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POWER UNITS FOR FUTURE - QUO VADIS?[1]

Dušan Gruden

UDC:62.1.436;504.75

Summary

For decades, discussions on vehicular traffic have been followed by forecasts on limited reserves of oil and other fossil fuels, by increased demands regarding the vehicle's economy and by more and more severe legal regulations regarding reduction of CO2 emission and toxic components in exhaust emission. The question of which power units will drive the automobiles of the future or which will replace the piston internal combustion engine in the next 10 to 15 years, has not lost its actuality even after 130 years of engine development. Energy supply has the key role in answering this question. Forecasts that the known reserves of oil will be spent in the next 30 to 40 years and that the four-stroke piston engine will be replaced by better power units have been repeated for several decades.

Key words: IC engines, downsizing, hybrid drive, fuel cell, electric motor, oil, biofuel, synthetic fuels

POGONSKE JEDINICE ZA BUDUĆNOST - QUO VADIS?

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Rezime

Diskusije o automobilskom saobraćaju su već decenijama praćene progno-zama o ograničenosti rezervi nafte i drugih fosilnoih goriva, povećanim zahtevima u pogledu ekonomičnosti vozila, kao i u sve strožijim zakonskim propisima u pogledu smanjenjea emisije CO2 i toksičnih komponenata u izduvnim gasovima. Pitanje, koji će pogonski agregati pokretati automobile u budućosti, odnosno ko će zameniti klipni motor sa unutrašnjim sagorevanjem u sledećih 10 do 15 godina nije izgubilo na svojoj aktuelnosti ni posle 130 godina razvoja motora. Ključnu ulogu u odgovoru na to pitanje ima snabdevanje energijom. Prognoze, da će poznate rezerve nafte biti istrošene za 30 do 40 godina i da će četvorotaktni klipni motor biti zamenjen boljim pogonskim agregatima, ponavljaju se već više decenija.

Ključne reči: motor SUS, downsizing, hibridni pogon, goriva ćelija, električni motor, motorna vozila, bio goriva, sintetička goriva

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POWER UNITS FOR FUTURE - QUO VADIS?

Dušan Gruden*[1], *Phd full professor

UDC:62.1.436;504.75

1. HISTORY - A VIEW AT THE PAST

For almost four decades, from the first issue of the journal „Motor Vehicles and Motors“ (1975), which name has become the name of the scientific meeting held here, in Kragujevac, the question has been set on which power units will drive the automobiles of future, that is which will replace the piston internal combustion engine (IC engine) in the next 10 to 15 years? This question has not lost its actuality even today and the generation of engineers and scientists currently engaged in these problems has an impression that development had never been as dynamic and diverse as nowadays.

The French writer, André Malraux, had said: “If you want to read the future, then you should scroll through the past!”

For over 130 years, the piston IC engine has been without the competition as a power unit, not only for motor vehicles - automobiles, but in many other areas of human activities. During historical development, many other power machines (Figure 1) have been tested, none of which, with a sum of its features, had succeeded to establish itself in any field of application of the IC engines [1].

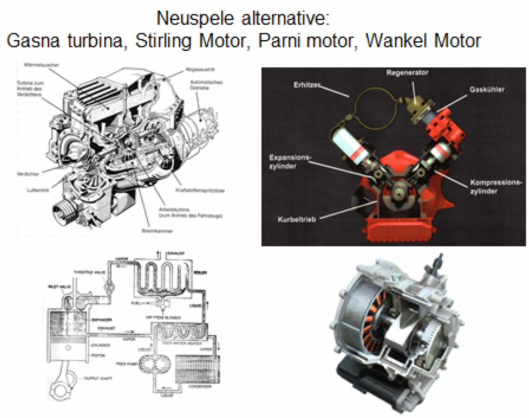

***Figure 1** Unsuccessful alternatives*

For several decades, discussions on vehicular traffic have been followed by forecasts on limited reserves of oil and other fossil fuels, by increased demands regarding the vehicle’s economy and by more and more severe legal regulations regarding reduction of toxic components exhaust emission and carbon dioxide (CO_2) emission.

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2. THE STATE NOWADAYS

From year to year, the modern vehicles are safer, more reliable and their negative impact on environment is getting smaller. In spite of this, emission and consumption must be further reduced so their impact on human environment would be as small as possible.

It is very interesting that, despite huge efforts to replace the conventional piston engine with some other, better unit, and despite the forecasts that it could not fulfil more and more severe legislation set in front of the automobile, it has shown a very great development potential that has enabled it to fulfil all demands set before it so far.

Pretty picture of the achieved engine progress is painted by Porsche 911 vehicle which has been evolutionary and continually developing for 45 years. At the first test of exhaust emission performed in 1966, a former Porsche 911 vehicle with the engine volume of V_H = 2.0 dm^3 and the power of 130 HP (96 kW), has had fuel consumption of 15.4 l per 100 km. The modern Porsche 911 vehicle has engine volume of 3.6 dm^3 and the power of 295 kW (400 HP) and the fuel consumption (at the same test) amounts to 8,2 l per 100 km, that is, it is 47% smaller than its predecessor (Figure 2).

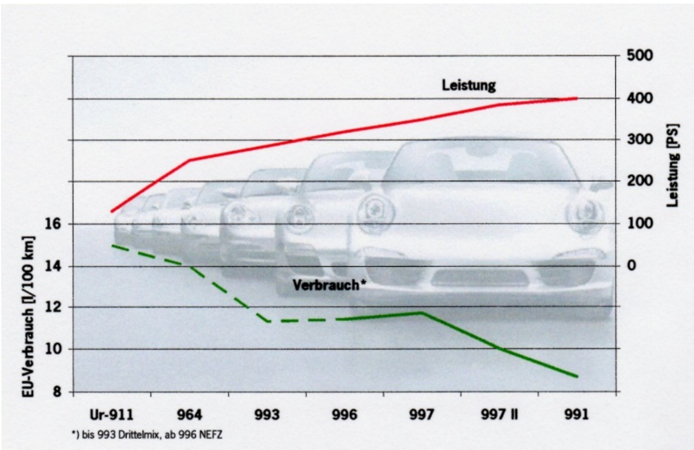

Figure 2 *Historical development of Porsche 911 fuel consumption*

Of course, the toxic components emission from the exhaust emission in modern engines is incomparably smaller than in their predecessors. The same amount of legally limited components (CO, HC, NO_x) emitted by a single automobile half a century ago is being emitted by 300 to 500 modern automobiles.

3. ECOLOGICAL DEMANDS SET BEFORE THE MOTOR VEHICLE

After the distance travelled for almost 140 years, IC engine, in its further development, must take more into account the social framework in which it is placed. The central place thereby belongs to:

- increasing anthropogenic emissions, in context with discussions on climate change,
- demographic changes in the world, with increasing number of population on Earth and with moving the age limits,

- increasing urbanization and
- limited reserves of fossil fuels that follow this development.

These social frameworks, as well as forecasts that both passenger and freight traffic will further grow in the next 20 years, explain the reason for more severe legislation set before the automobile and its power unit.

Firstly, there is reduction of energy consumption and closely related reduction of CO_2 emissions. Member countries of the European Union have adopted the so-called "20-20-20" program in 2007 or the program with the goal to reduce energy consumption by 20%, to reduce CO_2 emission by 20% and to cover 20% of energy needs with so-called green energy, *i.e.* energy from the renewable sources.

Satisfaction of the legislation on reducing CO_2 emissions is one of the greatest challenges in development of new propulsion systems for vehicles. The European Union and Japan lead today in regard to the law on the reduction of CO_2 emissions; China prepares stricter regulations, followed by the USA and the other countries (Figure 3).

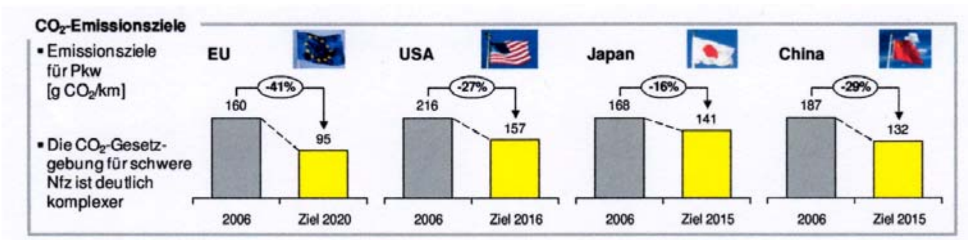

Figure 3 *CO_2 emission reduction goals*

According to current plan, the new passenger vehicles in the European Union will be allowed to emit, on average, just 95 gCO_2/km (≈ 3.8 l/100km) from 2020 and this value should be reduced to 70 gCO_2/km (≈ 2.8 l/100 km) from 2025. Currently, average value of CO_2 emission from passenger vehicles in Germany amounts to 148 gCO_2/km (≈5.9 l/100 km).

Of course, it is required that the automotive industry will continue to meet the ever harsher legislation on reduction of toxic components in the exhaust gases: carbon monoxide (CO), unburned hydrocarbons (HC), nitrogen oxides (NO_x) and particulate matter (PM). In the European Union, the strongest legal regulations for now, known as Euro 6, are predicted for the year 2014, while in the United States, with legal regulation LEV 2 (**L**ow **E**mission **V**ehicles), the California Air Resources Board (CARB) has introduced the world's sharpest legal rules defined as SULEV (**S**uper **U**ltra **L**ow **E**mission **V**ehicles), Figure 4.

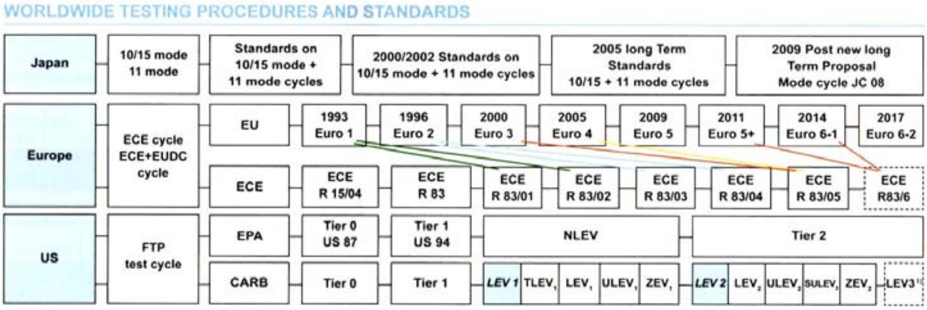

Figure 4 *Legal regulations on exhaust emission in the world*

In order to classify the vehicle as PZEV (**P**artial **Z**ero **E**mission **V**ehicle), the limit value recommended for SULEV vehicles must be guaranteed for 15 years of the vehicle operation and for the path of 150,000 miles (240,000 km) travelled. Vapour emission of the stationary vehicle must be 0 and the vehicle emissions are controlled by OBD II (**O**n **B**oard **D**iagnostic II) system.

At the end of their "life" (End of Live Vehicles, ELV), passenger vehicles must meet the legal requirements on recycling, which in the European Union say that, from 2015, only 5% of vehicle weight may go to landfills, while 95% of vehicle weight must be used, either as material or energy.

4. TOPICS IN MODERN DEVELOPMENT OF IC ENGINES

To respond to the demands that are set upon it, the IC engine is still continuously and evolutionary developing. Its future depends, above all, on the further increase of efficiency in all areas of work. Former studies show that there is still considerable potential for further reductions in fuel consumption and CO_2 emissions that is estimated as at least 15% for both Otto and Diesel engines in the short-term.

4.1. Downsizing

Under pressure from legislation, the main priority in the development of the engine is to reduce the fuel consumption. One of the attractive ways that is being increasingly used is the so-called "downsizing" or reducing the engine capacity, while maintaining the desired power with the help of engine turbocharging (Figure 5). The fact that it is more important what cylinders do and not how many of them there are in the engine is slowly being accepted by the professional world.

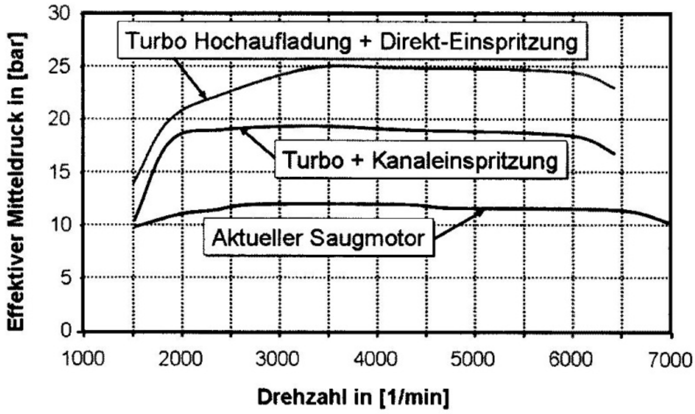

Figure 5 *Characteristic curves of engine torque with turbocharging and with free intake*

"Downsizing" process, *i.e.* a turbocharged engine with a relatively small volume, had allowed the mass use of diesel engines in passenger vehicles 20 years ago and completely had thrown out the free intake diesel engines from applications, even in the field of commercial vehicles. While this process in diesel engines have long been the standard, manufacturers of vehicles with Otto engines introduce in their programs more engines with reduced volume and charging, in order to meet CO_2 emission value of 130 gCO_2/km (≈ 5.2 l/100 km) required by the European Commission, Figure 6, Table 1.

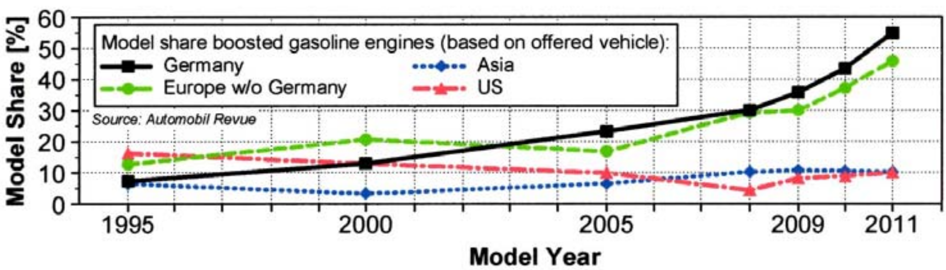

Figure 6 *Share of „downsized“ Otto engines in the world market*

For the current generation of "downsized" turbocharged engines, the displacement is reduced by about 75% compared with the displacement of the engine having the same power, but with a free intake.

Table 1 *The producers of turbocharged „downsized“ engines*

Producer	Displacement VH [dm^3]	Number of cylinders	Power [kW]	CO_2 [g/km]
Ford	1.0	3		
Kia		3		85
Fiat	0.875	2		
Nissan	1.2	3		95
Renault	0.9	3	66	

VW	0.8	2	35	

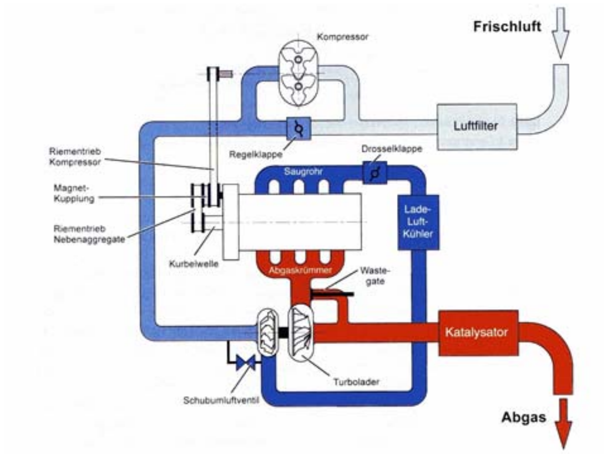

***Figure 7** Engine with double turbo charge (source: VW)*

While complex turbocharging systems with two (mostly turbo) compressors and direct injection at diesel engines have become almost standard, they begin to be more and more applied in Otto engines, more often combined with variable valve timing scheme.

4. 2 Working process

Success of internal combustion engines decisively depends on the combustion process, which takes place in the cylinder. Efforts to reduce the losses during engine operation have led to the development of valve drives with variable valve timing.

Direct fuel injection with pressures of over 2000 bars has become a standard for diesel engines, while direct fuel injection with pressures of over 100 bars is more and more applied in Otto engines as well. Otto engines with two injection systems increasingly appear on the market (Figure 8).

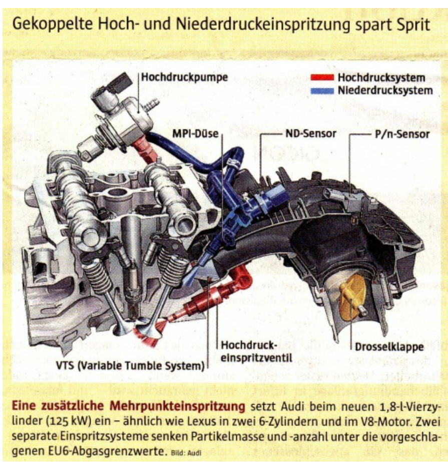

***Figure 8** Engine with two injection systems (Audi)*

The first system allows the injection of the main fuel quantity in the initial stage of realization of the mixture. It is used to achieve homogeneous, lean mixture of fuel and air. The second, smaller amount of fuel is injected just before the combustion of the mixture, in order to facilitate its safe ignition.

In a recent dispute between the automotive industry and an inventor, it has been found that modern diesel engines operate on the principle, which has been described by so-called thermodynamic cycle with isothermal expansion (Figure 9).

This thermodynamic cycle had been patented back in 1975. The goal was to achieve high efficiency with the least emissions of nitrogen oxides (NO_x) and unburned hydrocarbon (CH). In order to achieve this, it is necessary that the first, main amount of energy (fuel energy, Q_1') is introduced in the first phase of the cycle, at constant volume, in order to reduce maximum combustion temperatures, responsible for the creation of NO_x, while the second, smaller quantity of energy (Q_1") is introduced during the expansion, which provides relatively high temperature, important for further oxidation of unburned hydrocarbons during the expansion stroke and in the exhaust system.

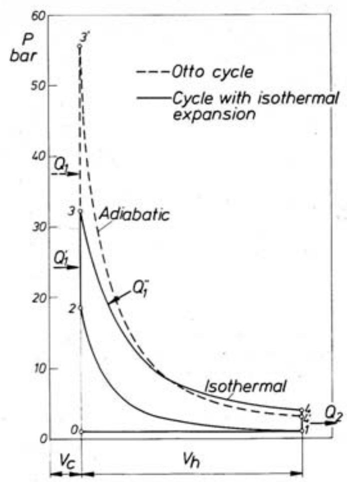

Figure 9 *Otto cycle and cycle with isothermal expansion [4,5]*

It was found that all modern Otto and Diesel engines with multiple injection operate in accordance with this thermodynamic cycle.

4. 3 Reducing mechanical losses

Many of the measures to improve IC engines have already been applied after years of effort, so that it becomes increasingly difficult to devise the ways for their further optimization. Especially for Diesel engines, options for the further improvement of the combustion process are almost exhausted, so it's harder to develop engines using conventional methods and all possible potentials where further optimization is still possible are searched for on the engine as a whole. Beside the efforts in improving the combustion process and reducing the engine losses, a great potential is still seen in reduction of mechanical losses and friction (Figure 10) [6].

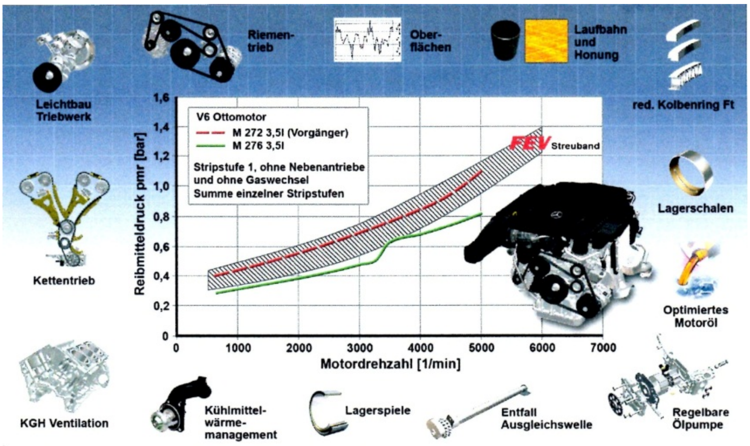

***Figure 10** Measures for reduction of engine mechanical losses*

In addition to new technological procedures in the processing of cylinder liners, piston group and bearings, this area also includes the reduction of piston mechanism weight, which significantly reduces the inertial forces at higher engine speeds, and thus also reduces the mechanical losses.

In addition to optimization of the engine, further development of all the important components that affect not only the combustion process, such as the injection system, but the entire operation of the engine, particularly the system for lubrication and cooling, is required. Significant potential for reduction of mechanical losses is also in variable drive of auxiliary units on the vehicle: oil pump and water pump, which are involved in the work according to engine needs. Neglecting these potentials will not allow further necessary development of IC engines.

4. 4 Modular design of engine

Of course, all these necessary measures to improve the engine, in addition to large investments in research and development, have resulted in the increase of production costs, including the price of the engine.

At the end of 1960-ies, before the introduction of legislation on reduction of exhaust emissions and fuel consumption, the production cost of one full 8-cylinder engine with the exhaust system in the U.S. was around 160 to 180 US \$. The production cost of modern engines that meet all legal requirements, rises to several thousand dollars.

Despite the steady increase in production costs, continuous efforts are being made to reduce, as far as possible, the production price of the engine. Similarly, as many years ago, when the principle of "common platform" has been introduced in the construction of vehicle in order to reduce production costs, the modular design principle is also observed in the construction of the engine (Figure11), which represents a common base to build a family of engines with different displacements and numbers of cylinders by one producer.

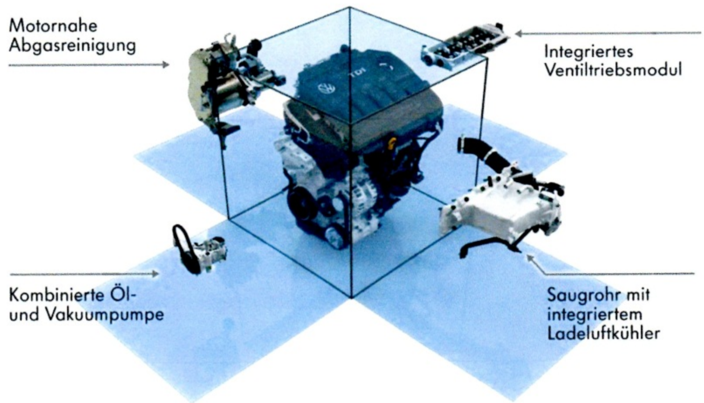

Figure 11 *Modular design system (source: VW)*

Modules - joint groups and sets - are divided into groups of the base engine (piston mechanism, cylinder head, valve mechanism) and sets that are built on the engine (elements for filtering of the exhaust emission, intake system with integrated cooler for cooling the air for turbocharging).

Even in the engine concept phase, it is taken into account to modularize the engine to the greatest extent. In addition to reducing the production cost of the engine, modular construction offers possibility to produce the engines in different places and for different markets and applications. With no major losses in time, the engines can be built according to the different requirements that are set in the world in the standardized production plants, with always optimal technical solutions.

5. HYBRID DRIVE

In the last ten years, the vehicle power units with two units: IC engine and electric motor have been recognized as the only alternative to a purely engine power unit that is produced in series.

5. 1 Micro hybrid

The initial step in this direction, the so-called “micro hybrid” or start-stop automatics (start-stop generator) has become an integral part of modern engines. In the city driving, fuel savings with this system amount to about 3% to 5%.

5. 2 Mild hybrid

A "Mild Hybrid", in addition to the start-stop generators, uses power of the electric motor at start-up and acceleration of the vehicle. The energy needed for the electric motor is taken from a relatively small battery. Starting the vehicle for a long distance is not possible for this system, because the battery is too small. Very often, braking energy recuperation is frequently used in this system during deceleration. Fuel savings during city driving and during new European driving cycle (NEUDC, **N**ew **E**uropean **U**rban **D**riving **C**ycle) are about 15% to 20%.

5. 3 Full hybrid vehicle

Unlike "mild hybrid" vehicles, "full hybrid vehicles" have a battery of sufficient size, so the electric motor can drive the vehicle on a relatively long distance. Electric motor power is between 20 kW and 60 kW. Fuel savings or CO_2 emission reduction in the new European test are between 20% and 50% (Figure 12).

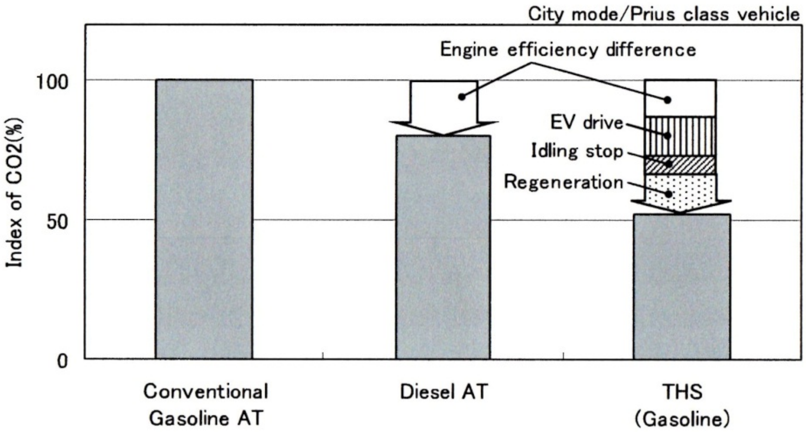

Figure 12 *Reduction of CO_2 emission in hybrid vehicle (Toyota)*

The key features of full hybrid vehicles are: pure electric driving ability at a certain distance, the exclusion of both engines operation at idle speed and braking energy recuperation.

5. 4 Range Extender, Plug in hybrid

High production costs and the price of full hybrid vehicles and their relatively small radius with pure electric drive, and the fact that most of the vehicle rides during the day do not exceed 30 to 60 km, led to hybrid vehicles solution with so-called "range extender" or "plug-in" systems.

Electric battery and electric motor power are dimensioned at these vehicles in a way that enables pure electric driving for about 50 to 60 km, for example, going to work and returning home, where the battery is charged during the night to ride the next day. If the vehicle has to take more distance or the energy consumption is increased (lights, wipers, heating, air-condition), then the additional IC engine is engaged in operation, which is so dimensioned that allows unobstructed started drive ("range extender").

"Range extender" - IC engine, as modular part of a hybrid concept, provides compensation of known deficits of purely electric drive. To support the electric motor, one relatively small and light IC engine (V_H = 0.3 to 0.8 dm3) with low production costs is used (Figure 13).

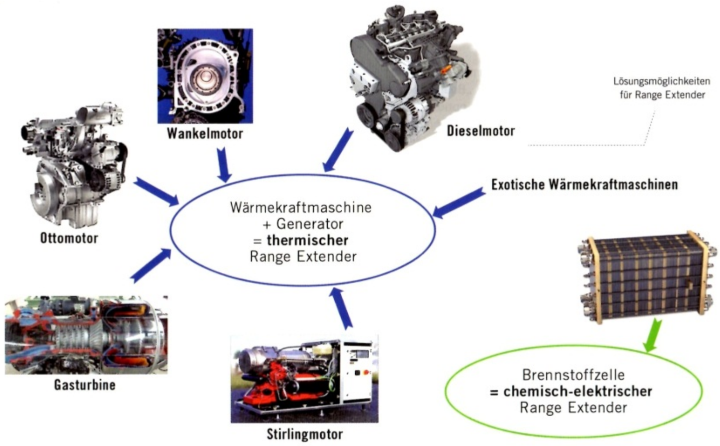

***Figure 13** Researched power units as „range extender“*

Almost all alternative power units that were once offered to drive motor vehicles now are being researched as possible variants for the position of the "range extender" engine (Vankel engine, two-stroke engine, gas turbine, etc.).

Despite all the efforts to promote the hybrid vehicles at the market as a serious alternative to a pure engine drive, their share in the total vehicle fleet is still very small: At the beginning of 2012, the number of new hybrid vehicles in traffic in Germany totalled 12.622. The share of hybrid vehicles in Germany in total number of passenger vehicles is now about 0.80%.

6. ELECTRIC VEHICLES

In its efforts to reduce CO_2 emissions, the current policy in many countries is very focused on electric drive motor vehicles. After the nuclear disasters in the USA (Three Mile Island), Ukraine (Chernobyl) and Japan (Fukushima), electro-mobility provides opportunities to reduce CO_2 emissions only if the electric energy is derived from the so-called renewable energy sources (water, wind, solar energy). Since more than half of electric energy today is generated using fossil fuels (coal, oil, natural gas), the switch to electric vehicle drive does not reduce CO_2 emissions, but, on the contrary, induces its global increase.

Electric automobiles are not the discovery of today’s techniques. The first electric drives were already built between 1835 and 1839 [9]. At the beginning of the 20th century, more vehicles were powered by electric motors than with IC engines. Advantages and disadvantages of electric vehicles at that time were similar to those that now follow the development of these automobiles.

Plans of some countries regarding the introduction of electric vehicles in the traffic sounded very seriously a few years ago. Federal Republic of Germany had a goal that 1 million vehicles in the traffic should be with pure electric drive until 2020. In 2030, that number should be increased to six million electric vehicles on German roads. China has planned that, by 2015, half a million electric vehicles should be found in traffic, and by 2020, this number should be increased by ten times - to five million vehicles [10].

The main drawbacks of electric vehicles are found, same as 150 years ago, with the battery or accumulator of electric energy or with its capacity, weight and production cost (Figure 14).

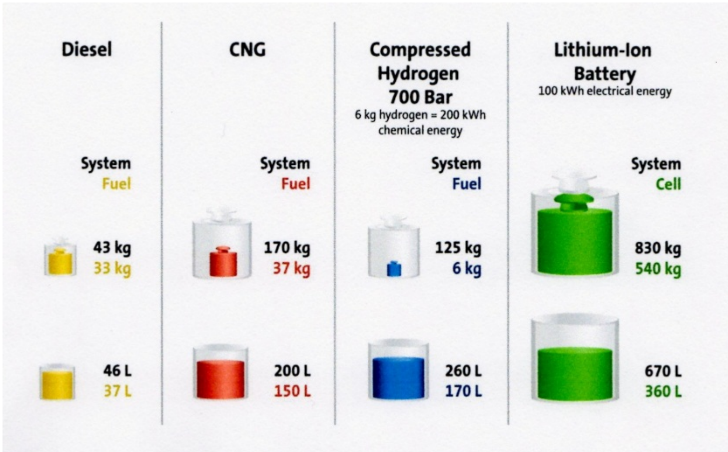

Figure 14 *Weight and volume of energy sources (source: GM)*

The capacity of the today's best batteries, Lithium-ion batteries, amounts to 0.5 kWh/l, which is 20 times less than that of gasoline and diesel fuel (9-10 kWh / l). Low capacity means great battery weight for any longer distances, which vehicle shall travel. Today, they amount to a maximum of 120 to 150 km at a flat, dry road, on a sunny, not too hot day. Turning on the lights, wipers, heating or air conditioning of the vehicle, reduces the potential radius of the vehicle by half.

The second major handicap of electric vehicles is the high production cost of battery that has been moving between 600 and 800 €/kWh for years, which is 20 to 25 times greater than for IC engines. The goal of battery price reduction is set between 200 and 300 €/kWh, which is expected to happen in about 10 years.

Electric vehicles producers are trying to reduce production costs for both electric and hybrid vehicles with using electric modules, which can be integrated into existing vehicle platforms. Thus, the modular building system is one of more important paths of development for electric automobiles.

Listed shortcomings of electric drive, along with a lack of necessary infrastructure and standards for supply stations, cables, sockets (Figure 15) and still great amount of time of several hours for charging the battery, have led to slow correction of ambitious plans on the share of electric vehicles in traffic.

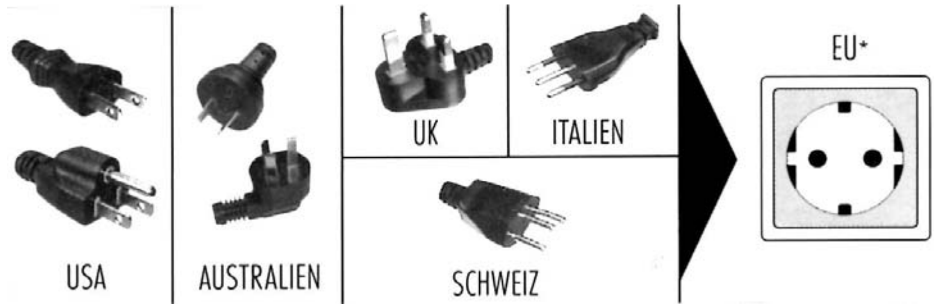

Figure 15 *After 150 years: different sockets for electrical devices in the world*

In early 2012, there were only 4541 electric vehicles in traffic in Federal Republic of Germany or 0.01% of the total number of vehicles in traffic. In China, until this year, 7.000 electric vehicles were sold, representing 0.02% of the vehicles in traffic.

Most experts agree that, in the next 15 to 20 years, electric vehicles will not make a greater breakthrough in the market, but they will be used only for small, local fleets. According to Professor Lenz (Technical University of Vienna), electric vehicle will not at all contribute to improving the state of human environment, while, for vehicle buyers, it means a considerable increase of the price of vehicle that is significantly inferior in terms of all properties compared to IC engine drive [7].

7. FUEL CELLS

At the beginning of the 1990s, especially under the influence of development in Daimler Corporation, the electric vehicle drive with so-called “fuel cell” as a source of electrical energy, which was supposed to be the solution to drive the vehicle in the near future, had been euphorically presented. Electrical energy that serves to start the vehicle is produced by joining the fuel (hydrogen H_2) and oxygen from the air (O_2). The other product of this chemical reaction is water (H_2O), which has no harmful effect on the human environment.

Almost all vehicle producers have invested heavily in research and development of fuel cells. Predictions about the introduction of fuel cells in mass production were very optimistic (Figure16).

Predicted introduction of vehicles with fuel cells in serial production

Firm	Star of serial production	Prediction from year
Honda	2003	(2000)
Daimler	2004	(2000)
GM	2004	(1998)
Ford	2004	(2001)
Chrysler	2007	(1997)

Figure 16 *Predictions on introduction of fuel cells in serial production*

Most producers claimed that they will start the serial production in 2003 or 2004 [8, 9, 10]. To date, however, the basic weaknesses of the system have not been solved. High production rate (100 to 150 times higher than that of IC engines of the same power), lack of fuel (hydrogen has proved to be the only possible solution) and a complete lack of the necessary infrastructure for the production, distribution and storage of hydrogen have meant that the chances for application of fuel cells in road traffic are considered much smaller than the chances for battery-powered electric vehicles.

8. VIEW AT THE FUTURE

Accepting the advice from Andre Marlowe that "If you want to read the future, then you should scroll through the past", the past development shows that, so far, no alternative power unit has managed to threaten the primacy of IC engines. Today's parameters also show that the IC engine remains unchallenged power unit for motor vehicles.

Experts around the world mutually agree that the piston IC engine will maintain its dominant status for several decades as the main power unit for vehicles

8. 2 Fuels

They also agreed that fuels based on oil and other fossil sources will not be able to meet the growing global demands for energy alone, so the answer to the question which sources of energy will be available will also carry the answer to the question of which power units will drive the future vehicles. Apparently, in the future, several fuels with similar characteristics, originating from the different sources will simultaneously exist at the market.

Conventional fuels derived from crude oil globally remain, according to today's forecasts, as the main source of energy in the next 30 years (Figure 17) - the prognosis that has been repeated for over 50 years.

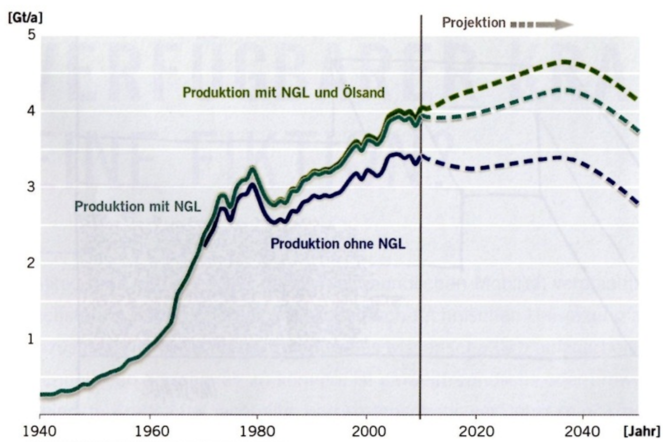

Figure 17 *Trends in exploitation of oil sources and natural gas sources*

Maximum exploitation of oil wells is expected in the next 15 to 20 years, after which there should be the reduction of oil supply, although the demands for energy continue to grow steadily. This fact means that the time of cheap oil is over. With the increase in prices of oil or fuel, many alternative fuels which could not, because of their high cost, be promoted at the market until now, become economically interesting.

Relying on the European Directive on fuel quality (2003/30/EG) and direction on the use of renewable energy sources (2009/28/EG), the European Commission, in its ambitious plan to support the use of renewable energy sources in traffic, follows a specific goal that, by year 2020, at least 10% of conventional fuels in the European Union is replaced by fuels from regenerative sources [14, 15].

In the first place, there are fuels from bio-mass as fuels from renewable sources. The goal of the efforts to introduce these fuels is to enable the traffic neutral in terms of CO_2 emissions. Since the gaining of the first generation of biofuels is potentially in competition with food production and is therefore subject to harsh criticism, experts are working intensively on the next generation of biofuels, where fruit of the plant will not be used to produce fuels, but only as plant waste material. Obtaining the biofuels from algae seems especially attractive looks and it has been intensively investigated.

Beside fuels from biomass, synthetic fuels, which are likely to have greater importance in the future than it is considered now, gain more and more importance. These include:

- ETBE - Ethyl-tertiary-butyl-ester,
- Synthetic fuel obtained from natural gas (GTL, **G**as **t**o **L**iquid) and
- Synthetic fuel obtained from coal (CTL, **C**oal **t**o **L**iquid).

Efforts to obtain synthetic or gaseous fuels from CO_2 emission, emitted by power plants and large industrial plants, by using excess electricity from regenerative sources or to conduct the "recycling" of CO_2, seem to be attractive (Figure 18).

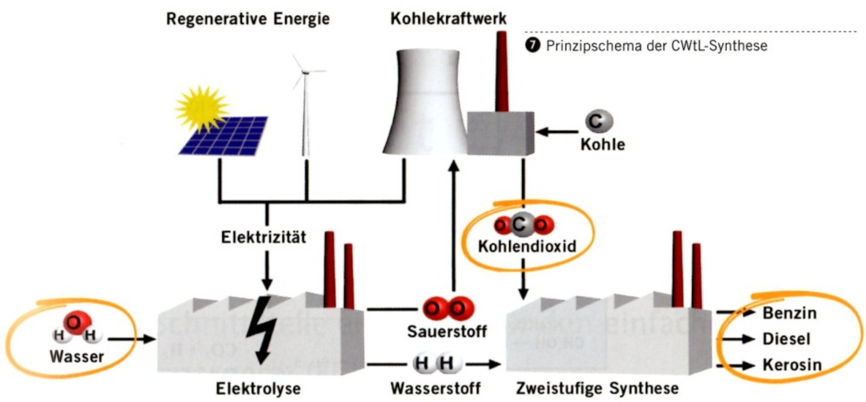

***Figure 18** Fuels from CO_2 and water*

9. CONCLUSIONS - MOTOR VEHICLE DRIVES IN THE NEXT 15 TO 20 YEARS

Since the liquid fuels from fossil and biogenic sources will be the main energy sources for road transport in the coming decades, the IC engine will keep its dominant position as a power unit for motor vehicles during that time. Of course, it will continue to be optimized in all phases of the operating cycles and in every detail of its construction (Figure 19).

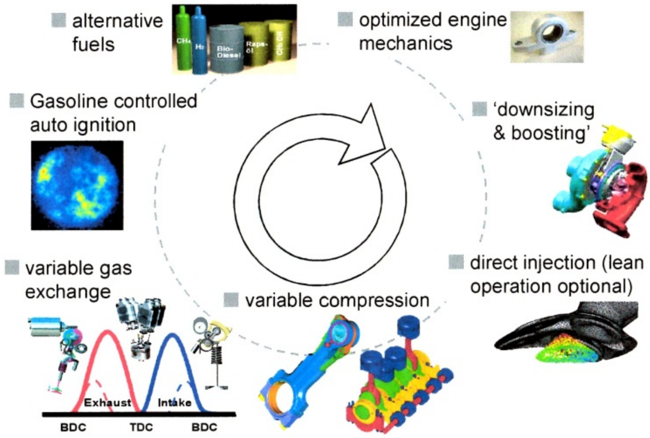

Figure 19 *Further optimization of IC engines*

The potential for further reduction of fuel consumption and CO_2 emissions is still large (Figure 20).

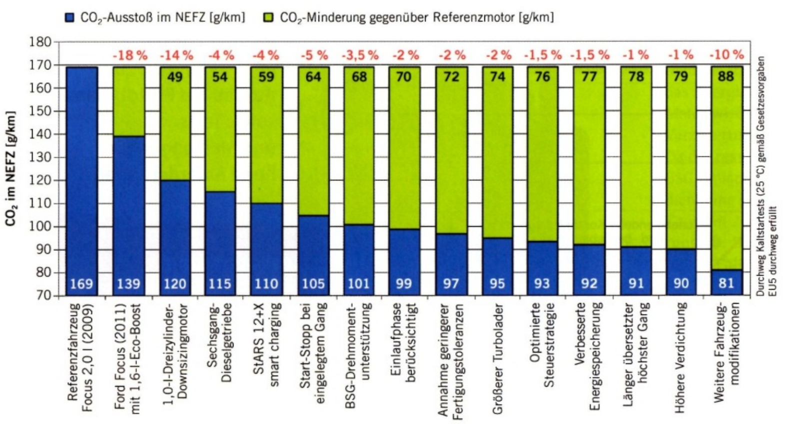

Figure 20 *Measures for further reduction of CO_2 emission and fuel consumption*

The start of power unit electrification, a hybrid drive, will probably remain a definitive solution for the foreseeable future.

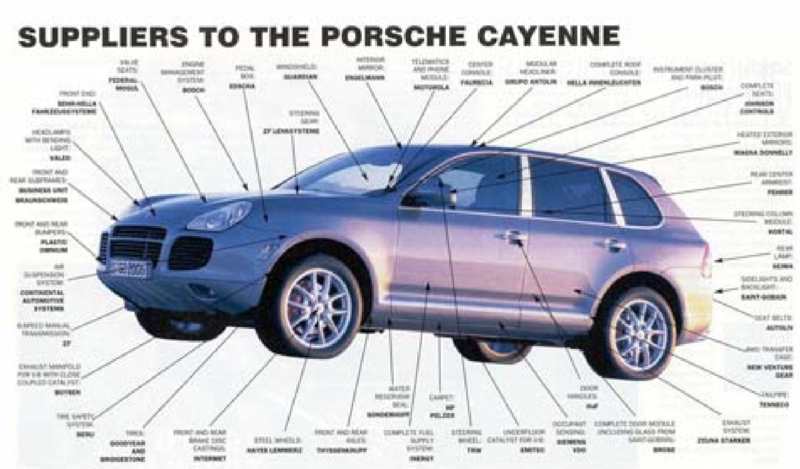

Figure 21 *Porsche and its suppliers (example of Cayenne)*

Shorter and shorter intervals available for product development and introduction of innovations in engine construction, along with the growing demands regarding the conservation of natural reserves of the planet, have led to the fact that collaboration between a producer and several partners from automobile industry has become an important factor in achieving success. Thereby, the cooperation with the so-called attached industry or the suppliers has the most important role. Over 70% of the value of an automobile is created today in the attached industry (Figure 21).

This picture will not be changed in the future.

Today, about 70 million of road vehicles are made annually in the world. In the meantime, China has become the largest producer, with annual production of over 16 million vehicles. In the next 5 years, the number of vehicles is projected to increase for more than 30% (Figure 22).

Reihung der Fahrzeugproduktion nach Ländern:

	2001		2011		2016	
Nr.	Produktions Land¹	Fahrzeuge p.a.	Produktions Land¹	Fahrzeuge p.a.	Produktions Land¹	Fahrzeuge p.a.
1	Japan	8.100.000	China	15.800.000	China	25.000.000
2	Deutschland	5.300.000	USA	8.400.000	USA	11.000.000
3	USA	4.800.000	Japan	7.500.000	Japan	8.800.000
4	Frankreich	3.100.000	Deutschland	5.800.000	Indien	6.200.000
5	Süd Korea	2.400.000	Süd Korea	4.400.000	Deutschland	6.200.000
6	Spanien	2.200.000	Indien	3.600.000	Süd Korea	4.600.000
7	Brasilien	1.500.000	Brasilien	3.200.000	Brasilien	4.400.000
8	UK	1.500.000	Mexiko	2.400.000	Mexiko	3.200.000
9	Kanada	1.200.000	Spanien	2.300.000	Thailand	2.800.000
10	Italien	1.200.000	Frankreich	2.200.000	Spanien	2.600.000
11	Belgien	1.000.000	Kanada	2.100.000	Frankreich	2.400.000
12	Russland	1.000.000	Thailand	1.700.000	Russland	2.400.000
13	Mexiko	1.000.000	Russland	1.700.000	Kanada	1.900.000
14	China	700.000	Iran	1.600.000	UK	1.700.000
15	Indien	650.000	UK	1.500.000	Iran	1.600.000

¹ (CBU & CKD)

Quelle: CSM Q2 /11

Figure 22*: Ranking-list of world vehicle producers*

More than 99% of the produced vehicles will be driven by further developed IC engines. This provides the engineers in the automotive industry with safe, interesting and intensive work in the coming time.

At the end, let's repeat the conclusions, which have been repeated for decades, since the first scientific meeting "Motor Vehicles and Motors":

- Four-stroke piston engine stays as main power unit for motor vehicles in the next 10 to 15 years.
- World oil reserves are secured for the next 30 to 40 years.

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NONLINEAR DYNAMICS OF HEAVY GYRO ROTORS[1]

Ljiljana Veljović, Dragan Milosavljević, Gordana Bogdanović, Aleksandar Radaković, Mirjana Lazić

UDC: 531.383

Summary

The rotors are basic parts in many mechanical systems, such as automobiles, airplanes, helicopters, and so on. The problem of rotor vibrations existed for a long time. Nevertheless, the nonlinear dynamic of motion of rotating parts in engineering such are rotors, coupled rotors, planetary gears or gyro–rotors is actual nowadays. Rotors are analyzed as discs which rotate around axes in various ways. In this paper the rotor is analyzes as a shaft – disc system. The support shaft is vertical while the rotor shaft is horizontal. They may be with section or without section. The disc is eccentric with eccentricity e. A system of nonlinear differential equations is determined. For different cases of eccentricity the motion character analysis is performed by means of phase trajectories. The method of phase plane and phase trajectories in researching nonlinear dynamics properties gives some phenomena such are their stability, transformation of homoclinic orbits, their appearance or disappearance.

For the special case when heavy disc is eccentrically and self rotation axis rotate in horizontal plane around vertical axis with constant angular velocity, a series of graphical presentation with changes of disk eccentricity are presented here.

Key words: rotor, nonlinear dynamics, eccentricity, phase plane, stability

NELINEARNA DINAMIKA TEŠKIH GIRO ROTORA

UDC: 531.383UDC: 621.112

Rezime

Rotori su bazični delovi mnogih mehaničkih sistema, kao što su automobili, avioni, helikopteri, itd. Problem vibracija rotora egzistira odavno. Bez obzira na to, nelinearna dinamika rotirajućih elemenata, poput rotora, spregnutih rotora, planetarnih prenosnika ili giro rotora aktuelna je i danas. Rotori se posmatraju kao diskovi koji mogu rotirati oko osa na različite načine. U ovom radu rotor se posmatra kao deo sistema osovina – disk. Osovina je vertikalna, dok je osa rotora horizontalna. Ove ose se mogu seći, a mogu biti i mimoilazne. Disk je ekscentričan (ekscentriciteta *e*). Dobijen je sistem nelinearnih diferencijalnih jednačina. Za različite slučajeve ekscentriciteta se analizira kretanje preko faznih trajektorija. Metod fazne ravni i faznih trajektorija, pri analiziranju nelinearne dinamike, omogućava analiziranje stabilnosti kao i transformaciju karaktera kretanja.

U specijalnom slučaju, kada ekscentričan disk rotira oko sopstvene horizontalne ose i zajedno sa njom oko vertikalne nepomične ose, dobija se serija faznih portreta za različite vrednosti ekscentirciteta.

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Ključne reči: rotor, ekscentricitet, fazna ravan, stabilnost, nelinearna dinamika

NONLINEAR DYNAMICS OF HEAVY GYRO ROTORS

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INTRODUCTION

Heavy bodies in motion, artillery projectiles in motion, rotors of turbines, different mobile installations on ships, aircraft propeller rotating possess special properties known as properties of gyroscopes. Tern or a toy top is just a simple well-known toy with unusual property that when it rotates by high angular velocity about its axis of symmetry, it keeps in the state of stationary rotation around this axis. This feature has attracted scientists around the world and as a result of year's research many devices and instruments are created; from simple to very complex structures, which operate on the principle of a spinning top that plays an important role in stabilizing the movement. Ability gyroscope that keeps the line was used in many fields of mechanical engineering, mining, aviation, navigation, military industry and in celestial mechanics. Gyroscopes are used to measure angular rotation rate in airplanes, spacecrafts, missiles, automobiles and even consumer electronics.

Gyroscope's name comes from the Greek words γυρο (turn) and σκοπεω (observed) and is related to the experiments that the 1852nd were painted by Jean Bernard Leon Foucault. The principle of gyroscope based on the principle of pseudo regular precession. Gyroscopes are very responsible parts of instruments for aircraft, rockets, missiles, transport vehicles, many weapons and robotics. Gyroscopes are used in several forms in transportation. Car engines act like big gyroscopes, so the racing industry takes them into consideration. Because the cars only turn in one direction, the gyroscopic force helps the car stay on the track. Motorcycle wheels also act as gyroscopes to make the bike easier to balance. Gyroscopic behaviour is used in the racing car industry because car engines act just like big gyroscopes. This has its uses, for example in the American Indy car racing. During the race the cars go round the circuit in one direction only. Because of the gyroscopic forces from the engine depending on whether the engine is spinning closewise or anti-clockwise the cars nose will be forced up or down. Providing the engine spins in the right direction it can help the car to stay on the track.

Gyroscopes are also used in monorails and ships to help them stay upright [17]. They are used in navigation to stabilize the movement of ships in a seaway, to change direction, and direction of angular and translator velocity projectiles, and in many other special purposes. Mathematical analysis of the two-wheeled vehicle gyroscopic stabilization problem first appears in [2], and more recently in [17], without derivation, or in [14], where the derivation is by use of bond graphs. The problem of gyropscopic stabilization of unstable vehicles in roll is considered in [17] using Lagrangian dynamics. Linearized versions of the equations of motion show that the stability conditions are dependent on turn rate and direction for the single gyro case, but not for the double gyro case.

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Majority of emerging safety and comfort system require yaw- and roll-rate sensing.Yaw-rate sensors for electronic stability control (ESC) brake systems, which is a major breakthrough in automotive safety have been the most significant. By automatically activating asymmetric individual wheel braking actions in an out-of-control vehicle to stabilize it to regain traction and driver control ESC helps prevent accidents. Automotive applications are known to impose quite harsh environmental conditions such as vibration, shock, temperature, and thermal cycling on inertial sensors [18]. Micro machined gyroscopes are known to be especially challenging to develop and commercialize due to high sensitivity of their dynamic response to fabrication and environmental variations. Meeting performance specifications in the demanding automotive environment with low-cost and high-yield devices requires a very robust microelectromechanical systems (MEMS) sensing element. [20].

There are many devices that are applied to the military, and their design is based on the principles of gyroscopes. This gives them a very important role, and they need to be under the strict control of the design because in case of damage they could lead to catastrophic consequences. High-yield devices require very robust microelectromechanical systems (MEMS) sensing element. MEMS angular rate sensors have implicated in the automotive field since the mid-1990s (yaw-rate sensors for electronic stability control, roll-rate gyroscopes to sense impending vehicle rollover conditions) [5]. Gyroscopes in lane-keeping systems assist the drivers steering action to preserve vehicle in existing highway lane.

Gyroscope is a homogeneous, axis-symmetric rotating body that rotates by large angular velocity about its axis of symmetry. It is one of the most inertial sensors that measure angular velocity and small (angular disturbances) angular displacements around the reference axis.

Technical applications gyros today are so manifold and diverse that there is a need to get out of the general theory of gyroscopes allocates a separate discipline, called “applied theory of gyroscopes.”

The online Museum of Retro Technology [19] cites many articles and examples of gyrocars, including a 1961 Ford Gyrocar concept called the Gyron and a concept from Gyro Transport Systems of Northridge, California that was on the cover of the September, 1967 issue of ”Science and Mechanics”.

Each mechanical gyroscope is based on coupled rotations around more axes with one point intersection. Most of the old equipment was based on rotation of complex and coupled component rotations which resulting in rotation about fixed point gyroscopes. The classical book [1] by Andonov et al. contains a classical and very important elementary dynamical model of the simple case of the gyrorotor, and presents an analogous and useful dynamical and mathematical model of nonlinear dynamics [21, 22]

This work is different in that we derive the equations of motion using vector method proposed by K. Hedrih [9], and propose stability analysis for the system based on the derived model.

The vector approach is very suitable to obtain new view to the properties of dynamics of pure classical task, investigated by numerous generations of the researchers and serious scientists around the world.

Using Hedrih’s (See [5–10]) mass moment vectors some characteristics members of the vector expressions of derivatives of linear momentum and angular momentum for the gyrorotor coupled rotations around two axes without intersection obtain physical and dynamical visible properties of the complex system dynamics. Between them there are vector terms that present deviation couple effect containing vector rotators whose directions are the same as kinetic pressure components on corresponding gyrorotor shaft bearings.

Organizations of this paper based on the vector method applications with use of the mass moment vectors and vector rotators for obtaining vector expressions for linear momentum and angular momentum and their derivatives of the rigid body coupled rotations around two axes without intersections. These obtained expressions are analyzed and series of conclusions are pointed out, all useful for analysis of the rigid body coupled rotations around two axes without intersections when system dynamics is with two degrees of mobility as well as with two degrees of freedom, or for constrained by programmed rheonomic constraint and with one degree of freedom. It is possible to obtain two nonlinear differential equations in scalar form for rotations about each axes and also corresponding kinetic pressures in vector form by using two vector equations of dynamic equilibrium of rigid body dynamics with coupled rotations around two axes without intersection for two degrees of freedom.

This paper presents a new concept of mechanical design gyroscope. It is expected that multi-DOF concept will lead to reliable, robust, and high performance-angular-rate sensors with high yields, ideal for the demanding automotive environment.

1. MATHEMATICAL MODELING

A number of researchers have devised a mathematical model of a rotor based on the classical engineering problems so it takes place in world scientific and engineering professional literature [21,22]. Many scientists have studied disc as a mathematical model for rotor. Problem of dynamics of eccentricity, skew positioned disc on the shaft rotation is classical problem with gyroscopic effect which take place in all text books of Dynamics and Theory of Oscillations with application in engineering. But, their presentation was finished only by nonlinear differential equations without solutions and expressions for kinetic pressure. Simes, Stodola et al. [4] linearized nonlinear dynamics problems and got some results. It was like they used simplified model. The use of the simplified model could limit the ultimate performance of the designed controller, especially if more aggressive or fast maneuvers are desired.

We will model the gyrorotor as a rigid eccentric disc (eccentricity is e with mass m and radius r which is inclined to the axes of its own rotation by the angle β . Although this is not physically true, this assumption helps us to collect all the various inertia effects such as main and tail rotor dynamics into a simple constant mass-inertia matrix. The angle of own rotation around moveable axis oriented by the unit vector $\vec{n}_1$ is φ_1 and the angular velocity is $\vec{\omega}_1$**.** The angle of rotation around the shaft support axis oriented by the unit vector $\vec{n}_2$ is φ_2 and the angular velocity is φ_2 (see Figure 1 a*). When the support shaft is vertical and the rotor shaft is horizontal and when they are without intersection, with orthogonal distance a we obtain that angular velocity of rotor is $\vec{\omega} = \vec{\omega}_1 + \vec{\omega}_2$ and in a form $\vec{\omega} = \omega_1 \vec{n}_1 + \omega_2 \vec{n}_2$.

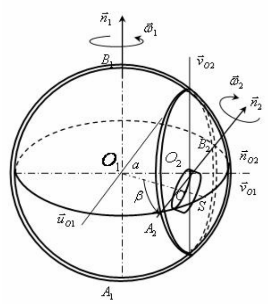


Figure 1 *Model of gyrorotor*

1.1 ***Coordinates and Frames***

As shown in Figure 1, there are two frames that we consider: the reference support frame and the body frame attached to the disc. The transformation between the two frames is given by the homogeneous transformation matrix where the rotation matrix ${}_0R^1$ represents the relative orientation between the two frames. The rotation matrix can be expressed in coordinates using yaw-pitch-roll Euler angles as:

$${}_0R^1 = \begin{bmatrix} \cos\varphi_2\cos\varphi_1 & \cos\varphi_2\sin\varphi_1 & \sin\varphi_2 \\ -\sin\varphi_2\cos\varphi_1 & -\sin\varphi_2\sin\varphi_1 & \cos\varphi_2 \\ \sin\varphi_1 & -\cos\varphi_1 & 0 \end{bmatrix} \tag{1}$$

1.2 ***Rigid body equations***

The angles φ_1 and φ_2 are generalized coordinates in case when we investigate system with two degrees of freedom. In this case φ_1 is generalized coordinate. The second angle φ_2 is a rheonomic coordinate which is defined by a time function. The tensor matrix of mass inertia moments of eccentric disc in a relation to the point O, (this point is cross section of rotor support axis and a plane which contains the support shaft axis and that is perpendicular to rotor shaft axis as it can be seen in (Fig.1), in the system coordinate axes coupled with support.

Velocity expression for elementary body mass particle is in a form $\vec{v} = [\vec{\omega}_2, \vec{r}_0] + [\vec{\omega}_1 + \vec{\omega}_2, \vec{\rho}]$, so by using basic definitions for linear momentum and angular momentum we can obtain main expressions in aim to get differential equations.

Vector expression for linear momentum is:

$$\vec{\mathrm{K}} = [\vec{\omega}_2, \vec{r}_0]M + \vec{\omega}_1 \vec{S}_{\vec{n}_1}^{(O_1)} + \vec{\omega}_2 \vec{S}_{\vec{n}_2}^{(O_1)} \tag{2}$$

The angular momentum of the system is:

$$\vec{L}_{O_1} = \omega_2 \vec{n}_2 r_0^2 M + \omega_2 \left[\vec{\rho}_C, \left[\vec{n}_2, \vec{r}_0\right]\right] M + \omega_1 \left[\vec{r}_0, \vec{S}_{\vec{n}_1}^{(O_1)}\right] + \omega_2 \left[\vec{r}_0, \vec{S}_{\vec{n}_2}^{(O_1)}\right] + \omega_1 \vec{J}_{\vec{n}_1}^{(O_1)} + \omega_2 \vec{J}_{\vec{n}_2}^{(O_1)} \tag{3}$$

Where $\vec{S}_{\vec{n}_1}^{(O_1)} = \iiint\limits_V [\vec{n}_1, \vec{\rho}] dm$ and $\vec{S}_{\vec{n}_2}^{(O_1)} = \iiint\limits_V [\vec{n}_2, \vec{\rho}] dm$, $dm = \sigma dV$ are coresponding body mass linear moments of the rigid body for the axes oriented by direction of component angular velocities of coupled rotations through the moveable pole O_1 on self rotating axis and $\vec{J}_{\vec{n}_1}^{(O_1)} = \iiint\limits_V [\vec{\rho}, [\vec{n}_1, \vec{\rho}]] dm$ and $\vec{J}_{\vec{n}_2}^{(O_1)} = \iiint\limits_V [\vec{\rho}, [\vec{n}_2, \vec{\rho}]] dm$ are coresponding body mass inertia moments of the rigid body for the axes oriented by direction of component angular velocities of coupled rotations through the moveable pole O_1 on self rotating axis.

We apply first derivative of angular momentum in the form

$$\begin{aligned} \frac{d\vec{L}_{O_1}}{dt} &= \vec{\omega}_2 r_0^2 M + \omega_1 \omega_2 \left\{ \left[\vec{n}_1, \vec{J}_{\vec{n}_2}^{(O_1)}\right] + \left[\vec{n}_2, \vec{J}_{\vec{n}_1}^{(O_1)}\right] + \right. \\ &\left. + \mathfrak{I}^{(O_1)} \left[\vec{n}_2, \vec{n}_1\right] \right\} + \omega_1^2 \left[\vec{n}_1, \vec{J}_{\vec{n}_1}^{(O_1)}\right] + \omega_2^2 \left[\vec{n}_2, \vec{J}_{\vec{n}_2}^{(O_1)}\right] + \dot{\omega}_1 \vec{J}_{\vec{n}_1}^{(O_1)} + \dot{\omega}_2 \vec{J}_{\vec{n}_2}^{(O_1)} \end{aligned} \tag{4}$$

The dynamical equations for motion on a straight inclined track can be easily obtained from the theorem of angular momentum derivative which in this case is in a form [16]:

$$\frac{d\vec{L}_{O_1}}{dt} = \overrightarrow{M}_{O_1}\left(\vec{F}_i\right) - \left[\left[\vec{\omega}_2, \vec{r}_0\right], \left[\vec{\omega}_1 + \vec{\omega}_2, \vec{\rho}_C\right]\right] M \tag{5}$$

This leads to the two vector equations of rigid body coupled rotation around axes without intersection [15,16]. These vector expressions can be used in general case when considered system has two degree of freedom. In case when the second angle φ_2 is a rheonomic coordinate which is defined by a time function $\varphi_2 = \omega_2 t + \varphi_{20}$, we have vector equation in a form:

$$\ddot{\varphi}_1 + \Omega^2 (\lambda - \cos\varphi_1) \sin\varphi_1 + \Omega^2 \psi \cos\varphi_1 = 0 \tag{6}$$

where constants in differential equation are in following form:

$$\Omega^2 = \omega_2^2 \frac{\varepsilon \sin^2\beta - 1}{\varepsilon \sin^2\beta + 1}, \qquad \varepsilon = 1 + \left(\frac{2e}{r}\right)^2, \qquad \lambda = \frac{g(\varepsilon - 1)\sin\beta}{e\omega_2^2 \left(\varepsilon \sin^2\beta - 1\right)},$$
$$\psi = \frac{2ea \sin\beta}{er\left(\varepsilon \sin^2\beta - 1\right)} \tag{7}$$

Transforming previous nonlinear differential equation into system of two first order nonlinear differential equations it is possible to obtain stationary values which correspond to the relative equilibrium positions of the disc on the self rotating axis. For each of relative equilibrium position Lyapunov criteria of stability can be applied and then it can be concluded about center or saddle points.

2. PHASE PORTRAIT OF THE GYROROTOR

Relative nonlinear dynamics of heavy gyrorotor disc around self rotation shaft axis is possible to present by means of phase portrait method. Phase trajectories forms and their transformations by changing of initial conditions and for different cases of disc eccentricity express nonlinear phenomena.

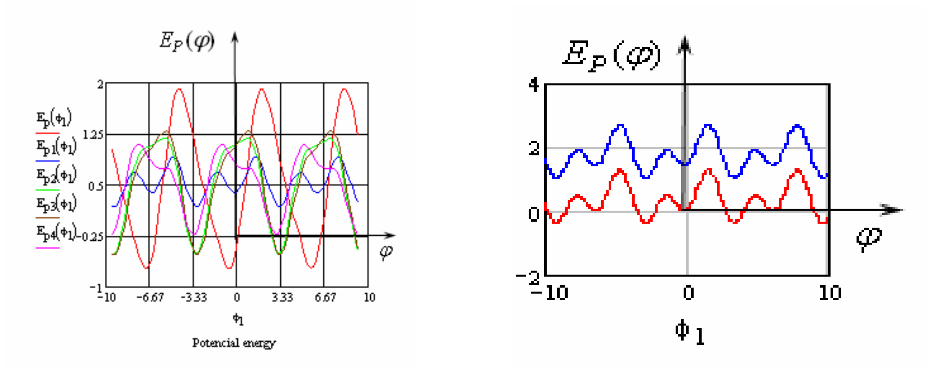
The solution-first integral of differential equation (6) with the initial conditions $t_0 = 0$, $\varphi_1(t_0) = \varphi_{10}, \dot{\varphi}_1(t_0) = \dot{\varphi}_{10}$, we obtain in the following form:

$$\varphi_1^2 = \varphi_{10}^2 + 2\Omega^2\left(\lambda\cos\varphi_1 - \frac{1}{2}\cos^2\varphi_1 + \psi\sin\varphi_1\right) - 2\Omega^2\left(\lambda\cos\varphi_{10} - \frac{1}{2}\cos^2\varphi_{10} + \psi\sin\varphi_{10}\right) \tag{8}$$

and it is the energy integral because the conservative system is analyzed [14-17].

$$\widetilde{E}_P = \Omega^2(\varepsilon)\left(\lambda(\varepsilon)(\cos\varphi_1 - \cos\varphi_{10}) + \frac{1}{2}\left(\cos^2\varphi_{10} - \cos^2\varphi_1\right) + \psi(\varepsilon)(\sin\varphi_1 - \sin\varphi_{01})\right) \tag{9}$$

Using MathCAD program on accomplished numerical experiment for researching of existence, like as number and character of stationary values of potential energy, as number of configuration of equilibrium positions and character of their stability, and transformations of phase trajectories with exchanging one of the kinematic parameter of system, graphs of potential energy exchange of corresponding basic system are obtain on numerical way. So, it was very interesting to analyze the influence of these parameter on nonlinear dynamics behavior of system. The potential energy exchange curves for different values of the system parameters (eccentricity e) are given on Figure 2 .



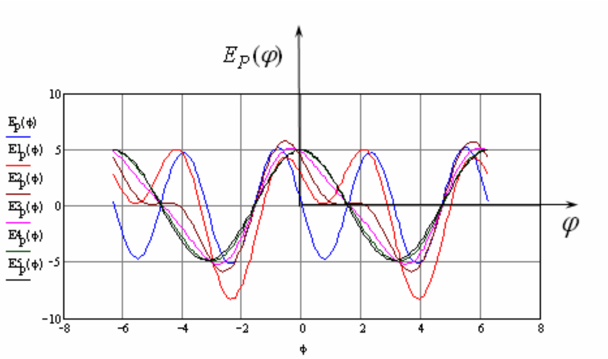


Figure 2 *Potential energy curves of system*

In the case when the dynamic relative equilibrium position is defined by φ_{10} the relative equilibrium position can be stable or unstable and the motion around that position may be defined as the sum of homogenous and particular solution. The character of stability depends on systems parameters.

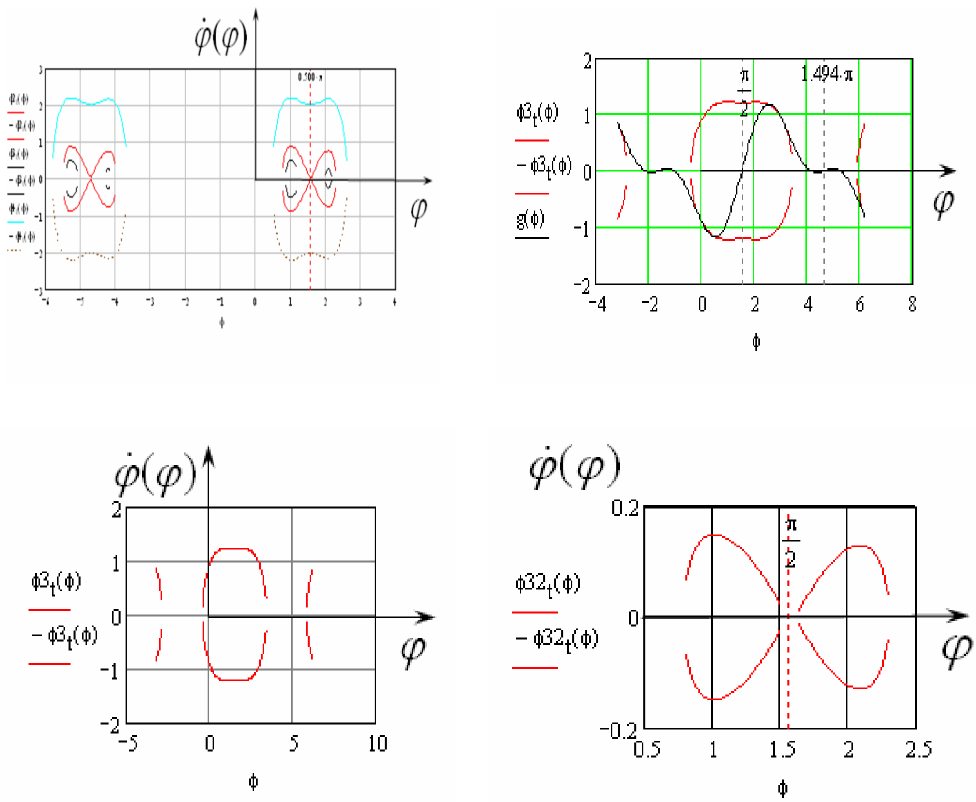


Figure 3 *Stable and unstable relative equilibrium position*

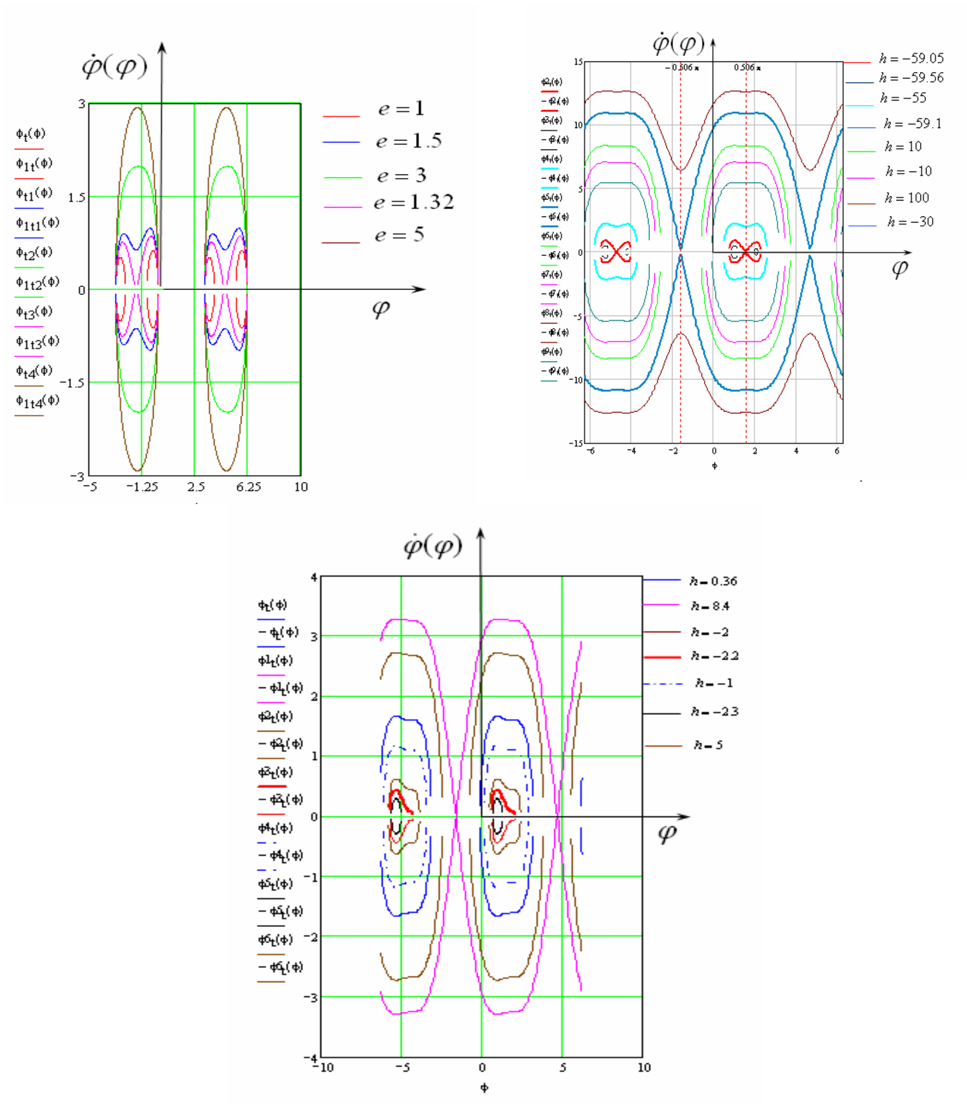


Figure 4 *Phase trajectories for different system parameters*

3. NUMERICAL EXPERIMENT AND GRAPHICAL PRESENTATION

In Figure 4. we can see characteristic homoclinic separatrix phase trajectories for different parameters values of the basic system correspond to the dynamic model. Examples of the trigger of the coupled singularities and coupled triggers of the coupled singularities are shown. The homoclinic trajectories in the form of the number eight are presented in Figure, as well as in the form of the duplicate number eight.

In Figure 4. we can see characteristic phase trajectories portraits for examples of the potential energy curves from Figure 2, and corresponding homoclinic separatrix phase for different parameters values of the basic system correspond to the gyrorotor dynamic model. Examples of the trigger of the coupled singularities and coupled triggers of the coupled singularities are presented on Figure 3. We can see more than five types of characteristic phase portraits which contains two types of singular points: by type stable center and unstable saddle

In Figure 4. transformations and layering of the homoclinic trajectories with change of the kinetic parameters values of the basic system correspond to the dynamic model are presented. Examples of the trigger of the coupled singularities and coupled triggers of the coupled singularities and homoclinic trajectories in the form of the duplicate number eight are, also, presented.

Characteristic potential energy curves and corresponding homoclinic separatrix phase trajectories for different parameters values of the basic system correspond to the dynamic model. These examples illustrate of the trigger of the coupled singularities and coupled triggers of the coupled singularities and homoclinic trajectories in the form of the number eight and also in the form of the duplicate number eight.

Figures 3 and 4. for different system parameters, we see that structures of phase portraits are different by types of phase trajectories and homoclinic orbits (phase trajectories of separatrix).

We can see on Figure 4, one-sided separatrix, which are “prolating”, and we see also open phase trajectories, which are comprising enclosed phase trajectories which are matching to the periodical oscillator motion-rotations system round stability configurations of equilibrium positions for specific initial conditions when initial angular velocity are small and small angles elongation of rotations, and when that condition are satisfying for any time.

In Figure 4. on phase portrait we notice augmentation of singular points, and we deduce by researching that for some kinetic parameters of system one stable equilibrium position loses stability and that positions now on phase portrait response to homoclinic point by type unstable saddle, but in symmetrical neighborhood appear two near-by stable equilibrium positions (configuration of masses), which on phase portrait response two singular points by center type. We can see also that all of three points are coupled in one “trigger” (trigger of coupled singularities, see reference [6] or [8]). Two stable singular points by type centers enclose one, and the new, closed homoclinic orbit which goes around three singularities, and passing trough one homoclinic point by type saddle in which it self-cross, that it is shaped like form of the number eight or in the form of duplicate of number eight or multiplication. Inside that new separatrix trajectory homoclinical orbit we notice a series of common closed phase trajectories which los instability relative equilibrium positions or relative rest positions, which correspond to periodic oscillatory motion for certain initial conditions, apropos oscillations around new stable position of equilibrium. We notice that homoclinic orbit shaped by number eight and multiplicities of number eight self-cross in points type by saddles which are issue from stable points type by saddle which is lose stability exchanging parameters of system an it is “disintegrate” on three, or even number which are trigger of coupled singularities or coupled triggers of coupled singularities. That point(s) is (are) also bifurcation point(s), because types of bifurcation, and define triple point. In Figure 5 for different system parameters it can be seen how the elongation changes with time.

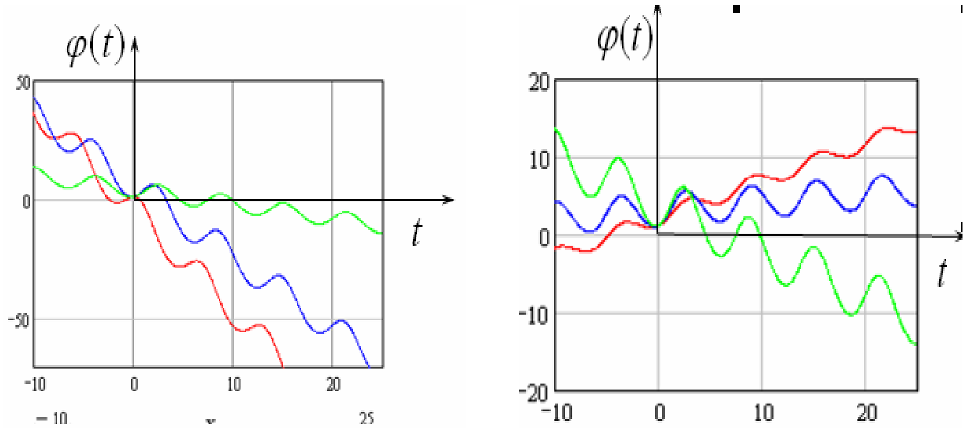


Figure 5 *Diagrams of elongacy-time*

4. CONCLUSIONS

We consider the problem of gyropscopic coupled rotation. We derive the full nonlinear equations of motion for the non-trivial case (coupled rotation around two orthogonal axes without intersection) using Hedrih's vector method dynamics, consider different configurations (different values of eccentricity), and derive numerical versions of the equations of motion. We consider nonlinear dynamics of gyrorotor which are dependent on system parameters. Here we take account only eccentricity but we can consider how another system parameters (angle of inclination and orthogonal distance between axes) influence at nonlinear dynamics of gyrorotor. Also, by using equations of phase trajectories some properties of nonlinearity are investigated. We analyzed homoclinic orbits and their transformation shaped by number eight, their appearance and disappearance by changing someone parameter of system. This is also verified by numerical experiment. Many applications of the discovered vector method by using mass moment vectors are presented for to express kinetic parameters of heavy rotors dynamics as well as of coupled multistep rotors dynamics and for gyrorotors dynamics.

Automotive gyroscopes are highly engineered inertial sensors, involving a large number of disciplines. Stability and robustness of the mechanical sensing element becomes crucial so to improve it multi-DOF sense systems are analysed. It is expected that multi-DOF concept will lead to reliable, robust and high performance angular rate sensors with low production costs and high yields that is ideal for demanding automotive environment. The automotive environment is a daunting combination of temperature, vibration, thermal cycling, shock, humidity, acoustic, etc. Gyroscopes typically operate in safety systems so detection of sensor failures is of utmost importance to avoid an unintended vehicle action that endangers people.

Future work includes further analysis of the equations of motion, and comparisons to other gyroscopic systems such as automotive gyroscope or ship stabilizers. In addition, we will also further analyze control properties, compare the performance of the early mechanical feedback systems to more modern approaches, perform simulations for a full scale vehicle, and analyze results from the scaled model experiments. We can't forget that automotive gyroscope must meet many requirements in addition to having extremely small size and low-unit production cost.

Gyroscopes are simply devices which measure rotation. For robotic applications the gyro output can be used to determine rotation rate, altitude or heading and can be combined with other sensor inputs to determine position. A wide range of robots and autonomous vehicles are currently used in fibre gyros, and many more are likely to be used in the future.

5. ACKNOWLEDGMENT

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CEPSTRUM ANALYSIS OF VIBRATION IN TRANSMISSION SYSTEM OF THE VEHICLE[1]

Aleksandar Đurić
Snežana Jovanović

UDC: 629.01;681.5.015.87

Summary
This paper presents cepstrum analysis of measured vibrations in vehicle transmission. The use of vibration analysis tools for transmission status is very hard, especially in real off-road conditions of vehicle exploitation. The main thing is the fact that vibration level is correlated with the loading level of engine and vehicle. In this paper we tried to adopt cepstrum analysis for transmission status determination in off-road vehicle testing.

Key words: vibrations, vibration analysis, Cepstrum

CEPSTRUM ANALIZA VIBRACIJA SISTEMA ZA PRENOS SNAGE VOZILA

UDC: 629.01;681.5.015.87

Rezime
U radu je predstavljena cepstrum analiza izmerenih vibracija sistema za prenos snage vozila. Primena alata za analizu vibracija kod sistema za prenos snage je veoma teška naročito u realnim vanputnim uslovima eksploatacije. Najvažnija je činjenica je da je nivo vibracija u korelaciji sa nivoom opterećenja motora i vozila. U ovom radu je pokušano da se formira cepstrum amaliza za stanje sistema za prenos snage u vandrumskim testovima.

Ključne reči: vibracije, analiza vibracija, Cepstrum

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CEPSTRUM ANALYSIS OF VIBRATION IN TRANSMISSION SYSTEM OF THE VEHICLE

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1. INTRODUCTION

1. 1 History of vibration

The origins of the theory of vibration can be traced back to the design and development of musical instruments (good vibration). It is known that drums, flutes, and stringed instruments existed in China and India for several millennia B.C. Also, ancient Egyptians and Greeks explored sound and vibration from both practical and analytical points of view.

The foundation of the modern day theory of vibration was probably laid by scientists and mathematician such as Robert Hooke, of the Hooke's law fame, who experimented on the vibration of strings. Sir Isaac Newton gave us calculus and the laws of motion for analysing vibrations. Daniel Bernoulli, Leonard Euler, Joseph Lagrange, Charles Coulomb, Joseph Fourier, Simeon-Dennis Poisson are some of the great scientists who work in vibration analysis. As a result of the industrial revolution and associated developments of steam turbines and other rotating machinery, an urgent need was felt for development in the analysis, design, measurement, and control of vibration. Motivation for many aspects of the existing techniques of vibration can be traced back to related activities since the industrial revolution. Much credit should go to scientists and engineers of more recent history, as well. Among the notable contributors are Rankin, Kirchhoff, Rayleigh, de Laval, Timoshenko, Crandall and others [1].

2. VEHICLE TESTING AND VIBRATION

Huge number of direct and indirect values has been measured and tracked in process of off-road vehicle prototype quality examinations methodology and testing. Analysis together with vehicle testing in real exploitation conditions has been improved, as measurement equipment quality has been increase. Highly sophisticated measurement technique is enabling direct and postprocess visualisation of all variations set up from load in extreme vehicle testing conditions.

Examination and testing has become an objective technical diagnose, where, as a part of support tools, has been import a vibration analysis test. This test represent diagnose of conditions by analyse of motion characteristics conditions of measured vibrations.

Method of vibration diagnosis enable re-establish correlation between dominant frequency (one or more) and mechanical defects which cause it. In that way, we can predict failure in vehicle transmission before it cause damage. In measurement and examination of vehicles that is very important, because it enable us to detect weak spots without dismantling transmission.

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Vibrations, that being research, often are not forced like consequence of dynamic force influence, but combination of force and natural, resonant and modal vibrations. They are being generate by single source, so that recorded vibration signal at characteristic measurement position are heterodyning of several elementary vibrations with different frequency and intensity.

It is very hard to identify individual vibration generator in recorded time domain, so, thanks to absolute correlation of time and frequency region, recorded signals has been analysed in frequency domain through power spectrum that present in frequency axis decomposed energy. Practical merit of frequency spectrum is that enables detection and behaviour description of component part, which reveal existential large or slight problem.

In diagnostic matters vibrations defines: magnitude (expressed like shift, velocity or acceleration), frequency, phase and cast. Vibration level is function of shift and frequency. Vibration velocity is function of shift and frequency too, so it is gauge of vibration level and best indicator of part an/or condition of entire scheme. Velocity gives the most uniform spectrum, so we us it for vibration analysis and monitoring [2].

3. VIBRATION AND TRANSMISSION

3. 1 One degree of freedom model

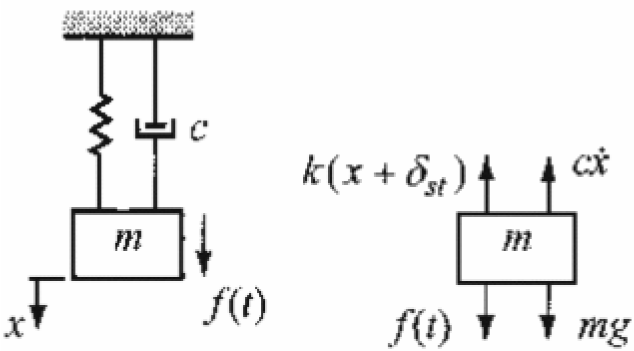

Figure 1 One degree of freedom model

The fundamental physical law governing all vibration phenomena is Newton's second law, which in its most commonly used form says that *the sum of the forces acting upon an object is equal to its mass times its acceleration.* Force and acceleration are both vectors, so Newton's second law, written in its general form, yields a vector equation. For the one degree of freedom (DOF) system, this reduces to a scalar equation, as follows:

$$F = ma \qquad (1)$$

Where:
- F – Sum of forces acting upon body;
- m – Mass of body;
- a – body acceleration.

For the system in Figure 1, equation (1) yields its differential equation of motion, as follows:

$$m\ddot{x} + c\dot{x} + kx = f(t) \tag{2}$$

The solution for the motion of the unforced one degree of freedom system is important in its own right but specifically important in laying the groundwork to study self excited instability rotor vibrations. If the system is considered to be unforced, then $f_{(t)}=0$ and equation (2) becomes the following:

$$m\ddot{x} + c\dot{x} + kx = 0 \tag{3}$$

This is a second order homogeneous ordinary differential equation. To solve for x $_{(t)}$, from Equation (3), one needs to specify the two initial conditions, $x_{(0)}$ and $\dot{x}_{(0)}$. As summing k and c are both positive, three categories of solution can result from equation (3), (a) under damped, (b) critically damped, and (c) over damped. These are just the traditional labels used to describe the three distinct types of roots and the corresponding three motion categories that equation (3) can potentially yield when k and c are both positive. Substituting the known solution form ($Ce^{\lambda t}$) into equation (3) and then cancelling out the solution form yield the following quadratic equation for its roots (Eigen values) and leads to the equation for the extracted two roots, $\lambda_{1,2}$, as follows:

$$m\lambda^2 + c\lambda + k = 0 \tag{4}$$

$$\lambda_{1,2} = -\frac{c}{2m} \pm \sqrt{(\frac{c}{2m})^2 - (\frac{k}{m})} \tag{5}$$

The three categories of root types possible from equation (4) are listed as follows [3]:

Under damped, $(\frac{c}{2m})^2 < (\frac{k}{m})$, complex conjugate roots, $\lambda_{1,2}=\alpha \pm i\omega_d$;

Critically damped, $(\frac{c}{2m})^2 = (\frac{k}{m})$, equal real roots, $\lambda_{1,2}=\alpha$;

Over damped, $(\frac{c}{2m})^2 > (\frac{k}{m})$, real roots, $\lambda_{1,2}=\alpha \pm \beta$.

4. GEARS AND VIBRATION

Vibration diagnosis is extending number of parameters about present and future condition and quality of component parts in dynamic system (in this case, elements in power transmission system). These parameters give better presentation of system for power transmission without dismantling parts of system.

Conditions for vehicle vibration measurement in exploitation working conditions are of non-steady nature (number of revolution n≠const.).

Vibrations are synchronised emitted with number of revolution of rotating mass. In frequent spectrum prominent characteristics synchronised components of vibrations are being generated – harmonic from kinematics frequency, that make harmonic series (orders).

Connection of rpm and order is:

$$OrderN(Hz) = N \times \frac{RPM}{60} \tag{6}$$

The first order is ratio of RPM and 60, and Nth order is multiplication of Ith order and integer, like in this example: impeller speed = 1200 rpm, I–order is at 20 Hz, and so forth.

Basic order is the first order. But, in gears, basic vibrations are consequence of RPM and gear-wheels. Te first shaft order is one characteristic for gearbox. Second characteristic order is I-order multiplicities with m (m-module of gear-wheel). That is I-order for gear in conjunction.

Potential cause of expanded vibrations can be battered, excenter or damaged gears.

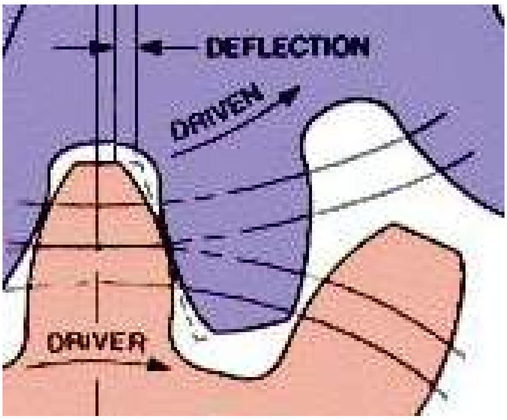

Figure 1 Conjugate gear action

Defect gears generate high and medium frequency vibration, which are suitable to multiplication of defect gear tooth number (r) and RPM of drive gear (n). Signal form from defect gear toot can be recognized.

$$f_{toothing} = f_{shaft} \times N_{tooth} \tag{7}$$

Variable vibrations are occurring in gear-wheels that work under low load, because loading is advancing from one tooth to another.

Beside characteristic frequencies coupled to basic number of revolutions of rotating elements, in spectrum there are also and fix frequencies combined with structural, resonant system characteristics. Impacts, which are results of load changing, will excite gear natural frequency. Frequency analysis has important part in failure prediction at gearbox, and gives us a non-destructive tool for determine time of maintenance in transmission elements.

Gear vibration can be observed like free damped vibrations where excite regenerate itself with every impact in every conjugate gear action next tooth's couple [2].

4. 2 Cepstrum analysis

Cepstrum analysis is a tool for the detection of periodicity in a frequency spectrum, and seems so far to have been used mainly in speech analysis for voice pitch determination and related questions. In that case the periodicity in the spectrum is given by the many harmonics of the fundamental voice frequency, but another form of periodicity can also be detected by cepstrum analysis is the presence of sidebands spaced at equal intervals around one or a number of carrier frequencies. The presence of such sidebands is of interest in the analysis of gearbox vibration signals, since a number of faults tend to cause modulation of the vibration pattern resulting from tooth meshing, and this modulation (either amplitude or frequency modulation) gives rise to sidebands in the frequency spectrum. The sidebands are grouped around the tooth-meshing frequency and its harmonics, spaced at multiples of the modulating frequencies, and determination of these modulation frequencies can be very useful in diagnosis of the fault [4].

The cepstrum exist in various forms but all can be considered as a spectrum of a logarithmic (amplitude) spectrum. This means that it can be used for detection of any periodic structure in spectrum, e.g. from harmonic, sidebands, or the effects of echoes. It is also shown, however, that effects which are convolved in the time signal (multiplied in the spectrum) become additive in the spectrum, and subtraction there results in a deconvolution.

The cepstrum was first proposed as far back as 1963 and was at that time defined as "the power spectrum of the logarithmic power spectrum". The intended application at that time was to seismic signals because it was realised that it would give information about echoes and that in turn would help to determine the depth of the hypocentre of a seismic event. The reason for defining the cepstrum as above is not entirely clear, as even in the original paper it is compared with the autocorrelation function, which can be obtained as the inverse Fourier transform of the power spectrum. Later, another definition of the cepstrum was given as "the inverse Fourier transform of the logarithmic power spectrum", thus making its connection with the autocorrelation clearer. At about the same time, another cepstrum like function was defined as the "inverse Fourier transform of the complex logarithm of the complex spectrum" and to distinguish it from the above cepstra it was called the "complex cepstrum", while they were renamed "power cepstra".

Cepstrum is normally defined as the power spectrum of the logarithm of the power spectrum. Quefrency is the independent variable of the cepstrum and has the dimensions of the time as in the case of the autocorrelation. The quefrency is seconds is the reciprocal of the frequency spacing in Hz in the original frequency spectrum, of a particular periodically repeating component. Just as the frequency in normal spectrum says nothing about absolute time, but only about repeated time intervals (the periodic time), the quefrency only gives information about frequency spacing's and not about absolute frequency [4].

Using the function terminology to indicate the forward Fourier transform of the bracketed quantity, the original definition of the cepstrum is:

$$c_{(t)} = \left| F\left\{ \log F_{xx}(f) \right\} \right|^2 \tag{8}$$

Where the power spectrum of the time signal $f_x(t)$ is given by:

$$F_{xx}(f) = \left| F\left\{ f_x * t) \right\} \right|^2 \tag{9}$$

The new definition of the power cepstrum is:

$$c_p(t) = F^{-1}\left\{ \log F_{xx}(f) \right\} \tag{10}$$

While the autocorrelation function is given by:

$$R_{xx}(t) = F^{-1}\{F_{xx}(f)\} \tag{11}$$

This can be interpreted as the square root of equation (8) or as the modulus of equation (10), since for real even function such as a log power spectrum, the forward and inverse transforms give the same result. The complex cepstrum may be defined as fallows:

$$c_c(t) = F^{-1}\{\log F_x(f)\} \tag{12}$$

Where $F_x(t)$ is the complex spectrum of $f_x(t)$ i.e.

$$F_x(f) = f\{f_x(t)\} = a_x(f) + ib_x(f) = A_x(f)e^{i\phi_x(f)} \tag{13}$$

In terms of real and imaginary components or amplitude and phase, respectively. From (13) the (complex) logarithm of F_x (f) is given by:

$$\log F_x(f) = \log A_x(f) + i\phi_x(f) \tag{14}$$

Where $f_x(t)$ is real, as is normally the case, then $F_x(f)$ is "conjugate even", i.e.

$$F_x(f) = F_x^*(f) \tag{15}$$

From the general theory of the Fourier transforms it is known that the spectrum should be of real even function is real and even, and of a real odd function is imaginary and odd. Since any real function can be divided into even and odd components it follows that the real part of the Fourier transform comes from the even part of the time signal, and the imaginary part from the odd part of the time signal.

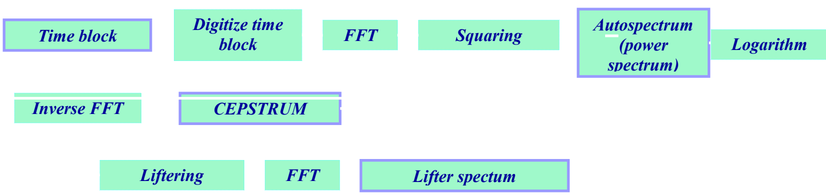

Figure 2 Data processing schemes

For determine period in spectrum and enable research of modulation, for gear set analysis is characteristically Cepstrum tool, that present spectrum in time domain.

5. CASE STUDY OF MEASUREMENT AND VIBRATION MONITORING

The vibration diagnosis and analysis is following steps:

- Analysis dynamic system, which is the object of measurement.
- Preparation for measurement.
- Vibration measurement and number of revolutions.
- Analysis of measured vibration signal (frequency and harmonic).
- Diagnostic conclusion.

Measurement object was gear set of vehicle, which is a prototype (off-road vehicle with regular and non planetary gearbox). All measurements have been made in mountain off-road conditions, in every gear. For measurement we used four acceleration transducers, CTC–131 type. There was four points for measurement. Precise measurement points were:

- Two points in input shaft (transducers 1 and 3), in place of input bearing.
- Two in output shaft (transducers 2 and 4), in place of output bearing.

Multi channel measurement, type NETdB 12, French manufacturer 01 dB Metravib, Areva Corporation. Measurement equipment is set with software also from 01-dB Metravib, Areva Corporation. Measurement place for transducers were bearing boxes of input and output shift. Measurement set signal was recorded to PC hard-disk (laptop), for future analysis. All measurements are being made at about 2000 rpm of vehicle engine, with calculation of transmission ratio.

In the beginning, nought state was measured, for determing starting parameters for measurement. After, only periodically measurements are being made (at about every 2000 km). Next parameters were measured:

- Absolute vibrations with four accelerometers in measurement points.
- RPM with laser tachometer.

Measurements were in real-time, what enable various on-line testing.

With post-process mathematic averaging we are processing random time signal, which we recorded in measurement of working load characteristics (vibration and number of revolutions) in off-road conditions drive. After all measurements, we made vibration analysis, which include:

- Time signal filtration.
- Recorded acceleration signal integration.
- FFT (Fast Fourier Transformation) transformation.
- Mathematic parameter selection (RMS, log) for presenting 2D I 3D (Waterfall) spectrum for analysis.
- Cepstrum analysis.

For gear vibration signal, magnitude modulation is characteristic, as a result of excenter and frequency modulation appearance, which cause variation of some gear wheel rpm. There are several characteristics for gear vibrations:

- Frequencies from errors in making gears, that are nonsensitive to load alteration.
- Sidebands from magnitude (tooth conjuction deformation) and frequency (rpm fluctuation) modulation.
- Low harmonics from extra impacts, at every rpm.
- Combination of frequency components of conjunction.

For this measurement we used laser tacho probe too, made by Monarch instruments, attached to gearbox output shaft. From numerous measurements we made, in paper we'll present spectrum for second gear in different off-road conditions, for transducer 4.

6. SECOND GEAR

All measurements were made in real off-road conditions, in different off-road types. We made measurements in off-road conditions at road at mountain Fruska gora and Zlatibor. Route and ground quality of road was very similar in both test roads. All the measurements were made in uphill driving, so the ground loading was the same in both ways.

6. 3 Second gear at Fruska gora

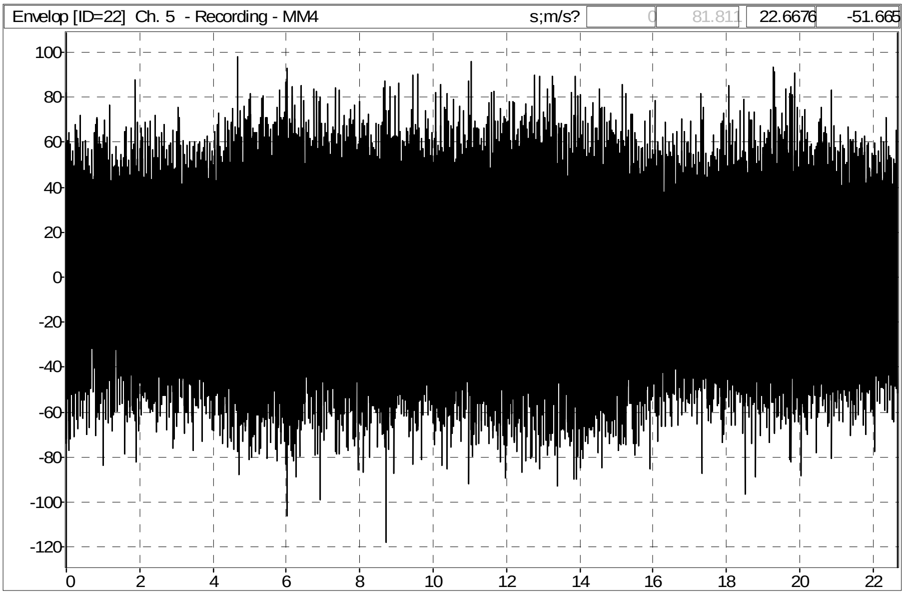

Figure 3 *Speed signal in time for the second gear, Fruska gora (m/s2 / s)*

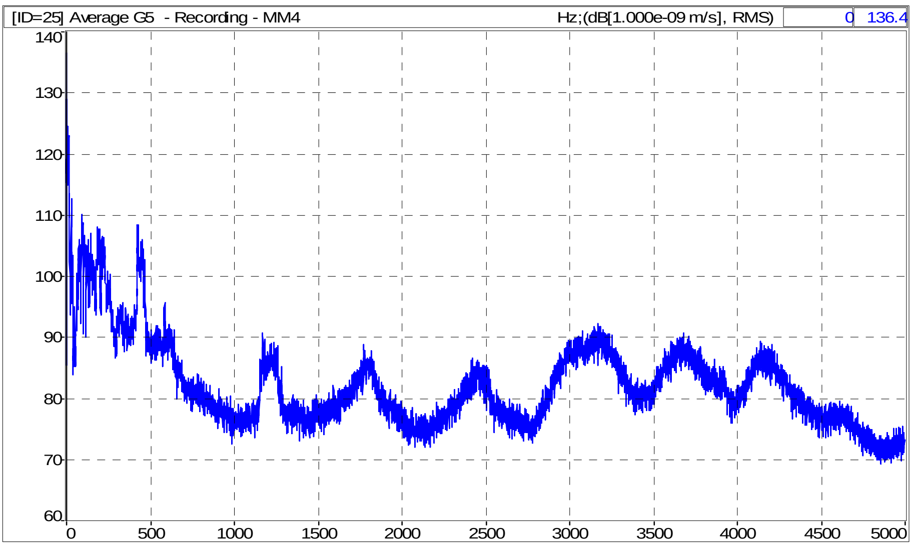

Figure 4 *Frequency spectrum, Fruska gora (dB/Hz)*

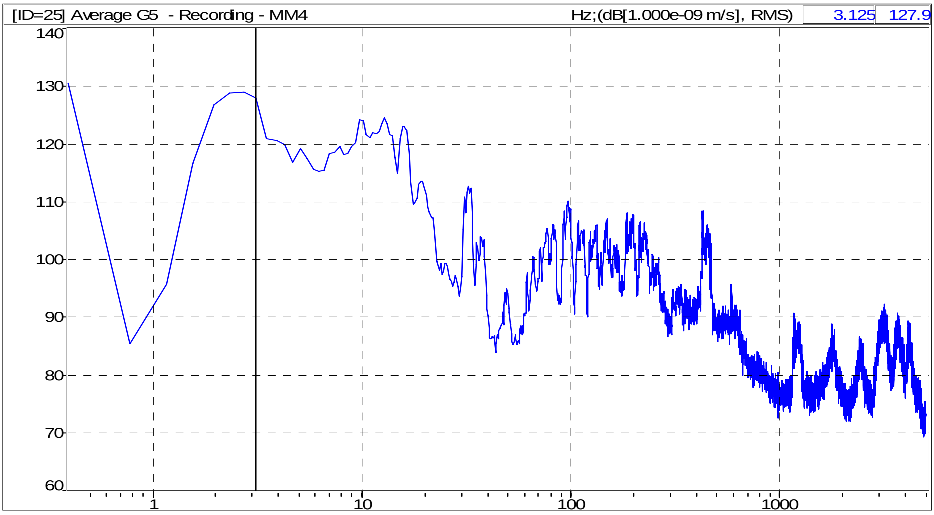

Figure 5 *Logarithm of the frequency spectrum, Fruska gora (dB / Hz)*

Frequency spectrum is giving us the levels of vibrations, picks and frequencies that are characteristic for some pick values. In frequency spectrum we can identify frequencies from output shaft, conjucted gears, gear mesh frequency and found some frequencies that are not expect to see in that signal. Logarithm of frequency spectrum is magnifaing the beggining of the spectrum, where are some working frequencies of the output shaft and it subharmonics.

Signal of speed (measured in m/s) in time (measured in s) is shown at figure 3. We can see that speed level for gear is from about $\pm$ 60 [m/s] (with some picks from about $\pm$80 [m/s]). The whole frequency spectrum of the time signal in fig. 3 is shown in fig. 4, and the logarithm of frequency spectrum is shown in figure 5. At figure 6, there are shown characteristic frequencies for conjucted output gear, with their subharmonics shown in figure 7 and 8.

Characteristic frequencies are:

- 6,4 Hz, frequency of the I-order of the output shaft (RPM of the output shaft is at about 386 rpm/60= 6,4 Hz).
- 15,4 Hz, gear mesh frequency of the output shaft and conjugated gears. All subharmonics we can get by multiplication of the first order.
- 554 Hz, the I-order of the conjugate output gear.
- Other subharmonics we can get by multiplication of the first order of output shaft frequency and gear mode.

Only time signal, with subharmonics from specific frequencies is the beginning of the signal analysis.

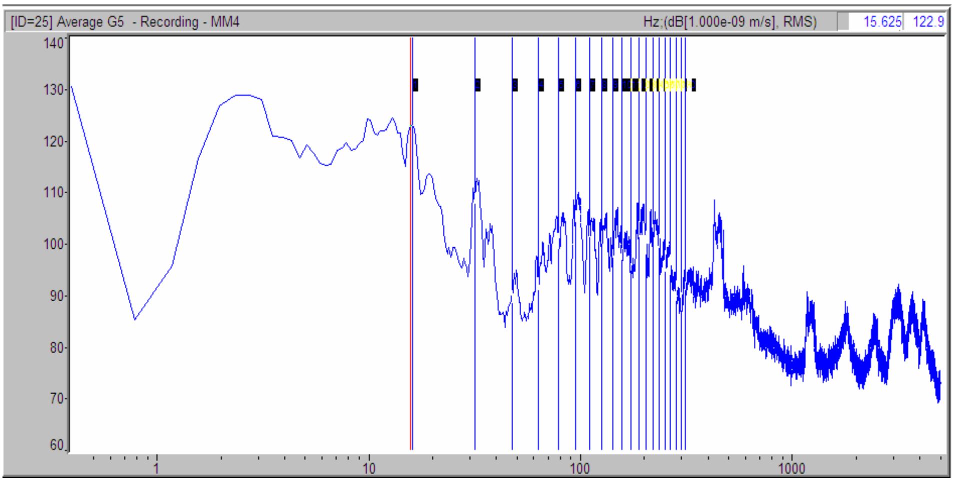

Figure 6 Logarithm of the frequency spectrum, Fruska gora specific frequencies (dB / H)

Display spectrum PUTNI DRUGI UZ.cmg		ID=25
FFT resolution	0.391 Hz	
Harmonic	Frequency	Level
1	15.6	122.9
2	31.3	109.4
3	46.9	88.9
4	62.5	94.3
5	78.1	100.7
6	93.8	103.4
7	109.4	101.6
8	125.0	99.5
9	140.6	95.5
10	156.3	97.3
11	171.9	96.0
12	187.5	102.9
13	203.1	101.9
14	218.8	102.4
15	234.4	100.0
16	250.0	96.3
17	265.6	97.2
18	281.3	91.9
19	296.9	88.0
20	312.5	91.2

Display spectrum PUTNI DRUGI UZ.cmg		ID=25
FFT resolution	0.391 Hz	
Harmonic	Frequency	Level
1	3.1	127.9
2	6.3	115.2
3	9.4	120.1
4	12.5	123.4
5	15.6	122.9
6	18.8	113.0
7	21.9	107.2
8	25.0	99.3
9	28.1	97.1
10	31.3	109.4
11	34.4	98.2
12	37.5	103.7
13	40.6	88.0
14	43.8	83.7
15	46.9	88.9
16	50.0	94.1
17	53.1	85.2
18	56.3	85.9
19	59.4	87.3
20	62.5	94.3

Figure 7 Subharmonic of the first gear mesh and the output shaft

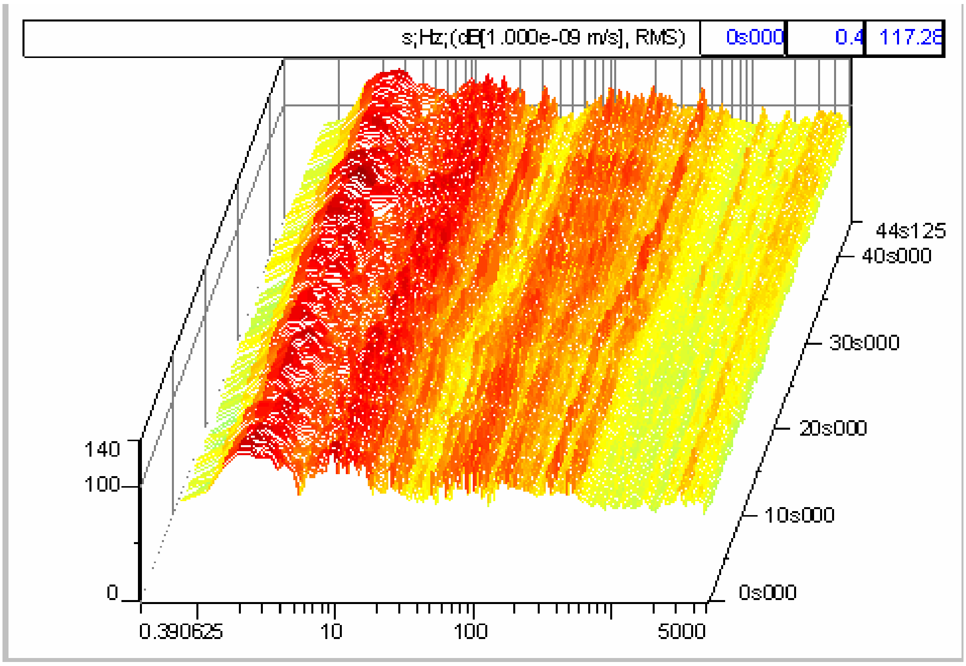

Figure 8 3D spectrums for second gear, Fruska gora (dB / Hz / s)

In 3d spectrum diagram we can see specific frequencies, in multispectrum conditions. Time, frequency and pick levels are shown in one diagram.

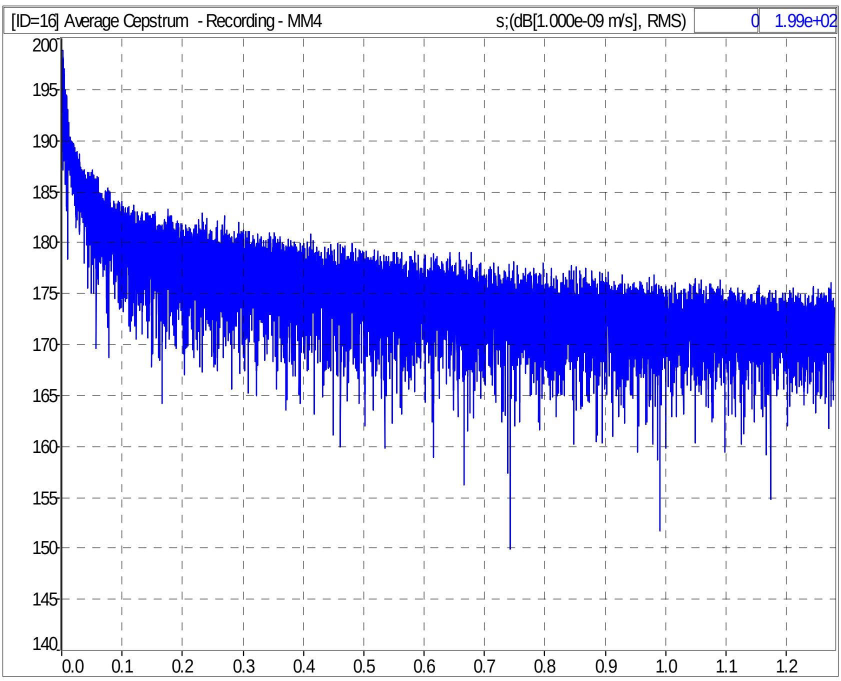

Figure 9 Cepstrum of recorded signal (dB / s)

Cepstrum of the signal of vibration measured in second gear of transmission at Fruska gora off-road conditions is shown in figure 9. The main advantage of cepstum is, because he's finding all periodic events (it is shown as diagram pick at some time). In this signal there was no characteristics picks at any time. In this signal we can see that there is no pick values that would made some suspicion that something is at (or will be in short time period) failure. The conclusion was that in that gear there is no specific problem with gears in second gear.

6. 4 Second gear at Zlatibor

Signal of speed (measured in [m/s]) in time (measured in s) is shown at figure 10, for the off-road road recorded signal in Zlatibor. We can see that speed level for gear is from about ± 60 [m/s] (with some picks from about ±80[m/s]). The whole frequency spectrum of the time signal in fig. 10 is shown in fig. 11, and the logarithm of frequency spectrum is shown in figure 12.

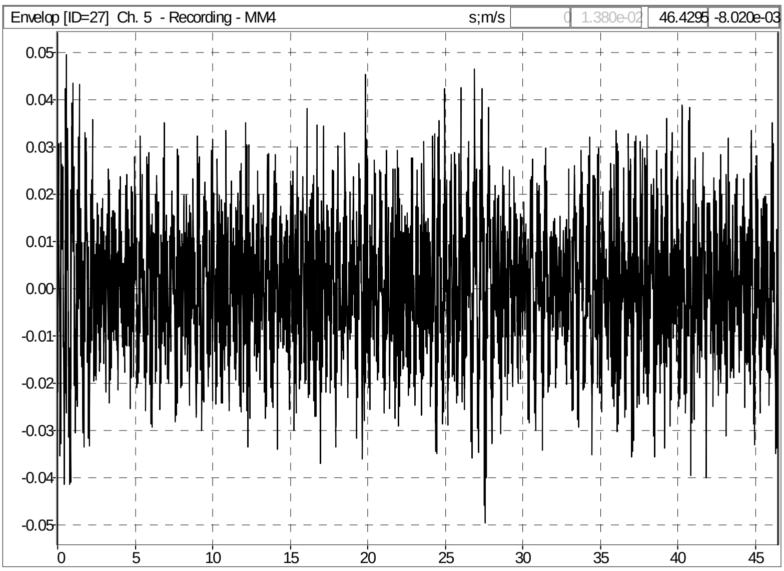

Figure 10 Speed signal in time, Zlatibor (m/s / s)

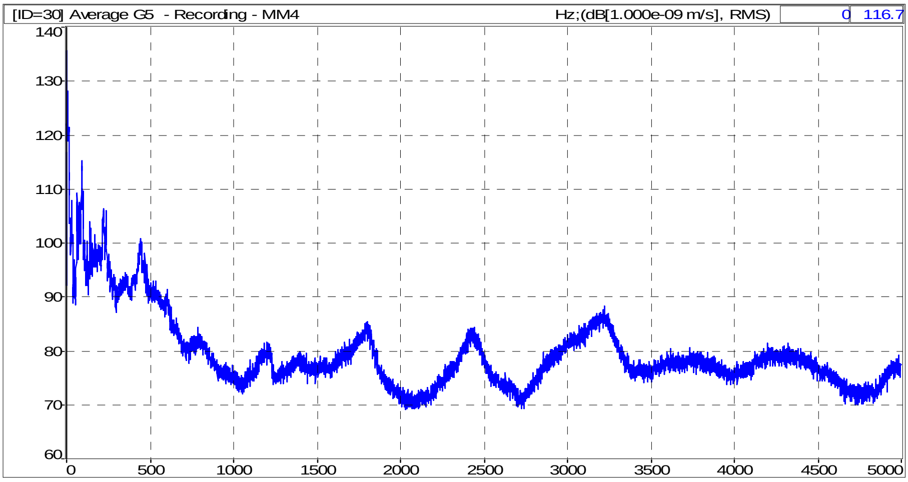

Figure 11 Frequency spectrum, Zlatibor (dB / Hz)

Characteristic frequencies are the same like in Fruska gora off-road road:

- 6,4 Hz, frequency of the I-order of the output shaft (RPM of the output shaft is at about 386 rpm/60= 6,4 Hz).
- 15,4 Hz, gear mesh frequency of the output shaft and conjugated gears. All subharmonics we can get by multiplication of the first order.
- 554 Hz, the I-order of the conjugate output gear.
- Other subharmonics we can get by multiplication of the first order of output shaft frequency and gear mode.

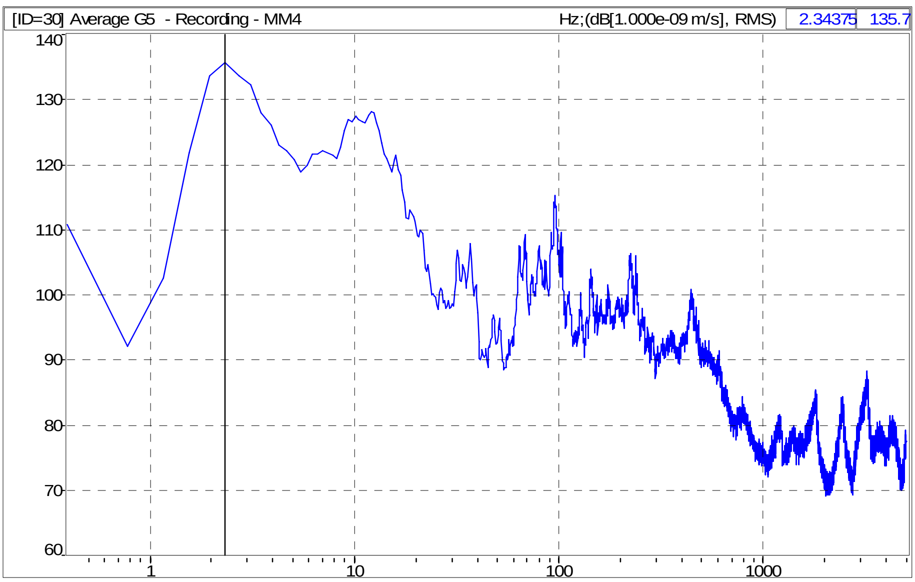

Figure 12 Logarithm of the frequency spectrum, Zlatibor (dB / Hz)

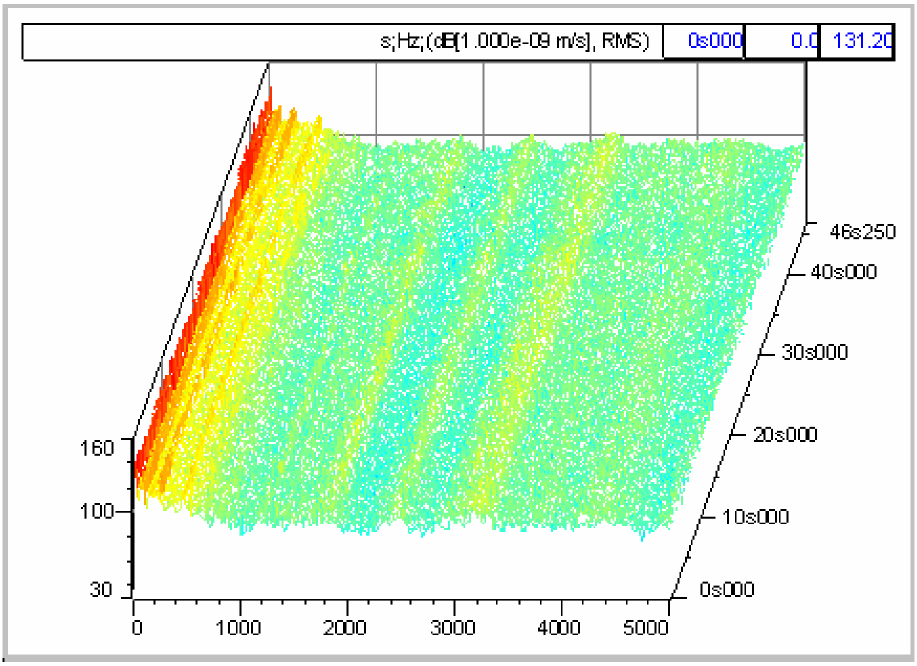

Figure 13 3D spectrum for second gear, Zlatibor (dB / Hz / s)

At figure 13 we can see 3D spectrum of measured vibration signal. If we make some comparation with 3d spectrum in Fruska gora, we can see that "setttling" of the spectrum, as the vehicle is being tested in time. The reason of the spectrum "setttling" is running in gears in transmission with time period of working. The gears are being "more conjucted" in working time, and that is shown in figure 8 and 13.

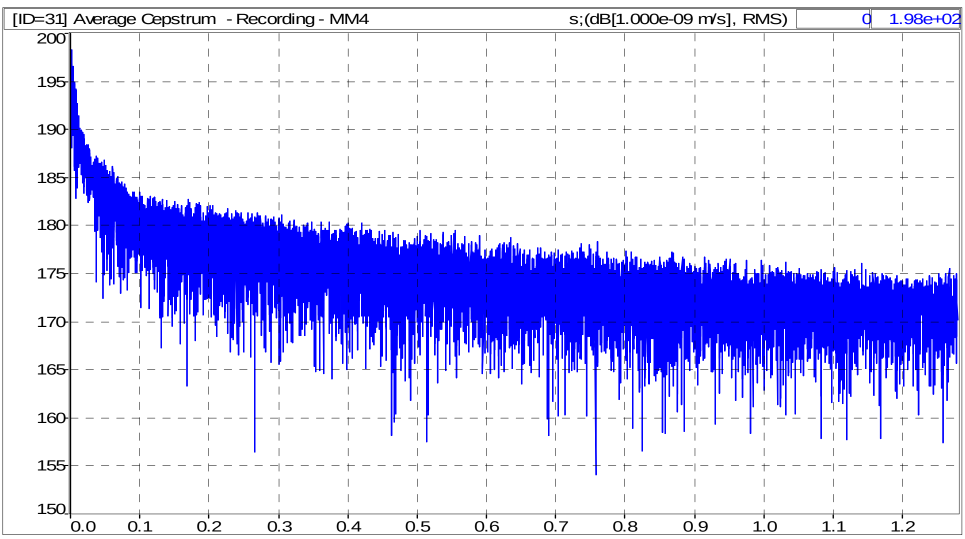

Figure 14 Cepstrum of recorded signal (dB / s)

Cepstrum of the recorded signal is shown in figure 14. Cepstum is finding all periodic events and showing it at diagram like a pick at some time. In this signal there was no characteristics picks at any time. The conclusion was that in that gear there is no specific problem with gears in second gear.

7. CONCLUSIONS

Cepstum tool for vibration analysis is finding all periodic events and showing it at diagram, like a pick at some time. It is wary suitable for non aggressive gear testing and condition analysis in real working conditions of vehicle transmission.

In our signals in second gear, that we recorded during vehicle testing in two off-road conditions in same conditions of testing we found no characteristics picks at any time. Main conclusion was that gear set is working without damages or failures that are going too happened. The measurements are being made in the beginning of the testing, at Fruska gora, and in the middle of testing, at Zlatibor. At the end of testing, we opened gear set and explore all gears in gear set of vehicle transmission. We didn't found out any failure. That gave our conclusions a form of evaluation.

The second characteristic of transmission that we can see is "setttling" in 3D spectrum, in time of testing. The 3D spectrum in Zlatibor, being measured in the middle of testing, is more "setttled" than the 3D spectrum recorded at Fruska gora, in the beginning of vehicle testing. Conclusion was, reason of the spectrum "setttling" is running in gears in transmission with time period of working (the gears are being "more conjucted").

Usage of cepstrum analysis in vibration analysis of vehicle gear transmission testing in validation and estimation of working conditions and failures (future failures) is a relative new approach in vehicle testing. With respect to it non damage nature, in future time it`s going to be more exploit tool in vehicle testing, and maintenance and monitoring of working condition in vehicle work time period. We can detect weak spots in early phase, without dismantling of transmission. All the testing we can make faster and more economic.

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[1]DYNAMOMETER FOR TESTING HIGH-FREQUENCY NOISE OF DISC BRAKES

Jasna Glišović, Miroslav Demić, Jovanka Lukić, Danijela Miloradović

UDC:629.1.07

Abstract

Brake squeal phenomenon has been studied for nearly 70 years now. During this period, majority of tests were based on subjective assessments and measurements with a moving vehicle on the road. Over the years, many laboratory tests have been developed with a wide range of options and approaches. The modern brake noise dynamometer has become a sophisticated test platform for the identifying the propensity of a brake to generate squeal and diagnosing squeal noise problems. Re-creating brake squeal is not an easy task. In many cases, brake noise occurs only during a portion of the process of deceleration or during braking with maintaining a constant speed - drag. Brake components often have to work in exactly the right conditions. These conditions may include speed, temperature, humidity, pressure and brake wear.

Key words: disc, brake, noise, laboratory, testing

DINAMOMETAR ZA ISPITIVANJE VISOKOFREKVENTNE BUKE DISK KOČNICA

UDC: 629.1.07

Rezime

Fenomen škripe kočnica se istražuje skoro 70 godina unazad. Tokom ovog perioda, većina ispitivanja bila su zasnovana na subjektivnim procenama i merenjima sa vozilom u pokretu na putu. Tokom godina, razvijena su mnoga laboratorijska ispitivanja sa širokim spektrom opcija i pristupa. Moderni dinamometri za ispitivanje buke kočnica su postali sofisticirane test platforme za identifikaciju sklonosti kočnica da generiše škripu i pri dijagnostikovanju problema buke. Ponovo stvaranje škripe kočnica nije lak zadatak. U mnogim slučajevima, buka kočnica javlja se samo tokom jednog dela procesa usporavanja ili tokom kočenja sa održavanjem konstantne brzine. Kočione komponente često moraju da rade u tačno određenim uslovima. Ovi uslovi mogu da uključuju brzinu, temperaturu, vlažnost, pritisak i habanje kočnica.

Ključne reči: disk, kočnica, buka, laboratorija, ispitivanje

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DYNAMOMETER FOR TESTING HIGH-FREQUENCY NOISE OF DISC BRAKES

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UDC:629.1.07

1. INTRODUCTION

Even if every noise during a vehicle test is recorded with a data acquisition device, the majority of the release procedures of a new brake system - at least in Europe - are based solely on the subjective noise ratings of the drivers during a vehicle test. The best known tests for researching the brake noise in road conditions are the Los Angeles City Traffic test (LACT) in urban conditions in the United States and Mojacar test in Europe. The length of Los Angeles City Traffic test is normally 5000 miles or 8000 km. An average of 250 miles or 400 km is driven per day. Some tests might run for more or less days than the normal 20 day length. On average, the number of braking per mile is 4 to 5. This is a slightly higher brake application rate than normal city traffic. As a result of years of experience, this test was unanimously accepted between automotive manufacturers and their suppliers in the United States. These vehicle tests can fully assess the actual brake noise performance and are very representative in terms of the end customer perception. However, all of these vehicle tests are expensive, time consuming, and usually occur too late to affect any structural changes if noise is detected on a test vehicle brakes. That's why the leading manufacturers have developed laboratory dynamometer tests that can shorten the brake noise development cycle and provide accurate and objective statistical data to evaluate brake noise performance. Results from the laboratory can be used to quickly affect the structural changes to optimize the brake noise performance.

Besides dynamometer tests, non-contact measurement techniques-laser metrology can be performed in laboratory conditions too. Holographic interferometry, Electronic Speckle Pattern Interferometry (ESPI) and laser Doppler velocity measurement are methodologies that are used to identify the cause of the brake noise and for evaluation of engineering solutions [1].

1.1 *Holographic interferometry*

Holographic interferometry is a proven technique for the measurement and analysis of the absolute displacement, both out-of-plane and in-plane, of a disc brake generating noise. The standard holographic interferometry technique has been developed such that both separate dynamic images of the in-plane and out-of-plane vibration, together with the combined dynamic image indicating absolute displacement can be represented. The technique makes use of a series of time-related holograms recorded from three different viewing perspectives of the brake. Each image records absolute displacement, but as each of the three holograms view the brake from a different viewpoint then each comprises varying degrees of out-of-

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plane and the in-plane vibration, dependant on their angular position. This varying degree of displacement allows their manipulation in order to identify and isolate the in-plane and out-of-plane contributions to the overall excitation so that each may be considered separately. The out-of-plane is represented as a three-dimensional image whereas arrows represent in-plane amplitude as a "quiver" plot. The results show that the disc modes are extremely complex for caliper brake system and further suggest that the in-plane vibration amplitudes are much larger in magnitude than out-of-plane vibration amplitudes.

The laser metrology has unique advantage and can be applied not only in case of body/chassis and power transmission, but also in research of complex issue of the brake noise. Engineers can find out information about basic causes of the noise and vibrations and provide guidelines for the optimal design [2].

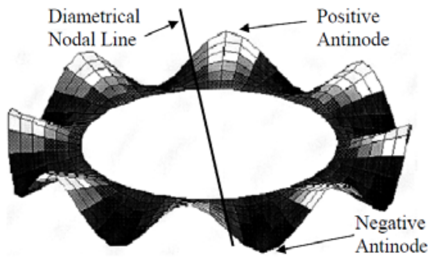
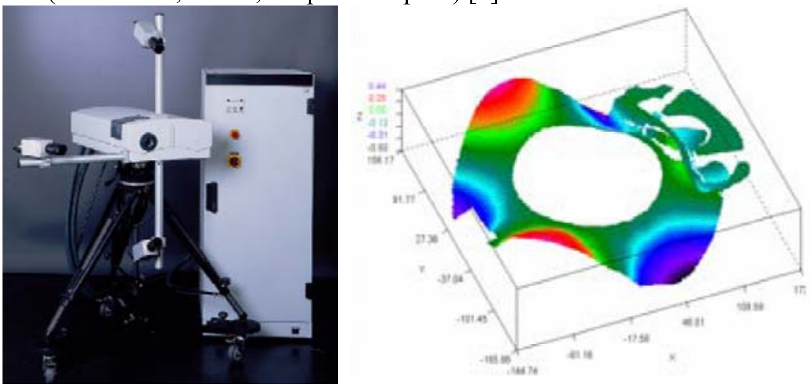


Figure 1: *Out-of-plane Vibration of a Brake Disc 8-Diameter Mode Order [2]*

Disc brake noise is generally associated with a friction pair of pad and disc. The vibration modes and natural frequencies have generally been associated with noise frequencies and there is strong evidence that the brake noise frequency is related to a specific natural frequency and out-of-plane mode of vibration of the disc - the frequency typically observed to be slightly lower than the corresponding free-free frequency. Previous research has mainly concentrated on the out-of-plane modes of vibration, where the disc is seen to exhibit purely diametrical modes of vibration, as shown in Figure 1. Therefore it can be said that there is a much greater appreciation of the contribution of out-of-plane displacement in a noisy brake system than in-plane displacement. The recent recognition of researchers to accept the importance of in-plane vibration during noise generation has therefore called for new techniques to be developed that enable complete field three-dimensional measurement [2].

1.2 *Electronic Speckle Pattern Interferometry*

In the area of structural testing, optical techniques have been successfully applied to the determination of dynamic response of the structure. Recently ESPI systems (Electronic Speckle Pattern Interferometry) using pulsed lasers are increasingly being used to replace conventional double pulse holography interferometers, and take not only the main advantages of holography relative to conventional measuring techniques, such as non-contact, full field measurement and sensitivity, but also use modern high speed cameras and computer techniques to capture and process the image (Figure 2). The ability of the pulsed ESPI has been further extended to the 3D measurement of dynamic response and modal analysis of vibrations. An important area of application is the field of the analysis of brake discs, with pulse ESPI systems providing complete maps of any component vibration during braking test (brake discs, drums, calipers and pads) [3].



a) *b)*

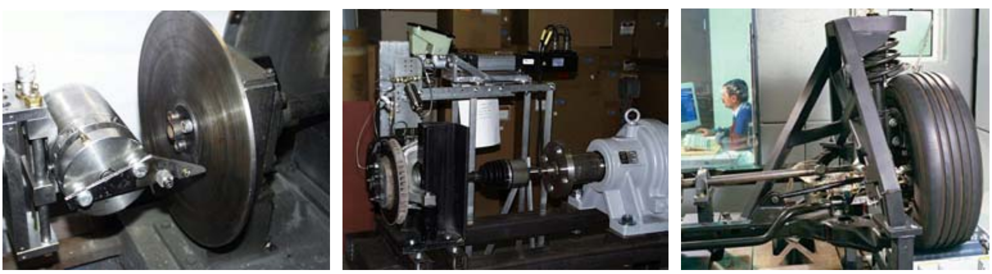
***Figure 2**: a) 3D-Puls ESPI System Q-600 b) Vibration mode of disc brakes [3]*

1.3 *Laser Doppler velocity measurement*

Scanning Laser Doppler Vibrometer - SLDV has some advantages over other measurement techniques because it is easy-to-use, applies low-power laser, has high spatial resolution, and it is non-contact technique. It can be used for the measurement of mode shape of components (rotors, pads and calipers) and modes of components/system. Operational Deflection Shape - ODS, or more precisely Instantaneous Squeal Mode- ISM of brake system rotor can also be recorded when the ISM is non-traveling mode. The procedure can be described as follows. When the microphone/acceleration sensor receives high-amplitude signal of single-frequency squeal/vibration, it sends a signal to the laser vibrometer to get it scanned. When the squeal disappears, scanning stops. This requires that an occurrence of squeal event takes a few minutes, which can be implemented with the brake dynamometer or drag vehicle test at a constant speed. Three-dimensional ISM can be obtained by combining the two laser heads/sensors for simultaneously scanning [4].

Disc brakes noise tests can be classified according to *the object of investigation*: the needle on the disc, the real disc brakes and disc brakes system with tire and suspension, Figure 3.

These test methods represent phases of approach to real structural connections of the braking system on the vehicle and the probability of generating a squeal in the case shown in Figure 3c) will have the greatest coincidence with the results in real road conditions.



a) *b)* *c)*

***Figure 3**: a) Pin on disc b) Disc brakes c) Disc brakes as part of the vehicle*

2. DYNAMOMETER

The objective of a brake noise dynamometer test is to reproduce the conditions and the noise that comes from the brake on the vehicle in motion. There have been many different approaches to this task, but the modern brake dynamometers are relatively similar in concept and operation. The current designs can be divided into two basic types. There is a brake or shaft-type dynamometer that drives the brake assembly from a shaft. The second category is a chassis dynamometer, in which a driven road wheel is used to drive a tire that drives the brake assembly.

2.1 *Shaft-type dynamometer*

Earlier noise dynamometers designed to reproduce brake noise were designed to performing only drag tests. These dynamometers did not have any means to represent vehicle inertia. With the implementation of SAE J2521 (SAE -Society of Automotive Engineers) [5,6], the first internationally recognized brake noise test procedure, these dynamometers were outdated. This test procedure requires both drag and regular stops. The application of inertial mass often gives smoother operation at low speeds, when the noise problems often occur. A typical example of such shaft-type dynamometer is shown in Figure 4 [7].

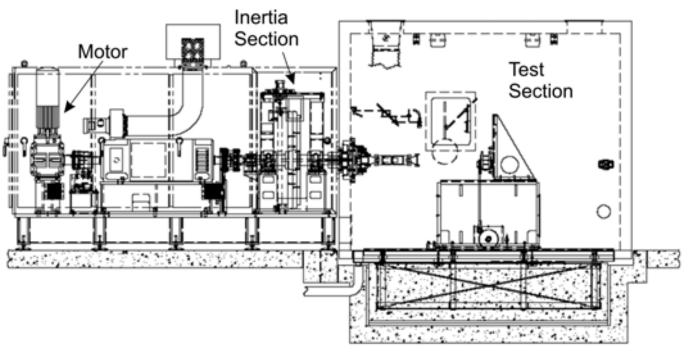


Figure 4: *Shaft-type brake noise dynamometer [7]*

2.2 *Basic Dynamometer Configuration and Operation*

An electric motor is used to spin the brake. For the modern automotive dynamometers the motor with power range from 75 to 300 kW is used. To perform realistic simulation of stopping a vehicle, the effect of the inertia of the vehicle must be represented. This can be done in two ways. The traditional approach is to use the discs on rotating shaft whose rotation inertia is equivalent to the linear inertia of the vehicle. A more recent approach is to use the electric motor to simulate the inertia of the vehicle. Modern brake dynamometers can provide both methods in order to give the user maximum flexibility in the simulating the vehicle braking system.

In order to test a wide range of vehicles it is necessary to simulate a range of inertia. The earliest approach to this problem was to provide a different inertia discs that can be manually mounted and removed on the dynamometer shaft. This is still done today on many dynamometers. In some dynamometers the inertia discs are automatically mounted and dismounted to the shaft. Depending on the size of the vehicle and the inertia increment being represented, these discs may weigh from several hundred to several thousand pounds.

Modern computers have enabled the simulation of the vehicle inertia through the control of the driving electric motor. Although for a number of reasons it is desirable to have at least one inertial disc on the dynamometer, it is now possible to provide extremely fine inertia increments and values above and below that physically mounted on the dynamometer with electrical simulation. There is a simple formula for predicting the amount of inertia that may be simulated on a particular machine [8]:

$$I_o = \frac{9549{,}3 \cdot P_m \cdot R_r}{a \cdot g \cdot \Omega}, \tag{1}$$

where

a - vehicle acceleration [m/s2],
g - gravitational acceleration - 9,81 m/s2,
I_o - available inertia [kg·m·s2],
P_m - maximum engine power [kW],
R_r - rolling radius of the wheel [m],
Ω - rotational speed of interest [rpm].

Whatever method is used, the goal is to match the vehicle inertia as closely as possible to assure the accurate reconstruction of the stop. Precise match will ensure that the brake under test sees the same torques, stop time, temperatures, and number of revaluation as seen on the vehicle.

It is also necessary to match the other operating conditions of the braking system as mentioned above. One such condition is the pressure applied on the brake. Not only is it necessary to match the application pressure using the same brake fluid as employed in the vehicle, but it is also necessary to provide the same rates of pressure application. In some cases, squeal can be sensitive to the rate of brake application or brake release [1].

Of course, all the same brake components must be used in dynamometer squeal testing as are used on the vehicle. The problem is in defining how far the match must extend. There is a clear agreement that the principal brake hardware must be the same. This includes brake pads or shoes, the caliper or drum, as well as their components.

The controversy occurs when one talks about the influence of the rest of the vehicle. Some researchers believe that it is necessary to include the impact of entire suspension system and the tire and wheel attached to the brake with the tire rolling on a surface at realistic vehicle loads. There is a general consensus that it is necessary for low-frequency noise. However, for high-frequency noise, such as squeal it is not clear that this is required. Requiring the tire rolling on a surface at regular vehicle loads requires a major change in dynamometer configuration. As can be seen in Figures 5 and 6, one can still have a basic shaft-type brake dynamometer with a roll added, or it may be necessary to go to a chassis dynamometer in which the vehicle or a vehicle corner is operated on a dynamometer [7].

A drawing of a passive roll dynamometer is shown in Figure 5. In this case, the road wheel may be removed for testing without the tire end wheel. With these dynamometers, the roll is said to be passive because it is being driven by the tire that is driven by the dynamometer shaft. As will be discussed later, the second approach is to use a chassis dynamometer in which the roll is driven (Figure 6). For squeal testing it is generally agreed that one should include the full suspension systems of the vehicle corner on the dynamometer.

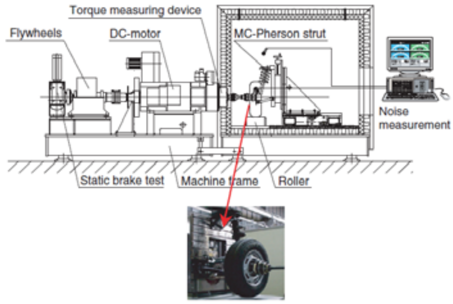


Figure 5: *Inertia Type Brake NVH Dynamometer [7]*

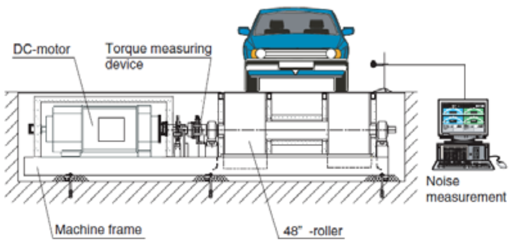


Figure 6: *Chassis Type Brake NVH Dynamometer [7]*

The environment in which the brake is operating is also important to reproduce the noise found on the vehicle. This includes both the acoustic environment and the environmental conditions. Acoustically, the environment of the vehicle operating on the road must be reproduced. This means an essentially free acoustical field above a reflecting plane. At the frequencies of concern, the road surface acts much like a reflecting plane. Otherwise, the brake sound energy is free to radiate with little impedance in all other directions. To approximate the free acoustic field in which a vehicle operates, the walls of the test enclosure are lined with an acoustically absorptive material. Materials used include glass fiber mats, mineral wool, and acoustic foams. Since squeal noise is defined to occur at frequencies of 1000 Hz and higher, the absorption of these materials is most important above 1000 Hz. A good approach is to maintain an absorption coefficient of 0.8 or higher above 500 Hz. This enclosure will have a floor of steel, concrete or other acoustically reflective surface. Such floors should approximate the acoustic characteristics of actual road surfaces [1].

To provide a sufficiently low background sound level to detect squeals, the walls of this room are also designed to provide very high sound transmission loss. Usually these rooms provide the background sound level below 50 dB (A). Double-wall construction is used to achieve the sound transmission losses necessary to assure sufficient transmission loss. Figure 7 shows the typical construction of chambers for recording brake noise [9]. In this case, two steel plates separated by 5 cm of foam or glass fiber insulation. Over the frequency range of interest, sound transmission losses of over 30 dB can be achieved.

As with any sound enclosure, it is crucial that flanking paths and seals at the door opening receive careful attention. If the enclosure is connected to the structure of the dynamometer of the laboratory floor on which noisy and vibration equipment are operating, all the work of careful enclosure design can be ruined. For more information on enclosure design, see [9,10,11].

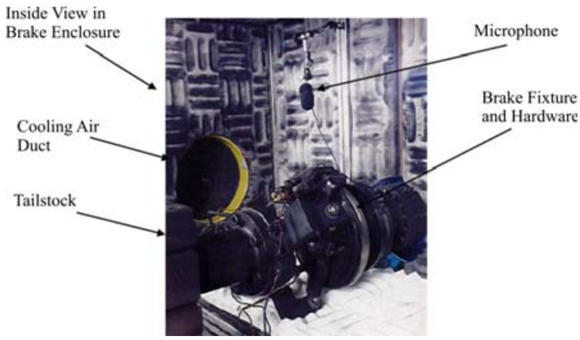


***Figure** 7*: *Typical Design of Sound Insulation [9]*

2.3 *Environment Control*

It is also necessary to represent accurately the environmental conditions under which the brake is operating. This may include both the temperature and relative humidity conditions. Later models of dynamometers provide the ability to control these parameters.

A problem that often arises is what is called "morning sickness". This condition occurs when the vehicle is parked outside overnight in the fall and winter months. The humidity rises overnight and may be higher than 90 % in the morning. At the same time, the temperature may drop to near freezing temperatures or below. These conditions of low temperature and high-humidity often encouraged squeal. In some cases, customers are experiencing noise under this scenario, when it is not noticeable under any other operating condition [1].

With an increasing awareness that environmental conditions can play a key role in squeal generation, most new noise dynamometers include systems to control the air temperature and relative humidity at the brake. Additionally, many manufacturers of components and vehicle manufacturers are clearly specifying environmental conditions that must be maintained during the noise test procedure. It is anticipated that these requirements will become more common.

2.4 *Data Acquisition and Control*

A critical part of the dynamometer system is the control and data acquisition system. A sophisticated control system is required to achieve the desired operating conditions and match the requirements of the current test procedures. The control system is required to follow prescriptions for the control of braking pressure, deceleration or braking torque. There may also be protocols setting desired starting temperatures and changes in pressure, speed, or other factors during the process of stopping [1].

To understand the occurrence of squeal properly, it is important to collect a large amount of experimental results. First the noise data must be acquired. In some instances this is as simple as collecting the data from a single microphone. In other instances, it may mean acquiring data from multiple microphones, accelerometers, and even laser Doppler vibrometers and optical holography systems. Often an accelerometer is located on the brake assembly for measurements of vibration in addition to noise. The data from the accelerometer can be compared with data from the microphone to determine whether the suspected squeals are from the brake and not extraneous noise from the dynamometer. In some cases coherence and coherent outpour power calculations are used for this assessment [12].

For the presentation of the actual vehicle operation and speed, the transition between tests, cooling air flow is typically provided to the brake assembly on the dynamometer. During normal operation of the brakes on the vehicle, especially at higher speeds, there is air flow over the brake that provides significant cooling. Furthermore, when the Initial Brake Temperatures - IBTs are specified, the cooling air allows faster cycles between tests. For instance, when an initial temperature of 50°C is specified, the next test may be at an initial temperature of 75 °C. Unfortunately, the 50 °C test may heat the brake at to over 200 °C. One can simply wait for the brake to cool down as a result of natural convection or, using cooling air flow, one can cool the brake to 75 °C in a few minutes.

During the noise test, it is important to record all the brake operating conditions to understand properly what was going on when the noise occurred. Typically this will include measurements of the brake pressure, one or more temperatures, speed, sound pressure levels and vibration amplitudes.

3. LABORATORY TESTING OF HIGH-FREQUENCY NOISE OF DISC BRAKES

Brake noise dynamometer developed in the Laboratory for testing the IC engines at the Faculty of Engineering in Kragujevac is shown in Figure 8. It is possible to comprehend the effects of vehicle inertia, due to the disc that is mounted on the rotating shaft. Disc's inertia is equivalent to the linear inertia of the vehicle, thus ensuring stable operation at low speeds that are relevant in terms of brake noise.

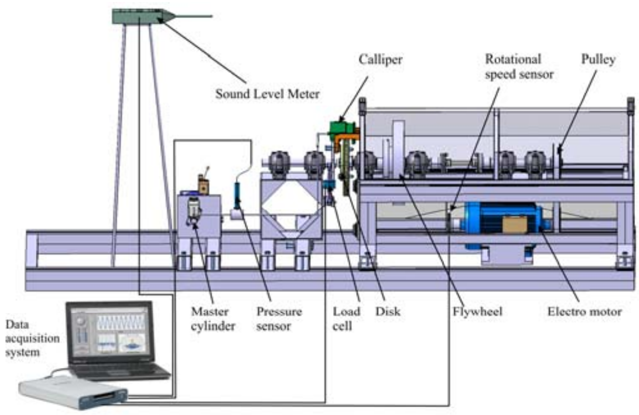


Figure 8: Brake Noise Dynamometer

3.1 *Description of the designed brake dynamometer*

Brake noise dynamometer consists from following functional units:

- Test bench with electric motor, power transmission and disc,
- Electric power installations,
- The installation for activation of disc brakes *i.e.* applying the braking pressure,
- Measuring equipment.

Detailed model of the developed brake noise dynamometer with measuring equipment is designed in software CATIA. Based on the 3D model, the technical documentation needed for building these technical solutions is made. Schematic view of measuring installations designed for testing the high-frequency disc brake noise is shown in Figure 9.

3.2 *Test bench with electric motor and disc*

Electric motor is manufactured by "SEVER" Subotica with nominal power of 4 kW at 2830 rpm. Torque on the drive shaft at the measuring point, including a dynamic component, must not be greater than the measuring range of torque sensor. The rotational speed of the drive shaft, *n,* is continually variable in range from 600 up to 3000 rpm with the stability $\Delta n<\pm$ 2% at the prescribed regime, with a maximum discrepancy δ_{max} = 1/30.

The electric motor is mounted stiffly on the base plate. The drive unit consists of: asynchronous electric motor fed by the frequency regulator (1), belt pulley transmission with ratio 1 (2), flywheel (3), disc brakes (4) (Figure 10). Rotation speed sensor (5) is mounted on the free end of the electric motor's shaft.

Flywheel on the input shaft is the disc with diameter R=0.35 m, a width of 0.045 m and mass 35 kg. Flywheel has the moment of inertia of 0.54 kgm^2 and it is corresponding to kinetic energy of the test vehicle, at low initial velocities, which are critical in terms of appearance of brake squeal.

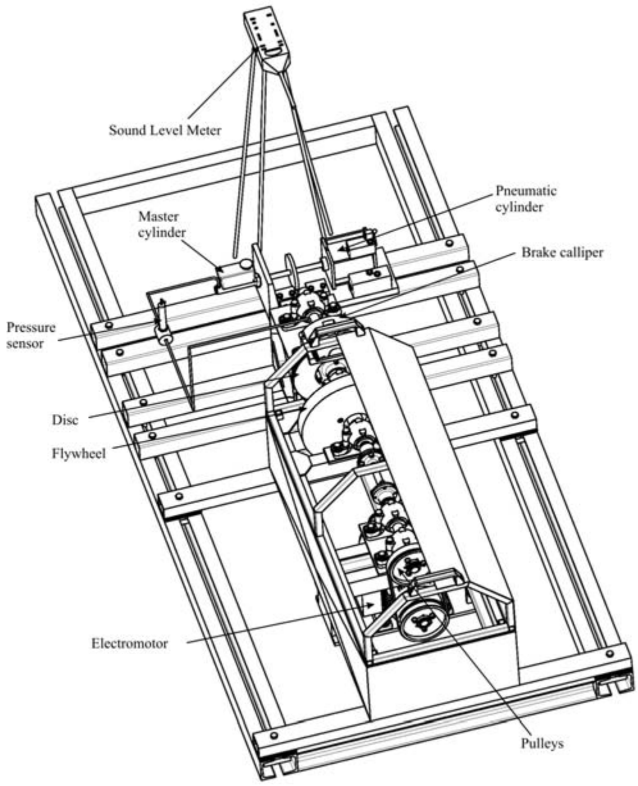
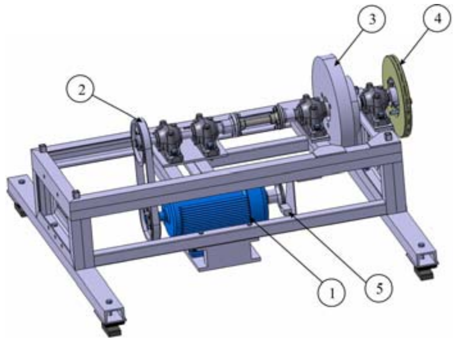


Figure 9: *Schematic view of laboratory installations designed for testing high-frequency noise of disc brakes*



1 – Asynchronous electromotor *2 –Friction Drive Pulley*
3 – Flywheel *4 – Disc brake*
5 – Rotational speed sensor

Figure 10: *3D model of the drive block of designed dynamometer*

3.3 *Measuring block*

The components of the measuring block formed to record the activation pressure in the cylinder of disc brake, *p*, brake torque, M_k, RPM of disc brakes, *n*, and the sound pressure level, *SPL*, as well as the connections of individual components are shown in photography in Figure 11 and the block diagram in Figure 12.

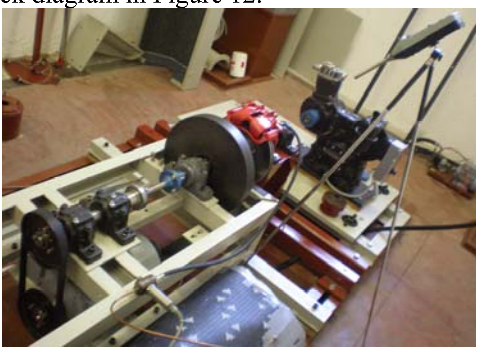


Figure 11: *Photo of laboratory installations designed for the brake noise investigation*

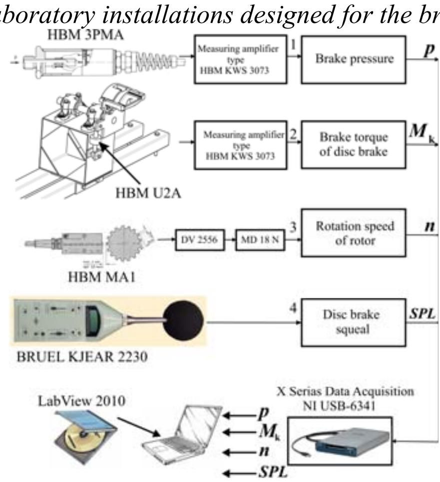


Figure 12: *The block diagram of measuring installation for disc brake noise investigation*

4. DISC BRAKE NOISE PROCEDURES

Basically, each vehicle manufacturer as well as each full brake system supplier has his own dynamometer test procedures and the standardization is quite poor. Today, two procedures are widely used and commonly accepted - the AK master (SAE J2522) for performance tests and the SAE J2521 [5], which was derived from it, for noise purposes. The SAE J2521 consists of an initial conditioning module after which the evaluation section is repeated three times with an optional fade and recovery module at the end. The evaluation module contains drag and stop brake applications as well as drag brake applications in forward and reverse vehicle direction. Nevertheless, an integral part of the release procedure of a new brake system is vehicle testing in Los Angeles (LACT) for US vehicles and in Mojacar for European vehicles.

One weakness that has been observed in the SAE J2521 procedure is the lack of a specification for environmental controls. A good example is the "morning sickness" problem. To replicate this kind of noise, J2521 would have to specify testing near 90% relative humidity and near freezing temperatures. It is anticipated that this may be one of the next steps in enhancing the capabilities of this matrix to screen for noise propensity. Summary of SAE J2521 procedure is shown in Figure 13.

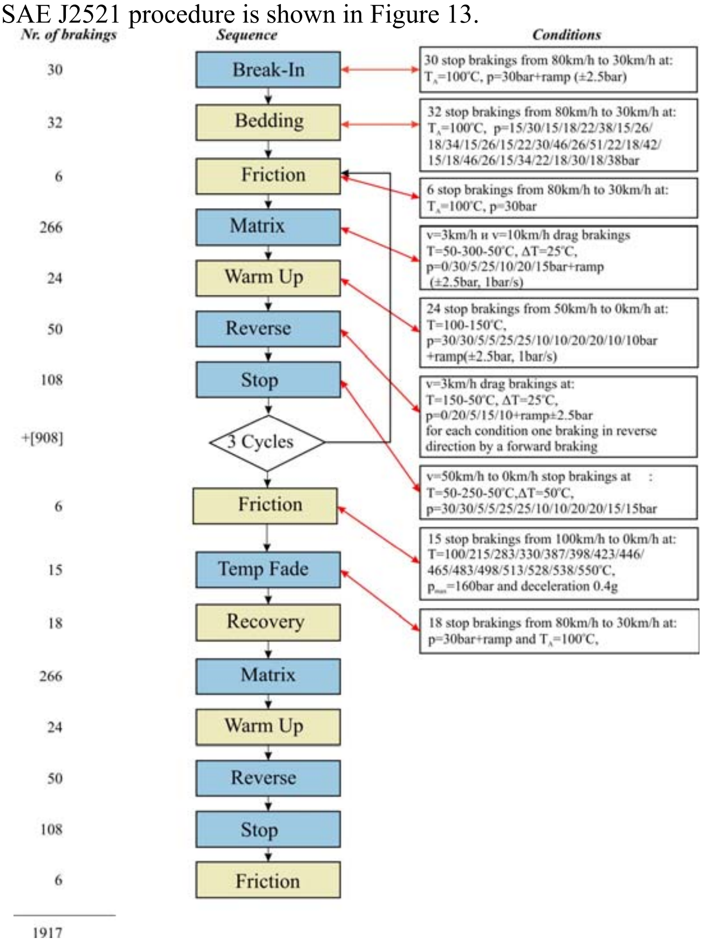


Figure 13: *SAE J2521 flow chart of the procedure [13]*

In the United States, the Los Angeles City Traffic (LACT) test is often used as the final evaluation of brake noise performance in vehicle development programs. This test on the streets of Los Angeles represents severe test conditions in an urban environment and is well known to generate significant brake squeal in susceptible vehicles. The advantage of this test procedure is that it has been shown to be effective in generating brake noise. The disadvantage is that it requires a vehicle and 2-3 weeks to run. It is difficult to incorporate such a lengthy test in the braking system development process.

The SAE J2521 procedure does not attempt to provide a substitute for the LACT test. Correlating a dynamometer test to a vehicle evaluation is difficult. There may be many differences in the components used in the two tests; tolerances and other variations may also have a substantial effect. Instead, SAE J2521 provides a fast process to screen for brake noise. Generally, without the optional fade-and-recovery section, J2521 can be carried out in 24-33 h on a modern chassis dynamometer. Many researchers go by the "rule of thumb" that the J2521 test is more severe than LACT. In other words, the J2521 matrix may detect the noise that does not appear during vehicle testing. Experience has shown that it is not often that a noise is found in LACT testing that is not seen on J2521 tests.

Although dynamometers have been used for brake noise evaluation for decades, the first internationally recognized standard was SAE J2521, published in June 2001. Therefore, this is a well-established area of testing with a relatively new standard for measurement procedure. There is still much to be learned and improved.

It is anticipated that there will be further refinements to dynamometer test procedures to improve their ability to screen for squeals found in actual vehicle operation. Some potential areas for improvements include:

- Establishing requirements for tests as well-defined environmental conditions,
- Refining the balance between the various matrix segments such as drags, decelerating stops, backward/forward sections, etc.
- Including modified burnish cycles for different types of friction materials or different applications.

SAE J2521 procedure proved to be a good tool for screening for the propensity of a brake to generate squeal. However, there is a need for further improvements, including the addition of environmental conditions. Experience has also shown that dynamometer testing does not correlate directly with LACT test results. The question to be addressed for the future is whether it is necessary and possible for a dynamometer-based screening procedure to match this particular vehicle procedure [5].

It is expected that dynamometer tests eventually become a better screening tools than LACT in that they will more accurately predict squeal, require less test time, and allows faster change-over. The dynamometer test will, however, be part of the development process, and not a final check of the braking performance.

4.1 *AK Noise Procedure*

The AK noise test procedure was developed by a working group in Germany that included automobile, brake systems and friction materials manufacturers. The fundamental concept consists of a matrix of tests that allows one to screen a braking system for its propensity to generate squeal. The AK noise procedure begins with a burnish phase in which new friction couple is permitted to accommodate itself. This is followed by a conditioning step. The essence of the noise monitoring is found in the next segment - city/country. Tests are run using two test speeds, 3 and 10 km/h. These low speeds are used because this is frequently where noise occurs. These are drag tests. The speed is held constant while a specified brake pressure is applied. A wide range of temperatures from 50 to 300 °C is utilized in this evaluation. During these tests, the pressure is variable instead of holding applied pressure at a constant value. The pressure is increased linearly from a value less than the normal value to a value greater than the nominal value and returned to the starting point. The triangular profile assures that if there is an applied pressure that is different from the nominal value at which noise occurs, this phenomenon is not missed.

After a conditioning segment, forward-and-reverse procedure is run. The concept behind this segment is to record the noise occurring when the vehicle is operating in the reverse direction. Testing is run at 3 km/h. It is not uncommon for a customer to complain about a squeal that occurs only when backing from a parking place.

Along with the city/country and conditioning segment, this component is repeated twice after the first run-through. The result is 990 stops total. Next, there is a short segment of deceleration stops. These stops are from initial velocity of 30 km/h to complete stop. Again, a range of temperatures and pressures are evaluated. A total of 54 stops is performed. The high heat conditioning section employs a brief series of high-temperature stops. This segment is taken from the AK master procedure. The city/country segment is then run again after the high-temperature segment. A key reason for this segment is to screen a greater propensity for noise after a high-temperature cycle. The forward/reverse segment is also repeated to evaluate the change in noise performance due to high-temperature cycling.

There are many ways to present the data from the AK noise test. One typical way is to plot the occurrence of noise. The number of noise occurrences can be recorded or, more frequently, the percentage of stops (or brake applies) in which the noise occurs is recorded. Generally, low percentages indicate good noise performance, while high percentages indicate poor noise performance. Every auto manufacturer and each supplier have algorithms for defining what acceptable level of noise is. Most would consider percentage below 1 as very good performance.

It must be noted that AK noise test method is not a published standard, but a standard prepared by a group in Germany with some restrictions on its distribution. Many versions of this procedure have been developed, in part because it is seen by some as proprietary. One of the cited limitations for this procedure has been that this does not incorporate enough deceleration sequences. Much of this procedure is focused on contact speed drags [1].

4.2 *The developed test procedure for screening high-frequency disc brake squeal*

The developed test procedure includes two categories of tests: braking with a constant rotation speed of the disc and different braking pressure and braking with a constant pressure and at different speeds. Tests that have been carried out correspond to braking with clutch-off *i.e.* with interrupted power transmission.

Test Modes:
Constant rotation speed of disc and different brake pressures:

- Constant speed in range from 250 to 1000 rpm.
- For every rotation speed, different pressures from 0.5 to 3 MPa with pressure increment Δp=0.5 MPa is applied.

Constant brake pressure and different rotation speed of the disc:

- Constant pressure in range from 0.5 to 3 MPa.
- For every pressure, different rotation sped from 250 to 1000 rpm with step Δn=50 rpm is applied.

In some braking regime (e.g. low brake pressure and high initial engine speed) there is only partial braking *i.e.* partial declining of disc's speed from the initial speed of v_1 to end speed of v_2 , while in other cases, it was braking until stopping of the disc brake.

Temperature measurement:

- Temperature was measured by a laser thermometer, Minolta, type Minolta-Land Cyclops Mini Laser for non-contact measurement of surface temperature. Thermometer has a platinum probe; temperature measurement range is from -50 ºC to 500 ºC, with accuracy of 1% of the measured value,
- Temperatures were measured at the midpoint of the brake disc.

It was important to make this temperature measurement in order to avoid overheating. If the system is cold, the working temperature should be raised up to 80 - 90 °C. This is achieved by braking with low pressure with engaged electric motor.

Measuring signals from all sensors are then led to National Instruments NI USB-type-6341 data acquisition system (Figure 14) which in interface with LabVIEW 2010 software, collects, analyzes and represents in real-time and stores the measurement results.

Figure 14: *National Instruments NI USB-type-6341 data acquisition system [14]*

During experimental research measuring value with the maximum change rate in time is the sound pressure level so its characteristic sets sampling speed. An examined high-frequency disc brake squeal has a maximum expected frequency of 20 kHz because it is the human audible range limit [15]. According to Nyquist-Shannon sampling theorem [16, 17] it is required that the sampling frequency must be at least twice higher than the frequency of the signal that we want to transfer to digital form, otherwise aliasing effect occurs. Therefore, the selected sample frequency was 50 kHz. So the data sampling period was $2\cdot10^{-5}$ s.

Selected time period of signal recording was 5 s, and this is, with respect that the duration of the braking process is maximum 1 s, a sufficient period to capture the processes before and after the brake (the level of environmental noise, influence of the drive unit noise, the effect of residual braking to stopping time of the disc).

The characteristics of the data collection process during the performed tests are shown in Table 2.

Table 1: *The characteristics of the data collection process*

Sampling period	Δt	$2 \cdot 10^{-5}$ s
Measuring time	T	5 s
The frequency band of signals	f_g	0 до 20 kHz
The minimum sampling rate	$f_{min}=2f_g$	40 kHz
The sampling frequency	$f_{\max} = f_o = \frac{1}{\Delta t}$	50 kHz
The number of samples in one measurement	N	250 000
Number of repeated measurements	n	3
Measurement Error	$\sqrt{\frac{2}{N}}$	0,0028

CONCLUSIONS

Even today, if a significant number of vehicle tests is needed to be performed for final verification of the brake design, dynamometer tests are used early in the development stage, when representative vehicles are not available or during problem resolution stages, when a better controlled design of experiment is needed. Inertia dynamometers are used to test the single brake assembly up to the whole mechanical brake system, often also including suspension. Dynamometers are used in different operational modes. During a test procedure, the time between the brake applications is either firmly specified (time controlled mode) or the next brake application is started when a specific (e. g. disc) temperature is reached (temperature controlled mode), independent of the cooling time in the latter case. For special test procedures, the control parameter during a brake application is either brake line pressure (pressure controlled) or the brake torque (torque controlled), besides the initial and final velocity and in the case of temperature control also initial temperature.

The objective of this paper is to develop original methodology for investigation the mechanism of high-frequency oscillations of a disk brake system that manifest themselves as squealing sound and their effect on indicators of braking efficiency and the overall vehicle noise levels. Comprehension of this phenomenon and elimination of the adverse effects is a priority in today's development and refinement of motor vehicle brakes. This goal is achieved by development and implementation of an experimental system for testing the brakes that can also verify the simulation results.

ACKNOWLEDGMENTS

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