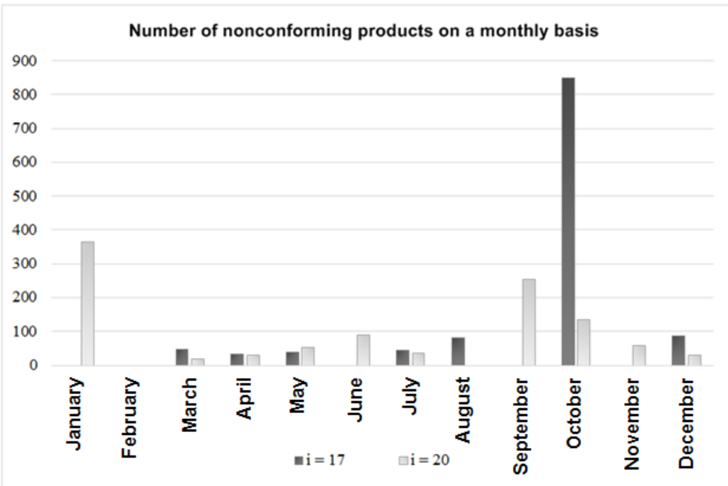

Figure 3 *Number of nonconforming products $i = 17$ and $i = 20$ on a monthly basis (Source: internal company documentation)*

Figure 3 shows a histogram of the quantity of nonconforming products $i = 17$ and $i = 20$ on a monthly basis. The highest number of nonconforming products for product type $i = 17$ occurs in January, while the highest number for product type $i = 20$ occurs in October. Based on these results, it can be concluded that organizing production was most demanding during these two months of the year under review. In other months, the impact of nonconformities of these two products on the effectiveness of the production process can be considered minimal. For example, issues with nonconforming products in the production process, aside from January and October, could have been easily resolved if there were safety stock of the necessary raw materials.

The number of nonconforming products primarily affects the effectiveness of the production process, which, in turn, impacts all business processes within the company. A high number of nonconforming products certainly leads to the inability to achieve set business goals.

4.2 Costs Arising from the Occurrence of Nonconforming Products

For the analysis of company operations, besides the quantity of nonconforming products, it is important to consider the costs incurred as a result of their occurrence. Figure 4 shows the costs associated with nonconforming products, specifically the costs directly related to the procurement of raw materials that remain unused as nonconforming products. These costs are expressed in euros and are presented for a period of one year.

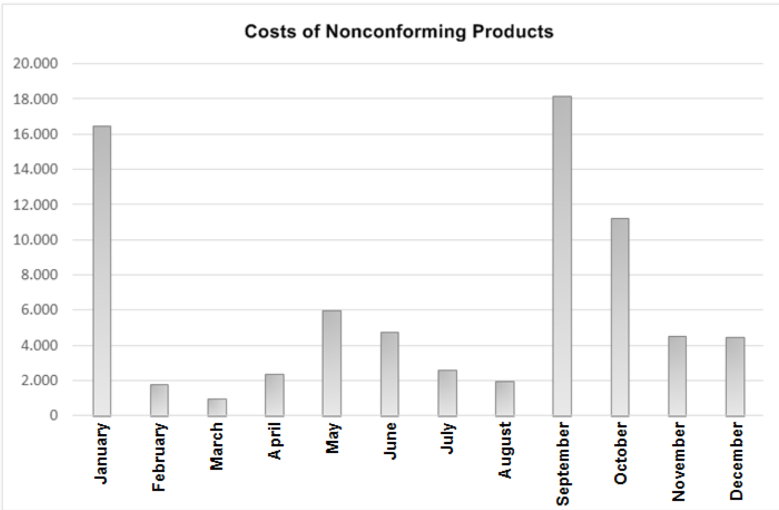

Figure 4 *Costs Arising from the Occurrence of Nonconforming Products, Expressed in Euros*

Figure 4 shows that the costs arising from the occurrence of nonconforming products are highest in September, January, and October. However, these costs should not be disregarded in other months. For the Finance and Administration Manager, this information is crucial as it directly impacts the overall business operations of the company.

Through further analysis of the costs of nonconforming products, their interval assessment has been determined. This analysis was performed using the descriptive statistics option within the Data Analysis package of Microsoft Excel, as shown in Figure 5.

Mean	6291,382423
Standard Error	1685,443922
Median	4510,46127
Standard Deviation	5838,549012
Sample Variance	34088654,56
Range	17208,65785
Minimum	991,04068
Maximum	18199,69853
Sum	75496,58907
Count	12
Confidence Level(95.0%)	3709,63706

Figure 5 *Analysis of Costs Arising from the Occurrence of Nonconforming Products Using Descriptive Statistics (Data Analysis, Microsoft Excel)*

From the analysis shown in Figure 5, the interval estimate for the costs arising from the occurrence of nonconforming products is:

$$6291{,}382 - 3709{,}637 \leq \mu \leq 6291{,}382 + 3709{,}637$$

$$2581{,}745 \leq \mu \leq 10001{,}019$$

Based on these results, it can be concluded that the expected value of the costs on a monthly basis may vary in the range from approximately 2500 to 10000 euros. In other words, these funds should be planned in advance and considered part of the financial plan.

4.3 Relationship between the Number of Nonconforming Products and Generated Costs

To improve product quality, the Quality Manager needs to take measures that will lead to a reduction in the number of nonconforming products. On the other hand, the implementation of these measures depends not only on the number of such products but also on the costs incurred due to their existence. To determine an appropriate set of measures, it is necessary to establish whether there is a correlation between the number of nonconforming products and the associated costs. In this study, the correlation was determined based on the correlation coefficient between these two variables (Tables 1 and 2).

Table 1 Quantity and Costs of Nonconforming Products at the Annual Level (Source: Internal Company Documentation*)*

i	Quantity	Costs (Euros)	i	Quantity	Costs (Euros)
$i = 1$	519	2711	$i = 14$	476	1571
$i = 2$	137	204	$i = 15$	173	501
$i = 3$	10	1	$i = 16$	29	87
$i = 4$	2	0	$i = 17$	1185	4235
$i = 5$	332	113	$i = 18$	0	0
$i = 6$	0	0	$i = 19$	0	0
$i = 7$	0	0	$i = 20$	1082	46425

$i = 8$	10	8	$i = 21$	334	11021
$i = 9$	30	7	$i = 22$	225	4051
$i = 10$	0	0	$i = 23$	179	2425
$i = 11$	0	0	$i = 24$	31	1397
$i = 12$	101	332	$i = 25$	20	404
$i = 13$	0	0			

Table 2 Correlation between Quantity and Costs of Nonconforming Products (Data Analysis, Microsoft Excel)

	Costs	Nonconforming Products
Costs	1	
Nonconforming Products	0.7	1

The correlation coefficient between these two variables is 0.7. This result indicates a strong correlation between the number of nonconforming products and the associated costs. As the correlation coefficient is positive, it can be concluded that products with a higher incidence of nonconformity also generate the highest costs. Therefore, it is essential to first address the measures for the products with the highest incidence, specifically products $i = 17$ and $i = 20$, for which the cost per nonconforming product is 3.57 and 42.9 euros, respectively.

In addition to the above, several other products also require special attention. The raw materials for products $i = 24$, $i = 25$, and $i = 21$ are relatively expensive, so preventing nonconformities in these products should be prioritized through appropriate control measures. Notably, product $i = 21$ has generated significant costs in the previous period.

Based on this analysis, the Quality Manager can optimize the resource usage necessary for addressing nonconformity issues. This optimization can be achieved by improving the quality control processes, organizing employee training, and implementing other preventive measures. Additionally, it is important to note that managing nonconforming products is a continuous process requiring the engagement of all business processes, from raw material procurement to production, quality, logistics, and sales.

5 CONCLUSIONS

In this study, an analysis of nonconforming products was conducted. Based on the records of nonconforming products, which were kept on a monthly basis over the course of a year, the trend of their occurrence was analysed. Additionally, an analysis of the costs arising from nonconforming products was performed. Correlation analysis revealed that the types of nonconforming products that occur most frequently also generate the highest costs. This indicates that implementing appropriate preventive measures to prevent these nonconformities would have a significant impact on reducing costs.

Some of the measures that should be undertaken to reduce the frequency of nonconforming products include adequate employee training, defining and strictly adhering to operational procedures, proper adjustment and preparation of machinery and equipment, and establishing a reliable quality control system. If any of these measures are not implemented or are not executed properly, nonconforming products will continue to occur.

To further enhance the effectiveness of the measures taken to address nonconforming products, it is crucial to establish a continuous improvement loop within the organization. This involves regularly reviewing and updating the preventive strategies based on feedback and new data. Engaging all relevant departments, such as procurement, production, quality control, logistics, and sales in this process ensures a holistic approach to problem-solving.

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