

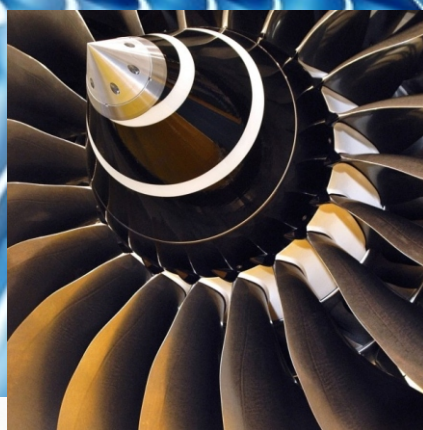
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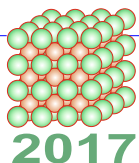
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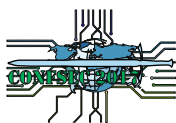
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UNDERWATER HULL OBSERVATION SYSTEM ARMUS

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Abstract: The ARMUS system presents a new approach for observation of the hull of large marine vessels providing fast, but at the same time very accurate and reliable method for underwater inspection, detection and identification of various threats. ARMUS is magnetic tracks based mobile robot capable to stick-on and move along iron surfaces and operate in extreme environmental conditions, including under the water surface. Among the main advantages of the ARMUS system is its ability to reduce significantly the time for hull inspection by providing in process estimation and assessment of the hull while the ship is in motion. This advantage reduces significantly the standby time of the ships when inspection of the hull is required before entering a port. On other hand the in process inspection allows the vessels to be observed continuously during the voyage. Another important advantage of ARMUS is its capability to stay underwater as long as it is necessary. The system is powered and controlled directly by the vessel it inspects and may operate in long cycles without maintenance and recharge.

Keywords: AUTONOMOUS VEHICLE, UNDERWATER ROBOT, OBSERVATION AND INSPECTION, SECURITY OPERATIONS

1. Introduction

The ARMUS robotic system is remotely controlled ship hull observation and inspection system based on magnetic tracks chaisee that guarantee the motion of the robot on the submersed and internal (cargo) part of the vesicle. This advantage makes ARMUS the desired observation system for large marine vessels like oil tankers, cruisers and military vessels. Also ARMUS can be used in inspection of see based oil drills and water channel gates. The system is designed for search and identification of limpet objects like naval mines, improvised explosives or contraband traffic attached to the underwater hull of the ship, but it also can be used in situations related to inspection of the hull condition not only externally, but also internally as it covers the ATEX requirements and can work in hazardous environment. ARMUS system consists of three main units: mobile observation robot, combined multipurpose cable and remote control unit. The robot control is intuitive and does not require specific skills from the operator. As feedback the robot provides live visual and specific sensorial information from the observed area directly to the remote control unit and to the onboard computer network, if necessary.

2. State of the art in marine vehicle inspection robots

The hull management has always been among the most important marine issues that if organized in a proper way can even decrease the fuel consumption of the vehicle. The hull roughness plays an important role in the ship's speed and that directly affects the fuel consumption making the ship less environmental friendly. The build-up of marine organisms on ship's hull, also called bio-fouling, reduces the ship's speed by up to 10%. To compensate the drag, it is said that, a ship may have to use about 40% more fuel [1]. Many research organizations and marine companies work on the development of different remotely operated and autonomous systems for hull observation and management, but only a few of them had reached a market accepted result. Among them is “HullBUG” (Fig. 1) developed by SeaRobotics and funded by U.S. Navy Office of Naval Research (ONR) to tackle this issue.

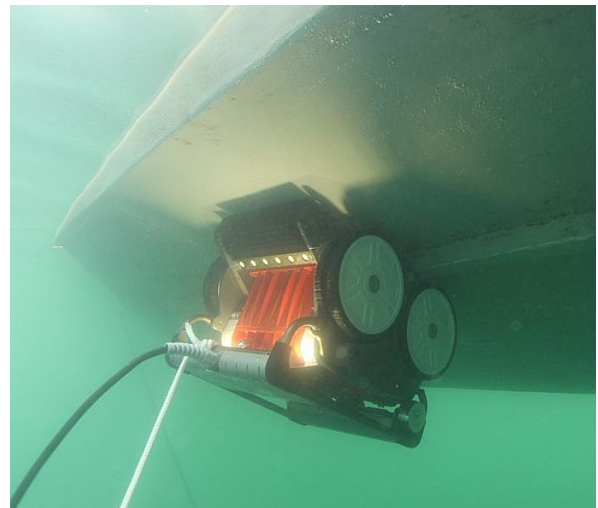


Fig. 1 HullBUG hull cleaning robot [1]

The Robotic Hull Bio-inspired Underwater Grooming tool (Hull BUG), is a small autonomous vehicle weighing 30 to 40 kg. It uses four wheels and attaches itself to the underside of ships, using a negative pressure device that creates a vortex between the BUG and the hull. It crawls on the hull surface and performs frequent grooming (light cleaning of fouling films). Sensors provide obstacle avoidance, path cleaning, and navigational capabilities. A fluorometer lets the robot detect biofilm and then it uses rotary brushes or water-jets to scrub the fouling film off [1]. The developer of the robot estimates that, if these robots are put into practice, there can be 5% improvement in fuel efficiency through proactive grooming, translating into a savings of \$15 billion annually for the shipping industry worldwide and reduction in 1 billion tons of greenhouse gases emitted by the fleet.

Similar efforts to develop hull cleaning robots is being done by Keelcrab (Fig. 2). The product I-keelcrab is semi automatic robot fitted with ip68 high-resolution camera and can be guided by smartphone or tablet. Keelcrab-one is an underwater robot, which can be controlled by a wired remote control with live video feed [1].



Fig. 2 Keelcrab hull cleaning robot [1]

The inspection of the hull roughness is not the only aspect of the hull management. Marine vessels are subject to numerous and regular inspections and maintenance measures. In general, inspection of huge cargo ships for cracks, corrosion or any wear to ensure that they comply with rising safety standards is a time-consuming task for surveyors. Moreover, a big part of the inspection cannot be performed by the surveyors from inside the ship and they have to dive in order to inspect the ship's hull under the water level. In most of cases, the surveyor performs only a visual inspection. In order to reach each spot on the ship, scaffolding has to be erected in the cargo holds. Typical heights of cargo holds are 15-20 m. The installation of the scaffolding usually takes several days, before the surveyor can start the inspection process. Every day the ship stays in the dock and out of service results in a significant loss of money for the ship owner, making this (currently necessary) preparation time is very expensive [2].

The EU-funded R&D project MINOAS [3] (Marine INspection rObotic Assistant System) addresses this challenge in an attempt to develop concepts for the automation of the ship inspection process. The key idea of the project is to develop and test a fleet of semi-autonomous robots which can provide visual data as well as thickness measurement data to the surveyor without the need for setting up scaffolding prior to the inspection process.

The result of the project is a lightweight and low-cost Ship Inspecting Robot (SIR). Its prototype is capable of conducting a visual inspection of ballast tanks and hard to reach parts in huge cargo vessels. Its four magnetic wheels and overlapping wheelbase enable SIR to navigate the I-beams and other awkward obstacles found inside ship ballast. These robots can be controlled via a wireless transmitter with live video feed and its four infrared distance sensors help in detecting edges and obstacles. Figure 3 shows the prototyped robot in action.



Fig. 3 Ship Inspecting Robot [3]

The biggest advantage of SIR is the reduction of time and cost for inspection of the cargo holders. But the SIR system unfortunately has some disadvantages. The most critical problem of SIR is the small contact surface that its wheels have with the walls cargo holder. Even small obstacles or rust on the surface can cause the fall down of the robot. Those series of fall downs of the robot limited its commercialization. The wheels based design (Fig. 4) does not allow a larger contact surface between the inspection robot and the inspected area. Another disadvantage of SIR is the limited by the battery pack operation time and its ability to inspect only the "dry" surface of the ship.



Fig. 4 Structure of the SIR wheel and the contact surface [3]

The problem of SIR's small contact surface is partially solved in MARC: Magnetic Autonomous Robotic Crawler (Fig. 5) [2].

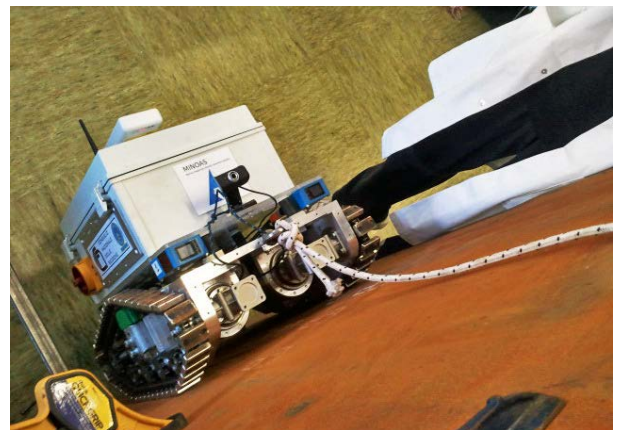


Fig. 5 MARC final release, during vertical climbing test [2]

As published by the authors [2] the MARC actuation system is basically constituted by a couple of magnetic tracks, actuated by an electrical motor coupled to a gear-head through a suitable mechanical interface. The magnets are lodged in dedicated housing connected to a chain to constitute a track. The weight of the robot is approximately 50 kg. and it is powered by batteries. The system is not capable to work underwater and to inspect the external part of the hull. It can operate only in the cargo holders. Magnetic tracks are selected as passive adherence system maximizing the contact area with the metal surface. In particular, in virtue of the exerted force, neodymium magnets are used. These magnets, also known as NdFeB, NIB, or Neo magnets, the most widely-used type of rare-earth magnets, are permanent magnets made from an alloy of neodymium, iron, and boron to form the Nd₂Fe₁₄B tetragonal crystalline structure. In the particular case, the employed magnets are characterized by a dimension of 40x22x9 mm and a 20 Kg attraction force. The driving element is composed by an aluminum frame where magnetic tracks, motors and traction gears are fitted. Magnetic elements are covered with anti-skid tape (Fig. 6) in order to increase the friction factor between the surface and the magnet itself, increasing the traction factor. A couple of small rear wheels are mounted in order to increase stability and adherence on vertical surfaces, avoiding possible rollovers.

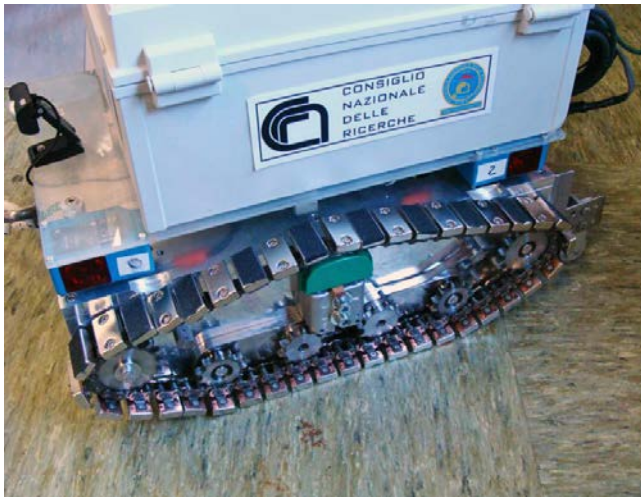


Fig. 6 Rectangular magnets and anti-skid tape [2]

What is easily noticeable from the tracks' configuration shown on Fig. 6 is that the suspension system of the robot is very much like the suspension of the systems that are forced to the surface by the gravity. Most of the track support wheels are placed on the lower part of the track that will be in contact with the inspected surface. But during the contact with the surface the force of the gravity and the weight of the robot will try to flip it over and the track will try to escape from the support wheels. Another disadvantage of the robot is that it does not use effectively all of its magnets. It is noticeable from the figures that the robot contact with only a quarter of its magnets to the surface. This approach requires use of stronger magnets that on other hand increase the overall weight of the systems and reduce its efficiency.

3. ARMUS underwater hull observation system

The ARMUS robotic system takes into consideration all current and past developments in the field of hull observation and inspection systems. The first generation of ARMUS robot (Fig. 7) is designed in a way that the robot reverently works under water. It is a three axes tracked system that attracts to the hull with the help of neodymium magnets. The total weight of the robot is 20 kg. and it is remotely controlled by cable that can be as long as 100 meters. The connection cable is used only for the control of the robot and transfer of video signal from the robot to the control unit.

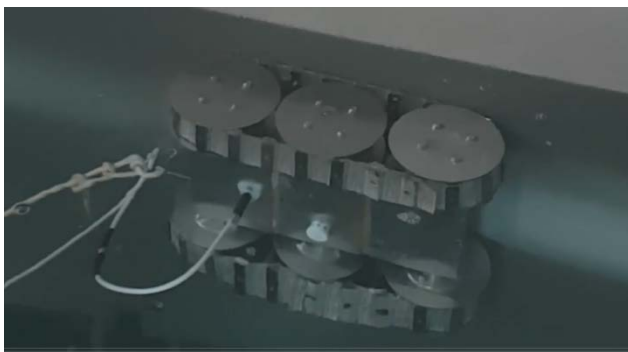


Fig. 7 First generation ARMUS robot inspecting ship's hull under the water line

The first generation of the ARMUS robot offers only visual inspection of the hull. The robot is powered by two DC motors coupled with gearboxes. The power is provided by batteries that are mounted inside the robot. The video signal is broadcasted as a radio signal through the connection cable. The same is with the control commands of the robot for speed and direction. Each track is controlled independently. The attraction force that the robot tracks provide to the surface is 336 kg.. This attraction force is enough to keep the robot limped to the hull surface even when the robot is operation while the ship is in motion. This is one of the major benefits of the robot as it reduces significantly the time for

inspection as it can start and continue during the approach of the ship to the port and does not require the ship to be in standby mode. The biggest disadvantage of the first generation ARMUS robot is the limited by the batteries operation time that is up to two hours, depending on the hull's condition. A second disadvantage is the limited maneuverability on a dry iron surface. Those disadvantages are overcome with the second generation of the ARMUS robot shown on Fig. 8.

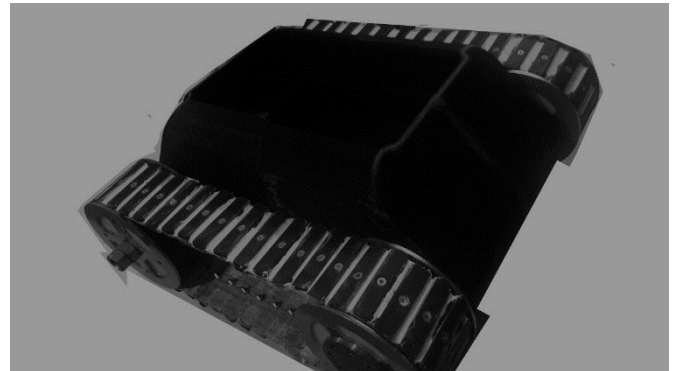


Fig. 8 Second generation ARMUS robot designed to work both underwater and on "dry" iron surface

The second generation of the ARMUS robot is designed in a way to be able to stay unlimited time underwater and to work on both sides (external and internal like cargo holders) of the ship's hull. The robot overall weight is 35 kg. and the attraction force of the tracks is 672 kg., doubled from the first generation. The robot uses a different approach from the first version. It is connected to the ship by a cable that supplies the robot with current between 110 V and 220 V. The same cable is used for the communication and control of the robot. The length of the cable can be up to 200 meters. The track system of the robot uses a two wheels type suspension without the use of extra support wheels. This type of suspension guarantees the successful maneuverability of the robot on a dry and wet surface and at the same time does not allow the robot to lose its attraction force and to fall down from the inspected surface. The good attraction between the tracks and the iron surface gives the robot the ability to overcome different obstacles even when it is climbing the walls of the cargo holder or the underwater part of the hull. The schematics of the robot submerge system is given on Fig. 9.

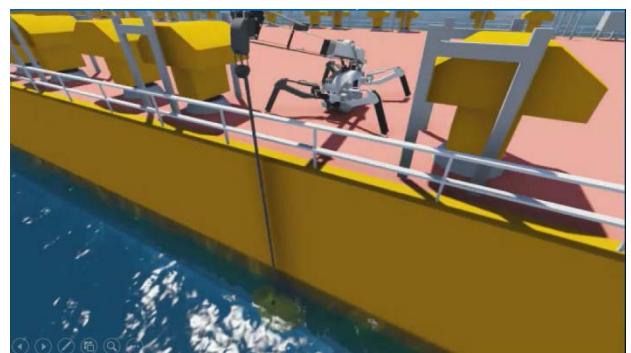


Fig. 9 ARMUS submerge system

The robot is controlled through a specific graphical user interface. The control system consists of two main parts – remote control unit and onboard control unit. The onboard control unit is a computer linked to a series of controllers that are used for the control of the motors, the communication and the observation systems. The robot generally carries one pair of cameras – a high resolution observation camera and a secondary rear camera. The robot can carry also a set of scanning sensors depending on the particular needs of the customer and the type of observation that is necessary to perform. The engines are powered by AC – DC converters. The second generation of ARMUS has a unique design

of the suspension as the motors and the gearboxes of each track are mounted outside the sealed area of the robot body. Each motor is placed separately in a sealed chamber and the torque is transmitted to the track leading wheel by an open type clutch. The use of simple DC motors allows the robot motion even the motor chamber is flooded. The motor drivers are designed in a way to work even if the motor is in a short connection. This will help the operator to control the evacuation of the robot from the water in case of decompression of any robot's seal.

The remote control unit is a box that provides the power a communication supply to the robot. It also broadcast a control communication signal on a private network to a tablet from which the operator drives the robot and observes the provided data. The control box is given in Fig. 10.



Fig. 10 ARMUS control unit

An important issue of the control system is that the graphical user interface of the control is designed to be user friendly and does not require a specific skills of the user in order to deal with the robot control. The control system has its internal procedures that prevent the operator of giving tasks that can cause a damage or malfunction of the robot.

Another advantage of ARMUS is that it can stay in hibernation as long as necessary. The system is of the type plug and play. The moment ARMUS control unit is plugged to the power supply of the ship and the connection cable of the robot is plugged to the control unit ARMUS is ready for action. The system for observation of the hull of large vessels is designed to search and identify limpet objects like naval mines, improvised explosives or contraband traffic attached to the underwater hull of the ship. It is able to perform tasks under various weather conditions during the ship motion. It has two main operational modes:

- Semiautomatic - in this mode the system motion is manually controlled while the automation supports the search and identification.
- Automatic - in this mode the system autonomously follows the predefined root.

Technical specification of ARMUS

- Capability to stick to an iron hull and travels underwater during the ship motion;
- Power supply to observe hulls with 275 meters in length and 16 meters depth;
- Maneuverability while ship sustains 20 knot water flow;
- Cognitive abilities to identify ship system (hull features) and hull structures and to define its own relative location;
- Generate hull refer point coordinates and transmit them for hull map generation;

- System is capable to avoid or traverse over typical hull shape obstacles (steps, grates, patches, weld seams);
- Remote control based on:
radio + umbilical cable communication.
- Up to 30 days standby mode (manual activation at any time);
- Up to 6 months hibernation mode (only programmed activation).
- Once activated - Autonomous Operation within 10 minutes;
- Complete ship survey within 72 hours;
- Extra a payload of 2 kg.;
- Operational Temperature: -5° to 25° C of water and -25° to + 50° C of the air;
- Storage temperature: -35° to 55° C.

4. Conclusion

The ARMUS underwater hull observation system is the only one known available at the moment robot capable to travel attracted to the ship surface both under the water and on the “dry” ship body like the cargo holders. ARMUS is the ideal tool for fast and reliable ship observation in order to perform hull management. Its ability to stay underwater as long as necessary and more important to perform its tasks during the ship motion makes ARMUS a desired tool for security measures. The use of ARMUS is organized in natural manner and does not require specific skills from the personnel. The complete system is designed in a way to be durable and easy for maintenance. All materials used are corrosion free and resistant to aggressive and hazardous environment. Moreover ARMUS is designed in accordance with the ATEX directive and can work in hazardous and explosive environment like oil tanks.

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IMPACT OF PARAMETERS OF POWER TRANSMISSION SYSTEM STATE ON VEHICLE'S WORKING LIFE AND POWER EFFICIENCY

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Abstract: This paper deals with a change of states of power transmission system elements up to the achievement of boundary conditions based on a material fatigue. Depending on the level of change in states of elements, an estimation of a remaining working life of a power transmission system was given. Additionally, an estimation of a change of conditions of power transmission system elements, as well as the impact of states of elements on a power efficiency of a vehicle are presented. Besides an analytical consideration, the simulation models were created and the results of simulations presented.

Keywords: WORKING LIFE, POWER TRANSMISSION SYSTEM, POWER EFFICIENCY, FATIGUE, SIMULATION

1. Introduction

The calculation of a working life of elements and power transmission systems in vehicles are the most often considered in reference literature starting from its state, before use, not taking into account the changes of states of elements during exploitation.

However, the changes emerge during the exploitation of the power transmission system influencing their working life on the basis of a material fatigue.

Thence the consideration of the change of states of elements and systems and impact of these changes on their working life, i.e. time of operation from the given initial state to the achievement of the working life is interesting here.

During the operation of a power transmission system in a vehicle, a mechanical, thermal, chemical and other loads act upon individual elements and on a system as a unity. Deformations, breakdowns, fatigues as well as destructions of individual elements occur under the influence of these loads leading to the changes of the basic parameters of states and parameters of functioning of the system. Additionally, it causes changes of physical and mechanical characteristics such as decrease of strength, resistance area, resistance to wearing etc.

Some of these processes such as deformations, wearing and corrosion emerge on the surfaces of elements, while the processes of weakening of materials occur inside these elements.

A particular importance for the remaining working life is related to the processes of weakening and processes of wearing.

Also, vehicle power transmission systems are complex functional and constructive systems which have to sustain a load exerted by both engine and surface.

The structural overview of a vehicle power transmission system can be seen on Fig. 1, [1].

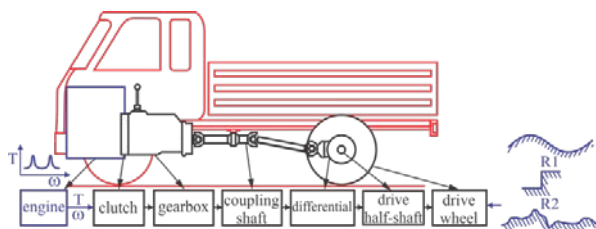


Fig. 1 Structural overview of a vehicle power transmission system; R1- surface (general case), R2- stair-step obstacle, R3- random shape surface irregularities, [1].

These loads create the resulting load with two main components: quasi-static and dynamic one.

During the operation of a power transmission system, all the above-mentioned loads are permanently present.

The basic types of stresses of materials leading to the changes of the states of elements in a power transmission system are the exceeding of a static strength, fatigue and wearing.

A fatigue destruction is a long-lasting process that most often begins in so-called weak zones in material. This process can, as the time elapses, be divided in the phase of occurrence of an initial fatigue crevices, phase of widening of the fatigue crevices and a phase of final destruction. Awareness of a fatigue process characteristics, namely a speed of expansion of fatigue crevices can help to foresee a time of an occurrence of failure, therefore, it can be concluded that a fatigue characterises the state of a mechanical part and can be used as a diagnostic parameter.

From the standpoint of the change in state of elements, friction and wearing of coupled surfaces of elements have a particular importance. Therefore, this type of changes of state (destruction) also characterises a technical state of the coupled elements and can be used as a diagnostic parameter.

Note that the state of elements of a power transmission system in vehicles changes during the exploitation due to wearing (dilapidation) of contact surfaces of coupled elements and deformation of elements.

A wearing is a process of a gradual change of sizes and shapes of the coupled elements, when during their mutual motion a phenomenon of friction between their surfaces occurs. A state and shape of frictional surfaces, speed of motion, loads of coupled elements as well as a range of other factors, significantly influence the process of wearing. An occurrence of clearance (play) in coupled elements representing a non-linearity with a zone of senselessness and linear parts.

A change of parameters of states of elements on the basis of clearance is considered below, and the estimation of remaining working life, as well as the impact of the state of elements on an vehicle's power efficiency are given.

Additionally, an analytical consideration of the remaining working life, simulation model and the results of simulation are also give.

2. Achievement of a boundary state on the basis of a material fatigue

In the analysis of the achievement of boundary state on the basis of material fatigue, a remaining working life of elements of power transmission system in vehicles from any determined state to the occurrence of a boundary state is considered.

The boundary state of elements of mechanical power transmission system in vehicles is mainly related to their boundary wearing.

The basic indicators of the wearing process are:

- Change of a size of elements (volume of wearing) in the process perpendicular to the friction surface – linear wearing h , Fig. 2a,
- Change of speed of wearing $v=dh/dt$, Fig. 2b.

There are three periods in the process of wearing: I - period of breaking-in, II - period of normal operation and III - period of breakdown wearing.

Dependence of wearing on an operation time of element, in a general case, is given by the following equation, [2,3]:

$$h = h_0 + c \cdot t^\beta \quad (1)$$

The value of h_0 characterises a wearing of an element on the completion of elaborating of the system, point 1 on Fig. 2. The coefficient β represents a deterministic value determined by a structural solution of contact surfaces of the coupled elements in the conditions of their work.

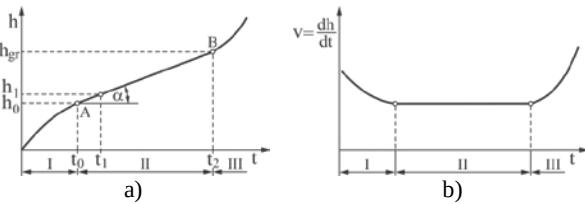


Fig. 2 The wearing process [2,4]; a) change of size of wearing in time, b) change of wearing speed in time

The boundary wearing according to the criterion of a flow of wearing curve is characterised by a critical point (point B on 2) dividing an area of an established and catastrophic wearing. For the elements having coatings resistant to wearing, or reinforcement with a chemical and thermal processing, the boundary wearing is determined by a thickness of a reinforced layer (finish).

A rise of load of elements, as well as a significant reduction of their working life up to the achievement of a boundary state, namely the loses of operative ability, occurs due to wearing of elements of power transmission systems in vehicles. The reduction of a working life is bigger if a vehicle is longer exploited in transitional modes.

Besides the technical criteria, a boundary state of the power transmission system in vehicles is also determined by economic criteria that are based on a minimising of specific costs.

3. Remaining working life

The remaining work life of the power transmission systems in vehicles represents a time of its operation from the given initial state to the achievement of the boundary state, when its further usage is unallowable, or not appropriate an assessment of the state as well as an estimation of the remaining work life of the transmission system are given according to technical and economic criteria.

With the ultimate wearing values defined, according to technical criteria ($h_{gr,T}$) and economic criteria ($h_{gr,E}$), the valid value is the lower one:

$$h_{gr} = \min(h_{gr,T}, h_{gr,E}) \quad (2)$$

Ultimate wearing value according to economic criteria ($h_{gr,E}$), in most of the cases, is close to the value of allowed wearing specified by the manufacturers of transmission elements.

Residual lifetime of transmission elements, from the moment t_1 , which corresponds to wearing h_1 until the moment t_2 , which corresponds to maximum allowed wearing h_{gr} , may be determined in the following manner.

Through wearing curve linearisation, in the part correlating to normal wearing, Fig. 2a, we get the approximately constant wearing speed which can be expressed as [3]:

$$v_H = \frac{h_{gr} - h_1}{t_2 - t_1} = \text{tg}(\alpha) \quad (3)$$

Residual lifetime of the element T_p is:

$$T_p = t_2 - t_1 = \frac{h_{gr} - h_1}{v_H} \quad (4)$$

Total lifetime of the element, until reaching maximum allowed wearing, including warm-up time is:

$$T = t_0 + \frac{h_{gr} - h_0}{v_H} \quad (5)$$

If h_{gr} is replaced with h_{doz} in the equations (4) and (5), a remaining work life of elements of the power transmission system in vehicles to the achievement of an allowable wearing is obtained.

The remaining, as well as the total working life of the power transmission system can be expressed by a traversed distance.

4. Impact of the state of a power transmission system on energy efficiency

As it has been mentioned, an occurrence of clearance is an indicator of the state of a power transmission system.

A clearance rises due to wearing, while a working ability of a vehicle, namely its working efficiency and energy efficacy decrease. The changes of its working efficiency and energy efficacies are techno – economic indicators of the change of the state of a vehicle, [5].

This impact can be illustrated by the following example.

Due to the presence of clearance in aligned link of a drive shaft a critical number of rotation of half-shaft decreases. The real value of the critical number of a drive half-shaft rotations n'_{kr} is less than the calculated value n_{kr} due to an insufficient strength a support, insufficient balance of a shaft and inaccuracy of the aligned joints. Given the abovementioned, a proper corrective coefficient K which, for a new drive half-shaft, is $K=0,9 \div 0,95$ is introduced. Having in mind this coefficient, a real value of the critical number of rotations of the drive half-shaft is:

$$n'_{kr} = K \cdot n_{kr} \quad (6)$$

Depending on the wearing of joints of shaft, the coefficient K decreases, Fig. 3, [5,6].

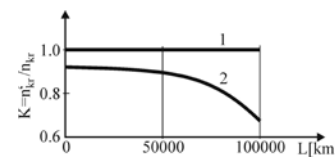


Fig. 3 The change of critical number of rotations of a drive half-shaft, [5,6]

5. Model of power transmission system and the results of the simulation

The system for a power transmission in vehicles, for the presence of a change of state in joints of elements (play – clearance) represents a complex elasto-inertial non-linear system with a torsional loading.

The total clearance in the system for power transmission in vehicles is divided into several places of kinematic chain, in clogged pairs, couplers, articular drive half-shafts and other joints of elements. A clearance in the power transmission system rises during the exploitation of a vehicle due to wearing. By the rise of clearance, each movement, change of speed and braking of the elements of the power transmission, especially when there is no its previous loading, cause impact stresses in the elements of the system for power transmission by an increase of clearance. For the

presence of clearance, the dynamics of loadings fairly rise and these can achieve the values that several times exceed the static loadings from resistance forces.

An impact of clearance on the dynamic loading of elements of the system for power transmission in the conditions of occurrence of impulse loadings is particularly expressed.

The impact of clearance and material fatigue reduces the reliability and work life of the power transmission elements.

For the purposes of determination of a loading of the power transmission elements, with the presence of clearance in some connection of elements, it is necessary to set a mathematical model of a torque oscillation of the power transmission as a non-linear system.

Depending on the simplification degree, the power transmission can be represented with a model having two or more masses, with one or more non-linearity elements like a senselessness zone representing a clearance.

An analytical determination of a loading of power transmission elements in a non-linear model for several masses and non-linearity is very complicated. Thence a computer simulation is applied in these cases.

In cases of presence of clearance in joints of elements in individual phases of movement, a breakage of a kinematic chain occurs, followed by collision of masses yielding additional loads.

The basic task for the determination of dynamic loadings of the power transmission system elements is the creation of a mathematical model of a dynamic behaviour of the power transmission system.

However, an elasto-inertial torque model with two masses can be used for studying of a dynamic behaviour of power transmission systems. This mode enables a finding of the solution in a closed shape that is particularly convenient for the analysis of the solution. Additionally, the quality of the solution is of an acceptable level.

For the model with two masses, the reduction of mass is executed in front of and behind a reference elastic connection for which a value of loading is searched for. Additionally, the first mass is taken as a driving, while the second mass is the driven one. There is an elastic connection between these masses.

In general case, model of damped torsional oscillations of power transmission from n rotation masses can be written in the form of matrix equation, [1,5]:

$$\mathbf{J} \cdot \left\{ \ddot{\varphi} \right\} + \mathbf{B} \cdot \left\{ \dot{\varphi} \right\} + \mathbf{C} \cdot \left\{ \varphi \right\} = \left\{ \mathbf{M} \right\} \quad (7)$$

where:

$\mathbf{J}$, $\mathbf{B}$, $\mathbf{C}$ are the matrices of inertia, damping and rigidity ($n \times n$ dimensions),

$\left\{ \ddot{\varphi} \right\}$, $\left\{ \dot{\varphi} \right\}$, $\left\{ \varphi \right\}$ are the vectors of an angular acceleration, angular speed and angular positions of the centres of rotation masses ($n \times 1$ dimensions)

$\left\{ \mathbf{T} \right\}$ is the vector of external exciting torques, originating from the engine and motion resistance ($n \times 1$ dimensions).

The solution of matrix equation (7) with multiple masses is analytically very complex, and therefore simplified models are used [7].

So, elasto-inertial torsion models with two masses can be used as well.

General elasto-inertial torsion model with two masses is shown below on Fig. 4, [5].

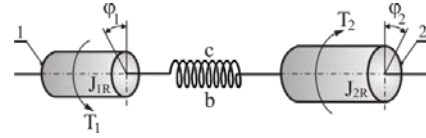


Fig. 4 Elasto-inertial torsion model with two masses.

The symbols on Fig. 4 have the following meanings: 1- mass of the drive part of the system, 2-driven mass, J_1 - reduced inertia torque of the masses of the drive part of the system, J_2 - reduced inertia torque of the masses of the driven part of the system, T_1 - drive torque, T_2 - load torque which may be either constant or variable (depending on the system position, time or speed), b - damping, c - reduced rigidity of the elastic system parts, φ_1 - angular position of the centre of the rotational mass of the drive part of the system, φ_2 - angular position of the centre of the rotational mass of the driven part of the system.

Differential equations of motion of this system in the warm-up phase are, [1,5]:

$$\begin{aligned} J_{1R} \cdot \ddot{\varphi}_1 + b \left(\dot{\varphi}_1 - \dot{\varphi}_2 \right) + c(\varphi_1 - \varphi_2) &= T_1 \\ J_{2R} \cdot \ddot{\varphi}_2 - b \left(\dot{\varphi}_1 - \dot{\varphi}_2 \right) - c(\varphi_1 - \varphi_2) &= -T_2 \end{aligned} \quad (8)$$

The detailed analytical description of the elasto-inertial model with two masses and clearance is given in reference literature [5], Fig. 5.

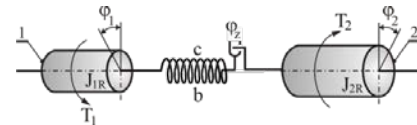


Fig. 5 Elasto-inertial torsion model with two masses and clearance [1,5].

The signs at Fig. 5 have the same meaning as the signs at Fig. 4, while φ_z is clearance.

The equation describing a motion of masses of the system presented at figure 5 is, [1,5]:

$$\begin{aligned} J_{1R} \cdot \ddot{\varphi}_1 + b(\dot{\varphi}_1 - \dot{\varphi}_2) + T_C &= T_1 \\ J_{2R} \cdot \ddot{\varphi}_2 - b(\dot{\varphi}_1 - \dot{\varphi}_2) - T_C &= -T_2 \end{aligned} \quad (9)$$

Where is the torsion torque in the elastic connection with clearance is:

$$T_C = \begin{cases} c \cdot \left(\varphi_1 - \varphi_2 - \frac{\varphi_z}{2} \right), & \frac{\varphi_z}{2} \leq \varphi_1 - \varphi_2 \\ 0, & -\frac{\varphi_z}{2} \leq \varphi_1 - \varphi_2 \leq \frac{\varphi_z}{2} \\ c \cdot \left(\varphi_1 - \varphi_2 + \frac{\varphi_z}{2} \right), & \varphi_1 - \varphi_2 \leq -\frac{\varphi_z}{2} \end{cases} \quad (10)$$

The equations (9) represent the basis for the simulation of the dynamic behaviour of the system, shown in Fig. 6.

Results of simulation

For the elasto-inertial model shown on Figure 6, we have adopted the following parameters: $T_1=5150$ [N·m], $J_{1R}=2.66$ [kg·m²], $J_{2R}=82.78$ [kg·m²], $c=4329$ [N·m], $\varphi_z=0.698$ [rad], [5,8,9], and $T_2=100$ [N·m], we performed the simulation of the movement of masses through the clearance using MATLAB-Simulink.

On Fig. 6, we gave the curves representing the torsion torque changes in the elastic connection with the clearance - curves 1 and

2, and without the clearance - curve 3, together with the torsion torque change M_1 . It is necessary to point out that we have considered an ideal case where there are no other disturbances in the system.

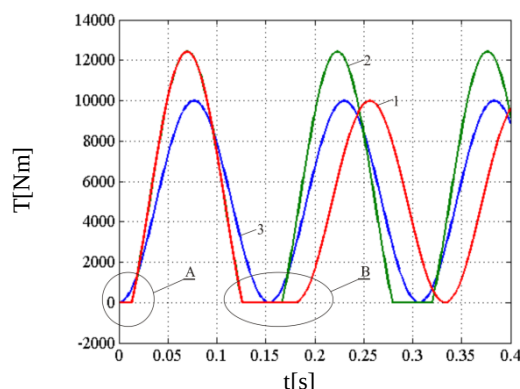


Fig. 6 The torsion torque changes in the elastic connection; 1- the curve obtained on the basis of the model of movement of masses through the clearance in five phases (in phase three the condition b is met); A and B phases merger, 2- the curve obtained on the basis of the simplified model where the clearance has been included through the initial conditions (when there is no a dumping of clearance value in an elastic connection); 3- the curve represents the case when in the model no clearance is present, [5,8].

Based on diagram at figure 6, the following can be concluded, [5]:

- Maximal value of a bending moment in an elastic connection with a clearance ($\varphi_z=40^\circ=0.698$ [rad]) – curve 1 is for 24.2 % bigger than in the case without a clearance – curve 3,
- Maximal values of a bending moment in an elastic connection with a clearance obtained by the use a model of mass motion through a clearance in five phases (curve 1) and a simplified model where a clearance is included through the initial conditions (curve 2) are the same,
- Amplitudes of curves 2 and 3 do not change with time; for curve 1, after the first amplitude of the maximal value, an oscillation with a constant amplitude whose value corresponds to the curve 3 occurs.

6. Conclusion

A friction and wearing between surfaces of coupled elements of power transmission system in vehicles have a particular importance for the change of states of power transmission elements in vehicles.

An achievement of a boundary state on the basis of a material fatigue in vehicles is related to a boundary wearing of coupled elements.

Due to a change of states of power transmission elements in vehicles, a rise of dynamic loads of elements, as well as decrease of a work efficiency and energy efficacy occur.

Also, the change of state of elements of power transmission system in vehicles decreases their remaining work live.

Based on a known state and speed of change of state of elements, a remaining work life of the power transmission system on the basis of the achievement of a maximal allowable wearing can be determined.

Based on an analytical consideration and a simulation model, it was found out that the change of state of power transmission system elements in vehicles in transitional modes multiply exceed static loadings.

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ANALYSIS AND MODELLING OF AUTOMOTIVE AIR FILTER AS A SUBSTITUTE FOR POWERCORE ITEM

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Abstract: Analysis of PowerCore automotive air filter was made and development of new alternative air filter was performed by increasing surface area of filtering material. New shape of construction of air filter was suggested by introducing double-layer filtering material. It was determined that effective surface area of the new design of air filter was increased 1.5 times comparing with the patented PowerCore item of the same size. Efficiency and strength of the new design air filter was evaluated calculating air flux rates, deformations and displacement in SolidWorks environment.

Keywords: AIR FILTER, AIRFLOW SIMULATION, DOUBLE-LAYER STRUCTURE

1. Introduction

The latest design of PowerCore produced by Donaldson's uses the same straight-through airflow and high-density filtration system as the original PowerCore, however is, reportedly, 30 % more environmentally friendly [1]. The latest design compared to other filtration technologies, is noticeably different in shape, colour and style. PowerCore is available in round and racetrack shapes, its compact, metal-free, cartridge-style design traps contaminants inside the structure for easier and cleaner servicing [2]. Donaldson PowerCore media technology can utilize options such as a secondary seal, protective wraps, and safety filter to enable increased airflow and system efficiency for peak performance in all conditions [3]. This happens that the axial flow air filter avoids turbulences by allowing the aerosol to flow straight through the filter. It leads to the decrease of potential pressure losses [4].

However, PowerCore air filters have disadvantages – they collect dust particles inside the tubes of the filtering material [4]. One of the possibilities that these particles could not move into the engine is a creation of nanofiber structure of filtering material [5-7]. Also the overlarge contaminating particle, depending on its size and shape, can jam the one or several of air filter tubes, thus shortening its lifetime.

The aim of our work was to design a new construction of the axial flow air filter with the elimination of the disadvantages mentioned above and improved performance.

2. Methodology

After reviewing and analyzing the technical literature, a round shape PowerCore design air filter was selected (Fig. 1, a). The filtering material of this filter is a specific one since it has a different design and is made from a plurality of tubes. On purpose to measure the surface area of the filtering material, it is necessary to count the number of tubes (equal: 3430 units), to model the profile of one tube, and to measure the area of its filtering surface as well (equal: 0.0018 m²). One tube 3D model (Fig. 2, b) was designed using Solid Works 2017 software. This software was also used for other necessary calculations that were needed to create a new design air filter. The surface area of the filtering material of PowerCore item was calculated:

$$S = \frac{3430}{2} \times 0.0018 = 3.087 \text{ m}^2$$

Calculated surface area of filtering material used in PowerCore item was used as an initial object for the further calculations creating a new air filter design.

Applied equipment for experimental research: 3D printer "EOS Formiga P110" for printing of plastic specimens for determination of mechanical characteristics; Tinius Olsen, a universal material testing machine H25 KT, designed to determine the mechanical properties of stretched and bent printed samples from plastic PA2200. The determined plastic characteristics were used for

modelling of the air filter, filtering material, fixing elements, for the calculation of their strength and deformation, and the influence of incoming air flows.

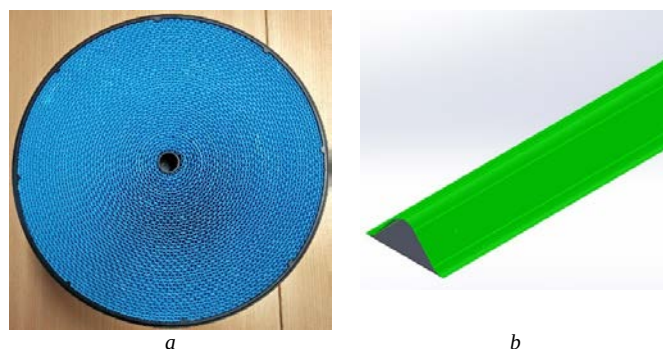


Fig. 1 Original PowerCore design air filter in round shape: a) ready-to-use product; b) model of filtering material single tube

3. Results and Discussion

3.1. Modelling of new design automotive air filter as a substitute for PowerCore item

Depending on the overall dimensions of the air filter, a round-shape design of an alternative air filter was designed, in which the Power Core filtering material structure was replaced with a folded double-layer filtering one (Fig. 2, 3). The modified air filter design allowed to increase the surface area of the filtering material. It was calculated to be 4.677 m², where the surface of the first row was 2.668 m² and the second was 2.009 m².

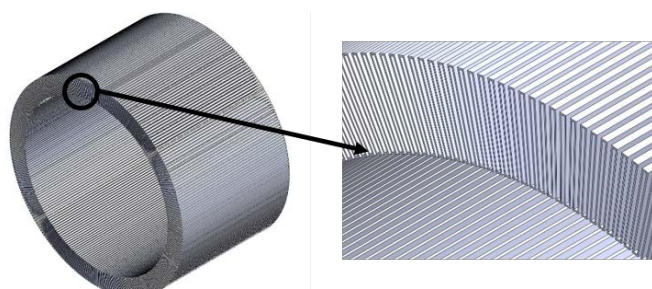


Fig. 2 3D model of pleated filtering material of round-shape outer ring

The surface area of our alternative structure of the air filter designed with a double-layer pleated filtering material was compared to the surface of the same shape Power Core item. It was found to be 1.52 times bigger comparing with the surface area of PowerCore item. Based on the results of simulation and computation, the proposed double-layer air filter has very light impact on the cost of production, but presents almost 1.5 times bigger area of the filtering surface. This area can be changed depending on the amount of pleats.

Simulated double-layer air filters should have two unique lids, two layers of filtering material, the inside of which is covered with

a plastic mesh for increasing of reliability. The exterior of the designed air filter is 20 mm smaller than the selected PowerCore air filter, so the airflow does not go along the filter, but covers the bottom lid. Newly engineered air filters are expected to increase durability due to approximately 1.5 times bigger surface area of the filtering material.

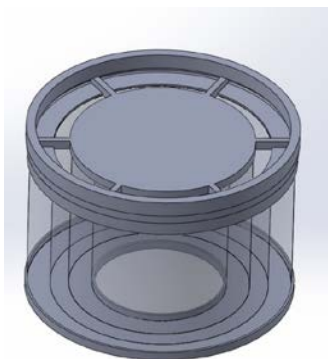


Fig. 3 Optimized design for a round shape double-layer air filter

When designing of a new construction of air filters, it is very important to study the distribution of airflow in the filtering material. This test is necessary in order to effectively use all surface area of the filtering material. Therefore, in order to investigate what processes are carried out in a simulated double-layer air filter, a calculating model with boundary conditions for analyzing airflow distributions was compiled. Computational schema for airflow calculation is presented in Fig. 4.

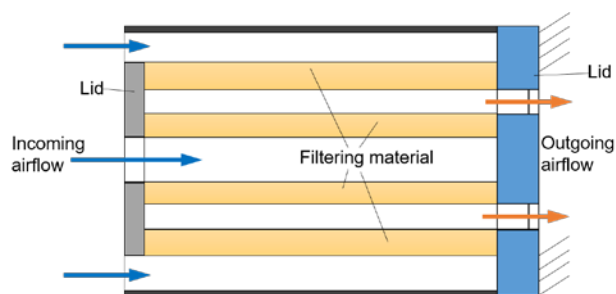


Fig. 4 Computational schema for airflow calculation

For the engineered air filter, a frame was created in which a new air filter was placed in the virtual space. During the calculation, the boundary conditions were set: air temperature 20 °C, air pressure 101325 Pa, air speed at the modelled frame output – 30 m/s. The result of the calculation is presented in Figure 5.

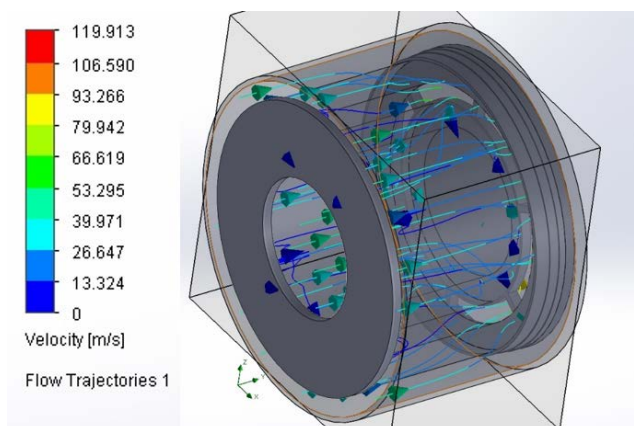


Fig. 5 Airflow rate distribution in a round shape filter

The results presented in Fig. 5 show that due to the sharp edges (marked in Fig. 6 by arrows), the maximum possible air turbulence will occur not only around the plastic lids, but also in the smaller filtration ring of the pleated filtering material. In addition, in this area, the surface of the hole is smaller than the surface area of the filtering material. However, this area is not less than the surface

area of the vehicle air intake port. Otherwise, the air filter will not provide the required amount of air for the internal combustion engine and will worsen its characteristics. It has also been observed that in this case, the maximum air flow rate is reached near the upper double-layer filter lid. By reaching this limit, the air flow changes the direction and crosses the filtering material, making it difficult to describe what type of air flow is. But in general, the air flow is evenly distributed.

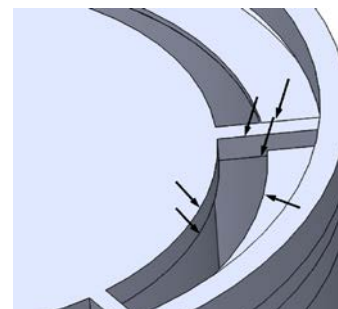


Fig. 6 Sharp edges of air filter lid

3.2 Calculations of air filter structural elements during static and dynamic loading

To perform calculations of strength of engineered structural elements and filtering material, it is necessary to determine the mechanical properties of the material used to produce these elements. The following figures (Fig. 7) show polymer specimens and their investigations.

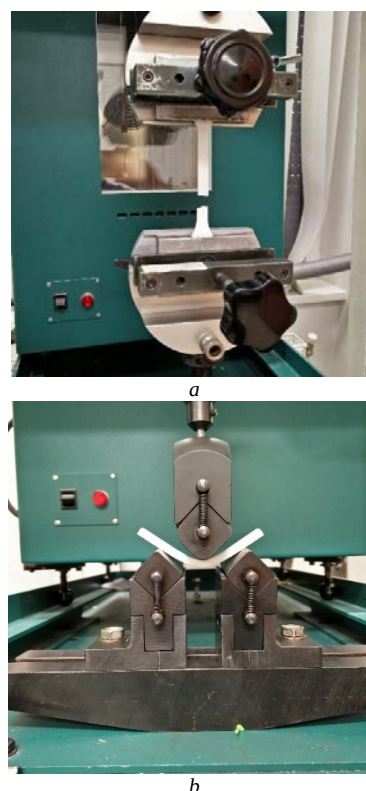


Fig. 7 Determination of mechanical characteristics of printed plastic samples: a) 3D PA2200 sample after tension test, b) 3D PA2200 sample after bending test

The experimental results revealed the mechanical characteristics of the test material, which were then used for theoretical calculations in the environment of the SolidWorks software. This polymer material was chosen for the available 3D printer that could be used to produce prototype polymeric structural elements. The results of the tests obtained are presented in Fig. 8.

The tensile test for polymer specimens has shown their strength ranges from 50-51 MPa. Meanwhile, the yield point was about 32 MPa.

The bending specimens reached a maximum force of about 200 N. This force was determined when the punch had moved 11 mm distance in the direction of the Z-axis (bending direction). The bending specimen did not crack and did not break, but remained deformed after unloading. This experiment showed that the material was sufficiently plastic to resist dynamic loads occurring during exploitation.

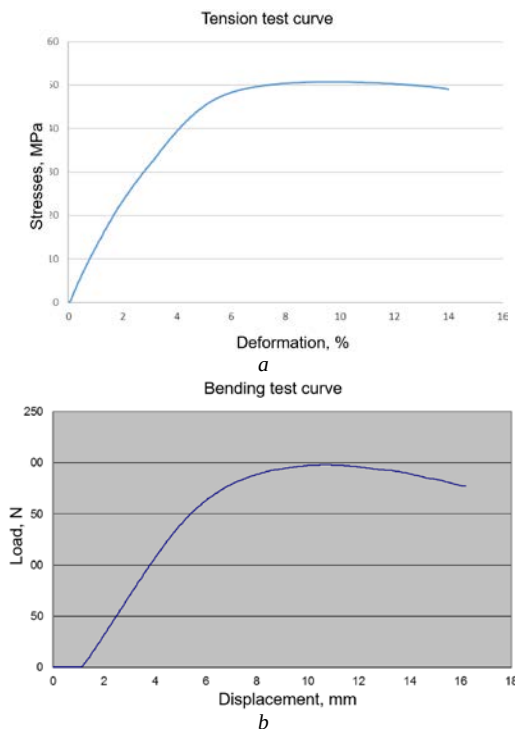


Fig. 8 3D PA2200 deformation results: a) tensile; b) bending

In order to investigate what processes took place in a simulated double-layer air filter, a calculating model with boundary conditions was engineered. It consisted of 62537 finite elements with 100750 nodal points. The filtering material of air filter of round shape double-layered design was loaded with 0.006 MPa. The selected load corresponded to the initial load generated by fully contamination of air filter. When this load value is reached, the air filter must be replaced by a new one.

Computational schema for stress-displacement calculation is presented in Fig. 9. Calculation results are shown in Fig. 10 and 11.

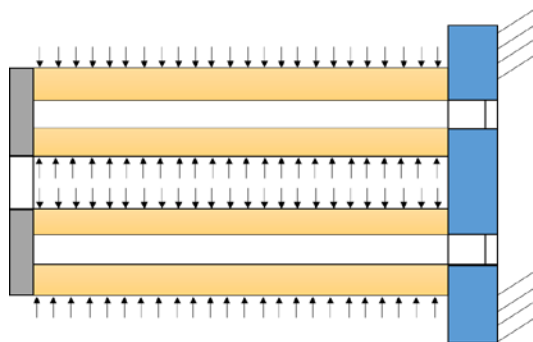


Fig. 9 Computational schema for stress-displacement calculation

The results obtained showed that the highest stresses are available in the upper and lower plastic lids, but they do not exceed the permissible ones. Meanwhile, the biggest displacements are determined inside the filtering material. These weak air filter locations require for additional reinforcement elements in order to ensure that the air filter will not be damaged during exploitation and will not damage the internal combustion engine.

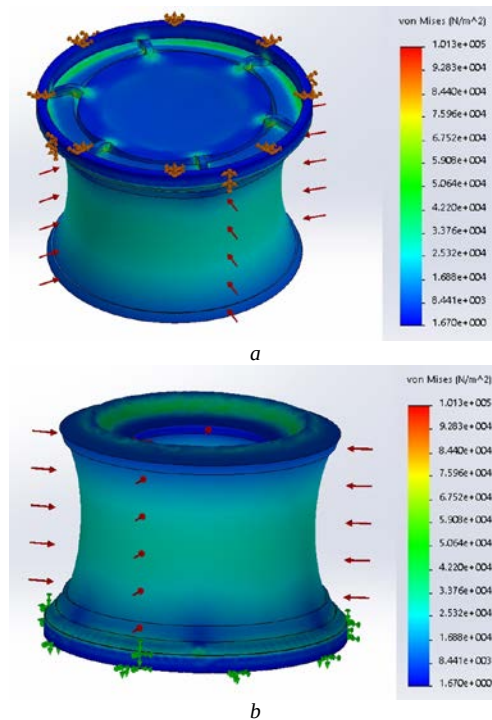


Fig. 10 Distribution of stresses in a double-layer construction in a round-shaped air filter: a) top view; b) bottom view

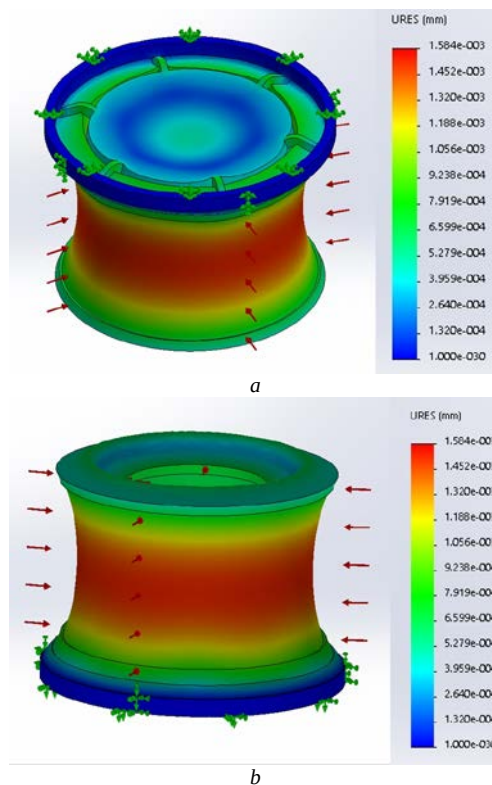


Fig. 11 The distribution of total displacements in a double-layer construction in a round shape air filter

4. Conclusions

Analyzing the results of experiments and sources of scientific literature, such conclusions were made:

- New design of double-layer round-shaped air filter has showed that it's filtering surface has been increased by 1.52 times compared to the similar PowerCore air filter. Increased filtering surface is likely to prolong the life of air filters. On the basis of the results, the proposed double-layer air filter will have insignificant influence on the production costs. The filtering material could be changed depending on the amount of pleats.

- The results of airflow dynamics calculations have shown that the highest possible air turbulence will be due to plastic lids and a smaller filtration ring in the pleated filtering material. It was found that the sharp edges of the lid surface had been one of the causes of the turbulence occurrence, and in addition, surface area of the lid air outgoing hole was smaller than the surface area of the filtering material. However, this area is not less than the surface area of the vehicle's air intake opening. However, in filtering materials, the air flow is laminar and evenly distributed.
- The tensile test of polymeric specimens showed that their strength ranges from 50-51 MPa. Meanwhile, the yield point is about 32 MPa. The analysis of the resulting curves showed that the material is sufficiently plastic to resist dynamic loads occurring during exploitation.
- The bending test of polymeric specimens showed that the material was able to withstand the maximum force of about 200 N, when the punch has moved 11 mm in the direction of Z-axis. The bending specimen did not crack and did not break, but remained deformed when was unloaded.

Acknowledgements. *The project was supported by Baltic Filter, Ltd.*

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DESIGN AND OPTIMISATION OF 2DOF VEHICLE SUSPENSION SYSTEM

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Abstract: In this paper is presented the optimal design of quarter car vehicle suspensions. During the optimisation the three design criteria which have been used are; vertical vehicle body acceleration, dynamic tire load and suspension working space. For implementing a optimization a comparison of two optimisation algorithms: Sequential Quadratic Program (SQP) and Genetic Algorithms (GAs). will be chosen five design parameters: sprung and un-sprung mass, spring stiffness, damping coefficient and tire stiffness. Through the simulation in Matlab it will be shown that GA is more powerful tool to find the global optimal point, without any restrictive requirements on the gradient and Hessian Matrix, while SQP has local convergence properties. The main focus of this research will be on minimising the vertical vehicle body acceleration subjected to a suspension working space and the dynamic tire load. At the end of this paper, it will be shown the comparison between the simulation carried out with nominal and optimal values of design parameters.

KEY WORDS: OPTIMISATION ALGORITHMS, DESIGN PARAMETERS, SQP, GA.

1. Introduction

Vehicle suspension design includes a number of compromises that have to do with suspension system which should be smooth to provide good levelling and ride comfort. On the other hand must be strong, to fix changes the behaviour of the vehicle and to ensure road holding for varying external conditions. Traditional design practices of vehicle suspensions have been based on trial and error approaches. Now the focus of vehicle suspension design has switched from pure numerical analysis to extensive design synthesis using optimization approaches. There are numerous methods available and even the choice of an efficient optimization algorithm is a non-trivial problem [2]. Genetic algorithms (GAs) have been used in various applications such as function optimization, system identification and control systems. GAs are general-purpose stochastic optimization methods for solving search problems to seek a global optimum. However, GAs is characterized by a large number of function evaluations [1]. The pattern search algorithm (PSA) is typically based on function comparison techniques. Most of these procedures are heuristic in nature and derivative evaluations are not needed. They can be used to solve problems where the objective function is not differentiable and continuous [3].

On the other hand, traditional methods, such as sequential quadratic programming (SQP), are well known to exploit all local information in an efficient way, provided that certain conditions are met and the function to be minimized is 'well-conditioned' in the neighbourhood of a unique optimum. These methods require adequate local information to be known (such as the gradient and Hessian matrix) [2, 3]. If the basic requirements are not satisfied, the reliability of the SQP method is greatly jeopardized [2]. By means of the ride quality analysis in the frequency domain, the vertical vehicle body acceleration (VBA), suspension working space (SWS) and dynamic tire load (DTL) can be obtained [1]. In this design optimization, the main objective is to minimize the VBA acceleration. In the meantime, the SWS and DTL are constrained. If the SWS is too small, the sprung mass will strike the un-sprung mass and this may lead to damage of the vehicle. If the DTL is greater than the static tire load, the vehicle's tires will bounce off the road [4] and this will result in unstable modes of vehicle motion. Therefore, it is necessary to optimize the suspension working space and dynamic tire load.

2. Vehicle System Modeling

The model of the simplified quarter-car active suspension system used in this paper with two degree of freedom is shown in Figure 1. The model represents a single wheel of a car in which the wheel is connected to the quarter portion of the car body through a hydro-pneumatic suspension.

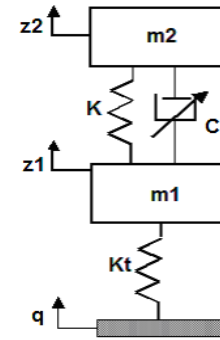


Fig.1. Quarter-car model

where:

$z_2 - z_1$: body displacement; $z_1 - q$: wheel displacement

$\dot{z}_2 - \dot{z}_1$: suspension velocity;

$\dot{z}_2$: velocity of the body; $\dot{z}_1$: velocity of the wheel

$m_2 = 637$ kg : body mass ; $m_1 = 104$ kg : wheel mass

$K = 3200$ N/m and $K_t = 100600$ N/m : respective spring constants

$C = 3200$ Ns/m : damping ratio

The equations of motion:

$$\begin{aligned} m_1 \ddot{z}_1 + C(\dot{z}_1 - \dot{z}_2) + K(z_1 - z_2) + K_t(z_1 - q) &= 0 \\ m_2 \ddot{z}_2 + C(\dot{z}_2 - \dot{z}_1) + K(z_2 - z_1) &= 0 \end{aligned} \quad (1)$$

Performing a Fourier transform of equation (1) yields:

$$\begin{aligned} z_2(-\omega^2 m_2 + j\omega C + j\omega + K) &= z_1(j\omega C + K) \\ z_1(-\omega^2 m_1 + j\omega C + K + K_t) &= z_2(j\omega C + K) + qK_t \end{aligned} \quad (2)$$

The amplitude ratio between the un-sprung mass displacement, z_1 , and the road excitation, q is given as follows:

$$\left| \frac{z_1}{q} \right| = \gamma \left[\frac{(1 - \lambda^2)^2 + 4\zeta^2 \lambda^2}{\Delta} \right]^{\frac{1}{2}} \quad (3)$$

The amplitude ratio between the sprung mass displacement, z_2 , and the road excitation, q is:

$$\left| \frac{z_2}{q} \right| = \gamma \left[\frac{1 + 4\xi^2 \gamma^2}{\Delta} \right]^{\frac{1}{2}} \quad (4)$$

Therefore, the amplitude ratio between the sprung mass acceleration, $\ddot{z}_2$ and the road excitation, q , can be expressed as:

$$\left| \frac{\ddot{z}_2}{\dot{q}} \right| = \omega \gamma \left[\frac{1 + 4\xi^2 \gamma^2}{\Delta} \right]^{\frac{1}{2}} \quad (5)$$

The suspension working space is the allowable maximum suspension displacement, f_d . The suspension working space in response to the road displacement input is:

$$\left| \frac{f_d}{\dot{q}} \right| = \frac{\gamma}{\omega} \lambda^2 \left[\frac{1}{\Delta} \right]^{\frac{1}{2}} \quad (6)$$

The amplitude ratio between the relative dynamic tire load, $\left| \frac{F_d}{G} \right|$, and the road input, q , becomes:

$$\left| \frac{F_d}{G_q} \right| = \frac{\gamma \omega}{g} \left[\frac{\left(\frac{\lambda^2}{1 + \mu} - 1 \right)^2 + 4\xi^2 \lambda^2}{\Delta} \right]^{\frac{1}{2}} \quad (7)$$

3. Stochastic road modeling

Road irregularity or unevenness represents the main disturbing source for either the rider or vehicle structure itself. The road profile elevation is usually expressed in terms of the power spectral density (PSD). The PSD of the road profile elevation is expressed as:

$$G_q(n) = G_q(n_0)(n/n_0)^{-w} \quad (8)$$

For the purposes of design optimization, according to James' principle, the root mean square (RMS) of the sprung mass acceleration $\ddot{z}_2$ can be expressed as:

$$\sigma_{\ddot{z}} = \left\{ \pi R V \left[\frac{K_t C}{2m_2^{3/2} K^{1/2}} + \frac{(m_1 + m_2) K^2}{2C m_2^2} \right] \right\}^{1/2} \quad (9)$$

The RMS of the suspension working space f_d is:

$$\sigma_{fd} = \left\{ \pi R V \left[\frac{(m_1 + m_2)(m_2 K)^{1/2}}{2m_2 C} \right] \right\}^{1/2} \quad (10)$$

The RMS of the relative dynamic tire load can be calculated as:

$$\sigma_{Fd/G} = \left\{ \pi R V \left[\frac{K_t^2 m_1}{2C(m_1 + m_2)^2} + \frac{(m_1 + m_2) K^2}{2m_2^2 C} - \frac{K_t K m_1}{C m_2 (m_1 + m_2)} + \frac{CK}{2m_1 m_2} \right] \right\}^{1/2} \quad (11)$$

4. Design and optimisation

In this section, the sprung mass vertical acceleration is minimized, while the design constraints on the suspension working space and dynamic tire load should be satisfied. To implement the design optimization, the two optimization algorithms, i.e, SQP and GA, will be applied, respectively.

4.1 Optimization based on SQP algorithm and GA

The SQP algorithm is a non-linear programming technique that is used for the purpose of minimizing a smooth non-linear function subjected to a set of constraints with upper and lower bounds. The objective function and the constraint functions are assumed to be at least twice continuously differentiable. This algorithm is a gradient-based search method [2, 3]. This algorithm is well-suited for constrained design optimizations.

The reliability for finding the optimum decreases with the increase of number of design variables when using SQP method. In contrast, whether the number of design variables increase the GA can still reliably find the optimum. This can be explained by the fact that GA works on a population of variables in parallel, not on a unique point. GAs are global search methods that are based on the Darwin's principle of natural selection and genetic modification. The GA has higher reliability to find the global optimum with minimum number of computational operations.

The RMS of the acceleration of a sprung mass $\sigma_{\ddot{z}_2}$ is frequently used to evaluate the riding quality of a vehicle. A rider's comfort improves as the acceleration decreases. Ride comfort is chosen to be the design criterion. The suspension working space and dynamic tire load σ_{fd} are selected as the design constraints. The design variables are m_1 , m_2 , K_t , K and C , respectively. Thus, the design optimization problem can be described as:

Minimise:

$$\sigma_{\ddot{z}_2}(m_1, m_2, K_t, K, C) = \left\{ \pi R V \left[\frac{K_t C}{2m_2^{3/2} K^{1/2}} + \frac{(m_1 + m_2) K^2}{2C m_2^2} \right] \right\}^{1/2} \quad (12)$$

Subject to:

$$\begin{cases} \sigma_{Fd/G}(m_1, m_2, K_t, K, C) \leq a = 0.5 \\ \sigma_{fd}(m_1, m_2, K_t, K, C) \leq b = 0.05 \\ 83.2 \leq m_1 \leq 124.8 \\ 509.6 \leq m_2 \leq 764.4 \\ 559440 \leq K_t \leq 839170 \\ 80480 \leq K \leq 120720 \\ 2560 \leq C \leq 3840 \end{cases} \quad (13)$$

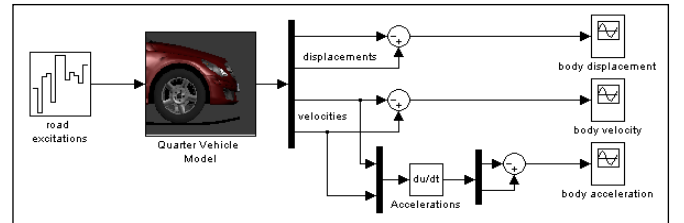


Fig 2. Simulation scheme of Quarter Vehicle Model

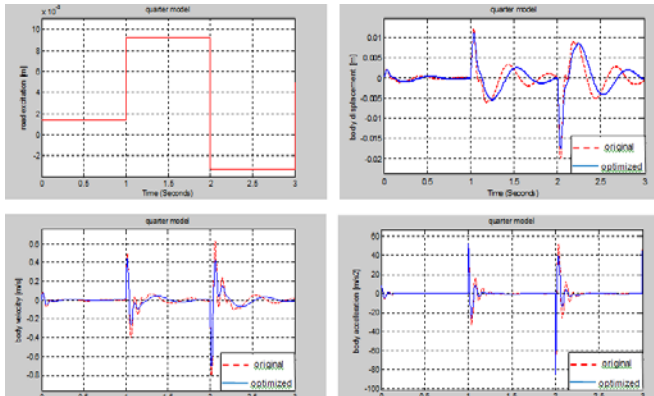


Fig 3. Simulation results for original --- (nominal) and optimized variables

In this sub-section, the optimization results are derived for a vehicle travelling at the speed of 40 m/s on the road with an irregularity coefficient of power spectrum taking the value of $6.5 \times 10^{-6} \text{ m}^3$.

Table 1. Optimal design variables based on the SQP and GA for minimizing the sprung mass vertical acceleration, vehicle speed 40 m/s

	Original values	SQP method	GA
Initial		[10 10 10 10 10]	[10 10 10 10 10]
m_1 [kg]	104	83.2	83.2
m_2 [kg]	637	764.4	764.4
K [N/m]	699300	80480	80480
K_t [N/m]	100600	559692	559692
C [Ns/m]	3200	3840	3840
$\sigma_{\ddot{z}_2}$	5.6206348677	1.0703660446	1.0703417681
		Optimum found: 47 iteration	Optimum found: 19 generates

5. Conclusions

A comparative study of two optimization algorithms (genetic algorithms, GAs and sequential quadratic programming, SQP), has been conducted through minimizing the vertical sprung mass acceleration subjected to a suspension working space and dynamic tire load. A quarter-vehicle model was used to implement the design optimization of the vehicle suspension systems. The SQP has very strong theoretical and local convergence properties. The numerical results demonstrate these features of the algorithm, since the SQP is located at local optimal points. The GA is more powerful to find global optimal points, without restrictive requirements on the gradient and Hessian matrix.

By optimizing the design parameters compared with the original design, the sprung mass (body) acceleration decreases. The suspension working space and the dynamic tire load satisfy the specified design constraints. Based on the simulation results the optimum found by GAs at 19 generations, while by using the SQP the optimum is found after 47 iterations. As shown from the numerical simulation results in Figure 3 the max amplitude of body displacement using optimized design variables is reduced for 9%, while maximal amplitude of body acceleration is reduced around 22%.

Numerical experiments reveal the fact that to improve vehicle ride quality and satisfy the specified suspension working space and relative dynamic tire load, different vehicle speed and road irregularity have different requirements on the design variables, in particular, the un-sprung mass. In order to solve this problem, application of multi-level optimization approach it is recommended and the resulting solutions will compromise the conflicting requirements on design variables for vehicles travelling at different speed and on roads with different irregularity.

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THE STRESS VARIATION BY CHANGING THE SUPPORTING POINT LOCATION IN THE MOTOR VEHICLE CLUTCH ASSEMBLY

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Abstract: The diaphragm spring as one of the most important parts of the motor vehicles clutch assembly provides the compressive force on the pressure disk. This force is needed for generating friction between the coupling of the flywheel and the pressure disk and transmitting the torque from the engine to the transmission. Therefore the diaphragm spring is subjected to complex loads. The nominal stress for dimensioning the spring is the tangential stress, which is calculated by the terms of Almen and Laszlo. The aim of this research is analyzing the location of the diaphragm spring supporting points of the clutch assembly and its effect on the caused stresses by using the Finite Element Method.

Keywords: DIAPHRAGM SPRING, FRICTION CLUTCH, EXPRESSIONS OF ALMEN AND LASZLO, TANGENTIAL STRESS

1. Introduction

The clutch coupling allows the engaging and disengaging of the vehicle transmission. The required compressive force for creating the friction necessary for torque transmission is achieved by the diaphragm spring. When performing its function, the diaphragm spring is subjected to dynamic loadings. The stresses occurring at the spring are compression and extension, figure 1. [1],[2].

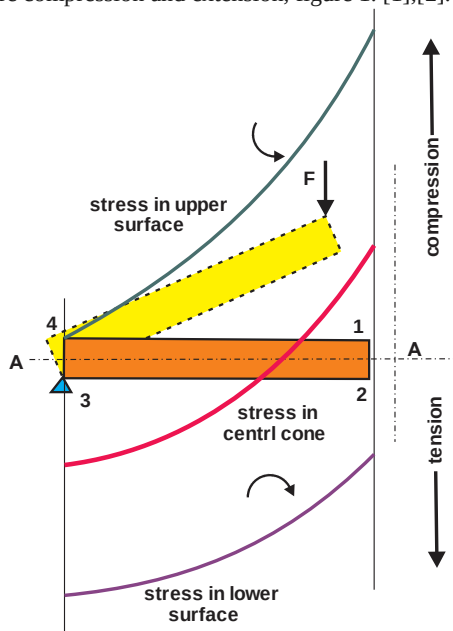


Fig. 1 Spring stresses

The diaphragm spring use the clutch assembly couplings has the location of the supporting points as shown on the picture or in the case of a vehicle those are a supporting edges, figure 2.

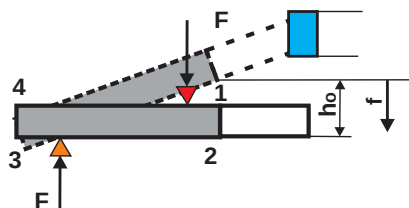


Fig.2 Diaphragm spring with its supporting points

By mounting the clutch in the vehicle, the shape of the spring changes, from its conical shape it becomes flat (it makes a deviation $f = h_0$), and then the process of spring loading starts. The spring has

two points of interest. Point 1 where the force F acts (created by the spring deflection) and the point 3 where the spring is supported, where the force is also equal to F . When the spring is in the flat position, the compression force allows the torque to be transmitted. Moving the release bearing for a certain path (deflection) causes the clutch to disconnect [3], [4].

2. Researching

The purpose of this research is by using the finite element method to determine the spring stress depending on the location of the supporting points in both the flat or mounted position and at the maximum deviation or disengaged position.

The analysis were carried out on a diaphragm spring for commercial vehicles with next dimensions : internal diameter of a diaphragm spring $D_i = 313$ [mm], outer diameter of a diaphragm spring $D_a = 395$ [mm], spring thickness $s = 5.2$ [mm], Module of elasticity of the steel $E = 206000$ [N /mm²], Poisson number of spring steel $\mu = 0.3$, internal diameter of the diaphragm spring with supporting points $D_{ip} = 336$ [mm], outer diameter of the diaphragm spring with supporting points $D_{ap} = 392$ [mm], path of the clutch while disengaging $\ell = 12$ [mm], release bearing diameter $d = 120$ [mm]. [5], [6]

Figures 3 and 4 shows the distribution of forces acting on the spring, with one and two supporting points used for the clutches.

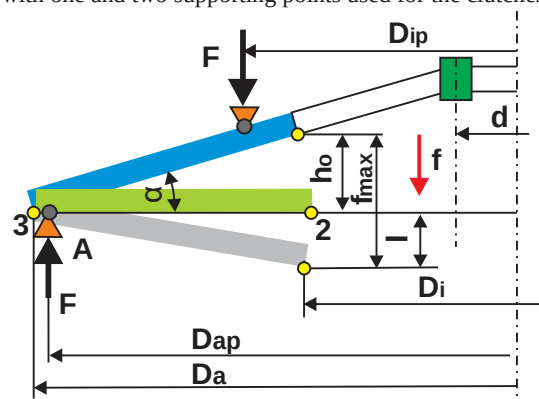


Fig. 3 Spring with one supporting point

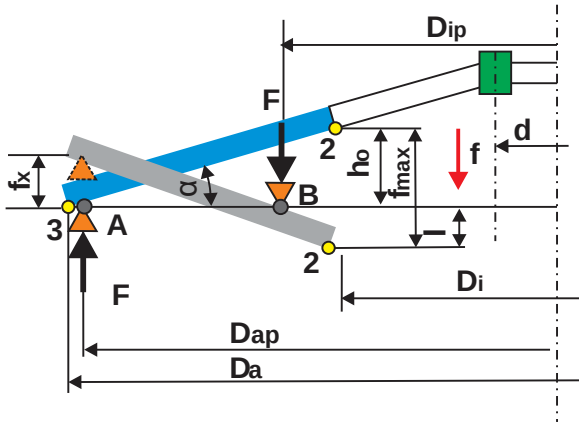


Fig. 4 Spring with two supporting points

In order to note the influence of the supporting points, the spring calculation was performed:

- supporting point A, by moving the spring from the initial or zero position to the flat position, the spring makes a deviation h_o .

The supporting point A is stationary, and the release bearing moves the spring from the flat position for the path ℓ that disengages the spring (maximum deviation is achieved).

- From the two supporting points A and B, in the flat position of the spring, the supporting point B is stationary. The release bearing moves from the flat position for the path ℓ , and the point A travels the distance f_x , and within this displacement the pressure disk is raised and the clutch is disengaged.

By using a Finite Elements Method (commercial software package), the spring cross-section is divided into 21 elements (Fig. 5) in which the stress is calculated. From the simulation results, the following diagrams for the tangential stress for two cases (Fig. 6 and 7) were obtained:

• The support point A is on a diameter of $\varnothing 392$ [mm], and the internal diameter $D_i = 313$ [mm] moves from the initial position to the flat position f (h_o) and to a maximum deviation of f_{max} .

• The support point A is on a diameter of $\varnothing 392$, and the supporting point B is on the diameter of $\varnothing 336$ [mm]. From the flat spring position, point B is stationary and point A moves in opposite direction from the direction of the release bearing that travels the distance ℓ , [7].

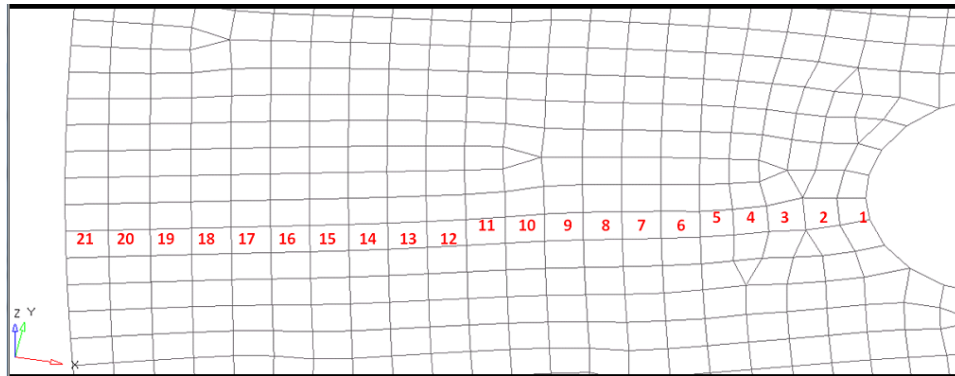


Fig. 5 FEM elements for stress calculation

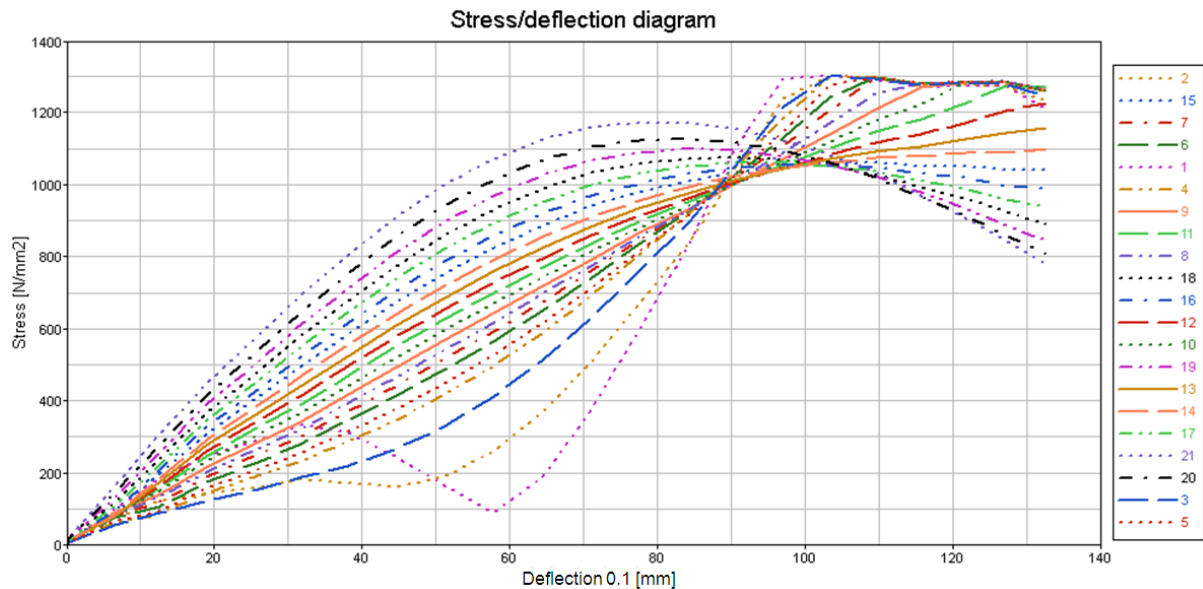


Fig. 6 Diagram of tangential stresses with a support point A,

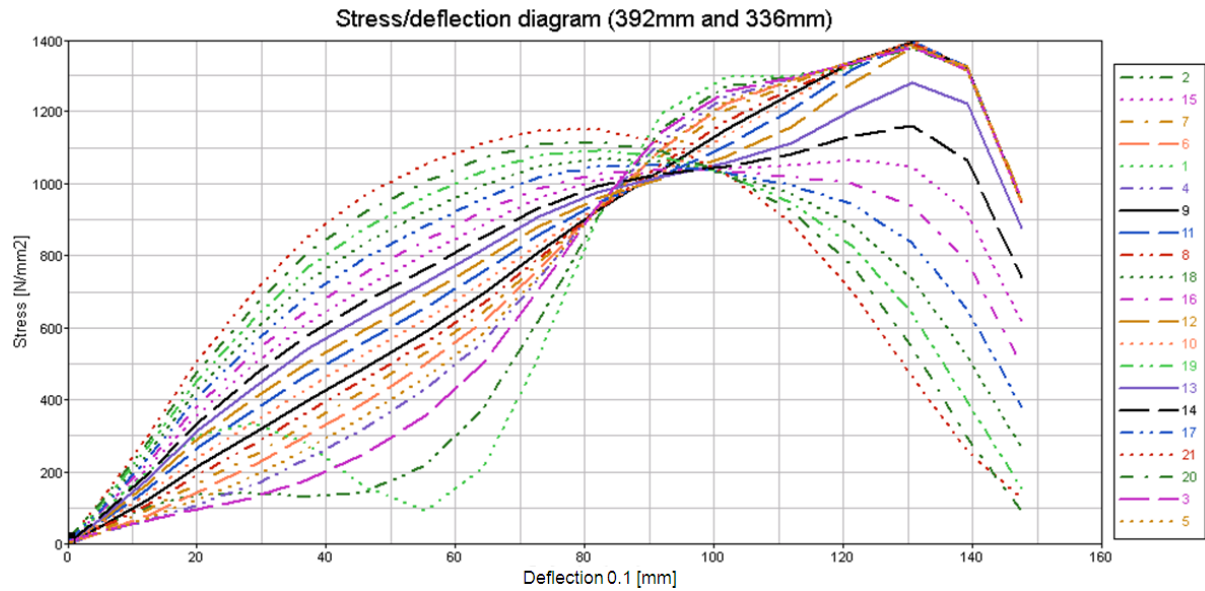


Fig. 7 Diagram of tangential stresses with two supporting points, A and B

Complex stresses are presented at figures 8 and 9 :

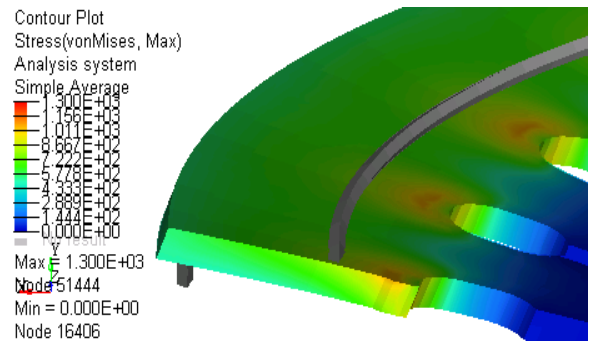
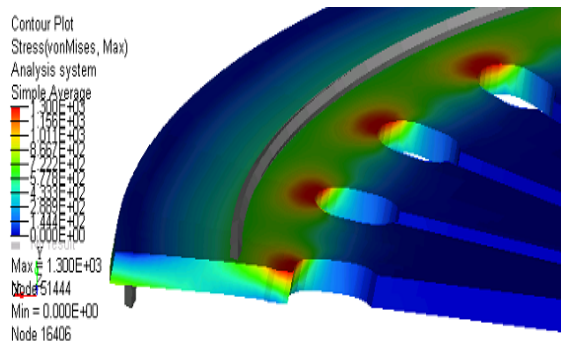


Fig.8 Complex stress at the flat and disengaged position Ø392/0

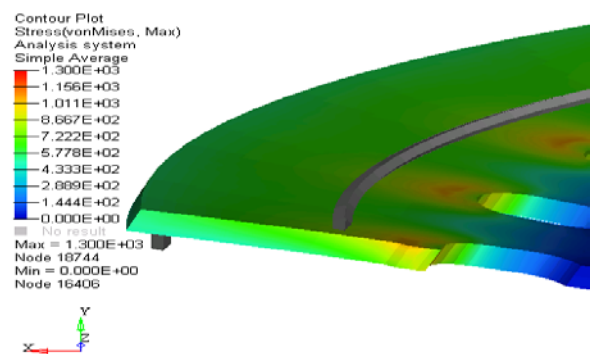
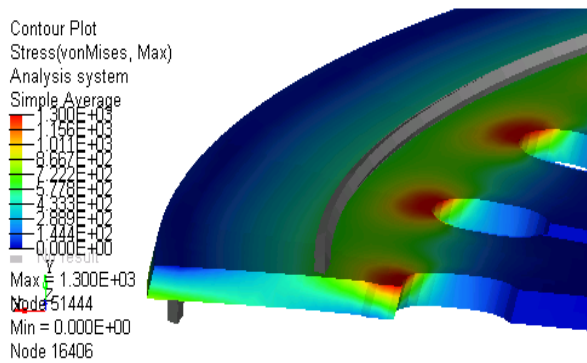


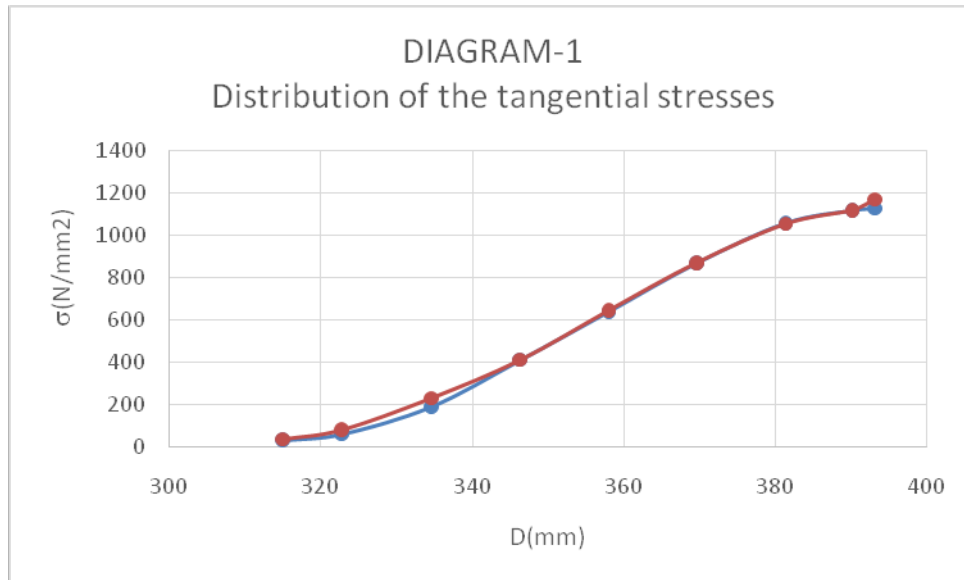
Fig.9 Complex stress at the flat and disengaged position Ø392/336

3. Analysis of the results and discussion :

The calculations were carried out by the FEM and the obtained diagrams (Fig. 6 and 7), for the change in the tangential stresses of the diaphragm spring body shown on the diagrams (Fig. 10 and 14) and the tables (1 and 2) were analyzed [4], [7].

Table 1: Tangential stresses at the spring flat position

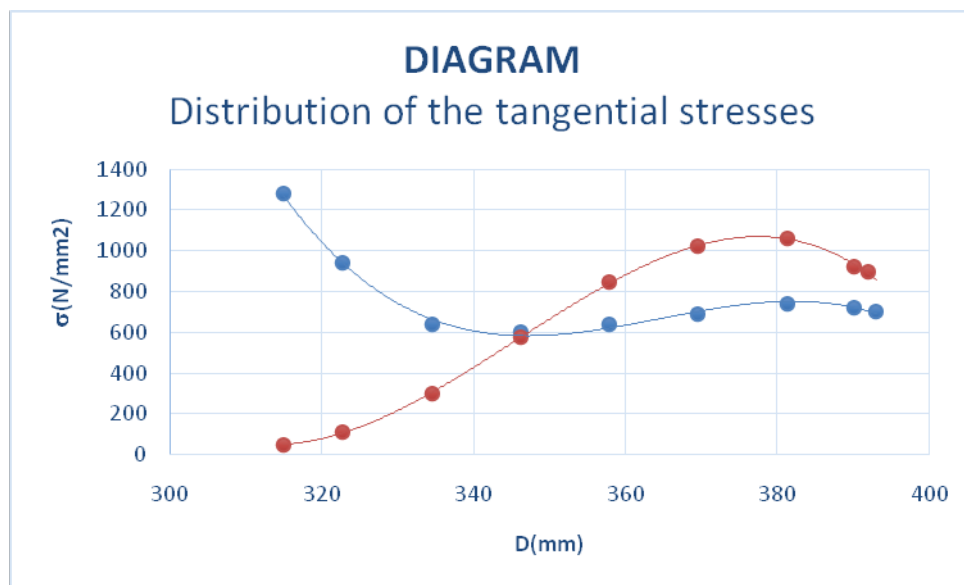
393,0	392,0	390,2	381,3	369,5	358	346,2	334,5	322,7	315	D(mm)
8,86	8,75	8,55	7,57	6,27	5,0	3,7	2,4	1,1	0,22	f(mm)
1130		1120	1060	870	640	410	190	60	30	Ø392/336 $\sigma(N/mm^2)$
1170		1120	1055	870	645	410	230	80	35	Ø392/0 $\sigma(N/mm^2)$



With one support —, with two supports —

Fig. 10 Distribution of the tangential stress at the flat position of the spring**Table 2:** Tangential stresses at the spring disengaged position

393,0	392,0	390,2	381,3	369,5	358	346,2	334,5	322,7	315	D(mm)
2,8		3,2	3,7	4,3	5	5,6	6,5	8,13	9,8	f(mm)
120		370	500	620	540	570	740	1120	1280	Ø392/336 $\sigma(N/mm^2)$
12,35	12,23	12,0	10,57	8,77	7,0	5,15	3,33	1,51	0,28	f(mm)
885		920	1060	1020	850	575	300	110	50	Ø392/0 $\sigma(N/mm^2)$



With one support —, with two supports —

Fig. 11 Distribution of the tangential stress at the disengaged position of the spring,

The highest tangential stress occurs in point 2 (if the supporting point is placed on the location of point 2), then it has the greatest deviation plus the deviation from the displacement of the release bearing traveling the distance of $\ell = 12\text{mm}$ ($f_{\max} = 7,85 + 4,9 = 13,7\text{mm}$). This can be seen from the picture of the complex stress (Fig.12).

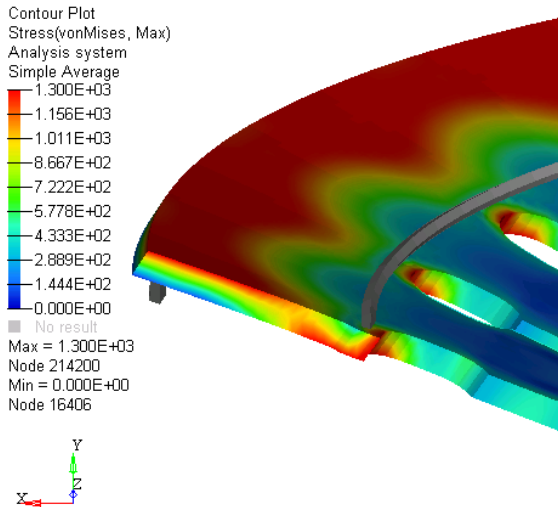


Fig.12 Complex stress at the disengaged position Ø392/ Ø313

4. Conclusion :

The following conclusions can be given:

- The stress at flat spring position with one and two supporting points has small deviation of one curve compared to the other.
- The stress at the maximum spring deflection has the mutual variation of the curves. With one support, the maximum stress occurs in the region of the point 3, and with two supports around the location of the point 2. The reason for this is that in the first case, the diaphragm spring is reclined like a beam fixed on one of its sides, and in the second case the spring is reclined on two supports.

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RESEARCH AND ANALYSIS OF DYNAMIC PARAMETERS IN V-BELTS

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Abstract: In this work is presented research about main dynamic parameters that have influence in the efficiency of trapezoid belts (V-Belts). Research is done in the testing machine with 3 wheels. Two main parameters analyzed are important dynamic parameters: Force of preliminary belt tightening (F_{pr}), and Coefficient of elastic slipping (K_{es}). Last coefficient is important factor for dynamic analysis of belt transporters while it defines clearly level of carrying capacity vs. defined force of preliminary belt tightening. Through laboratory testing has been researched influence of power that is transmitted and force of preliminary tightening in the coefficient of elastic slipping, with the aim to determine optimal limit of this coefficient. In practice it is recommended that values of this coefficient not to be higher than 2%.

Keywords: TRAPEZOIDAL BELT, DYNAMIC ANALYSIS, COEFFICIENT OF ELASTIC SLIPPING, FORCE OF PRELIMINARY BELT TIGHTENING, BELT DRIVES

1. Introduction

V-Belts or trapezoid belts are in the group of mechanical indirect transmitters, which are heavily applied for power transmission in all kinds of machines, industrial devices and motor vehicles. A specific problem during power transmission appears to be the non-constant value of transmission ratio because of the elastic slip of the belt. The elastic slipping appears on a part of the angle where belt embraces the pulley. The elastic slipping is a result of the action of forces with varying intensities on the belt embranchments.

2. Determining Force of preliminary belt tightening

In order to accomplish the work as required by the transmitter, the belt laying above its wheels needs to be fastened with appropriate force. This force that is created in the branches of belt in the condition when transmitter doesn't work is called *Force of preliminary belt tightening* (F_{pr}). Creation of this force is important to have friction in the surface of contact between belt and wheels of transmitter, and this force carries load from guiding (traction) wheel to guided wheel. No matter how tightening of belt is achieved, it is important to determine this force, while it matters for the carrying capacity of transmitter, respectively it matters for the power which can be carried by the V-belt transmitter.

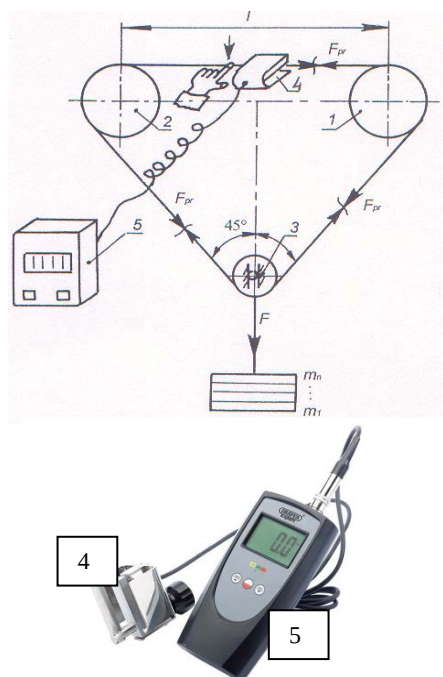


Fig.1. Methodology for measurement for preliminary tightening of the belt. 1- Guiding wheel, 2- Guided wheel, 3-Tightening Wheel, 4 – Sensor for measurement of frequency , 5-Digital measuring instrument

During the experiment are placed various masses in the wheel 3. Their placement has increased force F which acted in the tightening wheel 3, and at the same time the force of preliminary tightening of belt has increased. For various values of acting force in the wheel 3 are measured frequencies of belt oscillations, and their value is registered in the digital instrument 5 (Fig.1). Values measured are shown in Table.1

Mathematical dependence between force of preliminary tightening and frequencies of belt oscillations is given by the formula:

$$F_{pr} = 4 \cdot m_b \cdot (l \cdot f_L)^2 \quad (1)$$

Parameters in the formula are:

m_b (g/m) – Linear mass of belt, l (mm) – distance between center of wheels 1 and 2 of transmitter; f_L (Hz) – frequency of belt oscillations

Experiment is conducted with Belt type AV10x1200La, with linear mass $m_b = 71.87$ g/m and $l = 350$ mm.

Force in wheel 3, F (daN)	42	48	56	68
Frequency of belt, f_L (Hz)	78	87	110	120
Force in the belt, F_{pr} (daN)	21.5	26.7	42.6	56

Table 1. Dependence of force F_{pr} from F and f_L .

3. Changes in the Force of preliminary tightening in dependence of time

With experimentation is researched the issue of changes in the Force of preliminary tightening through time. With experiment has been concluded that this force decreases after few minutes and then stabilizes. This occurrence that happens in the belt is known as *relaxation of belt*.

Values gained with experiment for Force of preliminary tightening are shown in table 2, and diagram in fig.2.

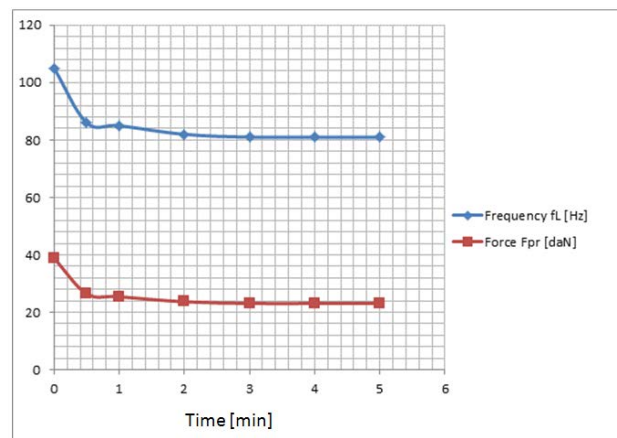


Fig.2. Changes of Force of preliminary tightening F_{pr} in time

Based on result from experiment it is concluded that *Force of preliminary tightening* in the belt drops increasingly in first 3 minutes and then stabilizes in the value 60% of initial value. this occurrence (relaxation) of belt has practical importance in the case of mounting of new belt in the transmitter.

Time [min]	0	0.5	1	2	3	4	5
Frequency [Hz]	105	86	85	82	81	81	81
Force of preliminary tightening, F_{pr} [daN]	38.83	26.45	25.45	23.68	23.11	23.11	23.11

Table 2. Changes of Force of preliminary tightening and frequency of belt oscillations in time

4. Determination of Elastic slipping coefficient

While angle of elastic slipping (γ) cannot be determined experimentally, then the phenomenon of elastic slipping is analyzed through the *Coefficient of Elastic Slipping* K_{es} . This coefficient represents the ratio between difference of the speed of belt in the traction branch and free branch compared with speed in the traction branch of belt:

$$K_{es} = (v_1 - v_2)/v_1 \quad (2)$$

Mathematical model that represents influence of Elastic slipping coefficient in the ratio of the transmission is given through formula (3):

$$i = \frac{n_1}{n_2} = \frac{d_{p2}}{d_{p1}(1 - K_{es})} \quad (3)$$

Based on the formula (3), for the defined number of rotations of guiding (traction) wheel n_1 , depending in the values of elastic slipping coefficient K_{es} , various number of rotations of guided wheel n_2 will be gained. As a result, during work there will be changes in the transmission ratio, depending on the dynamic conditions created in the transmitter.

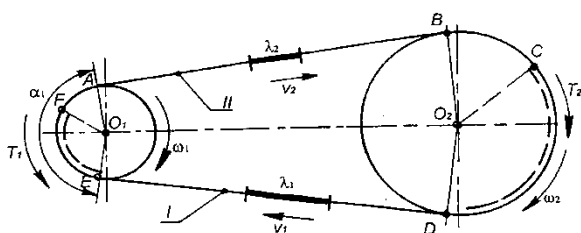


Fig. 3. Elastic Slipping of the belt during work

One element of the belt during the work of transmitter, in the traction branch has length λ_1 and speed v_1 . While free branch will have length λ_2 and speed v_2 (Fig.3). Field of elastic slipping is determined with angle of elastic slipping (γ). With the increase of difference of forces in the belt branch, angle of elastic slipping will increase.

5. Determination of Elastic slipping coefficient K_{es} with experiment

In order to better analyze influential of this coefficient in the work of transmitter, and dynamic conditions of changes on this coefficient, in the laboratory of dynamic research are conducted some experimental testing. Based on conducted research it it

concluded that this coefficient depends on the power that is transmitted P and Force of preliminary tightening F_{pr} . Frequency of oscillations of induced belt is measured with digital tensiometer (Fig.1). Force of preliminary tightening is calculated based on frequencies of belt oscillations, with formula (1). Experimentations are conducted in transmitter with three and two wheels. Also, research is done for the Elastic slipping coefficient of the belts if coating and with cutting. Research device had the option to change the carrying Power from 5 to 20 kW and Force of preliminary tightening through addition of masses $m_1 \dots m_n$ in the tightening wheel 3. Increase of carrying power is accomplished through loading of guided wheel 2. Results gained with experiments are presented in Tables 3 and 4, respectively in diagrams Fig.4, Fig.5, and Fig.6.

Belt AV10x1200 L _a (producer "Ballkan") $d_{p1}=d_{p2}=117.3 \text{ mm}$, $d_{p3}=57 \text{ mm}$ $n=4900 \text{ min}^{-1}$, $t=20^\circ\text{C}$, $\varphi=60\%$, $P_{EM}=20 \text{ kW}$, $B_M=3000 \text{ min}^{-1}$						
	$F \text{ (daN)}$	68	56	48	40	32
Carrying power $P \text{ (kW)}$	$f_{bz} \text{ (Hz)}$	123	112	102	96	86
	$F_{pr} \text{ (daN)}$	53.28	44.18	36.64	32.46	26.06
		5	0.25	0.25	0.32	0.32
		6	0.35	0.38	0.50	0.55
		7	0.47	0.50	0.70	0.75
		8	0.61	0.64	0.90	1.0
		9	0.77	0.82	1.2	1.5
		10	1.0	1.1	1.6	4.7
		11	1.2	1.4	2.2	10
		12	1.5	1.8	7	
		13	1.8	2.6		
		14	2.2	4.7		
		15	3.3	45		
		16	4.0			

Table 3. Elastic slipping coefficient K_{es} [%] for working belt with coating.

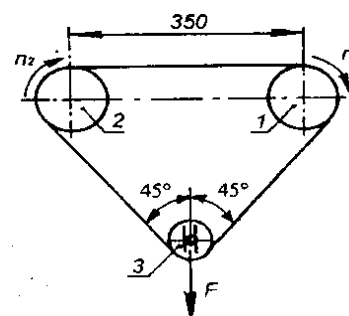


Fig.4. Scheme of experiment

Based on experimental results gained in table3, graph is shown in fig.4. From the recommendations in the literature [1] and [2], Elastic slipping coefficient K_{es} at the transmitters with V-belts should not past the value 2% during work. Therefore, based on fig.4, can be determined up the power of engagement in belt transmitters AV10 for given forces of preliminary tightening. These powers will be:

- For $F = 32 \text{ daN}$ $P = 8.27 \text{ kW}$,
- For $F = 40 \text{ daN}$ $P = 9.10 \text{ kW}$,
- For $F = 48 \text{ daN}$ $P = 11.00 \text{ kW}$,
- For $F = 56 \text{ daN}$ $P = 12.50 \text{ kW}$,
- For $F = 68 \text{ daN}$ $P = 13.90 \text{ kW}$.

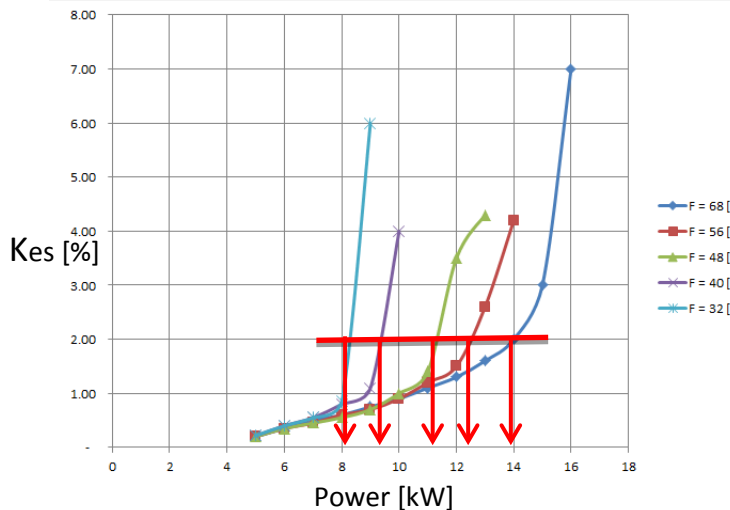


Fig. 5. Elastic slipping coefficient for the belt AV10 manufactured through cutting.

Power that can carry transmitter with V-belts AV10 manufactured through cutting, in order to gave stabile work, will be:

- For $F = 32$ daN..... $P = 8.27$ kW,
- For $F = 40$ daN..... $P = 9.10$ kW,
- For $F = 48$ daN..... $P = 11.00$ kW,
- For $F = 56$ daN..... $P = 12.50$ kW,
- For $F = 68$ daN..... $P = 13.70$ kW.

According to table 4 and diagram 5, for the belts manufactured through cutting, it can be concluded that same conclusion applies as for belts with coating, about the influential of Power that is transmitted and Force of preliminary tightening F_{pr} in the *Elastic slipping coefficient* but K_{es} for belts manufactured through cutting has smaller values. *Elastic slipping coefficient* K_{es} is researched also on the transmitter with two wheels and transmission ratio $i = 1$. results of the experiment are presented on the diagram fig.6.

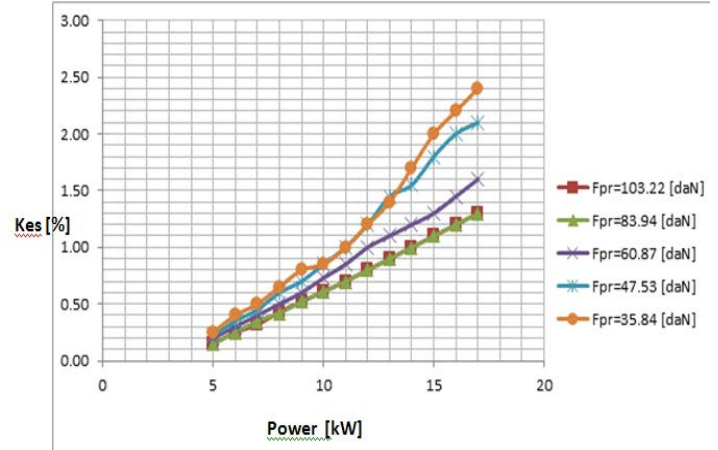
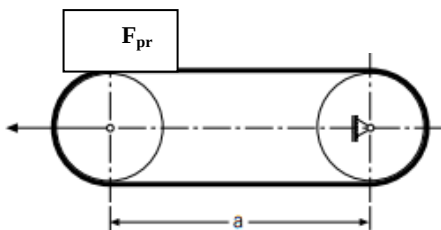


Fig. 6. Elastic slipping coefficient for the belt AV10 manufactured through cutting and transmitter with two wheels and $i = 1$.

6. Conclusions

Analyzing results gained through experimentation, which were represented in table and graphical form, can be concluded as follows:

- Measurement of Force of preliminary tightening first mounted in V-belts in the transmitter should be done after time of 3 minutes, while if this measurement is done immediately, results will not be accurate. Higher values of forces will be measured which are not real.
- By the increase of Force of preliminary tightening F_{pr} , will increase power that transmitter will need to carry,
- Elastic slipping coefficient increases with the increase of power that transmitter carries,
- This coefficient increases with decrease of Force of preliminary tightening,
- Belts manufactured through cutting have smaller Elastic slipping coefficient than Belts manufactured with coating,
- Angle of contact between belt and transmitter wheels has considerable influence in the Elastic slipping coefficient of the belt,
- Transmitters with transmission ratio $i = 1$ have greater carrying capacity than Transmitters with transmission ratio $i > 1$.

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METHODICAL ASPECTS OF DETERMINING THE PROFITABILITY OF THE RAILWAY ROLLING STOCK OPERATOR

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Abstract: This article analyzes and comparison of the various scientific and methodological approaches to determining the profitability of rolling stock operated by operating companies. We consider four different methods of profitability calculation, carried out their comparison revealed the advantages and disadvantages of each method. Recommendations on the field of application of each method.

KEYWORDS: PROFITABILITY, ROLLING, ROLLING STOCK OPERATOR, METHODOLOGY.

1. INTRODUCTION

Preserving stability in the market and the success of the enterprise depends on a clear understanding of - what types of products and some units offer a large profit, and what - are detrimental to efficient business enterprise. Costs and revenues, and the relationship between them, are two key aspects that ultimately determine the railway's future position in the freight transport market [1]. An investment in the rail sector represents costs and benefits for a wide range of institutions, companies and individuals. Actual flows of cash constitute the basis for the financial assessment of the project [2].

Identify errors in the work schedule of profit growth and to identify ways of further development of reserves helps to analyze profitability of the enterprise. The yield can be regarded as a general indicator of economic efficiency of the business processes of the organization.

What is the yield?

Based on different user, owner, and stakeholder expectations, the quality of the rolling stock can be defined in many ways [3]. According to the definition given number of sources (eg, - "BusinessTimes" - magazine about business in Russia and abroad), the yield (Yield), or rate of return - used in the economy (in finance) a relative measure of efficiency investments in certain assets, financial instruments, projects or business in general. Yield can often be classified as the ratio of the absolute value of income to a base, which is usually the amount of initial investments or investments to be carried out for this income.

The financial stability of the enterprise is entirely dependent on the sustainable development of its business activities. Currently the rolling stock industry is facing heavy consolidation pressure due to large global overcapacity [4]. For example, if railway achieves financial sustainability when it has sufficient longer-term financial resources to cover operational costs, to invest, and to meet debt service and other financing requirements [5]. In the US, rail operators have seen an accelerating decline in high margin coal business and McKinsey reports that none of the larger European carriers have achieved average margins of 3% in earnings in recent decades, and at least 7% is needed to cover their costs of capital [6]. Research shows that effective price management can increase margins by 2 to 7% within 12 months, and yield a 200 to 350% return on investment [7]. That is why an important condition for this is a professionally competent and timely implementation of the process of determining the yield to accurately describe the perspectives of the relevant investment or management decisions and is an effective mechanism for evaluating the profitability of transport operator.

2. PROBLEM STATEMENT

Speaking about profitability in the field of railway transport services operator examines the profitability of rolling stock - a relative indicator of the efficiency of rolling stock per day, defined as the ratio of the difference in revenue from transportation and relocation costs for empty wagon to wagon turnover.

On the basis of the definition that the profitability of the car work affects revenues from transportation, the cost of relocation of the empty car and the car turn.

Proceeds received from the transport operator, formed the basis of a carload component, taking into account the level of mark-up / discount rate to 10-01 Price List and is set per tonne of cargo carried in the car, or for a loaded one wagon-sending.

Costs of the operator in this case are the costs of empty runs of the car calculated in accordance with the Price List 10-01.

Wagon turnover is defined as the average time in days for which the car passes a distance equal to its full flight, taking into account the duration of parking in the loading / unloading stations.

At the present time, it has become imperative for rail systems to optimize the fleet size and freight car allocation in the presence of uncertainty. The problem can be considered as the problem of finding an optimal fuzzy regulator for a fuzzy linear system with fuzzy quadratic performance index and fuzzy random initial conditions [8]. Developed dynamic model of loaded and empty rail freight car flows explicitly treats state, control and station capacity constraints in presence of various freight car types under the partial substitutability among them. Demands and traveling times are considered as random variables [9].

Planning the structure and volume of the rolling stock is a key factor in achieving maximum efficiency of transportation by rail as well as forecasting the demand for these transport facilities [10].

The definition of rolling stock yields helps to assess the potential of optimizing freight operator base and, by ranking customers, allowing time to identify unprofitable transportation.

Income calculation of the car is possible by using several approaches, each of which corresponds to the basic questions:

1. What level of profitability provides a specific kind of rolling stock?

2. What is the rate of return provides a specific transportation / direction of transportation?

Wagon yield is calculated on a separate branch of the rolling stock. In this case the total revenue from the operating activity of the subject kind of rolling stock (hereinafter - RPM) minus the total cost of relocation of empty wagons considered RPMs divided by wagon-day.

In determining the profitability of operators in the costs can account for the different set of empty runs: one account for all the costs associated with empty flights, others - in costs do not account for empty flights to repair, attributing these costs for the maintenance of rolling stock, etc ..

The objective function of maximizing the average daily financial result relies heavily on the integration of logistics activities with an improved management of revenues. The operational policies chosen by the carrier have an important impact on the network yield and thus on global profitability [11].

In determining the wagon-days (the product under review, the park RPS operator the number of days in the reporting period) - and there are several approaches:

- Take into account the whole wagon-day inventory / usable / working park operator;

- Are not considered wagon-day related to the dregs.

In practice, operators expect to yield:

- The entire park operator wagons, i.e. total fleet operators of cars that are in operating;

- Fit cars park operator, i.e. wagons that may be repaired and used in operations, defined as the sum of non-working car fleet and working;

- Working park operator wagons - wagons, technically sound for the transportation of goods and used by the operator to provide transportation in accordance with the plan and requests of customers.

In general it can be noted that the methodological definition of return on CPF quite clear, and the variation calculation enables the operator to assess the impact on the profitability of individual factors such as the level of non-working of the park, the number of cars in the sludge and their impact on profitability, etc.

3. METHODS OF SOLUTION

Based on the need to assess each individual transportation, there are 2 questions to the operator:

Which revenues should bring the carriage to pay the total cost for the maintenance of rolling stock to ensure it? Total costs include costs associated with the maintenance of the car (including current and planned repairs), the costs of maintaining operating

company staff and direct costs of the subfamily under the carriage or transportation after removal of the empty wagon, etc.

What are the direct costs attributed to the considered transportation?

For the answer to both questions is in practice used alone or in combination, are some basic methods for determining the profitability of transportation.

Let us consider in more detail each of the 4 main methods of calculating the profitability of transportation:

3.1. Determining the yield based on the next flight or the empty set of the following empty flights

$$D = \frac{(\text{revenue} - \text{costs subsequent por.reys})}{(\text{wagon turnover})}$$

where *revenue* - revenue received by the operator of the transport in a particular direction (mainly depends on the RPS, the direction of transportation, cargo / cargo class);

the cost of the subsequent empty flight - the costs associated with the relocation of the empty carriage after the carriage in the direction being considered.

$$D = \frac{(\sum B_1 - \sum C_1) - (\sum B_2 - \sum C_2)}{(T_1 - T_2)} \quad \text{wagon laden and unladen subsequent flight of}$$

$$\sum C_1, \sum C_2 T_1, T_2$$

the car.

This method for determining yields for calculating the yield of the turntables (i.e. traffic returning to the loading station) as well as for the specialized rolling stock, which is usually drawn on a closed landfill and has empty run is equal to or close to unladen.

A significant advantage of the method is its simplicity of calculation and the minimum level of automation.

A significant drawback of this method is the inability to determine the level of profitability for optimizing traffic (i.e. "free" primary direction of following empty car traffic) and transport in complex logistics schemes (for example, universal rolling stock). Also, this method does not allow to determine the yield on the client as a whole.

Despite these shortcomings, the method is one of the common and is used by most operators.

3.2. Determining the yield based on previous unladen journey

$$D = \frac{(\text{revenue} - \text{costs por.reys previous})}{(\text{wagon turnover})}$$

where *the costs of the previous day. flight* - the costs associated with overseeding of empty wagons for carriage in the direction being considered.

In determining the turnover accounted wagon laden voyage and the previous empty runs. Thus, the turnover of the car in this case is calculated from the end of the previous discharge laden voyage to the end of discharge calculated laden voyage.

3.3. Determination of return by calculating it in the logistic scheme

This method is rarely used in practice, because of the complexity of billing the client on a wagon subfamily costs, especially in the period of balance or surplus rolling stock, as Determination of return by calculating it in the logistic scheme:

$$D = \frac{(\sum B_1 - \sum C_1) - (\sum B_2 - \sum C_2)}{(T_1 - T_2)}$$

where revenue $\sum B_1, \sum B_2$ - total revenue respectively, against all loaded flights included in the logistic scheme under consideration with a freight transport and without it;

$\sum C_1, \sum C_2$ - respectively, the total cost of all the empty flights included in the supply chain with consideration laden transport and without it;

T_1, T_2 - respectively, the time spent on the passage of cars of all chains logistic schemes considered a freight transport and without it.

Under the scheme is a sequence of logistic supply chains, which consist of loaded and empty flights. Logistics scheme begins and ends at the same station or loading region (eg, for the gondola Kuzbass). Thus, the rate of return on the 3 method is determined by comparing the logistics schemes under consideration transport and without it.

The main advantage of this method is the possibility of obtaining an answer to the question of the profitability level for any RPM, and in many areas, including optimization. This method can be applied to operators with regional logistics and transport with stable traffic areas.

For network operating companies and organizations that actively involve the new transportation from the market, a significant drawback of the method is the inability to perform the correct calculation for the entire logistics operator. For example, there is the possibility of uncontrolled referring to the optimization of direct transportation of a large amount of car traffic volume; logistics scheme operator not linked with each other; optimization options for calculations and linking logistic schemes are not computed. In addition, staff are required to the same understanding and deep knowledge of the logistics operator traffic, linking their settlements with a total logistics company - this requires an appropriate level of automation.

Despite the presence of serious adverse moments 3 method is the most accurate of the above in terms of determining the profitability of transport, but even with the appropriate level of automation it will concede 4 calculation method.

3.4. Method for determination of profitability by converting the entire logistics operator with the use of optimization models

The basis of the method - calculation of optimal logistics operator, taking into account the entire cargo base (option 1) and without consideration transportation or customer (option 2). This method is determined by the impact of the considered transport or

client on profit margins throughout the cargo base, the yield is calculated.

In fact, the method simulates a situation in which the considered transport / client would not be in the cargo operator based. Thus, it is possible not only to assess the transport, but also the client as a whole, to identify ways to improve the logistics operator.

The answers to these questions form the basis of mismatches ("gaps") between schemes zaadresatsii empty wagons to the facts of their relocation. These "gaps" due to the following reasons:

- The Convention's prohibitions on over seeding cars in the region of loading;
- Errors in the management of fleet cars;
- Rapid change in customer service technology.

A significant advantage of the method is the possibility of determining the existence of discrepancies between the actual management of the park empty wagons and optimization calculations that allows for management decisions aimed at eliminating suboptimal over seedings of empty wagons. Also, this method allows for the ranking of the client portfolio on profitability and make decisions aimed at:

- Changes in cargo base by searching transport, improve logistics;
- Change the principles zaadresatsii empty wagons for loading operator, including with measures aimed at reducing car traffic;
- Changes in pricing conditions in certain areas for the elimination of unprofitable operations;
- Refusal of transport.

The disadvantages of this method is the complexity of the calculations, and the presence of diverse geography of transportation - a significant length of computation (from tens of minutes to several hours). These drawbacks are solved by automating the calculation.

4. CONCLUSIONS

Thus, the first three methods of determining the yield can be used for rapid analysis and only if all the following conditions are met:

- No optimization model for calculating the logistics operator;
- The operator has a fleet of no more than 30 thousand cars;
- The operator is working on the free market, i.e., park operator wagons are not fixed service contracts;
- The operator is presented only in a limited range network - operator with limited transportation logistics.

The fourth method - at the appropriate level of automation - is used for the integrated assessment of the operator's activities well as - the complexity of predicting the direction of the subfamily of empty wagons.

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CONCENTRATION IN LINER SHIPPING AND ALLIANCES FORMATION: ISSUES AND CHALLENGES

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Abstract: *The present article analyses the present status of the container liner shipping and its market concentration. The forming of strategic alliances is considered in detail against the other forms of market cooperation. The specifics of strategic alliances in liner shipping is outlined, the latter being one of the drivers for operational and strategic efficiency of liner services. The results show that there has been a significant increase of market concentration as from 2014. The formation of strategic alliances ensures for lower operational costs although leading to, in the short-run, overcapacity and increased internal competition. From a long-term perspective, the formation of strategic alliances will lead to increased market concentration and, eventually, to more stable shipping markets and decreased competition.*

Keywords: LINER SHIPPING, CONTAINER TRANSPORTATION, MARKET CONCENTRATION, STRATEGIC ALLIANCES

1. Introduction

The development of international trade is dependent primarily by seaborne trade. Efficient liner shipping services is the basis for provision of competitiveness of maritime companies. During the last three decades liner operators have not only provided the physical movement of goods but have undertaken various commercial and marketing actions to further promote international movements of goods by sea.

A specific opportunity for liner shipping companies for gaining competitive advantage in a crisis-strained market is by entering into cooperation with competitors via strategic alliances. It should be mentioned that the liner shipping industry has pioneered with the concept of market cooperation in order to improve their strategic goals. The first strategic alliances have been set up at the end of the XXth century. Being a truly horizontal type of cooperation, integrating sharing of capacity and dedicated services, strategic alliances have proved to be a viable option for steadier market position. Alliance agreements can be characterized as technical agreements [6]. The marketing activity is retained within the control of the member as well as cargo documents and ship management.

There have been numerous agreements in the industry whereas only several companies participate in cooperation for the global network. The latter are presently forming the largest three strategic alliances concentrating mainly on the longest routes - East-West services. Liner shipping operators participate in the formation of different formats of market cooperation in newly established services and/or routes. The purpose of this article is to analyze the specifics and pertaining issues of strategic alliances in modern liner shipping. The specifics studied comprise structure of ownership, routes coverage and extent of cooperation with other liner operators. The specifics of the agreements of the largest three alliances are further outlined thus focusing on the development and re-structuring of services. Important features of the liner shipping companies that traditionally participate in alliances are also presented. The analysis encompasses market share, routes coverage and services organization, as well as the founding, conduct and structure of liner shipping strategic alliances.

2. Alliances in liner shipping – theoretical background

Liner shipping firms have a history of co-operation with the most prominent example being the price-fixing agreements between them in the context of the liner conference system [10]. The setback of the liner shipping conference system via the introduction of the US Ocean Shipping Reform Act in 1998 and the EU appeal of the exemption of anti-trust rules in 2008 have served as an impetus for establishment of other forms of cooperation in order to obtain

competitive advantages [2]. The latter process started as of the mid to late 1990s and throughout the last decade. Main feature of strategic alliances in liner shipping have been the coordination of liner shipping services and cooperation. There are several types of alliances in modern liner shipping. One of the most common types is the strategic (global) alliance which is considered as a new type of co-operative agreement in ocean shipping [4]. Such alliances were established in the mid-1990s aiming global cooperation on major trade routes (Europe – Far East, Transpacific, Transatlantic).

The purpose of strategic alliances, being a form of horizontal integration, is to achieve cooperation for the ships' capacity utilization on certain routes. The latter includes distribution of ships by type and size, timetables, joint use of terminals and empty container repositioning on a large scale. Furthermore, none of the activities carried out by alliances' members include canvassing, sales, marketing or tariffs fixing, common assets ownership, collecting in a pool of revenues or the profit/loss sharing. Alliance membership imposes restrictions on a member's use of a non-member carrier [12]. There are also specific terms for severance (withdrawal) via advance notice and pertaining penalties especially in a longer term agreements. Unlike conferences, alliances have no policy of common tariff fixing but instead complete cooperation is achieved as concerns capacity and offered services. Each individual member ensures for the complete client relationship management starting from the enquiry up to the final delivery to the destination. Only a few large companies, having larger fleet and service network "play" on their own (Maersk Line, MSC, etc.). As such companies possess considerable assets, they can achieve scale economies individually. [8] have outlined the concept of value chain whereas making a distinction between cooperation strategies depending on the type of the pooled resources. Thus members might include similar resources in the cooperation for economies of scope and decrease business risk. Similarly, members might participate in the alliance and devote additional resources to achieve competitive advantages. It can be claimed that the formation of, or even defection from, strategic alliances as well as the implementation of other strategies (such as M&A) are all driven by the need to accomplish corporate objectives [7]. These strategies include maximization of profit, sharing of capital investment, reduction of costs, attaining economies of scale, covering wider geographical scope and development of new markets. More efficient activities related to marketing (higher flexibility and service frequency, expanding of networks and reliability as well as various operation objectives are also achieved via entering in alliance agreements.

One of the strongest incentives for companies is attaining market power. [11] define market power as: "the ability of a market participant or group of participants to influence price, quality, and the nature of the product in the market place". There are various options for market power: elimination of barriers to market entry, product differentiation, increase of market share. There are two basic types of market cooperation in this respect. Offensive cooperation is set up for enhancing the competitive advantage of the

members by increasing the market share or decreasing of costs. On the other hand, defensive cooperation, typically favored by smaller companies, creates entry barriers thus securing the members' market shares and increasing profits. The establishment of alliances also created added value for customers. The main objective is to decrease transaction costs via timely and correct receipt of information using the members' benefits.

Irrespective of the obvious advantages of liner shipping alliances, there are certain issues of for some members related to strategic changes and company instability. The latter may be due to individual company's requirement to keep up with the company's objectives which may interfere with the proper cooperation within the alliance. In addition, other factors such as the number of partners in an alliance, the nature of their role and contribution to the alliance, the level of mutual trust and the complexity of the task itself may play a significant role in alliance instability [3]. There is also the issue of internal competition and conservative approach to the alliance strategy that also leads to possible instability of the alliance. [4] suggest that alliance stability and efficiency may be achieved by focusing on one or more of the following three measures: reduction in the number of partners, differentiation in their roles and contributions, and co-ordination of sales and marketing activities.

3. Global strategic alliances in liner shipping: structure and conduct

Figure 1 provides an overview of the fleet size and company characteristics. The companies are listed in accordance with their size based on capacity (TEU) volumes. As evident a vast majority of vessels are owned other than chartered in.

Rank	Operator	Total		Owned		Chartered		% Chart
		Teu	Ships	TEU	Ships	TEU	Ships	
1	APM-Maersk	3,527,257	657	1,711,114	245	1,816,143	412	51.5%
2	Mediterranean Shg Co	3,061,443	505	1,082,557	190	1,978,886	315	64.6%
3	CMA CGM Group	2,455,764	484	914,214	117	1,541,550	367	62.8%
4	COSCO Shipping Co Ltd	1,808,844	326	493,079	81	1,315,765	245	72.7%
5	Hapag-Lloyd	1,514,958	216	1,016,413	117	498,545	99	32.9%
6	Evergreen Line	1,045,713	193	548,041	105	497,672	88	47.6%
7	OOCL	655,746	100	427,574	54	228,172	46	34.8%
8	Yang Ming Marine Transport Corp.	587,815	98	209,150	45	378,665	53	64.4%
9	Hamburg Sud Group	555,943	102	313,508	46	242,435	56	43.6%
10	NYK Line	538,101	95	238,574	40	299,527	55	55.7%

Figure 1. Top 10 liner operators capacity structure [14]

The largest shipping companies are APM-Maersk, MSC and CMA-CGM are in control over 60% of operated ships among the top 10 companies. The same is valid for ships that are chartered in. This suggests that there are significant differences between the companies in terms of their owned versus chartered strategies. The companies choose different models to alleviate risks of ship owning in the liner shipping industry. Figure 2 presents the market share of global liner operators as of May, 2017 [14].

The purpose of this article is to outline the trends, specifics, differences and similarities of some of the leading strategic alliances in liner shipping.

Figure 3 presents an overview of the alliances development. Traditionally, the strategic alliances formation comprised agreements on the busiest service routes - East-West trades.

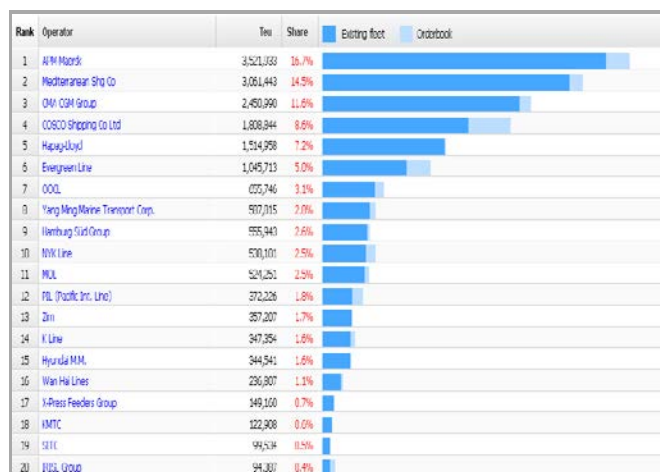


Figure 2. Top 20 liner operators by market share (%) [14]

However, during the last decade, cooperative activities have enlarged to encompass the north-south trades. As for the big three alliances, "Ocean Alliance", comprising CMA CGM, Cosco, Evergreen and OOCL is considered being most stable unlike the other two that have undergone some significant changes. There is a steady trend for alliances expansion in terms of number of ships operated and capacity. However, this development was supported by the individual growth of each alliance member [12].

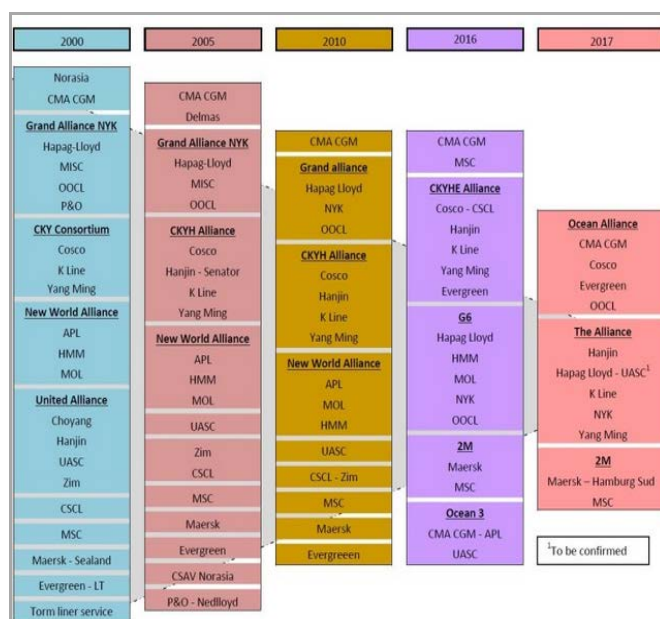


Figure 3. Alliances formation for the period 2000 – 2017 [9]

At the beginning of the 1990's the liner shipping industry was dominated by about thirty shipping companies (about 60% of the total fleet). Deregulations of liner shipping market concentration served as an impetus for capacity sharing in order to attain more flexibility on various trades. Gradually, to achieve lower costs and scale economies various forms of market concentration have become common - alliances, mergers and acquisitions. By the end of the 1990's only six alliances existed representing about 50% of the world container fleet (TEUs). By the end of the 2010 the overall fleet capacity of the top 30 operators has doubled, reaching about 11 mln TEUs. Even at that time there were only three alliances: Grand Alliance (Hapag-Lloyd, NYK and OOCL), CKYH Alliance (Cosco, K Line, Yang Ming and Hanjin) and New World Alliance (APL, MOL and HMM) while Maersk, MSC and CMA CGM, being the biggest players in the sector, still acted on their own. The latter

three companies represented about 50% of the fleet capacity. In 2014, the merging of CSAV and Hapag Lloyd was caused by the need to obtain additional resources and decrease losses. The same year, the P3 alliance was announced, comprising CMA CGM, Maersk and MSC but was further rejected by the merger control rules in China. The following year, in 2015, the Israeli controlled company Zim was the main individual carrier not participating in alliances, operating primarily via partnership and slot sharing on various routes except for Asia-North Europe [13]. By April 2017, three main alliances, The Alliance, Ocean Alliance and 2M, with a total fleet of 15.862.743 TEUs [5] are to be in operation, representing about 75% of the market. However, the “Ocean Alliance” is still undergoing formation by CMA-CGM, China COSCO, Evergreen and OOCL. “The Alliance” comprises Hapag-Lloyd, having merged with UASC, MOL, NYK line and Yang Ming. The two major market players - Maersk and MSC, are forming the 2M alliance. Lately, Hyundai Merchant Marine has also concluded an agreement thus forming H2M alliance.

For the purposes of concentration calculation the Herfindahl-Hirschman Index (HHI) is applied for demonstrating the specifics of the market. Until 2014 there was almost even distribution on the market with moderate concentration at about 1350 points (Figure 4). Gradually, only within three years the HHI has increased up to 2000 point while the number of companies decreased. In 2017 COSCO merged with CSCL and CMA CGM acquired APL (NOL).

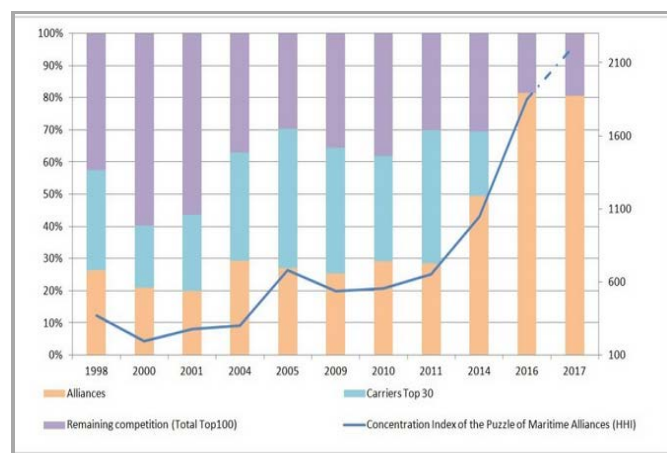


Figure 4. Distribution of fleet capacity and market concentration (1998-2017) [9]

According to [1] the present market is quite weak with low rates and overcapacity. The observations based on the HHI show a steady trend towards market concentration. As the number of companies entering into alliances will increase, the index is expected to rise exponentially.

The operations of the three biggest alliances are concentrated on the Europe-Far East and Transpacific routes. The announcements for the formation of CKYH Alliance, the Grand Alliance (GA) and the New World Alliance (NWA) for the 10 year period 2000-2010 have been studied in order to outline the companies' motives. On the basis of the announcements analysis made the motives for entering or leaving an alliance comprised firstly strategic reasons, secondly - operational reasons, enhancement of connectivity and optimization of capacity, new services establishment, slot sharing. Operational specifics – number of employed vessels, number of port calls, average voyage duration are similar among the biggest alliances. Services are constantly adjusted mainly due to strategic reasons despite the transfer of fleet capacity on the routes Asia. It should be noted that the management of the alliances requires internal negotiation of activities especially as concerns single services. Service agreements are being changed continuously according to the global demand for services – suspension or

introduction of services, merging of services, balancing of capacities. Due to the latter there were over 300 adjustments to the services within the last decade. This requires close cooperation between alliance members while the operations activities are still the responsibility of the individual members. The announcements overview demonstrated that members are not seeking forms of joint operations.

As the liner shipping industry is very dynamic there is no distinguishable pattern for service adjustment. Reasons for service adjustment can be seasonal factors or merging of parallel lines. It is a fact the significant changes on liner shipping services patterns are the indicator for a global shift in the world trade. Having truly global networks, alliance members can easily handle surplus in capacities and redirect same to areas with high demand (geographical diversification). Following the major economic downturn in 2008, as container liner shipping was also negatively affected, liner agreements have undergone significant changes – fleet capacity was shifted to the Europe – Far East routes from Transatlantic/Transpacific routes. Thus Europe-Far East services have been adjusted more frequently unlike the Transatlantic services. These results are another argument in favor of the dynamic. This observation reflects the high dynamics in the emerging Asian markets and their relationships with North American and European economies. Also the high flexibility in capacities geographical shifting justifies the high number of vessels ordered (unlike the dry bulk industry). Nevertheless, members have an individual understanding of the manner the joint services are organized. Companies would choose to have a central position in the alliance for two reasons: (1) enhance their internal influence within the alliance and thus decrease competitive instability and (2) ensure for maximization of access to strategic assets and decrease uncertainty of demand.

4. Conclusion

There are specifics pertinent to every alliance as concerns the variety of the partners, their market behavior and stability. These specifics are related to the alliance structure and size and also the individual conduct of the partners, namely partners' new-building strategy, the scale of the agreement, etc. The present current analysis shows that every membership in the alliance is affected mainly by operational motives. Most firms are prone to establishing alliance relationship with competing companies on a number of services. Medium size firms are generally more active in joining strategic alliances in liner shipping as they are most vulnerable to the uncertainty of demand unlike large or small companies. Large-scale firms dispose of more assets and resources and have better options to achieve strategic objectives thus coping with demand uncertainty on their own. The article has proved that the formation of liner shipping strategic alliances includes one member having a dominant position as concerns the fleet capacity contributions. Excluding the “2M” alliance where there are two major members contributing with resources generally the dominant member ensures for above 50% of fleet capacity in the alliance.

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