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DIESEL ENGINE'S COMBINED CARBURETION PROCESS

ПРОЦЕСС КОМБИНИРОВАННОГО СМЕСЕОБРАЗОВАНИЯ В ДИЗЕЛЬНОМ ДВИГАТЕЛЕ

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Abstract : The carburation process in modern diesel engines practically begins with the moment of fuel injection in into the cylinder, and ends at the same time as the combustion process. The improvement and development of the carburation process depend on the injection parameters, particularly on: the motion of charge in the combustion chamber; fuel properties; combustion chamber sizes; a surface temperature; interdependent motion of charge and fuel. With a view to carburation in diesel engines, we obtain a nonhomogeneous mixture in diesel engines. First of all, the internal carburation process cannot ensure uniform distribution of the injected fuel's steam and air. Besides, the improvement of carburation process is impeded by fact that as a result of the combustion process development there occur the fuel injection and carburation that increases that amount of fuel, which is burnt in the expansion line. All this causes increasing thermal heat losses and growing amount of toxic substances in exhaust emissions. Also, the improvement and efficiency largely depends on the length of self-ignition impeding period that directly defines the engine's dynamic magnification factor. The paper dwells also on the possibilities of partial eradication of negative phenomena and increasing the engine's efficiency, as well as reducing toxicity. In the engine's intake system, the electric injector is inserted, by means of which, at the beginning of the intake process, there is carried out the injection of a certain amount of fuel in front of the inlet valve, and together with the air coming into the cylinder it creates the mixture, which occurs during the filling and compression process, and at the end of the compression process, the main amount of fuel is atomized by means of basic injector.

By the end of the compression process, until the fuel is atomized from the basic injector, we have the depleted, but almost homogenous mixture, and therefore, this mixture is uniformly distributed in the combustion chamber. At the same time, the primary oxides, to a certain extent, are created for starting combustion. Immediately after the fuel atomizing from the basic injector, we will obtain the mixture required for combustion, and the combustion process begins earlier than during the process of the creation of a standard mixture, and this means that the length of self-ignition impeding period reduces, and consequently the efficiency goes up, and the amount of soot in exhaust emissions is reduced.

KEY WORDS : DIESEL ENGINE, CARBURATION, FUEL INJECTION, TOXICITY.

1. Introduction

The work process of ignition mixtures for diesel engine is conventionally divided into the following main phases: [1]

- initial stage of time τ_1 - during which the temperature rise is negligible, or it may be a decrease of temperature, so that the pressure does not increase. Completion by the end of this stage formed aldehydes, oxides of carbon, oxidants and decomposition products;

- Initial stage refrigerating plasma τ_2 - during which allocated 10% of the heat evaporation of burned fuel, whereby there is a slight increase in temperature and pressure. Suggest that at this stage of the process, take place the oxidation of acetaldehyde and achieved critical concentration of new types of oxides and the accumulation of their products.

- secondary cold-flame stage τ_3 - during the passage which chemical reaction has as endothermic and exothermic nature, There are a slight total heat allocation process, and it is accompanied by shades of blue flame.

Thus, each review stage, which is carried out in a specific time period, gives the ignition delay: τ_1 - Cold flame; τ_2 - blue flame, which also refers to a stage cold flame and τ_3 - thermal explosion. Thus, the ignition delay period is equal to $\tau = \tau_1 + \tau_2 + \tau_3$ which is shown in Fig. 1. Of the three stages of the ignition delay period long stage period τ_2 is long stage period - cold flame, that is 75-80% of the total ignition delay period. Thermal explosion spreads rapidly in the cavity of the combustion chamber and is characterized by the release of the total thermal energy, followed by Instant increase pressure and temperature. (Fig. 1).

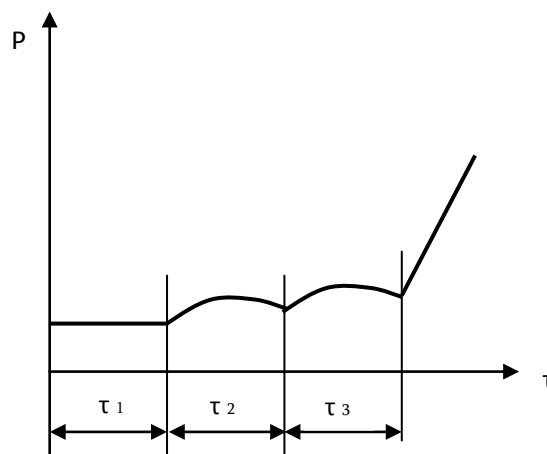


Figure 1. Schematic representation of the ignition process in the diesel engines.

Duration of the various phases of the autoignition delay period depends on the specific conditions. Thus, with increasing pressure during the ignition delay period, the first and the third stages are greatly reduced, at the same time, the initial process of the nature of the flame remains generally unchanged. respectively, for ignition of fuel in a diesel engine will not be necessary the existence of all three consecutive steps. So, for the τ_3 stage, with a very small period of time process is a two step process. In this case, the second and the third stage correspond to the initiation high cold ignition flame in the first stage. So at the same time periods second and third stages of $\tau_{2,3} = \tau_2 + \tau_3$ are considered as a hot flame delay. interpretation of this ignition delay period, comfortable enough, because somewhat difficult to distinguish (separate) the time of appearance of the blue flame and thermal explosion. Accordingly, the essence of the final process of auto-ignition in a diesel engine is the implementation of a possible auto-ignition of the working mixture provided when the working mixture with an initial low temperature.

2. Preconditions and means for resolving the problem

The duration of the time of the ignition delay period τ_i of diesel engines depends upon many factors, including temperature and air pressure in the cylinder at the fuel injection timing. The influence of these parameters for a ignition delay time is important because fuel injection moment into the cylinder, the process of mixing at the end of compression depends on the compressed air and temperature. This means that if the allowable temperature and pressure values are lower, then the ignition delay increases and therefore worsens mixing process which negatively affects on the efficiency of the engine and toxic properties.

Air pressure and temperature during the filling process - Increasing values of these parameters at the end of compression occurs with increasing temperature and pressure, respectively, with fuel spray. As a result decreases time of autoignition delay, therefore heating the incoming air is a good opportunity to ensure the normal development of the combustion process, but it should be noted that heating the inlet air reduces the quality of the degree of filling of the cylinder and, hence, efficiency engine performance; increase of quality of compression increases the pressure and temperature at the end of compression at the beginning of fuel atomization. Thus, the ignition delay time decreases. Heating the incoming air in the inlet system will cause thermal stress major parts, as increase compression degree, and the increase in of nitrogen oxides due to maximizing temperature in cycle.

Quality - hydrocarbon fuels quantitative increase carbon atoms shortens the delay period autoignition. The higher the cetane number of diesel fuel more likely to self-ignition propensity therefore reduced auto-ignition delay period.

The shape and dimensions of the combustion chamber - studies carried out with diesel engines with fuel injection can not reveal the precise regularity between the size of the engine and a latency period of auto-ignition; Therefore it is logical to consider the impact of the main dimensions of the cylinder on the value indicators of pressure and temperature at the start of the process the fuel injection. Usually taken into account the value of the specific heat area transfer on the impact the geometrical dimensions of the engine and the duration of the autoignition delay period, the more engine power, the less marked area and minimal amount of heat consumed from the mixture, respectively unto the completion of compression the temperature increases and reduces the period the ignition delay.

Pressure and saturation charge of vortex motion injection - Higher fuel injection pressure reduces the average diameter of the fuel spray droplet, Increases the total area of evaporation, evaporation rate Accordingly fuel carburetion, As a result, this leads to a decrease in the ignition delay period. It is difficult to create pressure the equipment fuel pressure with high-pressure spraying. Therefore, more and more widely used diesels with separated chambers where intensive mixture formation occurs mainly due to strong turbulence kinetic energy of the new charge, so do not require the use of sophisticated equipment for fuel. At the same time, remains the problem of unsolved undivided diesel engine with low efficiency. So now, more widespread received the diesel engines with combustion chambers separated by half. Thus at present, more widespread diesels, with a half-separated combustion chambers, used in the result a relatively high pressure for the fuel injection which significantly increases the rate of production of steam and the delay period of autoignition is reduced.

The intensity of the vortex motion charging - when it is possible to select slightly higher intensity motion to improve mixing. I.e The higher the intensity of the vortex motion, the lower the autoignition delay period. At the same time, increasing the intensity of the vortex motion can cause deterioration in performance of the engine.

As a result, one can observe, that analysis of the facts above shows that they affect the duration of the autoignition delay period, but do not provide an exhaustive answer.

Thus the development of fast transient diesel engines, the effective use of the working volume and opportunities undivided cameras, hinder-inferiority of the internal mixing of mixture, I.e Inability achieve a relatively uniform mixture, is a significant limiting factor in the excess air ratio (Rated power $\alpha \geq 1,35$).

To implementation partial elimination of side effects and increase efficiency and reducing the toxicity of the engine to the inlet cylinder system of the diesel engine, set fuel injector (Fig. 2) by means of which starts the injection of process a certain amount of diesel fuel in front of the inlet valve and begin the process of carburetion with air, entering the cylinder with air which occurs during the filling process and at the end of compression which occurs during the filling process and at the end of compression and the compression process in the combustion chamber by means of the main injector (in a prearranged lean fuel mixture) injected the amount of fuel

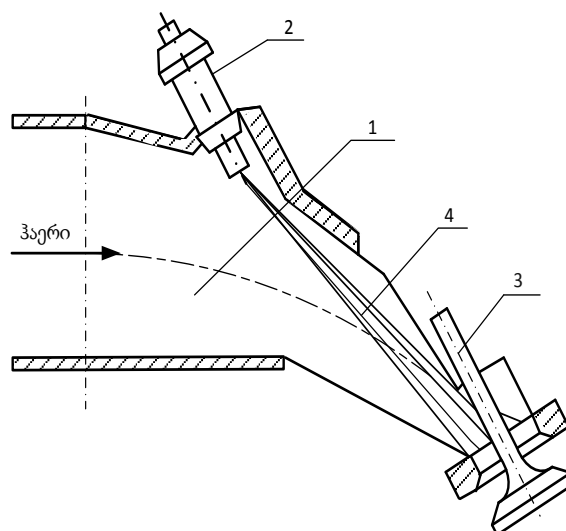


Figure. 2. atomization of the fuel circuit in the input system of cylinder;

1 - inlet channel; 2 - fuel injector or dispenser; 3 - inlet valve; 4 - soot from the atomized fuel

that is necessary for the selected mode. In this case, the fuel mixture formed as in the process of filling or compression process. At this time, the fuel vapor with air in contact with the heated surface of the engine and begins the process of mixing intensively before. Before reaching the top of the critical state of piston compression should spray from the main injector the amount of fuel that is required to implement the selected mode.

3. Conclusion

By the end of the compression process until spraying fuel from the main injector, in combustion chamber, we have lean mixture, but almost homogeneous mixture, which is uniformly distributed in the combustion chamber cavity. At the same time, to some extent primary oxides are formed for begin the combustion process, but insufficient for realization of the combustion process. As soon as a the fuel is sprayed from the ground sprayer, we could obtain necessary quantity of fuel mixture required for combustion and the combustion process starts earlier than the formation of the standard mixture, and hence declining value of the autoignition delay period.

4. Literature

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LOGISTICS LEGAL REGULATION PROBLEMS

ПРОБЛЕМЫ ПРАВОВОГО РЕГУЛИРОВАНИЯ ЛОГИСТИКИ

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Abstract : *Relationships that are the subject of logistics in general, in principle have still not been resolved from a juridical standpoint. Even individual types of logistics do not have holistic, integrated legal regulation. However, even the legal regulation of the transport logistics is a fragmented set of norms regulating individual types of transportation, but not a transport logistics in general.*

It is hard to imagine that in principle it is possible to develop a set of international legal standards, which would constitute a coherent legal system ensuring the functioning of macro-logistics systems, i.e. the so-called "the international logistics right" or "the right of international logistics systems". International practice is on the way of the development of legal regulation of individual logistics operations, but not of the logistics in general.

Many of the logistics spheres are not regulated by international law even on that level. Moreover, a large number of logistics operations is not regulated, even at the level of national law. Management of the goods distribution process, from the legal point of view, is one of the most "non-image" fields of economic activities.

There are adopted many international agreements in the field of international goods distribution (especially in the field of transport and customs regulation). However, majority of these agreements have hardly representative character (many of them, which are adopted quite a long time ago, still await either entry into force, because they do not have the required number of ratifications).

International trade interests, the process of international division of labor, and the internationalization of economic life have created a special tool of legal regulation - "Lex mercatoria" (international commercial law).

KEY WORDS : INTERNATIONAL LAW; LOGISTICS; LEGAL FLOW; TRADE TERMS.

1. Introduction

The progress of the logistics in all spheres of economic activities significantly exceeds its regulation by the state and state institutions. Functioning of the separate areas and spheres of logistical activities is subject to laws and subordinate legislation, whereas there is need for the methodologies of centrally unified regulation of the logistics chains as the set of subjects, objects and logistical activities for moving the material, information, financial and other flows from the initial point to the receiving point.

Insufficient readiness of the methods of legal regulation of the logistics processes in the supply chains, including the relevant legal framework, has a direct and indirect effect on: the activities of the logistics subjects, creation and application of documentation flows, execution of logistical operations and functions. In this context, there is required a scientific explanation of the processes of distribution and ownership exchange for the economic resources inside the logistical systems, as well as of the delegation of rights, duties and responsibilities of logistics subjects at all levels of logistics.

Guidelines for regulation of the interactions of the supply chain participants inside the logistics flows allow implementing more effectively the logistical management of material and followed flows inside and outside enterprise. It would be advisable to address the main logistics processes with a view to application to them the methods of legal regulation.

At present, such scientific areas began to take off as, on the one hand, legal economics, which combines an economic approach to legal aspects and, on the other hand, considering the economic and logistics theories from a juridical standpoint.

2. Preconditions and means for resolving the problem

The research data are based on the analysis of "boundary" zones of logistics as an area of the economics and other sciences – information technology, commerce, and employment and labor law. A logistical approach to the management of economic processes is based on the operation of information standards, business and other statutory acts in the field of logistics.

In the process of the development of logistics as a science, there have been studied the issues of the existence and functioning of

logistics flows. Along with the generally accepted – material, financial and information – flows, they also started to consider the supporting, accompanying flows, including the documentation and legal ones.

Some issues of the application of the statutory acts in logistics are considered in the majority of manuals on logistics. However, they dwell on administrative or legal accompanying of the logistics procedures at meso- and macro-levels, for example, conclusion of the supply agreements, transport operations, considering the basis terms of delivery, legal status of the trade agents within the distribution channels, transfer of ownership during the process of the delivery of goods and so on. Thus and so, it is time to pose the question of the elimination of gap in the theoretical-methodological tooling of investigation of logistics chains with a view to legal regulation. Based on the analysis of statistical data of the contract-based activities of enterprises, as well as on the information on the pre-action and legal controversies on the issues of non-fulfillment of obligations flowed from these agreements, taking the account for the legal constituent in the modeling of meso- and macro-logistical systems contributes to the decrease in the number of violation of administrative and contractual obligations of enterprise to their employees, contractors and state, and consequently, the reduction of the amount of losses incurred in connection with payments of penalties. For the clearer substantiation of the character and role of legal regulation of the activities of enterprises, there is introduced the concept of legal flow as a kind of logistics flows. It is defined as a combination of legal standards and in-house acts of the organizations, and legal actions based on them, which regulate the rights and duties of the subjects of logistics in the management and control of logistical operations and functions for reaching the efficient solution of goals and objectives at micro-, meso- and macro-logistics levels. There is a need for the inclusion of legal flow in synchronizing band of the logistics flows that allows, on the one hand, considering the legal aspects of the activities of enterprises as an element of the logistical system, and on the other hand, taking account for the legal constituent of the logistical management during the enterprise costing, including determination of transaction expenses [1].

The role of the legal flow within the logistical system consists in: legal regulation of participants (subjects) of the logistical systems; setting the norms and rules for execution of logistical operations and functions; accompanying other types of logistics flows; creating the regulatory environment acting within a single logistical system or going beyond its limits.

The structure of the legal flow is represented by the following elements: documents; rights and functional duties of the logistics subjects; logistical operations and functions as a basis for arising the rights and responsibilities of participants of logistical processes. These elements act at three levels of logistics: micro-, meso- and macro-logistics levels. Based on the classification and structure of flow, there are identified the following main functions of it: ensuring the principles of economic efficiency, legal environment and optimizing logistical processes; creating the integrated legal environment, in which the subjects and objects of logistical system are acting; creating the effective basis for authorizing the subjects of logistical systems with rights, duties and responsibilities during the process of logistical activities; ensuring the integrated system of interaction of material, financial, information and other flows; ensuring the coordinated operation of all logistical sections during execution of logistical operations and functions; efficient distribution of rights of logistics subjects during vertical and horizontal integrations; inclusion of legal foundation in vertical hierarchy of the supply chain management.

Legal regulation of logistical processes is based on the application of regulatory and legal framework as one of the instruments of centralized and objective impact on the activities of economic entities. An important thing is the correct definition of legal regulation principles of logistics management of the production-commercial activities of company at micro-, meso- and macro-levels of logistics that allows assessing the necessity of effective application of the regulatory legal acts in organization of logistical processes.

The methodology of legal regulation of the analysis of company's economic activities is based on the elements of general economic analysis with consideration of logistics principles. At each logistics level there are used the same particular principles. For example, the micro-level is represented by company's internal structure and combination of main stages of its activities. Taking into consideration of the matter of logistic analysis of legal flows, as well as their character, structure and link with other logistics flows as an algorithm of analysis, the outline of analysis of management of micrologistical system is applicable.

The meso-level is represented by commercial activities of enterprise, its relationship with the state, municipal organizations, legal and natural persons during the execution of all logistical business-processes.

The macro-level is represented by the external environment of enterprise, which is conditioned by economic, social, political and other factors of regional, federal and international importance. The highest significance is given to the analysis of the activities of enterprise at a meso-level, which consists of the following stages:

1. Defining purpose and performance evaluation criterion of legal ensuring with account for peculiarities of logistical system of a particular enterprise, tied with logistics subjects at meso-level, and strategic and tactical plans of the company;
2. Decomposition of purpose, its breakdown into sub-purposes according to strategic, tactical and operational plans, functional area of logistical activities and subject matter of existing commercial relations of the company.
3. Evaluation of objectives and factors with a view to time, resource and financial costs, conditions of environment and interdependence of objectives.
4. The analysis of the structure of mutual relations – studies of subject and object matters, and logistics tooling of commercial activities of enterprise with account for legal constituent.
5. Identification of problems in functional areas of logistical activities – studies of drawbacks in logistics chains of different functional areas of logistics, in commercial and contracting documentation, violations of requirements of legislative and regulatory acts. Identification of problems includes finding and exact defining of problem, analysis of its statistical and dynamic state, defining the ties with other problems.

6. Decomposition and analysis of sources of problem – determining the factors influencing on origin, development and transformation of problem, its breakdown with respect to object and functional signs.
7. Determining the means for problem solving:
 - the availability of possibilities of problem solving;
 - defining and evaluating the options of solving;
 - selection of option (options) for problem solving.
8. Forming of the program and control include elaborating the program, projects and development plans of legal logistical system, their regular succession and stage-by-stage approach, determination of executives, their rights and duties, and control of program performance.

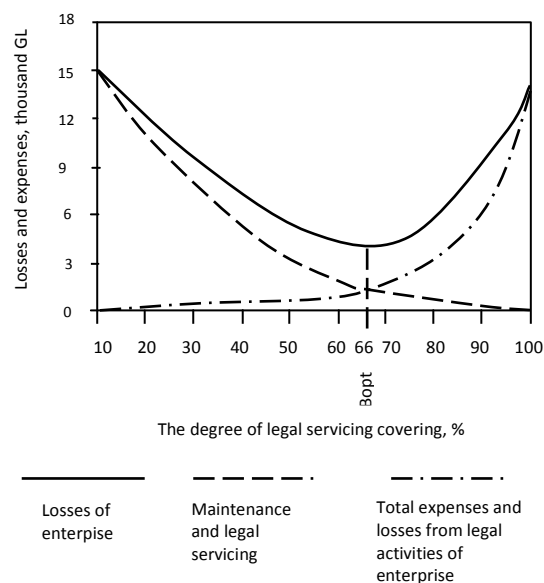


Fig. 1. Nature of the change of total cost structure of the company for legal support of the production-commercial activities.

The increase of the effect occurs owing to decreasing, but not excluding expenses, including for legal support of the activities of enterprise. With effective legal servicing, the economic losses of company go down, but there are appeared the service costs. Thus and so, there is a certain optimum relationship of the volume of legal ensuring and total costs incurred by enterprise (Fig. 1).

Performance evaluation of legal support of the production-commercial processes in logistical systems has a complex nature of dependency on the degree of covering the costs for this support. The legal ensuring of the company's activities is carried by legal service, and it can be evaluated by using a number of legal indicators. Thus and so, it is possible to set a question about performance evaluation of the activities of legal service in the enterprise with a view to relation of costs for its maintenance and savings, which is ensured by it for the enterprise.

The formation process of the model of logistical activities of enterprise comprises conducting the following steps:

- 1) Analysis of exiting legal regulation of logistical processes within supply chains;
- 2) Planning of statutory-legal support of company's logistical system.
- 3) Developing the project of public administration of processes company's logistical system, including legal support.
- 4) Approbation and application of proposed logistical system.
- 5) The process of using and evaluating of logistical system with account for its state regulation.

- 6) Approbation and application of proposed logistical system.
- 7) The process of using and evaluating of logistical system with account for its state regulation.

3. Conclusion

The model of logistical activities of company with account for legal elements fosters identification of economic, legal, production and other factors influencing on passing all stages from the initial point to the point of destination by material and followed flows, with account for logistics subjects, objects and activities at all levels. Taking account for the combination of these factors should allow lowering the negative impact of these factors and increasing efficiency of company's logistical activities due to reducing costs, decreasing the number and volume of penal sanctions and losses caused by legal infringements in the activities of logistical systems at micro, meso and macro levels.

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A ROLE OF INTERNATIONAL TOURISM IN REGIONAL DEVELOPMENT OF GEORGIA

РОЛЬ МЕЖДУНАРОДНОГО ТУРИЗМА В РАЗВИТИИ РЕГИОНОВ ГРУЗИИ

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Abstract: *The role of international tourism in economic development of Georgia and its regions is very large, which involves such significant tourism objects as customer-travel agents-tour operators- tourist transport services providers. With account for interests and demands of customers, the key things here are correct planning and operative management of such complex material-information processes, as tour formation and realization. International Tourism is a highly effective basis for foreign currency inflows to the country. An advantageous form of tourism export is capable of solving a number of social and economic problems for those countries, which actively develop tourism sector*

KEY WORDS : *International tourism; transport services.*

1. Introduction

Tourism came into everyday life of hundreds of millions of people, and became an integral part of modern life. Due to the rapid pace of development of tourism it is often called a phenomenon of 20th and 21st centuries. In 2012, about 1 billion of international tourist arrivals were recorded by World Tourism Organization (WTO) that exceeds a similar indicator for 1950 by 40 times. International tourism in many countries became one of the leading sectors of the economy and the reliable source of incomes to the budget. International Tourism is a highly effective basis for foreign currency inflows to the country. An advantageous form of tourism export is capable of solving a number of social and economic problems for those countries, which actively develop tourism sector. The share of tourist servicing in total revenues of international trade is about 7%, and they remain behind just such sectors as oil products and motor-cars export. According to international data the revenues of international tourism are expected to be doubled during the period of 2008-2014.

About 5 thousand companies are registered on the World Travel Market in London. Practice shows, that the strengthening of the role of international tourism concerns with Georgia as well, where the development of international tourism gathers the rapid paces.

2. Preconditions and means for resolving the problem

of 20th century, the international tourism acquired universal nature that resulted in formation of the world tourism market, in which all the countries are involved. On average 65% of international tourist travels is made to the European countries, 20% - to USA, 15% - to other regions. According to the World Tourism Organization, 21st century will be a tourism century. The influence of international tourism on the development of the country and its regions as well, (Fig At the beginning.1)

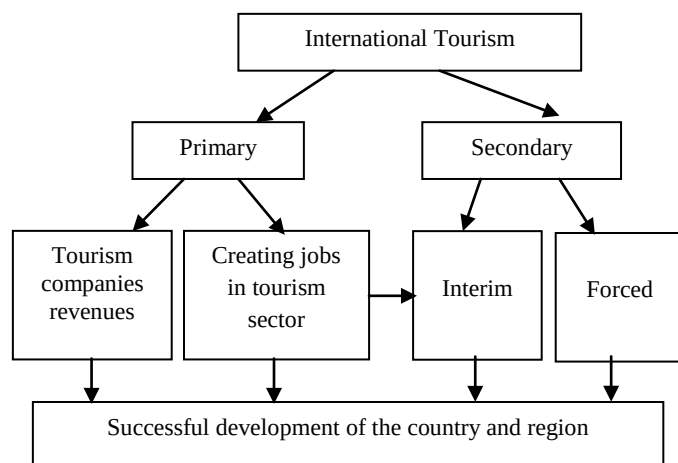


Fig. 1. The influence of international tourism on the development of the country and its regions

By tourist service cost per capita the countries are subdivided into three groups: over \$500, \$150-500 and \$150 and lower (Table 1).

Average annual tourism costs per capita (\$)

Table 1

Over 500	150-500	150 and lower
Kuwait	Great Britain	Korea
Austria	Finland	Iceland
Norway	Canada	Hungary
Switzerland	Australia	Ukraine
Singapore	Italy	Argentina
Belgium-Luxemburg	France	Rusia
Denmark	Japan	Romania
Netherlands	Czech Republik	Thailand
Israel	USA	Turkey
Ireland	Poland	China
Germany	Georgia	

The complex factors influencing the development of tourism industry subdivided into two groups as follows:

1. Political, economic, socio-demographic and cultural

2. Factors influencing tourism development and they are often used by tourism organizations in their activities (construction of new tourism infrastructure, personnel development, technical modernization of tourism facilities, and rational use of material resources, objects and tours).

The paces of development of international tourism over the recent 60 years, its prospective development by 2020, as well as revenues generated from international tourism (USD) are shown in Table 2.

The paces of development of international tourism

Table 2

Years	Number of Tourists	Foreign currency inflows (USD)
1950	25 mil	2100000000
1960	69 mil	6900000000
1970	159 mil	17900000000
1980	284 mil	97 bin
1990	415 mil	230 bin
2000	698 mil	476 bin
2010	1 bil	1100 bin
2020 (forecast)	1,6 bil	2 trillion

With its economic-geographical location, Georgia is one of the most distinguished countries. Despite smallness of territory, there are combined both plain and mountainous areas, sea and the land. Georgian resorts and tourist destinations long ago had gained international acceptance.

Due to beautiful nature, famous Caucasus Mountains and unique view of the Black Sea Coast, the number of people wishing to receive treatment in Georgian seaside and mountain resorts grows rapidly.

Millions of people have relaxed in Georgia for many years, and over 500 resort-tourism establishments rendered services to them. It should be noted that simultaneously over 120 thousand people were able to receive treatment in our sanatoriums, boarding houses, rest homes, holiday camps and other recreation-purpose establishments. At present, tourism industry in Georgia is redeveloping, so tourism is one again one of the main source of economic revival of Georgia.

Georgia is known with resorts of universal important in both plain and mountainous areas. There are more than 100 well-known resorts in Georgia, some of which have international importance, such as: Bakuriani, Borjomi, Kobuleti, Bichvinta, Akhali Atoni, Mtsvane Kontskhi, etc. Georgia is especially distinguished by existence of balneotherapeutic health resorts such as Tskhaltubo, Abastumani, Nabeglavi, Utsera, Sairme, etc.

According to recent data of National Statistics Office of Georgia, the number of international travelers and volume of revenues generated from them have

increased threefold in recent four years. The main goal of the development of international tourism in Georgia is to create a product and promote it in Asian and European countries. In recent five years, the number of international tourists inflowing into the country and revenues from international tourism has been increased considerably (Table 3).

The number of international tourists in Georgia and revenues from international tourism

Table 3

Years	The number of international tourists	The volume foreign currency inflows (USD)
2009	1 500 049	470 285
2010	2 031 717	659 245
2011	2 822 363	938 297
2012	4 389 256	1 800 550
2013	5 000 000	2 500 000

Statistical analysis of international tourists in Georgia in 2013 has shown that the total number of arrivals is as follows:

a) Tourists - 1 935 767 arrivals (39% of total number, growth in comparison with similar indicator of previous year is 17,3%);

b) Transit - 1 094 159 arrivals (22% of total number, growth in comparison with similar indicator of previous year is 55,6%);

c) Other - 1 943 706 arrivals (39% of total number, growth in comparison with similar indicator of previous year is 14,2%).

International tourism industry grows considerably in the regions of Georgia.

In 2013, especially large number of tourist flows was registered at Bagrati Temple. The total number of tourists was 120 795, including foreign travelers – 15 066. With the number of visitors, the first quintuple consists of the following countries: Ukraine – 3120; Israel – 2803; Poland – 1914; Russia – 1233; Japan – 1135.

The marked interest in the Imereti region is observed in the recent period in tourist routes as follows:

1. Kutaisi – Chiatura – Sachkhere;
2. Kutaisi – Tkibuli – Shaori;
3. Kutaisi – Tskhaltubo – Tsageri;
4. Kutaisi – Bagdati – Sairme;
5. Kutaisi – Khoni – Gordi;
6. Kutaisi – Kharagauli – Nunisi;
7. Kutaisi – Gelati;
8. Kutaisi – Sataplia Nature Reserve;
9. Kutaisi – Kumistavi Cave;
10. Bagrati Temple.

At present, under support of Shota Rustaveli National Science Foundation, there are carried out studies of mentioned routes and destinations, on the basis of which there will be created the transport-visitor passports, which will be widely available and they will undoubtedly

increase the capacities of both individual and tourist groups to access these tourist destinations.

The Georgian Tourism Association (GTA) was founded in 2006. It represents the private tourism companies, hotels and guest houses organization, which is aimed to develop such significant aspects in tourism sector, as follows:

- Successful cooperation between the tourism companies in Georgia;
- Cooperation between private and public sector;
- Quality management in tourism industry;
- Accessibility of tourism information;
- Sustainable tourism development in Georgia.

The major business lines of the association are incoming, domestic and outgoing tourism, conduction of educational seminars and trainings, training of professional guides. GTA assists to join to international hotel booking system, dissemination of tourism information, and lobbying, as well offers web-site consultancy, preparing video-materials, marketing support, seminars and trainings.

The Georgian Tourism Association carries out different projects, which assist to tourism development in Georgia. Within these projects, GTA cooperates closely

with Department of Tourism and Resorts of Georgia, Agency of Protected Areas, International Donor Organizations (EPF: USAID/SME; GTZ; SDC), local partners (Elkana, GeoLand) and its member companies.

3. Conclusion

It is supposed that for the next fifty years, tourism industry will be one of the real sources for generation of new jobs in number regions of the world, including Georgia. The humanitarian importance of international tourism is increasing that is one of the means for improving the living conditions for population.

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SIMULATION ON SINGLE CYLINDER DIESEL ENGINE AND EFFECT OF COMPRESSION RATIO AND EGR ON ENGINE PERFORMANCE AND EMISSION

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Abstract: In this research, the one dimensional (1D) CFD modelling of four-stroke direct injection diesel engine is developed by AVL Boost software. The performance of a diesel engine increases with increase in compression ratio. Variable compression technologies in IC engines are used to increase fuel efficiency under variable loads. Exhaust gas recirculation is a common way to control in-cylinder NOx production and is used in most modern high speed direct injection diesel engines. However the effect of EGR on performance, combustion and emissions production at different compression ratios are difficult to depict. In the present work an attempt was made to study the effects of exhaust gas recirculation on performance, combustion and emissions of a variable compression Diesel Engine. The test was conducted at different compression ratios with different loads and for different EGR rates.

Keywords: SIMULATION, DIESEL ENGINE, MODEL, AVL BOOST, PERFORMANCE, SINGLE CYLINDER, EGR, VCR

1. Introduction

Diesel engines are characterized by low fuel consumption and very low CO emissions. However, the NOx emissions from diesel engines still remain high. Hence, in order to meet the environmental regulations, it is highly desirable to reduce the amount of NOx in the exhaust gas. The Diesel engines by small power have a wide range of application for the mechanization of the most activities in industry and agriculture. Owing to their low fuel consumption, they have become increasingly attractive for smaller lorries and passenger cars also. But higher NOx emissions from diesel engine remain a major problem in the pollution aspect. In order to reduce emission levels, some external engine features can be applied, such as EGR or after-treatment systems. EGR systems have been used to reduce emissions of nitrogen oxides (NOx) from diesel engines.

Cylinder charge dilution with exhaust gas can be classified into internal EGR and external EGR. With external EGR, exhaust gas is taken from the exhaust port and supplied into the inlet port. Internal EGR is achieved by increasing NVO (negative valve overlap) during exhaust stroke, which requires an improved cam that can rapidly switch cam profiles to achieve any variable valve timing, otherwise it's impossible to independently and effectively control EGR ratio. This greatly limits the application of internal EGR. As a result, external EGR has become widely used on today's automobile engines. External EGR has a relatively low cost. It only needs to use dedicated EGR control valve, which can control EGR rate effectively under all work conditions of engine [1–4].

When EGR is applied, engine intake consists of fresh air and recycled exhaust gas. EGR (%) usually represents the percentage of the recirculated exhaust gas. The percentage of exhaust gas recirculation is defined as the percentage of recirculated exhaust in total intake mixture [5]. Where m_i is the mass of total intake mixture and m_{EGR} is the mass of the EGR.

$$E(\%) = \frac{m_{EGR}}{m_i} \cdot 100 \quad (1)$$

The principal source of NO formation is the oxidation of the nitrogen present in atmospheric air. The nitric oxide formation chain reactions are initiated by atomic oxygen, which forms from the dissociation of oxygen molecules at the high temperatures reached during the combustion process.

Shahadat.et.al [6] studied the combined effect of EGR and inlet air preheating on engine performance in diesel engine. They found that at medium load conditions, oxides of nitrogen (NOx), carbon monoxide (CO), engine noise, and brake specific fuel consumption decreased when inlet air preheating and EGR were applied together as compared to those during normal operations of the engine.

Arjun Krishnan et.al [7] studied the Prediction of NOx reduction with EGR using the flame temperature correlation technique. They developed a procedure to calculate the reduction in NOx levels due to EGR, given only engine base-data. This approach utilised the flame temperature correlation technique to obtain NOx predictions with sufficient accuracy – 6.5% at part loads.

Avinash kumar.et.al [8] studied the effect of EGR on exhaust gas temperature and exhaust opacity in compression ignition engines. They found that that the exhaust gas temperatures reduce drastically by employing EGR. Thermal efficiency and brake specific fuel consumption are not affected significantly by EGR. However particulate matter emission in the exhaust increases, as evident from smoke opacity observations.

AVL Boost is based on 1D gas dynamics which account for fluid flows and heat transfer. Each component in an AVL Boost model is discretized or separated in many smaller components. These components have very small volumes and the fluid's scalar properties in these volumes are assumed to be constant. The scalar properties of a fluid include pressure, temperature, density and internal energy. Each volume also have vector properties that can be transferred across it's boundaries. This properties include mass flux and fluid velocity.

2. Data Needed for Building an Engine Model

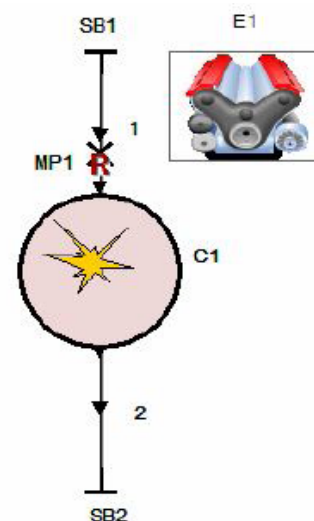


Fig. 1 Model of Single Cylinder Diesel Engine: SB-system boundaries, MP-measuring points, C-cylinder.

AVL Boost is a software tool that consists in a pre-processing program, used for initial data entry and technical characteristics of the engine to be designed as model. After forming the engine assembly with annexes systems, mathematical equations and algorithms of the model with the graphical user interface (GUI) will analyze and calculate the processes that are required during simulation [9, 10]. The model for the engine designed in AVL BOOST application is shown in figure 1.

A list of information that is needed to create a model in AVL BOOST is included in library. The main features of the diesel engine that have been used as initial data to define the cylinder parameters are presented in table 1. Cylinder (C1) of the model in AVL Boost is connected with element Engine (E1), and it defines the type of engine used, operating speeds on it, moments of inertia and break mean effective pressure (BMEP). Combustion method is Mixing Controlled Combustion model that predicts the rate of heat released (ROHR) and NOx emissions on the quantity of fuel in the cylinder and the turbulent kinetic energy introduced by the injection of fuel [8].

Table 1: Specification of the engine.

Engine Parameters	Value	Unit
Bore	76	[mm]
Stroke	65	[mm]
Displacement	295	[cc]
Power	4	[kw]
Speed	3000	[rpm]
Compression ratio	17:1	-
Valve lift	8,5	[mm]
Piston pin offset	10	[mm]

The pre-processing step of AVL Boost enable the user to model a 1-Dimensional engine test bench setup using the predefined elements provided in the software toolbox. The various elements are joined by the desired connectors to establish the complete engine model using pipelines

3. Result and Discussion

After definition of the engine parameters has been run a series of simulations, and then plot the results in Impress Chart were was analyzed.

The running simulation result is all of the engine performance data with the constant engine speed (rpm) by varying the compression ratios. This model was running at speed 2000 rpm.

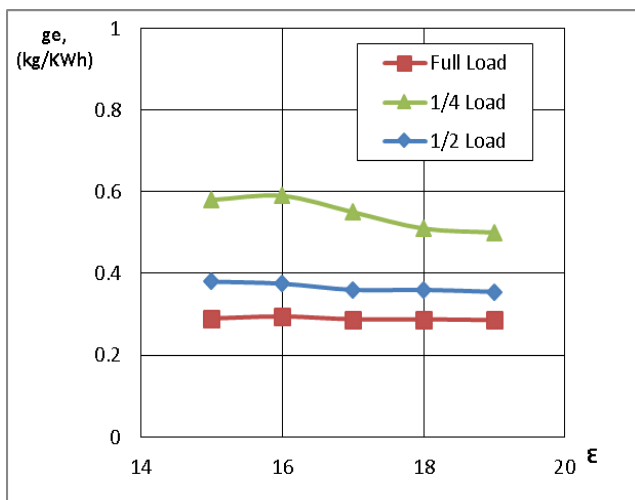


Fig. 2 Influence of Compression ratio on Specific fuel consumption at different loads

In Fig. 2 the specific fuel consumption at various loads and compression ratios are presented. With increase in compression ratio the specific fuel consumption decreases irrespective of the load except at compression ratio 16 due to less ignition delay and more combustion duration.

Figure 3 shows the effect NOx at different loads and compression ratios. With increase in load the NOx value also increased due to more heat release and caused for dissociation of gases.

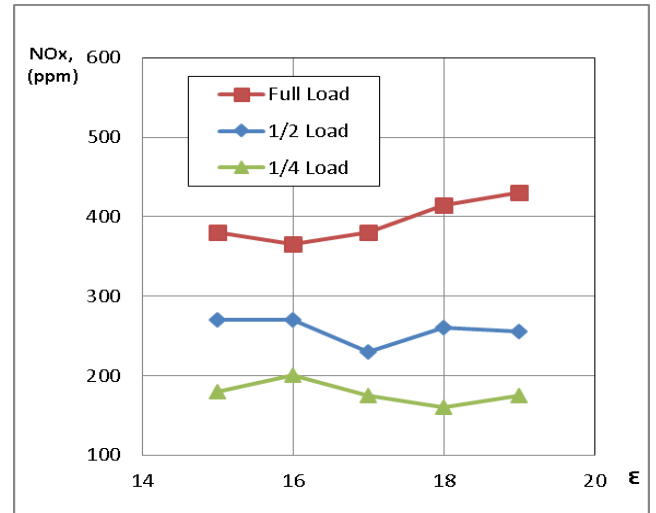


Fig. 3 Influence of Compression ratio on NOx emission at different loads

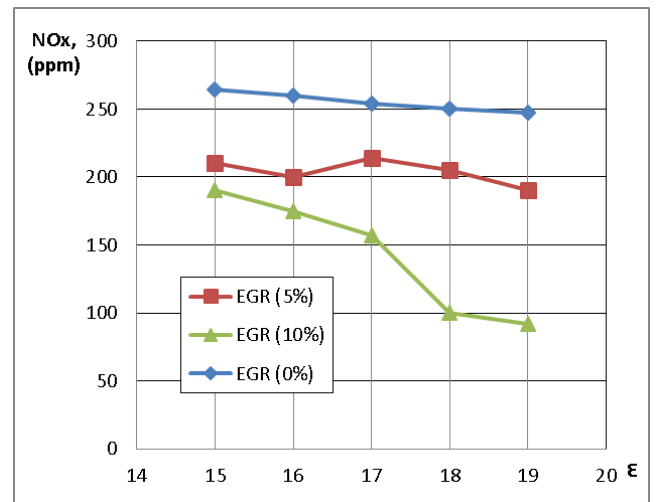


Fig. 4 Influence of EGR on NOx emission at different compression ratio

Figure 4 shows the effect of exhaust gas recirculation (EGR) on NOx emissions. The model simulation result shows that with increase in percentage of EGR the NOx emissions are decreased. These are mainly due to the lower combustion chamber temperatures caused by EGR and because the exhaust gas replaces some of the excess oxygen in the pre-combustion mixture.

3. Conclusion

This paper present the results of the engine cycle simulation of a single cylinder, direct injection diesel engine with different compression ratios, percentages of EGR and loads to estimate

performance, combustion and emission characteristics of the engine, using AVL Boost software. It was found that with increase in compression ratio the specific fuel consumption decreases. The results obtained indicated that with increase in % EGR the NO_x emissions was gradually decreases at different compression ratios due to less flame temperatures and low oxygen content in the combustion chamber. The high degree of recirculation is suitable for higher compression ratio because at compression ratio 19 and 10% EGR the percentage reduction of NO_x was 36%.

Acknowledgment

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STUDYING OF NIOSE, ACTING OF THE DRIVERS OF THE ROAD-BUILDING MACHINERY AND AUTOMOBILES

ИЗСЛЕДВАНЕ НА ШУМА, ДЕЙСТВАЩ НА ВОДАЧИТЕ НА ПЪТНО-СТРОИТЕЛНИ МАШИНИ И АВТОМОБИЛИ

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Abstract: Drivers of road-building machinery are subjected to noise pollution variable nature of the noise. The purpose of this study is to show the values of noise experienced by drivers of different types of road-building machinery (trucks, excavators, tractors).

Keywords: NOISE, ROAD-BUILDING MACHINERY, AFFECT DRIVERS.

1. Introduction

The noise on its health matter takes one of the first places among the adversely existing physical work environment factors. The increasing production of noise which is often accompanied by deterioration of the noisy parameters, leads to increase in both occupational groups exposed to excessive noise levels and occupational risk of noise injury and illness.

The purpose of this study is to show the values of noise experienced by drivers of different types of road-building machinery (trucks, excavators, tractors). The machines are randomly selected, and only one condition of this research is more common species in the region of Smolyan.

Therefore the noise standards apply to protect the health of the drivers of the machines, the values obtained for the noise in this article are considered as noise production. About the measured values of noise are measured at Ordinance № 6 of 15.08.2005 on the minimum requirements for ensuring the health and safety of workers at risks related to exposure to noise [2]. Methodology of measurement complies with BS ISO 1999:2004 Acoustics. Determining the impact of noise at work and assessing hearing damage reasons, but noise [3].

2. Theoretical formulation and methodology of the study

In accordance with Ordinance № 6 of 15.08.2005, the limits of the exposure and the exposure action are determined with based daily rates of exposure to noise and peak sound pressure as follows:

1. Exposure limit values: Lex,8h = 87 dB (A) and p_{peak} = 200 Pa, respectively 140 dB (C);

2. Upper values exposure action: Lex,8h = 85 dB (A) and p_{peak} = 140 Pa, corresponding to 137 dB (C);

3. Lower values exposure limits for action: Lex, 8h = 80 dB (A) and p_{peak} = 112 Pa, respectively 135 dB (C).

Peak sound pressure level represents ten times the logarithm of the ratio of the square of the peak sound pressure to the square of the reference sound pressure, where the peak sound pressure is the maximum absolute value of the instantaneous sound pressure for a specified period of time at a standard frequency weighting or standard width of the measuring tape.

Sound pressure is a variable which changes the frequency of sound waves. Sound pressure is the difference between the pressure at the time and static. Sound pressure characterizes the intensity of the sound wave at a given point of space and represents the variable component of the pressure, arising as a result of tremulous

movements of the sound source and accumulation on pressure. Sound pressure is denoted by P and its benchmark (atmospheric pressure) - P_c.

In BS ISO 1999:2004 is a methodology for determining the "level of exposure to noise, standardized for nominal working day of eight hours - (Lex,8h)", which according to Ordinance № 6 of 15.08.2005 on a 'daily level noise exposure (Lex,8h) '.

The methods include direct measurement of "daily level of exposure to noise (Lex,8h)", standardized for nominal working day of eight hours or calculation of the "level of daily noise exposure (Lex,8h)" for the time of impact. Standard requires measurements to be made with equipment conforming to IEC 804 Class 2 or better. The device must be directly measured equivalent continuous A-weighted sound pressure level L_{Aeq, T}. This parameter is calculated automatically by the device according to the following formula:

$$L_{Aeq,T} = 10 \lg \left[\frac{1}{t_2 - t_1} \int_{t_1}^{t_2} \frac{p_A^2(t)}{p_o^2} dt \right], \text{ dB}, \quad 1)$$

where:

t₂-t₁ is the period T for which the averaging is made, starting from t₁, and ends in t₂;

p_A - A-weighted sound pressure in Pa;

p_o - reference sound pressure - (p_o = 20 μPa in accordance with ISO 1683);

If the noise is unchanging level for the specified period, the L_{Aeq, T} [dB] is equal to L_{pA}, which is level A-weighted sound pressure is given by the formula:

$$L_{pA} = 10 \lg \left(\frac{p_A}{p_o} \right)^2, \text{ dB}. \quad 2)$$

Calculation of "Level of noise impacts, standardized for nominal working day of eight hours - (Lex,8h)" is carried out according to the time impacted action. Calculation is made using the following formula:

$$L_{EX,8h} = L_{Aeq,T} + 10 \lg \left(\frac{T_e}{T_o} \right), \text{ dB}, \quad 3)$$

where:

T_e is the effective length of the working day (time of impact) in h;

T_o - length comparison (8 h).

By that method, it is clear that for longer time impact, the driver will be exposed to higher exposure and at eight-hour working day, the exposure will be equal to the figure of the device equivalent continuous A-weighted sound level pressure $L_{Aeq, T}$.

When the duration of the impact is less than 8 hours, it is necessary to do calculation of "Level of noise impacts, standardized for nominal working day of eight hours - (Lex,8h)".

The methodology allows short high equivalent continuous A-weighted sound pressure level $L_{Aeq, T}$ over the limits of exposure, since calculating the impact of noise, standardized earlier for nominal working day of eight hours may be reduced values below-the-limit levels.

Methodology in BS ISO 1999:2004 requires the measurement noise, the microphone of the device to be placed at a distance 10 cm from the ear of the driver. Under this condition, the reported noise will be a total noise of the engine, transmission, movement of vehicles on the aerial resistance, etc.

The device used is an integrated meter volume 2238 Mediator, which was calibrated by an accredited laboratory for a period of calibration until 2016. Before each measurement, the device is controlled by an acoustic calibrator, which is also calibrated by an accredited laboratory valid until 2016.

The device automatically detects the peak sound pressure level and equivalent continuous A-weighted sound pressure level $L_{Aeq, T}$.

3. Analysis of the results of the manufacturing test

Therefore the objective of the study is to show the levels of noise to which drivers are exposed to machinery, without focusing on the duration of exposure, the research focussed reported and registered by the device "an equivalent continuous A-weighted sound level pressure" $L_{Aeq, T}$ and "peak sound pressure"

Conditions under which the measurements were performed are the same for the groups of machines – type and state of road surface, movement of the car (loaded or unloaded, in no time flat, horizontal gradients) and instantaneous technical condition of machines.

During the measurement, the nature of the noise was variable.

As the researches did not focus on determining of the exposure, which requires measurement of the noise level for the time of impact, ie for the entire period of operation of the machines, the data presented in this article apply only to the noise emitted by machines measurement time of 30 min.

Each machine has been tested several numbers, as the reported and registered by the device parameters are averaged.

The object of study in this article is the noise when working on these types of road-building machinery and vehicles:

- Excavator: excavator "JCB" - 4 pcs.; wheel excavator "ATLAS 1304" - 2 pcs.; front loader "ATLAS 52 D" - 2 pc.; mini excavator "Bobcat" - 2 pcs.;

- Tractors wheel - "Universal" 651 M - 2 pcs.; UMZ 6l - 2 pcs.; "TK-80" - 2 pcs.;

- Tractors chain - T 170-1 pc.; DT 75 – 1;

- Trucks: KAMAZ 5511-12 issue.; DAF cf 85-5 pc.; MAN TGS - 10 pcs.

In the process of manufacturing the experimental study are reported and the following parameters:

- Peak sound pressure (ppeak);

- Equivalent continuous A-weighted sound pressure level $L_{Aeq, T}$;

- Type and model of trucks;

- Type and model of road construction equipment.

Results obtained from the production study of changing noise of road-building machinery for the region of Smolyan region are subjected to statistical processing and are summarized in tabular and graphical relationships.

4. The value recorded noise

Table 1. Results of measurements

Types of machines	Source of noise	Equivalent continuous A-weighted sound pressure level $L_{Aeq, T}$, dB	Peak sound pressure (ppeak), dB
Excavators	Backhoe „JCB”	73	93
	Wheel excavator „ATLAS 1304	73	92
	Front loader „ATLAS 52 D	72	93
	Mini excavator "Bobcat"	85	104
Wheeled tractors	Universal 651 M	78	110
	UMZ 6JI	79	88
	TK-80	81	93
Crawler tractors	T 170	81	92
	DT 75	75	98
Trucks	KAMAZ 5511	78	91
	DAF cf 85	69	90
	MAN TGS	73	89

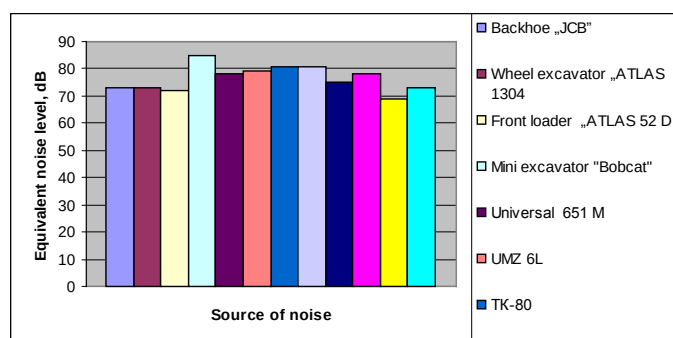


Fig. 1. Equivalent noise level for various types of road construction machinery

FIG. 1 shows the change of the equivalent noise level of the different road construction machinery in Smolyan. Added by the

histograms shows that the equivalent noise level is highest for tractors.

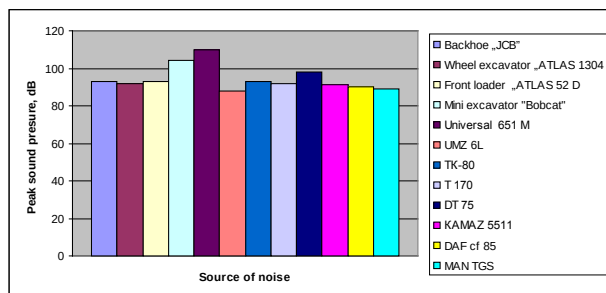


Fig. 2. Peak sound pressure for different types of road construction machinery

FIG. 2 shows the change in peak sound pressure various road construction machinery. From the displayed histogram shows that diesel engines of an older generation exhibited higher values of peak sound pressure.

5. Conclusion

1. It was made production experimental study of the noise effect on load drivers for various types of road-building machinery of Smolyan.

2. Incurred research and built graphical relationships indicate that drivers of tractors are subjected to higher noise levels than other drivers (Figure 1).

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RESEARCH ON DYNAMIC TURNING PROCESS OF TRACTORS TRAILERS AGGREGATE WITH MOUNTED DAMPERS AGAINST SUDDEN BENDING

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Abstract: This paper introduces cycle dynamic model of combined tractors - trailer with mounted damper against sudden bend in order to increase stability when turning round. The research results can make rational basis for selection of appropriate parameters of the damper.

Keywords: TURNAROUND, DYNAMIC MODEL, DAMPER.

1. Introduction

During rotation of the tractor - trailer aggregate (TTA), there is sometimes the case that the trailer pushes the tractor. Thrust of trailers has the effect of turning tractor in the same direction with the rotating direction, increasing the relative rotation between the tractor and the trailer, it means that TTA will be self- folding its body. This leads to an instability TTA's motion in rotation [1], particularly dangerous when turnaround times, which can lead to loss of control.

In order to limit the sudden bending of TTA, the authors have proposed mounting a hydraulic damper at joints (see Figure 1) to increase safety while rotating motion.

2. Research Methodology

Using the basic equations of solid dynamics, deflective motion features of elastic wheel, the methods of solving nonlinear equations.

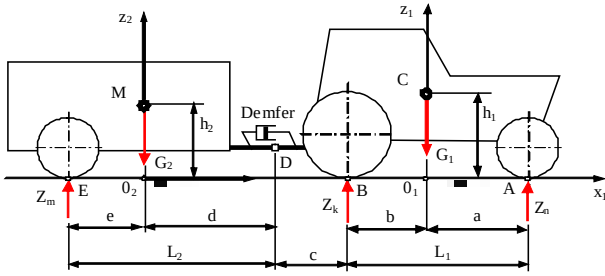


Figure 1. TTA tractor diagram in vertical plane

The model parameters are denoted as follows :

$\delta_1, \delta_2, \delta_3$ – deflection motion angle of the tractor's wheel and trailers;

V_{x1}, V_{x2} – axial velocity components of tractors and trailers;

β – rotation angle of the steering wheel of the tractor.

α_D – the angle between the velocity of the juncture of the vertical axis hooks and trailers;

δ_D – the angle between the velocity of mechanical joints and vertical axis of the tractor ;

θ – the angle between the longitudinal axis of the tractor and trailer's longitudinal axis ;

φ_1, φ_2 – rotation angles of traktor and trailer.

3. Research Results and Discussion

3.1. The revolving dynamic model TTA

To simplify in building revolving dynamic model without loss of generality of the problem, in this paper we accept some of the following assumptions:

- The flat line;
- Axial velocity of the tractor is not changed;

- The focus of tractors and trailers distributed along the plane of symmetry of it and ignore the weight distribution on the bridge.

With the above assumptions, giving the revolving dynamic model TTA with a trailer to flat model, a trace, shown in Figure 1 with the coordinate system: absolute coordinate system associated with road base $OX_0Y_0Z_0$; relative coordinate system attached to the tractor $CX_1Y_1Z_1$ and relative coordinates attached to the trailers $MX_2Y_2Z_2$, where points C and M are centers of garvity of traktor and trailer.

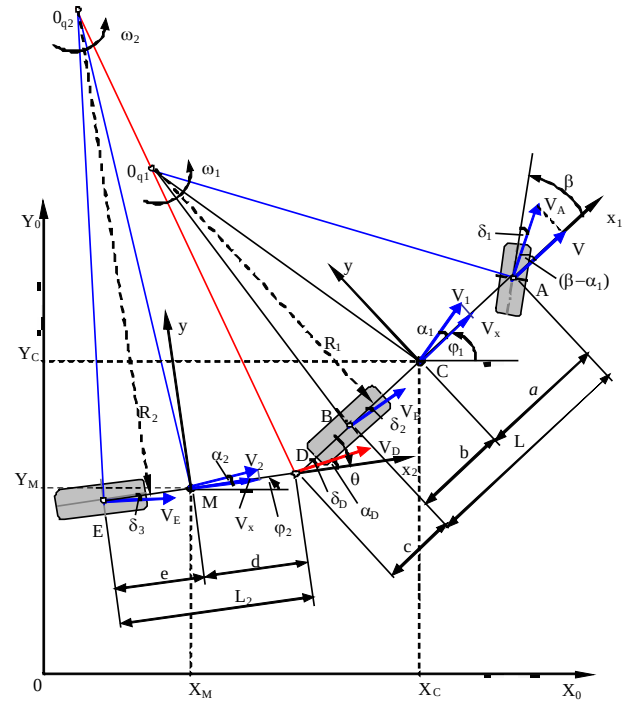


Figure 2. TTA kinetic diagram in the horizontal plane

The speed vector of the tractor consists 2 components: the axial component (in X_1) V_{cx} and side-oriented component (in Y_1) V_{cy} . V_{cy} is calculated by the formula:

$$V_{cy} = \frac{V_{x1}}{L_1} [btg(\beta - \delta_1) - atg\delta_2] \quad (1)$$

Conversion of velocity vector of tractor's center of gravity from relative coordinate system $CX_1Y_1Z_1$ to the absolute coordinate system $OX_0Y_0Z_0$, then differentiate in time we obtain the absolute acceleration of the center of gravity. Project absolute acceleration vector to the axes X_1 and Y_1 we obtain the components of absolute acceleration in the relative coordinates of the tractor $CX_1Y_1Z_1$.

$$\begin{cases} a_{Cx} = \dot{V}_{x1} - \frac{V_{x1}}{L_1} \omega_1 [b \cdot \operatorname{tg}(\beta - \delta_1) - a \cdot \operatorname{tg} \delta_2] \\ a_{Cy} = \frac{\dot{V}_{x1}}{L_1} [b \cdot \operatorname{tg}(\beta - \delta_1) - a \cdot \operatorname{tg} \delta_2] + \\ + \frac{V_{x1}}{L_1} \left\{ L_1 \omega_1 + \left[\frac{b}{\cos^2(\beta - \delta_1)} (\dot{\beta} - \dot{\delta}_1) - \frac{a}{\cos^2 \delta_2} \dot{\delta}_2 \right] \right\} \end{cases} \quad (2)$$

Rotation speed of the tractor is calculated using the formula:

$$\omega_1 = \frac{V_{x1}}{L_1} [\operatorname{tg}(\beta - \delta_1) + \operatorname{tg} \delta_2] \quad (3)$$

Then angular acceleration determined by the formula:

$$\begin{aligned} \dot{\omega}_1 = & \frac{\dot{V}_{x1}}{L_1} [\operatorname{tg}(\beta - \delta_1) + \operatorname{tg} \delta_2] + \\ & + \frac{V_{x1}}{L_1} \left\{ [1 + \operatorname{tg}^2(\beta - \delta_1)] (\dot{\beta} - \dot{\delta}_1) + (1 + \operatorname{tg}^2 \delta_2) \dot{\delta}_2 \right\} \end{aligned} \quad (4)$$

Similarly, we define the absolute velocity components for trailers.

Combines the formulas for the tractor and trailer, we obtain

$$\begin{cases} a_{Cx} = \dot{V}_{x1} - \frac{V_{x1}}{L} \omega_1 [b \cdot \operatorname{tg}(\beta - \delta_1) - a \cdot \operatorname{tg} \delta_2] \\ a_{Cy} = \frac{\dot{V}_{x1}}{L_1} [b \cdot \operatorname{tg}(\beta - \delta_1) - a \cdot \operatorname{tg} \delta_2] + \\ + \frac{V_{x1}}{L_1} \left\{ L \omega_1 + \left[\frac{b}{\cos^2(\beta - \delta_1)} (\dot{\beta} - \dot{\delta}_1) - \frac{a}{\cos^2 \delta_2} \dot{\delta}_2 \right] \right\} \\ \dot{\omega}_1 = \frac{\dot{V}_{x1}}{L} [\operatorname{tg}(\beta - \delta_1) + \operatorname{tg} \delta_2] + \\ + \frac{V_{x1}}{L_1} \left\{ [1 + \operatorname{tg}^2(\beta - \delta_1)] (\dot{\beta} - \dot{\delta}_1) + (1 + \operatorname{tg}^2 \delta_2) \dot{\delta}_2 \right\} \\ a_{Mx} = \dot{V}_{x2} - \frac{V_{x2}}{L_2} \omega_2 [e \cdot \operatorname{tg}(\theta - \delta_D) - d \cdot \operatorname{tg} \delta_3] \\ a_{My} = \frac{\dot{V}_{x2}}{L_2} [e \cdot \operatorname{tg}(\theta - \delta_D) - d \cdot \operatorname{tg} \delta_3] + \\ + \frac{V_{x2}}{L_2} \left\{ L_2 \omega_2 + \left[\frac{e}{\cos^2(\theta - \delta_D)} (\dot{\theta} - \dot{\delta}_D) - \frac{d}{\cos^2 \delta_3} \dot{\delta}_3 \right] \right\} \\ \dot{\omega}_2 = \frac{\dot{V}_{x2}}{L_2} [\operatorname{tg}(\theta - \delta_D) + \operatorname{tg} \delta_3] + \\ + \frac{V_{x2}}{L_2} \left\{ [1 + \operatorname{tg}^2(\theta - \delta_D)] (\dot{\theta} - \dot{\delta}_D) + (1 + \operatorname{tg}^2 \delta_3) \dot{\delta}_3 \right\} \end{cases} \quad (5)$$

The kinetic link

The binding kinetics at joints D, here is the hinge to the speed of the tractor and trailers are equal. On that basis, we define:

$$\begin{cases} V_{x2} = V_{x1} \cos \theta + \frac{V_{x1}}{L_1} [(L_1 + c) \operatorname{tg} \delta_2 + c \cdot \operatorname{tg}(\beta - \delta_1)] \sin \theta \\ \delta_D = \arctg \frac{(L_1 + c) \operatorname{tg} \delta_2 + c \cdot \operatorname{tg}(\beta - \delta_1)}{L_1} \end{cases} \quad (6)$$

Taking the derivative in time we will get the acceleration components:

$$\begin{aligned} \dot{V}_{x2} = & \dot{V}_{x1} \cdot \cos \theta - V_{x1} \cdot \dot{\theta} \sin \theta + \frac{\dot{V}_{x1}}{L_1} [(L_1 + c) \operatorname{tg} \delta_2 + c \cdot \operatorname{tg}(\beta - \delta_1)] \sin \theta + \\ & + \frac{V_{x1}}{L_1} \left\{ (L_1 + c)(1 + \operatorname{tg}^2 \delta_2) \dot{\delta}_2 + c [1 + \operatorname{tg}^2(\beta - \delta_1)] (\dot{\beta} - \dot{\delta}_1) \right\} \sin \theta + \\ & + \frac{V_{x1}}{L_1} [(L_1 + c) \operatorname{tg} \delta_2 + c \cdot \operatorname{tg}(\beta - \delta_1)] \dot{\theta} \cos \theta; \\ \dot{\delta}_D = & \frac{(L_1 + c)(1 + \operatorname{tg}^2 \delta_2) \dot{\delta}_2 + c(1 + \operatorname{tg}^2(\beta - \delta_1)) (\dot{\beta} - \dot{\delta}_1)}{L_1 \left\{ 1 + \left[\frac{(L_1 + c) \operatorname{tg} \delta_2 + c \cdot \operatorname{tg}(\beta - \delta_1)}{L_1} \right]^2 \right\}} \\ \dot{\theta} = & \omega_1 - \omega_2 = \frac{V_{x1}}{L_1} [\operatorname{tg}(\beta - \delta_1) + \operatorname{tg} \delta_2] - \\ & - \frac{V_{x2}}{L_2} [\operatorname{tg}(\theta - \delta_D) + \operatorname{tg} \delta_3] \end{aligned} \quad (7)$$

3.2. Dynamics of TTA

The forces on the tractor:

- Weight of tractor G_t .
- Forces in the coupling point: F_{Dx} ; F_{Dy} ; F_{Dz}
- The normal (radial) reaction forces of the wheels R_{z1} , R_{z2} ;
- The lateral reaction forces of the front and rear tires R_{y1} , R_{y2}
- Longitudinal reaction forces of tire R_{x1} , R_{x2} ;

External forces acting on the trailer are:

- Weight of the trailer G_M .
- Forces in the coupling point: F_{Dx2} ; F_{Dy2} ; F_{Dz2}
- The normal (radial) reaction force of the wheels R_{z3} ;
- The lateral reaction force R_{y3} ;
- Longitudinal reaction force of tire R_{x3} ;

The cross-reactions of the front and rear tires of tractor R_{y1} , R_{y2} và R_{y3} of trailer depend on sideslip angle δ_i of wheel, can be calculated: [2,3].

Where: k_1 , k_2 and k_3 is cornering stiffness of the wheel correspond with the front axle, rear axle and trailer. The coefficients k_i with nonlinear model itself depends on the deflection angle δ_i .

Damper:

Dampers have to achieve the requirements to create torque at small θ angle with the aim not obstruct the driving in normal conditions, but in the large θ angle to create great moment to prevent sudden bending of TTA. With the above requirements, author suggests the working principle diagram of a hydraulic damper as shown in Figure 3.

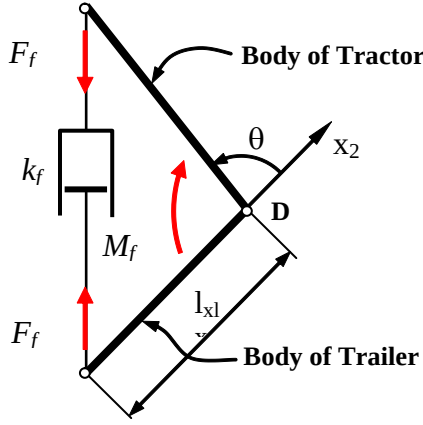


Figure 3. Principle diagram of the damper

Damping moment M_{demp} is placed at the point of the hook, generating by damping force F_{dmf} of the hydraulic cylinder and is calculated by the formula:

$$M_f = k_f * (l_{xl} \cdot \sin(\theta/2))^2 \cdot \dot{\theta}, \quad (9)$$

Where k_f e damping coefficient of the cylinder;

l_{xl} – hydraulic cylinder arm

The force component links hook points at F_{Dx2} , F_{Dy2} và F_{Dz2} , in the coordinate system of the trailer, which is determined by the formula:

$$\begin{aligned} F_{Dx2} &= -F_{Dx} \cdot \cos \theta + F_{Dy} \cdot \sin \theta \\ F_{Dy2} &= -F_{Dx} \cdot \sin \theta - F_{Dy} \cdot \cos \theta \\ F_{Dz2} &= -F_{Dz} \end{aligned} \quad (10)$$

Apply Dalamber principles for mechanical system, we obtain system of differential equations:

$$\begin{cases} \dot{V}_{x1} - \frac{V_{x1}}{L_1} \omega_1 [btg(\beta - \delta_1) - atg\delta_2] = \frac{F_{Dx} + F_k - f \cdot (R_{x1} \cos \beta + R_{x2}) - k_1 \delta_1 \sin \beta + G_{Tx}}{m_T} \\ \frac{\dot{V}_{x1}}{L_1} [btg(\beta - \delta_1) - atg\delta_2] + \frac{V_{x1}}{L_1} \left\{ L_1 \omega_1 + \left[\frac{b}{\cos^2(\beta - \delta_1)} (\dot{\beta} - \dot{\delta}_1) - \frac{a}{\cos^2 \delta_2} \dot{\delta}_2 \right] \right\} = \\ \frac{F_{Dy} - F_{f1} \sin \beta + k_1 \delta_1 \cos \beta + k_2 \delta_2 + G_{Ty}}{m_T} \\ \frac{\dot{V}_{x1}}{L_1} [tg(\beta - \delta_1) + tg\delta_2] + \frac{V_{x1}}{L_1} \left\{ [1 + tg^2(\beta - \delta_1)] (\dot{\beta} - \dot{\delta}_1) + (1 + tg^2 \delta_2) \dot{\delta}_2 \right\} = \\ \frac{-F_{f1} a \sin \beta + k_1 a \delta_1 \cos \beta - k_2 b \delta_2 - F_{Dy} * (b + c) + M_{demp}}{J_{yT}} \\ \dot{V}_{x2} - \frac{V_{x2}}{L_2} \omega_2 [etg(\theta - \delta_D) - dtg\delta_3] = \frac{-F_{Dx} * \cos \theta + F_{Dy} * \sin \theta - f * R_{x3} + G_{Mx}}{m_M} \\ \frac{\dot{V}_{x2}}{L_2} [etg(\theta - \delta_D) - dtg\delta_3] + \frac{V_{x2}}{L_2} \left\{ L_2 \omega_2 + \left[\frac{e}{\cos^2(\theta - \delta_D)} (\dot{\theta} - \dot{\delta}_D) - \frac{d}{\cos^2 \delta_3} \dot{\delta}_3 \right] \right\} = \\ \frac{-F_{Dx} * \sin \theta - F_{Dy} * \cos \theta + k_3 \delta_3 + G_{My}}{m_M} \\ \frac{\dot{V}_{x2}}{L_2} [tg(\theta - \delta_D) + tg\delta_3] + \frac{V_{x2}}{L_2} \left\{ [1 + tg^2(\theta - \delta_D)] (\dot{\theta} - \dot{\delta}_D) + (1 + tg^2 \delta_3) \dot{\delta}_3 \right\} = \\ \frac{(-F_{Dx} * \sin \theta - F_{Dy} * \cos \theta) \cdot d - e \cdot k_3 \delta_3 - M_{demp}}{J_{yM}} \end{cases}$$

Where J_{yT} , J_{yM} is the moment of inertia of the tractor and trailer in the vertical axis passing through the center of it.

$\dot{\delta}_3$, $\dot{\theta}$, $\dot{\delta}_D$, F_{Dx} , F_{Dy} . From the values of $\dot{V}_{x1}$, $\dot{V}_{x2}$, $\dot{\delta}_1$, $\dot{\delta}_2$, $\dot{\delta}_3$, $\dot{\theta}$, $\dot{\delta}_D$ can be solved to find the value of them in time by Runge-Kutta and it's modified method. Based on the basis, we can examine the effects of some structural factors and the mode used to target of TTA dynamics when turnaround.

3.3. Some illustrative results

With the purpose of examining the effects of damping to the basic parameters of the TTA when turnaround, we have developed the program to solve the problem and applied for KUBOTA 18L tractor pulling a shafting trailer, mounted with hydraulic demper absorbers against sudden frame folding with demper arm $l_{xl}=0,4$ m;

Rotaring angle of front axle is assumed to be sin function:

* When TTA entering the corner:

$$\begin{aligned} \beta &= \frac{\beta_{max}}{2} + \frac{1}{2} \beta_{max} * \sin \left(\pi \cdot \frac{t}{t_{max}} - \frac{\pi}{2} \right) \\ \text{for } 0 \leq t \leq t_{max} \\ \dot{\beta} &= \frac{\pi}{2 * t_{max}} * \beta_{max} * \cos \left(\pi \cdot \frac{t}{t_{max}} - \frac{\pi}{2} \right) \end{aligned}$$

$$\begin{aligned} \text{for } t_{max} < t < t_2 \quad \beta &= \beta_{max}; \\ \dot{\beta} &= 0 \end{aligned}$$

* When TTA exiting the corner:

$$\begin{aligned} \text{for } t_2 \leq t \leq t_{max} + t_2 \\ \beta &= \frac{\beta_{max}}{2} - \frac{1}{2} \beta_{max} * \sin \left(\pi \cdot \frac{t - t_2}{t_{max}} - \frac{\pi}{2} \right) \\ \dot{\beta} &= -\frac{\pi}{2 * t_{max}} * \beta_{max} * \cos \left(\pi \cdot \frac{t - t_2}{t_{max}} - \frac{\pi}{2} \right) \\ \text{for } t > t_2 + t_{max} \quad \beta &= 0; \\ \dot{\beta} &= 0 \end{aligned}$$

Where $t_{max}=4$ time to drive wheel; $\beta_{max}=10^\circ$ - maximum rotation angle of the steering wheel; $t_2=20$ s - time when exiting the corner.

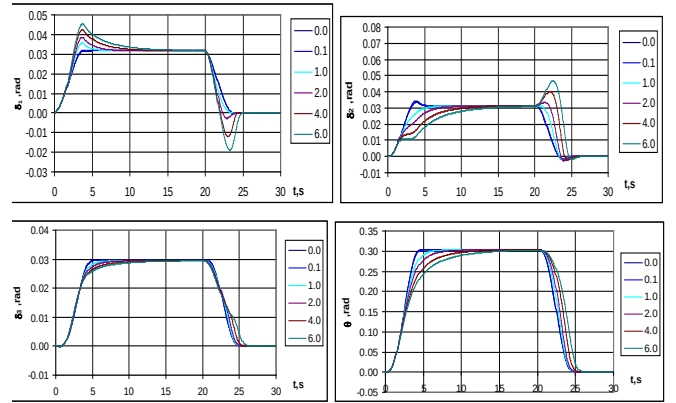


Figure 4. Changes result of the basic parameters in time of the TTA pulling Kubota trailer with $i_T = 30$; $V_{x1} = 3.53$ m/s; $k_f = 0-6$ MNS/m

In Figure 4 and 5 are graphs showing changing process of δ deviation moving angle of the wheel TTA and θ folding angle between tractors and trailers when changing the damping coefficient $k_f = 0-6$ MNS/m with 2 transmission ratio $i_t = 30$ and it = 20, corresponding to the initial velocity of the tractor $v_{Ix} = 3,53$ m/s and $v_{Ix} = 5,28$ m/s.

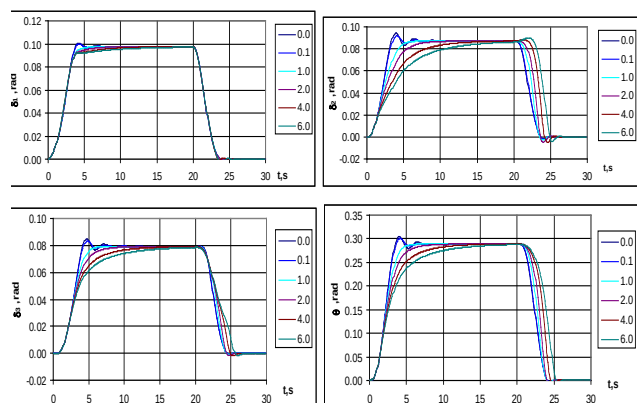


Figure 5. Changes result of the basic parameters in time of the TTA pulling Kubota trailer with $it = 20$; $VX1 = 5.28 \text{ m/s}$; $k_f = 0-6 \text{ Nm.s/m}$

The results showed that:

- Damper in some case has improved pretty well the stability of tractor when cornering. However in all cases when exiting corner, it absorber adversely affects the stability of TTA features. This suggests that, damper have to be one-side working.

- When increasing the damping coefficient k_f in the process of increased rotation $\beta=0$ to $\beta=\beta_{\max}$, the angles of deviation motion of δ_1 , δ_2 , δ_3 and folding angles are more likely downward. In contrast, in the process turning back steering, β angle down from $\beta_{\max}$ to 0, then δ_1 , δ_2 , δ_3 angles and θ increase relatively with increasing damping coefficient k_f .

- When changing the speed of movement of the tractor V_{IX} , then the angle δ_1 , δ_2 , δ_3 and θ and also vary with covariates.

4. Conclusion

1 - Model has shown quite sufficient the relationship between kinetics and revolving dynamics of combined tractor - trailer fitted with hydraulic dampers against sudden bending, allowing to survey effects of structural elements and properties used to control rotation of the TTA.

2 - Damping coefficient can significantly influence the folding angulation of TTA and therefore also affect the angle of deflection motion of the wheels, when the conditions in steering angle rotation the same. This reflects the effects of hydraulic shock absorber, increasing the safe in the sudden turnaround.

3 - The graph of the change of folding angle and deviation movement angle can be used as a basis for choosing appropriate damping coefficient for a specific combined tractor.

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CORROSION RESISTANCE OF LASER WELDED JOINTS OF TP347HFG AND VM12-SHC STAINLESS STEELS

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Abstract: The CO₂ laser welding technique was suggested as a method of joining dissimilar materials. The corrosion resistance at high temperatures of the joint of TP347HFG and VM12-SHC stainless steel was investigated. The stainless steel was butt welded. Materials were examined by the thermogravimetric method. The surface and microstructure of the sample were observed in a scanning electron microscope (SEM). The results showed the substantial intermixing of both substrates within the fusion zone. The thermogravimetric data indicate that at high temperatures and air condition the joint and the stainless steel undergo chemical corrosion. The joint has acquired medium resistance for chemical corrosion in relation to both steels.

Keywords: Laser welding; Joint, Austenitic, Martensitic, Cross-section, Thermogravimetric analysis, Chemical corrosion

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1. Introduction

Stainless steels are Fe-Cr-Ni alloys which are widely applied in various industrial sectors such as marine, chemical, desalination and petrochemical industries. Austenitic and martensitic stainless steels are most widely used to produce modern energetic devices. They can work at high and low temperatures [1-5]. These steels are usually used for a wide range of applications such as steam generators, pressure vessels, mixer blades, cutting tools and offshore platforms for oil extraction [6]. The joining of dissimilar materials is one of the most challenging tasks facing modern manufactures. Dissimilar metal joints are widely used in various industrial applications for both technical and economic reasons. Stainless steels are seldom welded because of their high hardenability and susceptibility to hydrogen induced cold cracking. Austenitic and martensitic stainless steels with their chromium and carbon contents are resistant to various environmental conditions. Moreover, nickel and molybdenum content provides the elevated-temperature strength through the formation of stable carbide both metals [7]. The laser welding technique was suggested as a modern method of joining two metals of different properties. Laser welding is a high energy density process and well-known for its deep penetration, high speed, small heat-affected zone, fine welding seam quality, low heat input per unit volume [8]. A butt joint is one of the most common laser welded materials used to produce tubes and tailor welded blanks. It is of profit to all welding, but even more important to laser welding. Unfortunately, in the case of dissimilar materials the butt joint is subject to chemical and electrochemical corrosion at high temperatures or in an aggressive environment, particularly in the presence of chloride ions [9-16].

The present paper deals with the corrosion resistance of the CO₂ laser welded joint of TP347HFG (austenitic) and VM12-SHC (martensitic) stainless steels. The samples were examined using a thermogravimetric analyzer (TGA) in air atmosphere. The surface and microstructures were observed by a scanning electron microscope (SEM).

2. Experimental

2.1. Materials

The materials of TP347HFG and of VM12-SHC stainless steels were designed for the laser welding. Samples for welding were prepared by cutting to the size of 150×200×5 mm from the base material.

2.2. CO₂ laser welding system

The CO₂ laser welding system was used for the melting of two stainless steels. A TRUMPF LASERCELL 1005 (TLC 1005) with 6 kW power made it possible to produce short and long series of different materials at low costs and in a short time.

The welding velocity was 160 mm/min in X-direction for the trial tests to perform a successful welding. The laser beam was

positioned at the joint (and moved) 2 mm in relation to the surface of a sample. Helium gas at the pressure of 200 Pa and speed of 18 L/min was used as a shield gas to protect the heated surface against oxidation.

2.3. Laser welded joint

The materials were butt welded. The full joint penetration was applied. The joint was initially examined in an optical microscope. The cross-section of the weld fillet was observed to determine the geometry, depth of penetration and cracks in the heat-affected zone (HAZ) as well as in the welded zone (WZ) of the joint. Moreover, no special heat treatment was carried out after laser welding.

2.4. Thermogravimetric measurements

Thermogravimetric measurements were carried out on an NETZSCH STA Jupiter 449 thermogravimetric analyzer (TGA). The sample in the furnace was heated up from 25 to 1380 °C with the heating rate of 5 °C min⁻¹. A analyzer was applied for the thermal analysis of TP347HFG or VM12-SHC stainless steels, and the joint of both steels.

2.5. Additional measuring instruments

The cross-section of the surface and microstructure of a sample was observed in a scanning electron microscope (SEM), Joel, type JSM-5400.

To evaluate the mechanical properties, microhardness values were determined along the cross-section by the Vickers (HV) method using a Microtech MX3 tester under of 0.4 N load.

3. Results and discussion

3.1. Thermogravimetric measurements and kinetic studies

The thermogravimetry curves for TP347HFG, VM12-SHC stainless steels and the joint are presented in Figure 1.

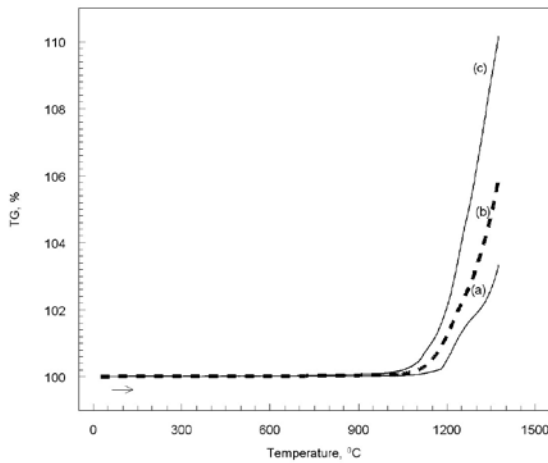


Fig. 1. Thermogravimetry curves for stainless steel and joint: (a) TP347HFG, (b) joint, and (c) VM12-SHC. Heating rate of $5^{\circ}\text{C min}^{-1}$.

The weight gain of samples were observed after the achievement of the critical temperature conversion in which the increase of the mass of a sample achieved the value of 0.1% in relation to the mass of the initial specimen, Table 1. TP347HFG was the most resistant to oxidation whereas VM12-SHC stainless steel was the least resistant to it. It is noteworthy that the critical temperature conversion for the joint was placed between both stainless steels. Moreover, Table 1 contains the weight gain of samples after the measurement.

Table 1. Critical temperature conversion, weight gain of stainless steel and joint.

Materials	Critical temperature conversion ($^{\circ}\text{C}$)	Weight gain sample (mg)
TP347HFG	1145	20.01
Joint	1101	41.71
VM12-SHC	1049	62.69

The largest increase in mass was registered for VM12-SHC, but the smallest for TP347HFG stainless steel. However, for the joint a medium increase in the mass of the sample was observed.

Kinetic studies, based on the change of mass, were obtained by the TG curve analysis. The corrosion rate of materials was calculated from the following equation:

$$k = \frac{\Delta m}{A t} \quad (1)$$

where Δm is the change mass of sample, A is the surface area of the test specimen, and t is the time of measurement. The corrosion rate was expressed in $\text{mg cm}^{-2} \text{h}^{-1}$.

The corrosion rate of samples were calculated on basis of Equation (1). The results are shown in Figure 2. The corrosion rate of samples increased with increase of temperature. The joint achieved the medium values of corrosion rate in comparison with both steels.

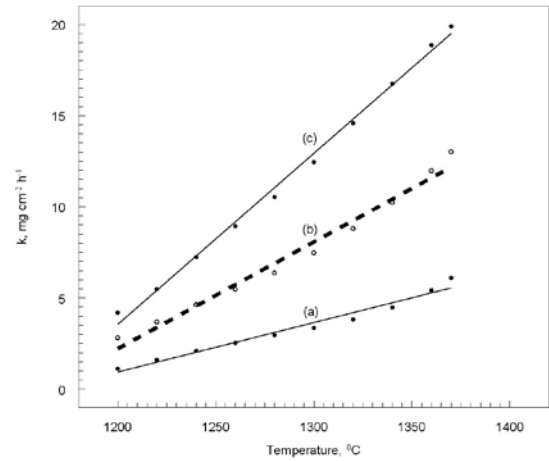
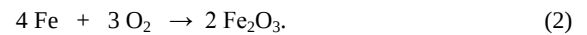


Fig. 2. Corrosion rate as function of temperature for stainless steel and joint: (a) TP347HFG, (b) joint, and (c) VM12-SHC. Heating rate of $5^{\circ}\text{C min}^{-1}$.

The thermogravimetric process in the atmosphere of air depends on the oxidation of the sample surface. The process of forming an oxide layer on the surface of the clean metal begins from the adsorption of oxygen:



The product of the iron corrosion (reaction (2)) at high-temperature has a stratified structure. The spatial structure of layers of oxide is of ion character. The process of forming a layer of oxide on the metal depends on the diffusion of Fe^{2+} through the layer in the direction of the gas phase (*from pithy diffusion*). Moreover, the diffusion ions of oxide from the layer in the direction of the metallic basis (*to pithy diffusion*) is also possible [17]. The tight and suitable thick of the oxide layer should effectively protect the metal against the significant progress of the corrosion process. The high temperature (higher than 1300°C) causes the dissociation pressure of the oxide layer according to the equation:



The compositions of scales for multiple alloys are usually varied.

In the case of TP347HFG and VM12-SHC stainless steels the compiled composition concerns the scale of the joint because it contains oxides of chrome, nickel, or cobalt.

Figure 3 shows the images of the joint, TP347HFG and VM12-SHC stainless steels after thermogravimetric measurements. It can be seen that on the surface of TP347HFG the compact, uniform and hard oxide layer was formed, which protects the steel prior to high-temperature corrosion.

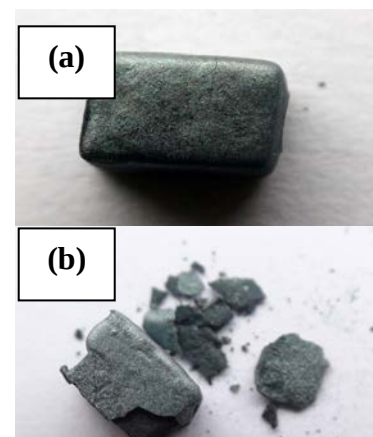




Fig. 3. Images stainless steel and joint after thermogravimetric measurement: (a) TP347HFG, (b) joint, and (c) VM12-SHC.

However, in the case of VM12-SHC and the joint, due to the dissociation pressure the top oxide layer undergoes cracking on small pieces, exposing the deeper layers of materials.

3.2. Microhardness profiles and microstructures

The microhardness profiles of the specimen were analyzed using the Vickers method. The results are presented in Figure 4.

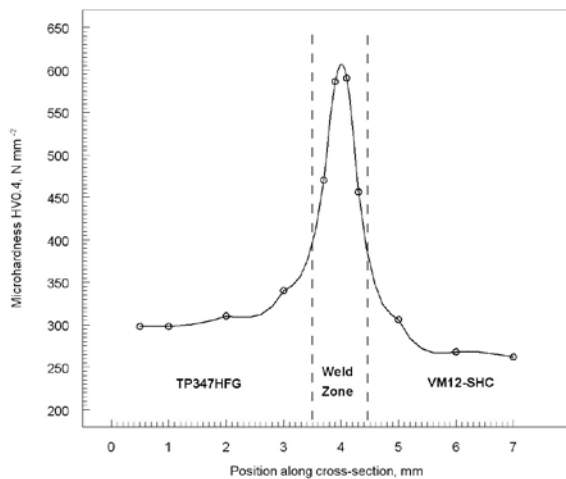


Fig. 4. Vickers microhardness profile along the cross-section in specimen.

In the heat-affected zone an increase in microhardness of both steels was observed. The maximum hardness value is located in the weld zone (about 580 HV0.4). The hardness of the joint increased about twice as much in comparison to the hardness of steels. The HV value indicates that the joint is fragile, and has low strength on hitting in relation to stainless steels. In this case special heat treatment should be applied after laser welding.

The scanning electron microscope images of the microstructure of the sample is shown in Figure 5.

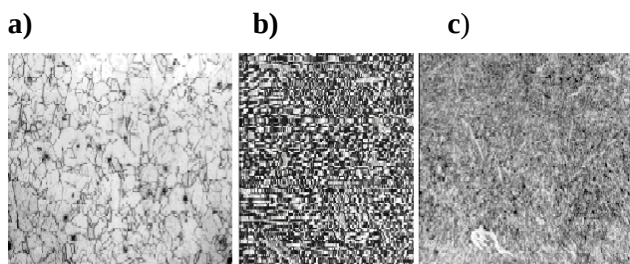


Fig. 5. Scanning electron microscope images of microstructure: a) TP347HFG, b) joint, and c) VM12-SHC stainless steel. Magnification 1000×.

The characteristic austenitic microstructure (Fig. 5a) is observed for TP347HFG as well as the martensitic microstructure (Fig. 5c) for VM12-SHC stainless steel. Moreover, observations revealed a fine grained microstructure, and basically dendritic in the weld zone (Fig. 5b). The material which possesses dendrite microstructure is usually very hard. However, this type of microstructure is a result of high cooling rates typical of the laser welding process.

4. Conclusions

1. The CO₂ laser welding technique was suggested as a method of joining dissimilar materials.
2. The thermogravimetric data indicate that at high temperature under air conditions the joint and stainless steels undergo chemical corrosion. The product of the corrosion of iron, Fe₂O₃, has a stratified structure at high-temperature.
3. The joint has the medium resistance to chemical corrosion in relation to both steels, and the chemical composition of the joint is different in comparison to both steels.
4. The weld zone revealed a fine microstructure and was basically dendritic, due to the high cooling rate which is characteristic of the laser welding process.
5. The value of Vickers microhardness indicates that the joint is fragile, and has low strength on hitting in relation to stainless steels.

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COMPLEX ANALYSIS OF PERFORMANCE CHARACTERISTICS OF SINGLE PHASE SHADED POLE MOTOR

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Abstract: Development of automatic, robotic and computer science has lead to development of wide range special motors (micro motors) which have large application and are interesting for research purposes. One of them is single phase shaded pole motor having a wide application in large number of household devices. Motor performance characteristics are analyzed using method of symmetrical components. For research purposes motor mathematical model is developed enabling prediction of motor operational characteristics. Obtained results are compared with experimental ones. Motor dynamic characteristics are obtained by building the simulation model in MATLAB/SIMULINK. Adequate conclusion regarding accuracy of developed motor models is derived.

Keywords: SINGLE PAHSE SHADED POLE MOTOR, SYMMETRICAL COMPONENTS, SIMULATION MODELS

1. Introduction

Single Phase Shaded Pole Motor (SPSPM) is well known for its simple constructions as well as for the complexity of electromagnetic processes which occur inside the motor due to existence of three mutually coupled windings which produce elliptical electromagnetic field in motor air gap. In this paper SPSPM, AKO-16 product of company MICRON is analyzed (Fig. 1). Motor rated data are: $U_n=220$ V, $f_n=50$ Hz, $I_{1n}=0.125$ A, $P_{1n}=18$ W, $n_n=2520$ rpm and $2p=2$.

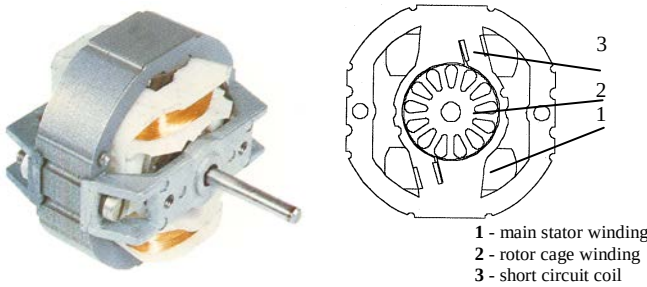


Fig. 1 Physical layout and motor cross section

Complex analysis of motor performance characteristics is applied by developing motor mathematical model based on symmetrical components method. Analysis is extended with experiments as well as with motor simulation model which gives inside in motor dynamical behavior at different operating regimes.

2. Motor performance characteristics based on method of symmetrical components

SPSPM is one of the most frequently used micro motors due to its simple construction and technological process for production. There are numerous advantages of this type of as motor short circuit current at main stator winding (locked rotor position) is very close to the rated current meaning that this type motor can sustain overloading or full load at locked rotor position without any overheating or motor damaging. SPSPM is very reliable during different operating regimes and overloading condition. From research purposes this type of motor is very complex as rotating electromagnetic field inside motor air gap, produces by stator windings is elliptic one meaning that there is a large inverse electromagnetic field which considerably contributes to the resulting elliptic rotating electromagnetic field and there is no standardized procedure for calculation of motor parameters and consequently motor performance characteristics. Method of symmetrical components is commonly accepted is literature as method for calculation of motor operational characteristics [1]. Mathematical model based on method of symmetrical components is developed which enables calculation of symmetrical components of currents in main stator winding i.e. direct component I_1^+ and

inverse component I_1^- and consequently rotor windings I_2^+ and I_2^- [2]. Two phase unsymmetrical system of vectors of currents in main stator winding I_1 and in short circuit coil I_3 which have different amplitudes and are displaced in space for arbitrary angle α are decomposed to two symmetrical systems each containing two vectors equal in amplitude and spatially apart for an angle of 90° . When this space angle between windings α is not equal to 90° for stator currents relation $I_3 \neq I_1$ is always valid.

Current in main stator winding is calculated from:

$$I_1 = -j \left(\frac{I_1^- e^{j\alpha}}{\sin \alpha} - \frac{I_1^+ e^{-j\alpha}}{\sin \alpha} \right) \quad (1)$$

Current in short circuit coil is calculated from:

$$I_3 = j \left(\frac{I_1^-}{\sin \theta} - \frac{I_1^+}{\sin \theta} \right) \quad (2)$$

Calculation of motor performance characteristic based on method of symmetrical components utilizes equivalent circuits for main stator winding for direct and inverse system (Fig.2).

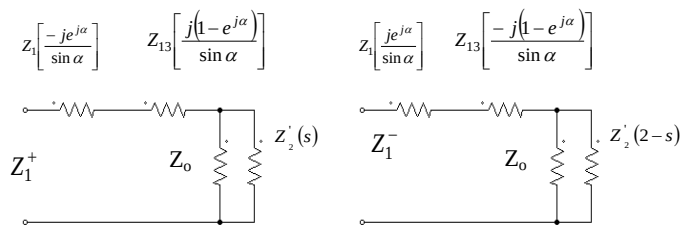


Fig. 2 Equivalent circuit of main stator winding direct and inverse system

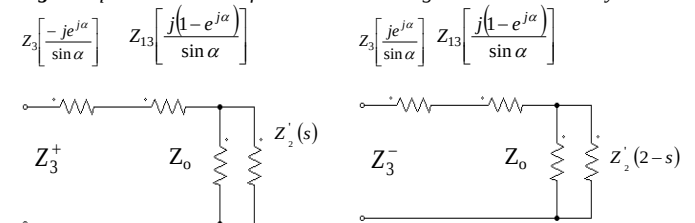


Fig.3. Equivalent circuit of short circuit coil direct and inverse system

Z_1^+ and Z_1^- are impedances of main stator winding and short circuit coil respectively. Z_{13} is the mutual reactance between main stator winding and short circuit coil. Impedance Z_0 is found from mutual reactance between stator and rotor winding. Z_2' is impedance of

rotor winding referred to main stator winding and s is denoting motor slip.

Symmetrical components of rotor currents are found from:

$$I_2^{+'} = I_1^{+'} \frac{Z_0}{Z_0 + Z_2^{+'}} \quad (3)$$

$$I_2^{-'} = I_1^{-'} \frac{Z_0}{Z_0 + Z_2^{-'}} \quad (4)$$

$Z_2^{+'}$ and $Z_2^{-'}$ are rotor impedances for direct and inverse system. Consequently motor electromagnetic torque is calculated from:

$$M_{em} = M_{em}^{+} - M_{em}^{-} \quad (5)$$

where:

$$M_{em}^{+} = \frac{9,55}{n_s} \frac{2|I_2^{+'}|^2 R_2'}{s} \quad (6)$$

$$M_{em}^{-} = \frac{9,55}{n_s} \frac{2|I_2^{-'}|^2 R_2'}{2-s} \quad (7)$$

where n_s is motor synchronous speed and s is motor slip.

And resulting electromagnetic torque is calculated from:

$$M_{em} = M_{em}^{+} - M_{em}^{-} \quad (8)$$

Power factor $\cos\varphi$, input power P_1 , output power P_2 , torque at motor shaft M_2 and efficiency factor are calculated from:

$$\cos\varphi = \frac{\text{Real}(I_1)}{|I_1|} \quad (9)$$

$$P_1 = U_1 |I_1| \cos\varphi \quad (10)$$

$$P_2 = \frac{P_{meh}}{1,15} \quad (11)$$

where P_{meh} is mechanical power on motor shaft.

$$M_2 = \frac{9,55 P_2}{n_s (1-s)} \quad (12)$$

$$\eta = \frac{P_2}{P_1} \quad (13)$$

Based on above mentioned methodology motor's electro-mechanical and operation characteristics are calculated and presented: current in main stator winding I_1 , rotor current I_2 , current in short circuit coil I_3 , power factor $\cos\varphi$, and efficiency factor η in dependence of motor slip s on Fig. 3, input power P_1 and output

power P_2 as function of motor slip s on Fig. 4, and motor electro-mechanical characteristic of output torque $M_2=f(s)$ and electromagnetic torque $M_{em}=f(s)$ on Fig. 5. Motor operational characteristics power factor $\cos\varphi$, efficiency η , input power P_1 , motor speed n and input current I_1 in dependence of output power P_2 are presented on Fig.6.

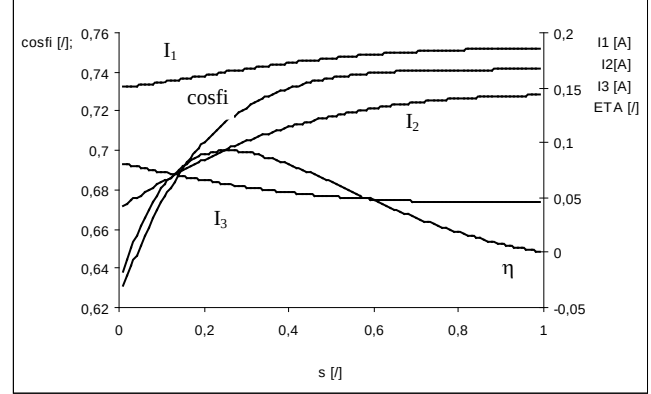


Fig. 3 Electromechanical characteristics $I_1=f(s)$, $I_2=f(s)$, $I_3=f(s)$, $\cos\varphi=f(s)$ and $\eta=f(s)$

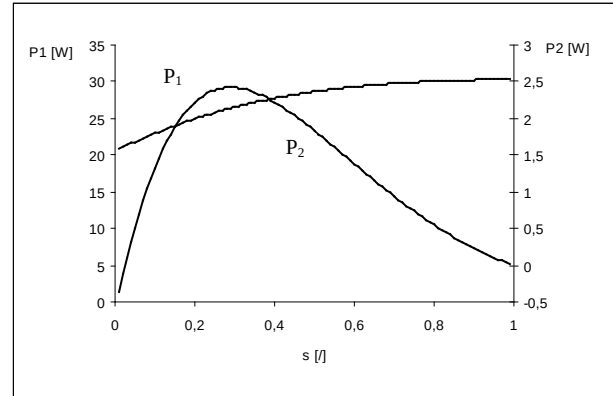


Fig.4. Electromechanical characteristics $P_1=f(s)$ and $P_2(s)$

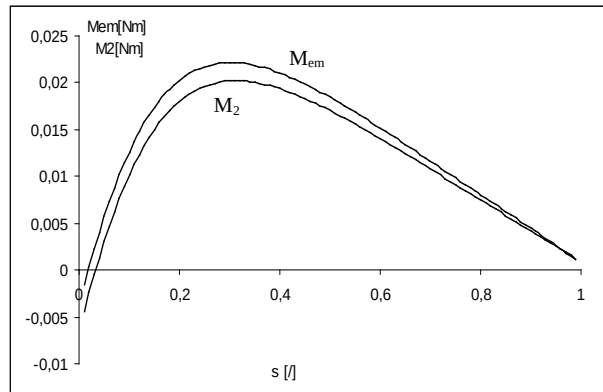


Fig.5. Electromechanical characteristics $M_{em}=f(s)$ and $M_2=f(s)$

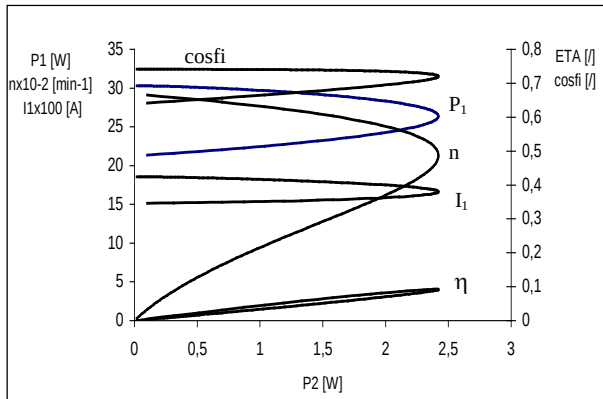


Fig.6. Motor operational characteristics versus output power

134.8	0.06125	5.225	0.634
145.4	0.0697	6.425	0.64
154.4	0.076925	7.6125	0.642
165	0.084625	9.1	0.649
174.14	0.091375	10.35	0.653
184.6	0.099375	11.975	0.657
195.2	0.107	13.775	0.659
204.02	0.1136	15.325	0.662
213.15	0.12085	17.1	0.6645
222.9	0.128875	19.1	0.666
230.7	0.13625	20.975	0.667
243.96	0.14775	24.15	0.6675

Parameters and characteristic obtained by analytical calculations and compared with data from experiment and producer. Results of comparison are presented in Table 4.

Table 4: Comparative analysis of results

Experiment	Analytical calculation	Producer
$I_1=0.125$ A	$I_1=0.1259$ A	$I_1=0.18$ A
$P_1=18$ W	$P_1=18.114$ W	$P_1=22$ W
$\cos\varphi=0.6545$	$\cos\varphi=0.6538$	$\cos\varphi=0.555$
$R_1=498$ Ω	$R_1=493$ Ω	$R_1=480$ Ω
$I_0=0.111$ A	$I_0=0.1134$ A	$I_0=0.13$ A
$P_0=13.8$ W	$P_0=10.21$ W	$P_0=13.5$ W
$\cos\varphi_0=0.5651$	$\cos\varphi_0=0.4092$	$\cos\varphi_0=0.472$
$I_k=0.135$ A	$I_k=0.181$ A	$I_k=0.221$ A
$P_k=19.87$ W	$P_k=29$ W	$P_k=31$ W
$\cos\varphi_k=0.6683$	$\cos\varphi_k=0.7389$	$\cos\varphi_k=0.6376$

3. Motor experimental data

Laboratory set-up for measurements of motor operational characteristics at no-load, locked rotor and rated load was done. Experiment of no-load was performed during free rotation of motor shaft without any connected load and motor supply voltage was varied in range $(0.45 \div 1.2)U_n$. Displays of all instruments are recorded, voltage at motor stator winding U_0 , no-load current I_0 and motor input power P_0 . At each operating point motor speed n is measured as well. Results from measurement at no-load operation are presented in Table 1.

Table 1: Measurement at no-load operating regime

n [rpm]	U_0 [V]	I_0 [A]	P_0 [W]	$\cos\phi_0$ []
2490	125,1	0.045	3.375	0.581
2620	144,35	0.05585	4.675	0.577
2700	164,8	0.069625	6.525	0.571
2740	185	0.0845	8.8875	0.568
2790	204.2	0.101875	11.875	0.575
2780	225.5	0.11975	15.375	0.583
2720	242.5	0.144	20.15	0.577033

Experiment of locked rotor is performed when motor is in the locked position and voltage is varied until rated current is reached. Measurements of all instruments are recorded and consequently short circuit current I_k , voltage at short circuit U_k and power factor $\cos\varphi_k$ are obtained. Results are presented in Table 2.

Table 2: Measurement at locked rotor regime

U_k [V]	I_k [A]	P_k [W]	$\cos\phi_k$ []
75.9	0.02275	1.125	0.703
109.6	0.04195	2.9375	0.688
140.1	0.06445	5.8	0.674
164.9	0.08475	9.225	0.661
186.8	0.105375	3.275	0.655
206.5	0.1244	17.6125	0.685615
235	0.15725	25.625	0.693434

Results from experiment at rated operational regime when motor is loaded with fan are presented in Table 3.

Table 3: Measurement at rated load operating regime

U_1 [V]	I_1 [A]	P_1 [W]	$\cos\phi$ []
123.6	0.053125	4.15	0.627

In Table 1 is also presented main stator winding active resistance R_1 . Results of comparison between calculated and measured current of main stator winding as well of measured and calculated input and output power are presented on Figs. 7 and 8 respectively.

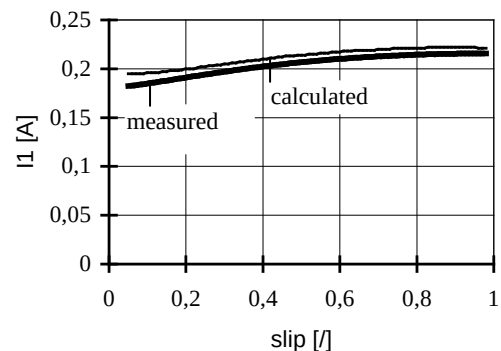


Fig. 7. Comparison of characteristics of input current

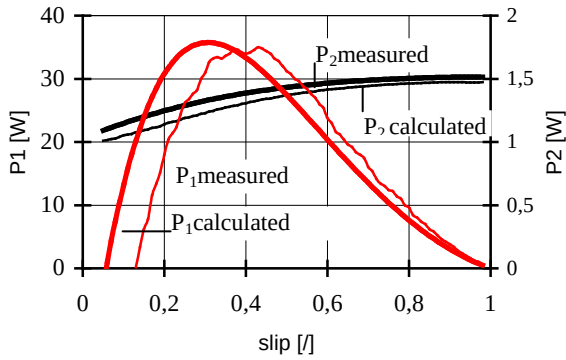


Fig.8. Comparison of characteristics of motor input and output power

3. Dynamic performance characteristics

In order to have an insight in motor dynamic behavior simulation model of the motor is built in SIMULINK [3]. Simulation model is consisted of four main parts: motor power supply (voltage sources-two phase system a,b), transformation of input variables (voltages) from two phase system into reference d,q system which is stationary, motor model where as direct output variables are obtained motor speed and torque and transformation from d,q reference system into two phase a,b system in order to be obtained motor current in main stator winding, short circuit coil as well as in rotor winding. On Fig.9, 10, 11 and 12 consequently are presented motor dynamic characteristics of current in main stator winding I_1 , electromagnetic torque M_{em} motor speed n_r and flux linkage- λ_1 at rated load operating regime.

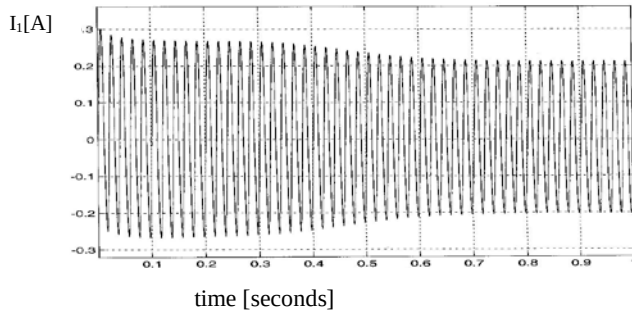


Fig.9. Transient performance characteristics of current in main stator winding at rated load

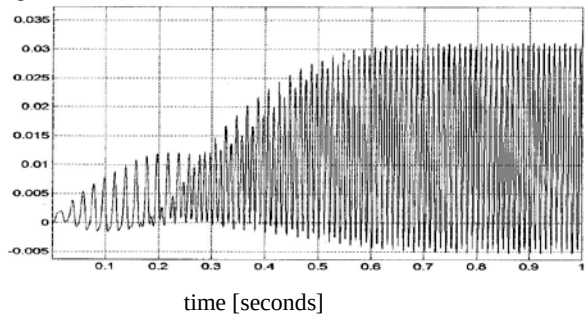


Fig.10. Transient performance characteristics of electromagnetic torque at rated load

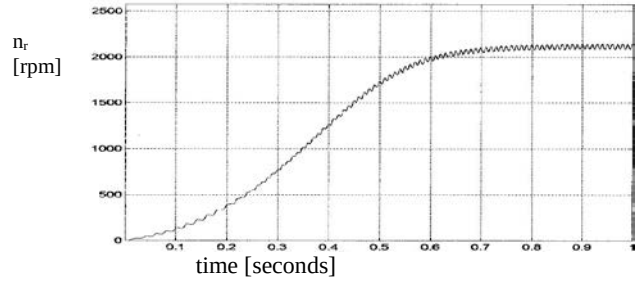


Fig.11. Transient performance characteristics of speed at rated load

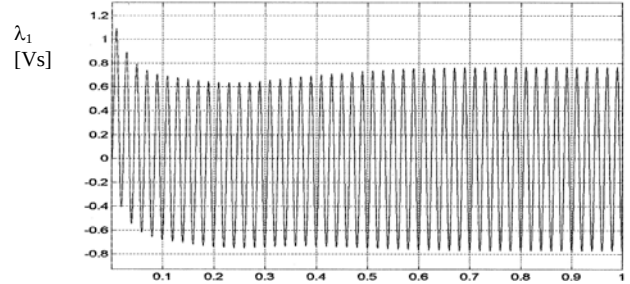


Fig.11. Transient performance characteristics of flux linkage at rated load
Comparison between results from simulation and calculation are presented in Table 5.

Table 5: Comparison between simulated and calculated values

Param.	Rated load operating regime		
	Simul.	Calcul.	Producer
n_r [rpm]	2857	2856	2523
M_{em} [Nm]	0.0198	0.018075	/
I_1 [A]	0.1259	0.134	0.18

In Table 6 is presented comparison between obtained values of electromagnetic torque from simulation and calculation and on the base of obtained values motor electromechanical characteristics in Fig. 12 are plotted.

Table 6: Comparison between simulated and calculated values

slip [l]	Calculation	Simulation
	M_{em} [Nm]	M_{em} [Nm]
0.047	0.005211	0.004
0.09	0.01145	0.011409
0.31	0.022166	0.02023
0.56	0.016509	0.01639
0.81	0.007681	0.0088
0.98	0.001568	0.002

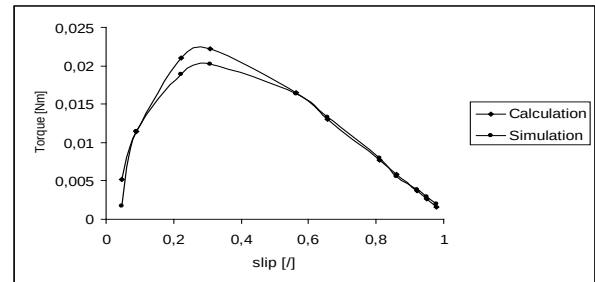


Fig.12. Comparative characteristics of electromagnetic torque from calculation and simulation

4. Conclusion

Methodology for analytical calculation of operational characteristics of SPSPM is developed based on method of symmetrical components. Experiments are performed regarding motor characteristic operating regimes: no-load, rated load and locked rotor. Obtained results of analytical calculation are compared with values from experiment and producer data confirming the accuracy of proposed methodology for calculation of motor parameters and performance characteristics. Motor dynamic performance characteristics are evaluated with simulation model built in Simulink at different operating regimes. Obtained values from motor transient characteristics of current in main start winding, electromagnetic torque and speed after dynamic process of motor acceleration is terminated are compared values obtained from calculation and producer showing reasonable agreement and proving the accuracy of developed simulation model. Mean value of motor electromagnetic torque from transient performance characteristic for different slips, meaning different operating regimes is compared with values from analytical model and agreement of results is satisfactory.

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APPLICATION OF NUMERICAL METHODS IN CALCULATION OF ELECTROMAGNETIC FIELDS IN ELECTRICAL MACHINES

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Abstract: Finite Element Method has been proved as valuable tool for solving different electromagnetic problems inside electrical machines. Calculation of magnetic flux density and its distribution in machine cross-section is difficult to be calculated by analytical methods. Therefore Finite Element Method is implemented for solving set of Maxwell equation which enables precise calculation of electromagnetic field and magnetic flux density in three different electrical machines: three phase squirrel cage motor type 5AZ801-4 prodct of company Rade Koncar, three phase distribution transformer type product of company EMO, and single phase capacitor motor FMR-35/6 product of company MikronTech. Distribution of magnetic flux density in all three machines is calculated for different operating regimes.

Keywords: Finite Element Method, three phase squirrel cage motor, three phase distribution transformer, single phase capacitor motor

1. Introduction

Calculation of electromagnetic fields inside machine cross-section is always a challenging task and often it implements numerical calculations based on Finite Element Method (FEM) which enables solving the set of Maxwell equations and obtaining the value of magnetic vector potential and magnetic flux density in all parts of machine cross section. On that way possible “weak” parts of machine construction can be discovered i.e. parts where flux density reaches high values and machine operates near to saturation point of magnetic core. In this paper three different machines are analyzed: three phase squirrel cage motor 5AZ801-4 product of company Rade Koncar from Zagreb single phase capacitor motor FMR-35/6 product of company EMO MikronTech from Prilep (Fig.1), three phase transformer product of company EMO from Ohrid (Fig.3).

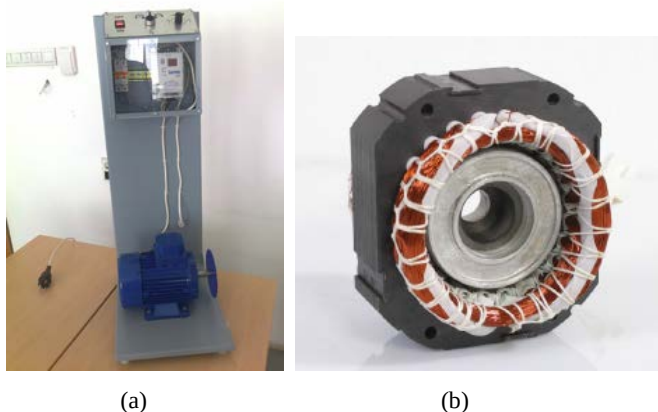


Fig. 1. Physical layout of (a) squirrel cage motor (b) single phase motor

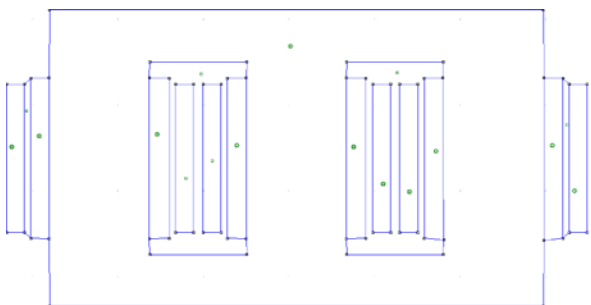


Fig. 2. CAD drawing of three phase transformer

Rated data of three phase motor 5AZ80-1/4 are: stator winding connection: D/Y, rated voltage 230/400, rated current 2,8/1.6 , rated power 0.55 kW, power factor $\cos\phi=0.76$, rated speed 1390 rpm. Rated data of single phase capacitor motor FMR-35/6 are: rated voltage 220-240 V, rated power 180 W, rated current 1,32 A, rated speed 2880 rpm. Rated data of transformer are: rated power 50 kVA, voltage 10/0.4 kV, Yzn5, short circuit voltage 4 %.

2. Methodology of FEM

Finite Element Method (FEM) is a numerical method used for solving relatively complex electromagnetic problems where material nonlinearity and anisotropy is included in analyzed domain. Method involves discretization of the whole analyzed domain in small triangle surfaces, which are called finite elements. By applying Maxwell's equations into FEM it is possible to calculate the stationary distribution of magnetic field inside electrical devices. The FEM analysis of the analyzed object is divided into three parts: pre-processing, processing and post-processing part. In pre-processing part the object geometry, as well as boundary conditions are defined. For all machines are chosen Dirichlet boundary conditions e.g. $A=0$. The most common use of Dirichlet-type boundary conditions in magnetic problems is to define $A=0$ along a boundary to keep the magnetic flux from crossing the boundary. Properties of all materials are input in object model. Beside inputting the magnetization curve as $B=f(H)$ also the lamination of magnet material is input according to Figure 3 (a) as well as fill factor. The result of this model is that one can account for laminations with hysteresis and eddy currents. In order the value of the magnetic vector potential A to be determined it is necessary the whole domain i.e. object cross-section to be divided into a certain number of elements.

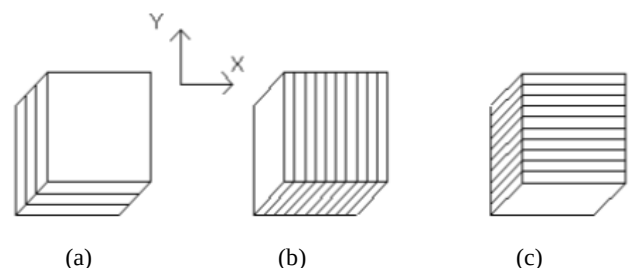


Fig. 3 Laminations of magnetic steel sheets in FEM models

The number of elements is problem dependent and for transformer model the finite element mesh is consisted of $N=17833$ nodes and $E=35089$ elements (Fig.4) while for the three phase motor model $N=34\,751$ nodes and $E=74724$ elements (Fig.5). For single phase capacitor motor finite

element mesh is consisted of N=51 638 and E=102850 elements. (Fig.6).

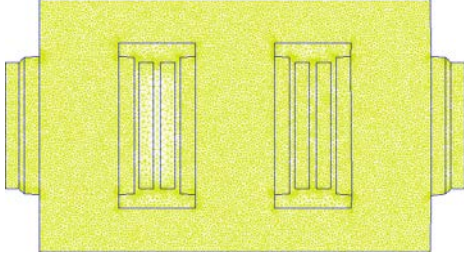


Fig. 4 FEM mesh in transformer

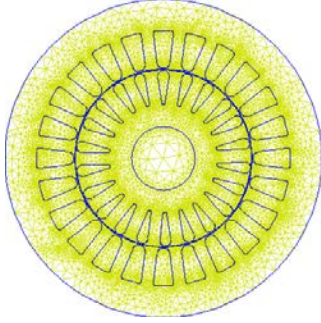


Fig. 5 FEM mesh in three phase motor

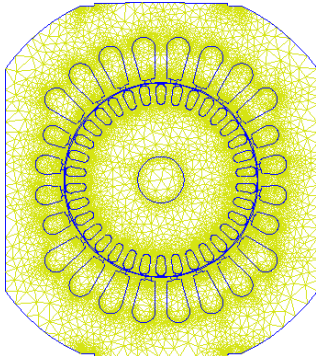


Fig. 6 FEM mesh in single phase capacitor motor

In order processing part to be executed and output results to be obtained system of Maxwell's equation should be solved. They differ in both cases: magnetostatic and time-harmonic.

Magnetostatic problems are problems in which the fields are time-invariant [1]. In this case field density $\mathbf{H}$ and flux density $\mathbf{B}$ must obey:

$$\nabla \times \mathbf{H} = \mathbf{J} \quad (1)$$

$$\nabla \times \mathbf{B} = 0 \quad (2)$$

subject to a constitute relation between $\mathbf{B}$ and $\mathbf{H}$ for each material:

$$\mathbf{B} = \mu \mathbf{H} \quad (3)$$

FEM goes about finding a field that satisfies (1)-(3) via a magnetic vector potential. Flux density is written in terms of vector potential $\mathbf{A}$ as:

$$\mathbf{B} = \nabla \times \mathbf{A} \quad (4)$$

For the magneto-static problem the non-linear B-H relation FEM solves the equation:

$$\nabla \times \left(\frac{1}{\mu(\mathbf{B})} \nabla \times \mathbf{A} \right) = \mathbf{J} \quad (5)$$

The advantage of using the vector-potential formulation is that all the conditions to be satisfied have been combined into a single equation. If magnetic vector potential $\mathbf{A}$ is found, $\mathbf{B}$ and $\mathbf{H}$ can be deduced by differentiating $\mathbf{A}$. After exact machine geometry is input boundary conditions are defined. For this specific motor model are chosen Dirichlet boundary conditions e.g. $\mathbf{A}=0$. The most common use of Dirichlet-type boundary conditions in magnetic problems is to define $\mathbf{A}=0$ along a boundary to keep the magnetic flux from crossing the boundary. When analysing induction machines, considering their AC excitation the air gap magnetic field is always a time-varying quantity [2]. In materials with non-zero conductivity eddy currents are induced, consequently the field problem turns into magnetodynamic i.e. non-linear time harmonic problem. When rotor is moving, the rotor quantities oscillate at slip frequency. In this case the rotor bars conductivity σ is adjusted corresponding to the slip. Consequently following partial equation is going to be solved numerically:

$$\nabla \times \left(\frac{1}{\mu(\mathbf{B})} \nabla \times \mathbf{A} \right) = -\sigma \dot{\mathbf{A}} + \mathbf{J}_{src} - \sigma \nabla V \quad (6)$$

Where $\mathbf{J}_{src}$ represents the applied current sources. The additional voltage gradient ∇V in 2-D field problems is constant over the conduction body. Strictly speaking the permeability μ should be constant for harmonic problems. However, FEM retains a nonlinear relationship in the harmonic formulation, allowing the program to approximate the effects of saturation on the phase and amplitude of the field distribution. FEM also allows for the inclusion of complex and frequency-dependant permeability in time harmonics. These features allow the program to model materials with thin laminations and approximately model hysteresis effects. Program is run at constant frequency $f=50$ Hz. In motor model is input only current in main stator winding while currents in short circuit coil and rotor winding are freely induced. For the transformer model current density is input in primary and secondary winding in all three phases and problem is analyzed at frequency 50 Hz.

3. Results from FEM

Three phase motor represents the standard motor construction which has symmetrical rotating electromagnetic field due to three symmetrical phase windings, placed in stator slots having voltages and currents shifted with a phase of 120° between each other. Magnetic flux density distribution in machine cross section at no load is presented at Fig. 7 while in Fig.8 is presented distribution of magnetic flux density in machine air gap.

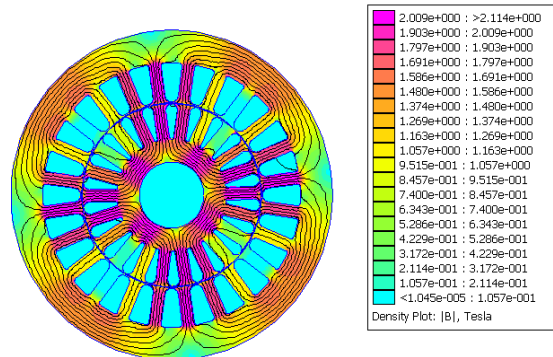


Fig. 7 Magnetic flux distribution in three phase motor-no load.

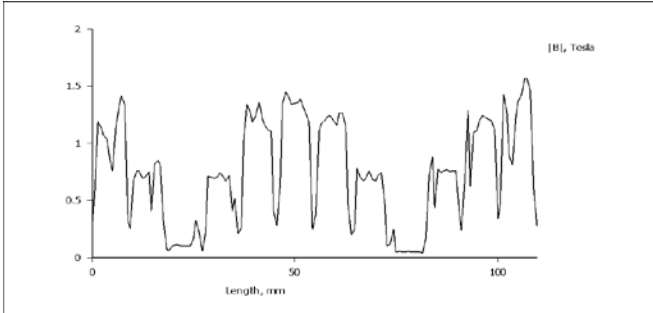


Fig. 8. Magnetic flux density distribution in air gap three phase motor at no-load

In contrast, single phase capacitor motor has an auxiliary element capacitor placed in auxiliary stator winding which enables motor start and enables rotating electromagnetic field in machine air gap. In this specific case capacitor of 6 μF is placed in auxiliary winding and it is a permanent capacitor which stays in operation after motor start-up. Motor supply is from single phase network, but from its operating principle there are always two winding in stator of the motor (main and auxiliary) which are connected to voltage supply (Fig. 9). Magnetic flux density distribution in cross section of single phase motor at no load is presented at Fig. 10 while in Fig.11 is presented distribution of magnetic flux density in motor air gap.

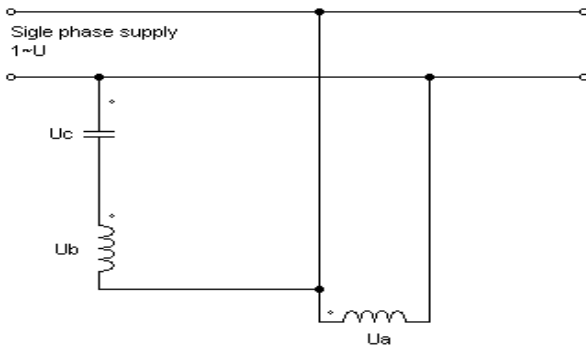


Fig. 9 Electrical circuit of single phase capacitor motor

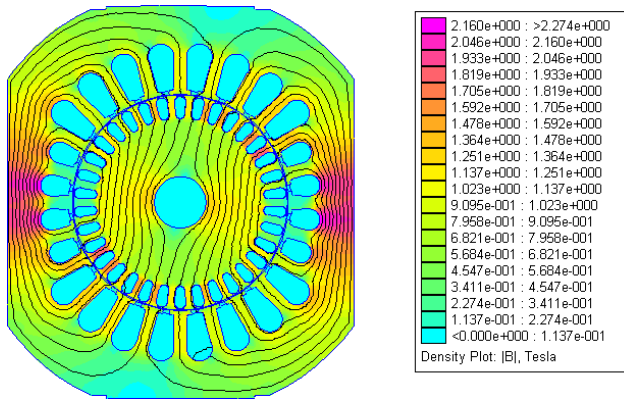


Fig. 10. Magnetic flux distribution in single phase motor-no load.

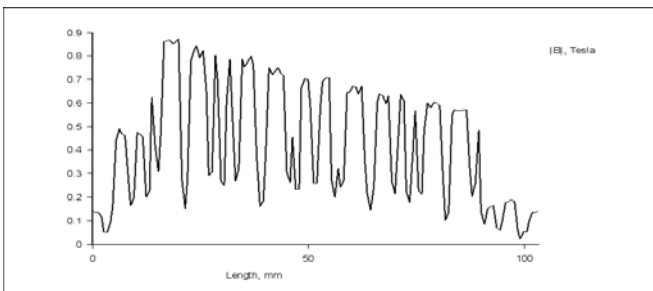


Fig. 11. Magnetic flux density distribution in air gap single phase motor at no-load

Three phase transformer represents the static electromechanical device i.e. there are no moving parts in machine construction. Electromagnetic field is symmetrical produced by three phase windings in primary winding coupled electromagnetically with secondary winding. Since there are no rotating parts in machine construction conductivity of material in secondary winding remains unchanged and same with the conductivity of primary winding. In no-load condition secondary winding is open, there is no current flowing, while primary winding is supplied with rated voltage. Magnetic flux distribution in machine cross section at no-load is presented in Fig 12 while distribution of magnetic flux density in first leg of transformer is presented in Fig. 13 and in second leg in Fig 14.

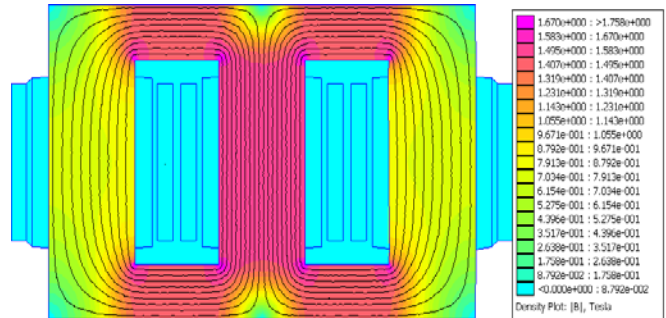


Fig. 12. Magnetic flux distribution in transformer-no load.

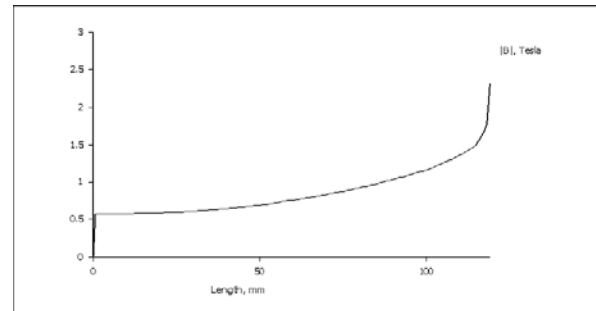


Fig. 13. Magnetic flux density distribution in transformer first leg at no-load

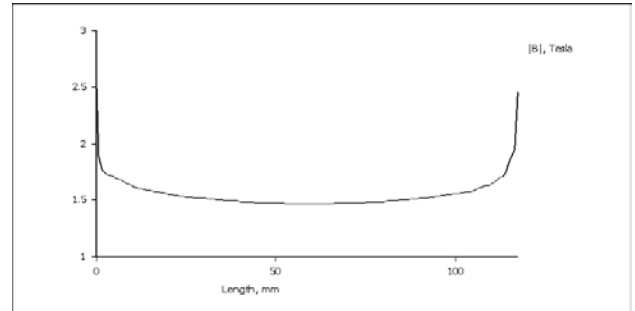


Fig.14. Magnetic flux density distribution in transformer second leg at no-load

In addition, rated load operating condition of machines is analyzed with respect to, magnetic flux distribution in machine cross-section as well as distribution of magnetic field in certain parts of machine construction, subject to relation (3). For modeling the rated load operating conditions, appropriate densities of currents in stator windings for motors must be input while currents in rotor winding are induced and conductivity of rotor winding material is adjusted in correspondence to motor slip:

$$s = \frac{n_o - n_n}{n_o} \quad (7)$$

where n_o is motor synchronous speed and n_n is motor rated speed.

For transformer rated load operating regime, rated load transformer currents are input in primary and secondary winding of transformer and problem is analyzed at frequency $f=50$ Hz. Magnetic flux distribution in all three machines is presented in Figs. 15, 16 and 17 respectively.

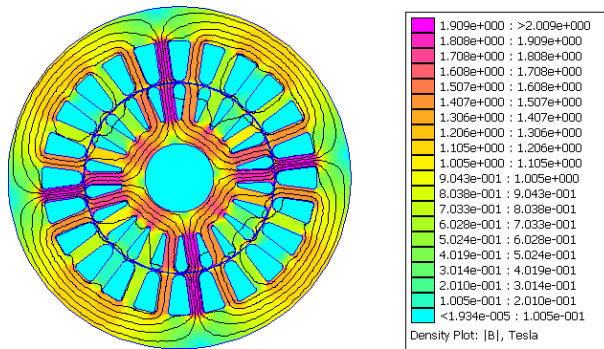


Fig. 15. Magnetic flux distribution in three phase motor-rated load.

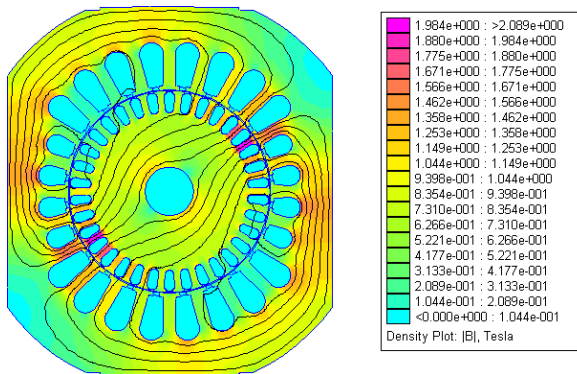


Fig. 16. Magnetic flux distribution in single phase motor-rated load.

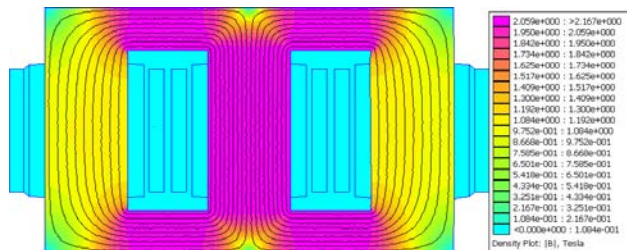


Fig. 17. Magnetic flux distribution in transformer-no load.

Magnetic field distribution in air gap-H taking into account non-linearity of magnetic materials at rotating machines is presented in Figs. 18 and 19 for three phase motor and single phase respectively while for transformer in first leg and second leg in Fig. 20 and 21. First and second leg of three phase transformer differ in value of electromagnetic field and flux density since second leg is always subject to higher values of flux density with respect to first and third leg of transformer core.

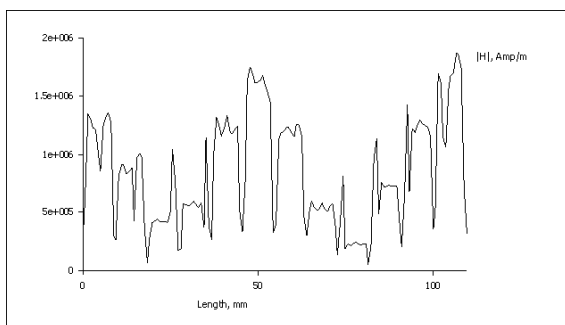


Fig. 18. Magnetic flux distribution in three phase motor-rated load.

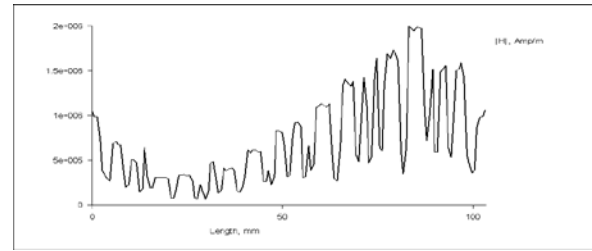


Fig. 19. Magnetic field distribution in single phase motor-rated load.

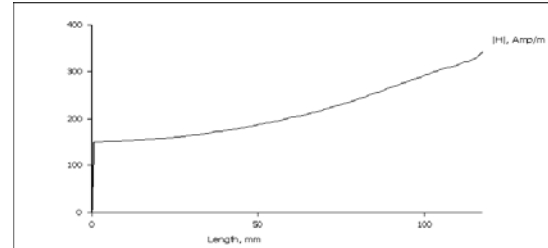


Fig. 20. Magnetic flux distribution in transformer at first leg- rated load.

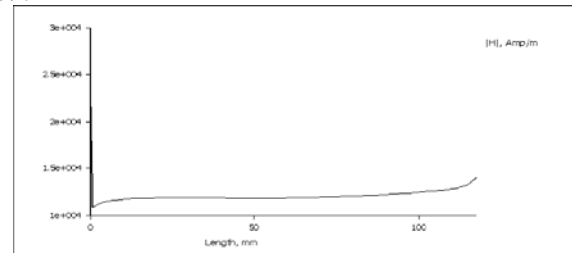


Fig. 21. Magnetic flux distribution in transformer at second leg- rated load.

4. Conclusions

Numerical methods are widely used in calculation of electromagnetic fields inside electromechanical devices. Finite Element Method has received wide popularity in numerical solving the Maxwell set of equation which defines the magnetic flux density and electromagnetic field in every point of machine cross section. Calculation of above mentioned parameters is difficult to be done analytically due to nonlinearity of magnetic permeability of magnetic material implemented in construction of machines and they are often subject of assumption and implementation of figures based on practical experience. FEM can be implemented on any machine during the phase of designing and construction. In this paper FEM is implemented on three different machines from three different producers. Further more two of them are rotating electrical machines: three phase squirrel cage motor with symmetrical rotating electromagnetic field and single phase capacitor motor which is considered to be the more special case for modelling due to asymmetrical electromagnetic field in electrical machine and existence of capacitor in auxiliary winding. Three phase transformer is stationary electrical machine with symmetrical electromagnetic field. Magnetic field density distribution in all machines is obtained as well as electromagnetic field for no-load and rated load operating regime. Further authors work will be focused on developing 3D FEM models of analyzed machines.

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ANALYSIS OF THE PROCESS OF BENDING A TWO-LAYER SHEET

PASCAGE

АНАЛИЗ ПРОЦЕССА ИЗГИБА ДВУХСЛОЙНОГО ЛИСТОВОГО ПАКЕТА

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Abstract: The analysis of two-layer sheet bending moment package under plane strain conditions, taking into account the strain hardening. Approximate analytical dependence for the largest radial compressive stress and the relative displacement of the radius of the neutral surface of the inner layer. The possibility of increasing the permissible degree of deformation and reduce the relative radius of curvature of the inner layer.

Keywords: SHEET MOLIBDENUM, BENDING, STRESS, STRAIN HARDENING, NEUTRAL RADIUS

1. Introduction.

Molybdenum is one of the most perspective heat resistant metal for creating structures used in various technologies of engineering. Products made from molybdenum alloys work in high temperature, radiation exposure, plasma and high-speed gas flows in contact with aggressive environment and melts. These products work in contact with liquid metals and their vapors in power systems of spacecraft, at temperatures up to 2000° C [1]. A similar situation occurs in the high temperature crystallization from melt of refractory dielectrics. In these processes, crucibles made of sheet molybdenum, working in contact with a melt at temperatures 2100° C [2].

A heat resistant construction from sheet molybdenum, in most cases are the type of products of thin shells that are currently produced using a bending flame heating in an air atmosphere combination with welding [3,4]. According to State standard specification (GOST) 17431-72 and "Product Specification Plansee" (Molybdenum Sheet PH-IHF 012, date of issue: 2011-08-02) molybdenum sheets with thickness of 0,25-0,65 mm used for fabrication of heat resistant constructions, have the following mechanical properties: yield stress $\sigma_{0,2} = 590 \text{ MPa}$ (60 Kg / mm^2), tensile strength $\sigma_b = 690 \text{ MPa}$ (70 Kg / mm^2), total elongation 5%.

These characteristics are defined by a uniaxial tensile test, in the first approximation allow to evaluate the ability of the material to forming also under complex stress state.

One of the main performance of sheet metal, that deformed in conditions of a plastic bending is the minimum value of the relative radius, which is determined from the condition of equality of the largest relative deformations in the linear tension and circumferential strain tension of the outer layer [5]. From the specified equality, without taking into account pressing of the layers to each other, determined the minimum value of the relative radius $(r/s)_{\min} \geq (1-\delta)/2\delta$ (where r, s - accordingly, the inner bending radius and the thickness of the sheet). Calculations show that for sheet molybdenum $(r/s)_{\min} \geq 9,5$, i.e. with a further decrease of the

relative radius on the outer side of bending molybdenum sheet value of circumferential tension strain reaches the limit value and there will be destruction. According to the technological recommendations [5] the relative minimum bending radius (ratio) depending on the direction of rolling,

the limits are set to $1 \leq (r/s)_{\min} \leq 2$, which significantly less than the permissible value for the molybdenum sheet.

Resource tests of heat-resistant products show that they fail due to fracture in welding areas, and their widespread application is constrained by another strong oxidizability molybdenum at elevated temperatures, and therefore the process of growing refractory single crystals carried out under high vacuum ($5 \cdot 10^{-5}$ Torr) [2]. Consequently, the development of new methods and technologies forming of heat-resistant products made of molybdenum sheet, providing the required design and performance and preventing the emergence of micro-defects in the finished product is an actual scientific and technical problem.

Difficulties of applying traditional general machine-building technologies forming of sheet molybdenum consist in the fact that in the process of deformation, due to the limited ductility, in certain parts of products tensile stresses occurs exceeding the permissible values, therefore destruction of material and forming process stops.

2. Statement of the problem and analysis of the stress state.

One of the feasible forming of molybdenum sheet is its deformation under a two-layer sheet package, consisting of internal and external molybdenum layers made of sheet steel with high technological ductility [6].

In forming a two-layer sheet package from the outer layer on a sheet molybdenum contact stresses are hindering the growth circumferential tensile stresses and contribute to increase the permissible degree of deformation. It is obvious that contact stress values should depend on the mechanical properties and thickness of the outer layer. Quantitative assessment of these stresses, due to the uncertainty of stress fields in the general case of forming thin shells is quite difficult task, so in this paper we consider the simplest case of forming - plastic bending moment of a two-layer sheet package.

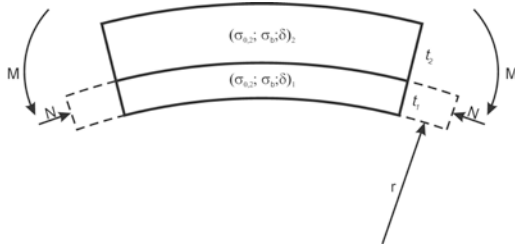


Fig.1 Scheme of a two-layer sheet bending package

Let us consider the bending moment of a wide double-layer sheet package, consisting from the internal, molybdenum layer with thick t_1 and an outer, steel with thickness t_2 . We assume that the change in curvature of a median surface of the package sheet comes under plane strain conditions. Replace the influence of the outer layer of radial contact stress and consider plastic equilibrium Mo layer under the combined action of bending moment and radial contact stress (**Fig.2**)

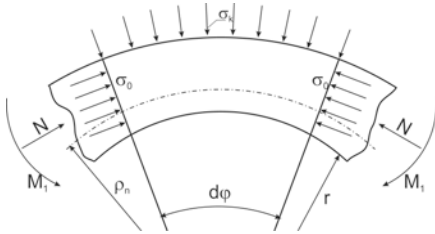


Fig.2 Scheme of bending moment and longitudinal load

From **Figure 2**, it follows that to comply with the equilibrium conditions in the presence of radial contact stress is necessary that the thickness of the sheet in the longitudinal direction acted some compressive load N , the effect of which should be reflected in the stress redistribution in thickness and on the location of the neutral surface. When the proportion of the thickness in which circumferential stresses have a negative value should be increased, and more than half the thickness of the sheet, and the neutral surface to be displaced towards the outer surface, thus reducing the tensile deformation of the outer circumferential layer and an increase in allowable degree of deformation.

A longitudinal compressive load N can be expressed in terms of a conditional, the average thickness, compressive stress σ_0 , which is established on the basis of the relationship between the radial contact stress and conditional $\sigma_k = \sigma_0 t_1 / R$.

Analysis of the stress state will spend without the influence of shear (transverse) forces on the field of stress and strain hardening take into account the power dependence of yield stress of the equivalent strain [5,7].

$$\sigma_s = A \varepsilon^n, \quad (1)$$

where A and n - strain hardening parameters, depending on the mechanical properties of deformable material and determined from the relation $n = \ln(1 + \delta)$, $A = \sigma_b e^n n^{-n}$ (σ_b and δ accordingly, the tensile strength and relatively uniform elongation at a linear tensile of deformed material) [5,7,8].

Following [5], we assume that the deformations in (1) may be replaced approximately true (logarithmic) tensile strains

(compression) in the circumferential direction $\varepsilon_\theta = \pm \ln(\rho / \rho_n)$ (ρ and ρ_n - accordingly, curvature radii of the element and the neutral surface).

The equilibrium equation in the polar coordinate system, taking into account equation plasticity hypothesis of constant maximum tangential stresses ($\sigma_\rho - \sigma_\theta = \pm \sigma_s$) and the relation (1) is given to the differential equation

$$\rho \frac{d\sigma_\rho}{d\rho} = \pm A \left(\pm \ln \frac{\rho}{\rho_n} \right)^n, \quad (2)$$

where, the plus sign refers to the area of circumferential tensile and the minus sign - to the compression of circumferential zone. The integration of equation (2) using the boundary conditions at which $\sigma_\rho = \sigma_k$ when $\rho = R$ and $\sigma_\rho = 0$ when $\rho = r$, leads to dependency for the extension zone

$$\left. \begin{aligned} \sigma_\rho &= -\frac{A}{n+1} \left[\left(\ln \frac{R}{\rho_n} \right)^{n+1} - \left(\ln \frac{\rho}{\rho_n} \right)^{n+1} \right] - \sigma_k, \\ \sigma_\theta &= A \left(\ln \frac{\rho}{\rho_n} \right)^n - \frac{A}{n+1} \left[\left(\ln \frac{R}{\rho_n} \right)^{n+1} - \left(\ln \frac{\rho}{\rho_n} \right)^{n+1} \right] - \sigma_k \end{aligned} \right\}, \quad (3)$$

for the compression zone

$$\left. \begin{aligned} \sigma_\rho &= -\frac{A}{n+1} \left[\left(\ln \frac{\rho_n}{r} \right)^{n+1} - \left(\ln \frac{\rho_n}{\rho} \right)^{n+1} \right], \\ \sigma_\theta &= -A \left[\left(\ln \frac{\rho_n}{\rho} \right)^n + \frac{A}{n+1} \left(\ln \frac{\rho_n}{r} \right)^{n+1} - \frac{A}{n+1} \left(\ln \frac{\rho_n}{\rho} \right)^{n+1} \right] \end{aligned} \right\}. \quad (4)$$

Depending (3) and (4) characterize the change of stress σ_ρ and σ_θ stress in thickness of bending of molybdenum sheet with the influence of strain-hardening and radial contact stresses.

3. Radius of the neutral surface

In [5] it is shown that the bending moment of a wide band radius of the neutral surface, defined on the basis of various models of deformable material (perfect rigid-plastic, ideal rigid-plastic with linear and power hardening) is expressed identical dependence $\rho_n = (Rr)^{1/2}$ on the basis of what it is made reasonably be concluded that the actual conditions of bending strain hardening will not have a significant effect on the position of the radius of the neutral surface.

From the continuity condition of the radial stresses on the neutral surface when $\rho = \rho_n$, the first dependency (3) and (4) we have

$$\left(\ln \frac{R}{\rho_n} \right)^{n+1} + (1+n) \frac{\sigma_k}{A} = \left(\ln \frac{\rho_n}{r} \right)^{n+1}. \quad (5)$$

The fractionality of degrees obtained equation can not convert it and get radius of dependence for the neutral surface in the form of analytical function. However, numerical analysis when $1 \leq r/t_1 \leq 2$ and $n \leq 0,05$ in that the assumption of equality of the type $(\ln R/\rho)^{n+1} \approx (n+1) \ln(R/\rho)$ **error in** (5) does not exceed 10%, on the basis of which it becomes possible to obtain approximate dependence

$$\rho_n \approx \left(Rr \exp \frac{\sigma_k}{A} \right)^{1/2}. \quad (6)$$

To evaluate the possible value of the radial contact stresses assume that sheet dual layer package conventionally is deformed as a whole, and the interface of layers coincides with the neutral surface. Assume further that the limits of strength of materials of layers are approximately the same, and the strain hardening exponent for the outer layer are much larger than the inner layer $n_2/n_1 \gg 1$. Solving a similar problem for conditionally deformable sheet package, you can install the distribution of stress dependencies (when $\sigma_k = 0$) similar to (3) and (4).

Figure 3 shows the distribution of stresses during bending of double-layer sheet package, conventionally deformed as a whole, with a relative minimum bend radius equal to 2.5. When determining the values of stress it is assumed that the respective values of the limits strength $\sigma_{b_1} = \sigma_{b_2} = 690 \text{ MPa}$, and the exponent in equation (1) for the outer layer is 0.2. Relevant parameters of strain hardening calculated by the method described in the previous section.

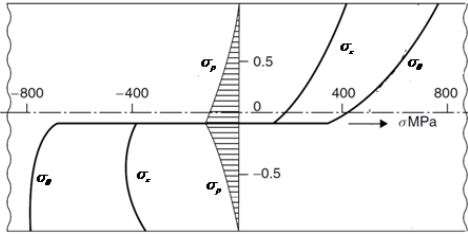


Fig. 3. Stress distribution in bending double-layer sheet package

Stress distribution shown in Figure 3, is somewhat arbitrary, both in connection with (both due to) the accepted model of deformable material, and assumptions about the finite size of the sheet bending under plane strain conditions. These conventions discussed in detail in [9], in connection with which the present paper these issues are not discussed.

The highest relative magnitude (value) of the radial compressive stress on the neutral surface of the deformable sheet shareware package is defined as follows

$$\frac{\sigma_{\rho \max}}{\sigma_b} = - \frac{e^{n_2}}{n_2^{n_2} (1+n_2)} \left[\ln \left(1 + \frac{t_2}{R} \right) \right]^{1+n_2}. \quad (7)$$

From the analysis of the expression (7) that the highest value of the radial compressive stress depends on the parameter of strain hardening of the outer layer, the relative radius of bending, and has an extremum. Figure 4 shows the graphs of this dependence for different values of the relative radius of the bending, from which it follows that a decrease in the relative location of the bending radius of the extremum is shifted to larger values of the parameter of strain hardening.

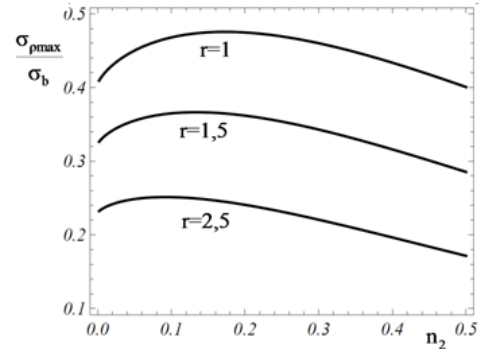


Fig.4 Change of the dimensionless radial stresses depending on the parameter n_2

When calculating the numerical values indicated $\ln(1+t_2/R) = h(t)$, and taken values of three inner bending radius r mm (1, 1.5 and 2), with respective thicknesses of the of layers $t_1 = 0,6$ mm and $t_2 = 0,8$ mm.

On the basis of previously received equality $|\sigma_{\rho \max}| = |\sigma_k|$, substituting (7) into (6), we obtain the analytical dependence for the radius of the neutral surface (the subscripts 1 and 2 refer respectively to the inner and outer layers).

$$\rho_n(\sigma_k) = \left\{ Rr \exp \frac{A_2}{(1+n_2)A_1} \left[\ln \left(1 + \frac{t_2}{R} \right) \right]^{1+n_2} \right\}^{1/2}. \quad (8)$$

4. Results and discussion

The value of relative displacement of the neutral surface radius in the direction of the outer surface and the corresponding increase in the allowable degree of deformation (decrease of the radius of curvature) can be evaluated by comparing its finite value with the initial $\Delta\rho_n/\rho_n$ ($\Delta\rho_n = \rho_n(\sigma_k) - \rho_n$)

$$\frac{\Delta\rho_n}{\rho_n} = \left\{ \exp \frac{e^{n_2} n_1^{n_1}}{(1+n_2)e^{n_1} n_2^{n_2}} \left[\ln \left(1 + \frac{t_2}{R} \right) \right]^{1+n_2} \right\}^{1/2} - 1. \quad (9)$$

Such an assessment is made possible on the basis of sound in the previous section, the independence of the radius the neutral surface of the accepted model of deformable material and assumptions of equality $\sigma_{b_1} = \sigma_{b_2}$.

Figure 5 presents a three-dimensional graph of the relative displacement of the radius of the neutral surface of appropriate indicators hardening n_1 and n_2 layers at a predetermined radius of bending $h(t_2) \approx 0,405$, ($r = 1$ mm, $t_1 = 0,6$ mm, $t_2 = 0,8$ mm).

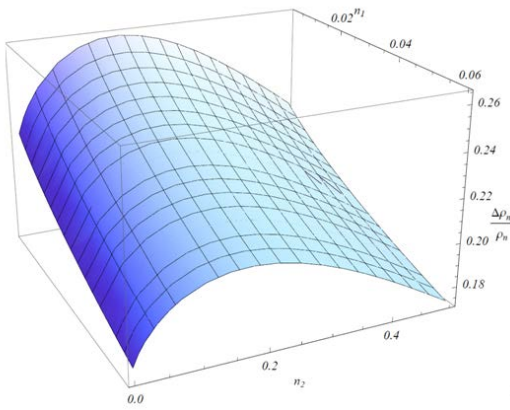


Fig. 5 Change the radius of the neutral surface depending on the parameters n_1 and n_2

From the analysis of the relationship (9) that it has an extremum that does not depend on the index n_1 . Equating the result of differentiating (9) to zero we obtain

$$(1 + n_2) \ln n_2 - (1 + n_2) h(t_2) + 1 = 0, \quad (10)$$

where indicated $\ln(1 + t_2/R) = h(t_2)$. According to equation (10), given the $h(t_2)$ determined value of the index n_2 at which the dependence (9) takes extreme value ($n_2 \approx 0,178$)

Our results imply that the outer layer is most appropriate according to GOST 1542-71, the use of sheet alloy structural steel general purpose mark 25ХГСА 0.8 mm thickness ($\sigma_b \approx 690 \text{ MPa}$, $\delta \approx 0,18$).

Obviously the relative values of greatest compressive stresses and radial displacement of the radius of the neutral surface in bending actual conditions due to the relative slipping layers are significantly less than the values computed dependencies (8) and (9). For this reason, slightly increase the permissible value of the relative bending radius.

Despite some arbitrariness of the analysis, the results obtained allow us to estimate the influence of different parameters and to some extent control the process of forming.

Conclusions

1. Analysis of a two-layer sheet bending package under plane strain taking into account the strain hardening in the absence of relative slippage.
2. Analytical dependences of the limiting values of the radial compressive stresses acting on the interface between the layers and the relative displacement of the radius of the neutral surface of the inner layer.
3. Shown that these dependences have extremums depending both on the relative bending radius, and the parameters of strain hardening of the outer layer.
4. Quantitative evaluation set relative ability to reduce the bending radius of the molybdenum sheet, and on the basis of the choices displayed for the outer layer of the material with predetermined mechanical properties.

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DETERMINATION OF THE ENERGY NECESSARY FOR CARS BODY DEFORMATION BY APPLICATION OF THE NHTSA STIFFNESS COEFFICIENT

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Abstract: One of the main problems in appraiser work is to determine speed the car immediately before the accident, basing only on the collision of two vehicles or collision of a car with an obstacle. One of the research directions providing a solution to this problem are analysis of the energy necessary to cause permanent deformation. Since the early '70s used was the method based on the analysis of the permanent deformation depth, that are defined as proportional to the force causing deformation.

Keywords: NHTSA, DEFORMATION ZONE, PERMANENT DEFORMATION.

1. Introduction

One of the main problems in appraiser work is to determine speed the car immediately before the accident, basing only on the collision of two vehicles or collision of a car with an obstacle. One of the research directions providing a solution to this problem are analysis of the energy necessary to cause permanent deformation. Since the early '70s used was the method based on the analysis of the permanent deformation depth, that are defined as proportional to the force causing deformation and approximated by the following relationship.

$$W_{def} = \frac{1}{2} \cdot b \cdot h \cdot k \cdot f_{tr}^2 \quad (1)$$

where:

b - width of deformation zone [m],
h - height of deformation zone [m],
f_{tr} - depth of deformation zone [m],
k - stiffness coefficient or energy-consumption [N/mm²].

In literature [1] the following ranges of the stiffness coefficient can be found:

- $k=(9,0 - 11) \cdot 10^5$ N/m²m, when the vehicle strength structure was broken as a result of the deformations
- $k=(2,0 - 4,0) \cdot 10^5$ N/m²m, when the deformations are located only in the skin plate elements
- when the vehicle strength structure was broken:
 - $k=(13,5 - 22,6) \cdot 10^5$ N/m²m for small vehicles,
 - $k=(9,1 - 13,5) \cdot 10^5$ N/m²m for medium cars,
 - $k=(5,2 - 7,2) \cdot 10^5$ N/m²m for big car.

2. Crash test model by Campbell

Essential part in ensuring vehicle safety are the crash tests. The basic type of investigation is vehicle frontal crash test with stiff roadblock. During this test, the crash velocity is measured. Crash test model proposed by Campbell [2] introduces the proportionality between deformations and speed collision.

$$V = b_0 + b_1 \cdot C \quad (2)$$

where:

V – collision velocity when the deformation C occurs [m/s],
b₀ – limit velocity where the permanent deformation do not occur [m/s],
b₁ – characteristic inclination [m/ms],
C – depth of the permanent deformation [m].

On beginning of 70's R. McHenry [3] proposed linear dependence between a force acting on the car and permanent

deformations appearing as its acting result. According to McHenry it can be illustrated by equation (3) and graphic (figure 1).

$$F = A + B \cdot C \quad (3)$$

where:

F - unit force of deformation,

A - unit boundary force, after transformation of which it plastic deformation occurs,

B - directional coefficient of linear body model deformation, defining its lengthwise unit stiffness,

C - Permanent body deformation.

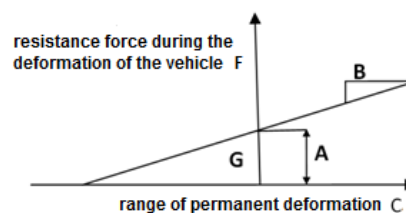


Fig. 1 Stiffness model of the vehicle during a collision by McHenry.

The value of the C coefficient is obtained from the measurements of deformation's depth. In the crash test used was procedure of measuring its value in *n* points, that are evenly disposed on length of the indentation, as shown on the figure 2.

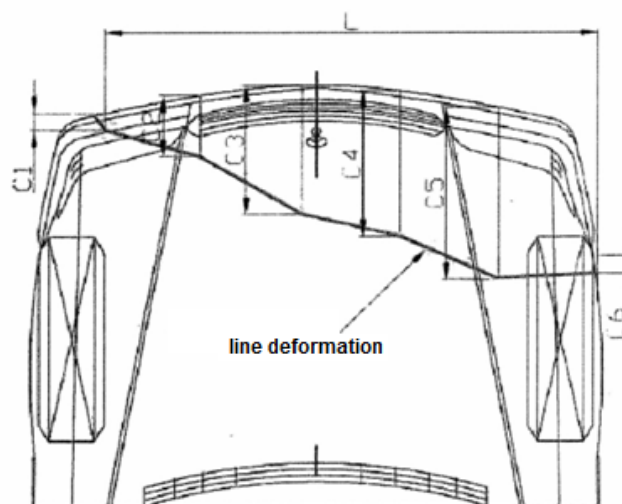


Fig. 2 Method of depth dimensioning of the permanent deformation.

3. Depth of deformation

The effective depth of deformation is defined as mean value on the length of indent and can be presented by the following relationship:

$$C_{sr} = \frac{\frac{C_1 + C_n}{2} + \sum_{i=2}^{n-1} C_i}{n-1} \quad (4)$$

The energy which causes permanent deformation on length L is expressed in the following way:

$$\int_0^L \int_0^C (A + B \cdot C) dcdll = \int_0^L (A \cdot C + B \cdot \frac{C^2}{2} + G) dl \quad (5)$$

where:

G - integration constant equal,

$$G = \frac{A^2}{2 \cdot B} \quad (6)$$

L - width of the deformation area [cm].

After integration we obtain:

$$E = A \int_0^L C dl + B \int_0^L \frac{C^2}{2} dl + \frac{A^2}{2 \cdot B} \cdot L \quad (7)$$

Individual modules of the equation (7) can be defined in the following way:

- $\int_0^L C dl$ as indent projection's surface,
- $\int_0^L \frac{C^2}{2} dl$ as moment of the first degree of projection of the indent surface in relation to primary body profile (product of surface of projection and distance of its center of gravity from the first outline of the car body).

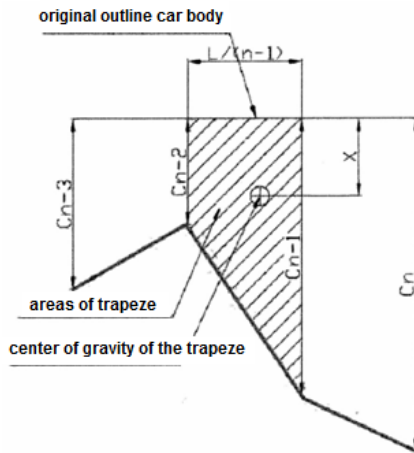


Fig. 3 Method of the determination of the deformed surface gravity center location.

As show in figure 3, it is assumed that all deformation areas have shape of trapeze. The surface of deformation unit is equal:

$$P = \frac{L}{n-1} \cdot \frac{C_{n-2} + C_{n-1}}{2} \quad (8)$$

while the distance of centre of gravity from primary profile of body is obtained from:

$$X = \frac{C_{n-2}^2 + C_{n-2} \cdot C_{n-1} + C_{n-1}^2}{3 \cdot (C_{n-2} + C_{n-1})} \quad (9)$$

The energy necessary to cause the indentation of the surface L and of average depth C_{sr} can be expressed by:

$$E = \frac{L}{n-1} \cdot \left(\frac{A \cdot \alpha_1}{2} + \frac{B \cdot \beta}{2} + (n-1) \cdot G \right) \quad (10)$$

where:

$$\alpha = C_1 + C_n + 2 \cdot \sum_{i=2}^{n-1} C_i$$

$$\beta = C_1^2 + C_n^2 + 2 \cdot \sum_{i=2}^{n-1} C_i^2 + \sum_{i=1}^{n-1} C_i \cdot C_{i+1}$$

In case of symmetrical deformation, the equation (10) takes the form:

$$E = \left[A \cdot C + \left(\frac{B}{2} \right) \cdot C^2 + \frac{A^2}{2 \cdot B} \right] \cdot L \quad (11)$$

If then, the energy absorbed by the vehicle for the deformation work is equivalent to the car kinetic energy, that is lost during the crash, then we can to assume, that :

$$\frac{m \cdot (\Delta V)^2}{2} = E \quad (12)$$

$$(\Delta V)^2 = \left(\frac{2 \cdot A \cdot L}{m} \right) \cdot C + \left(\frac{B \cdot L}{m} \right) \cdot C^2 + \frac{A^2 \cdot L}{m \cdot B} \quad (13)$$

$$\Delta V = \sqrt{\frac{B \cdot L}{m} \cdot \left(C + \frac{A}{B} \right)} \quad (14)$$

Therefore, the conclusion is that the border speed of the crash that do not result in permanent deformation and for which the $C = 0$, has the form

$$\Delta V_{gr} = b_0 = A \cdot \sqrt{\frac{L}{B \cdot m}} \quad (15)$$

Conducted by NHTSA investigations show, that border speed b_0 contains in range $b_0 = 2 \div 4$ m/s = 7,2 ÷ 14,4 km/h.

Figure 4 presents graph which is speed characteristic in function of depth of body deformation (coefficients b_0 and b_1 correspond to coefficients defined by Campbell).

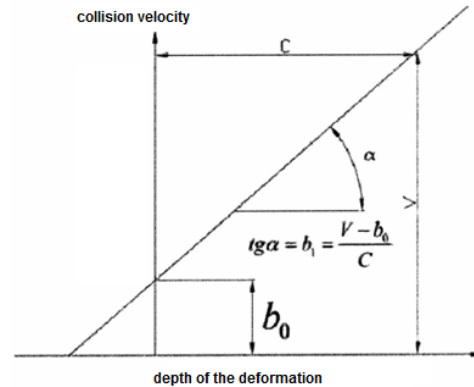


Fig. 4. Campbell dependence of the velocity of the body at the moment of the crash form the depth of the deformation.

Coefficients A , B and G are defined during cars crashes test in such way, that the surface of barrier in which the vehicle hits is perpendicular to longitudinal or transverse vehicle axis. In the case of diagonal crashes, tangent component of friction force contained among barrier and the vehicle increases the stiffness efficiency in direction of main impulse of force.

Coefficients A and B can be found in literature [4,5,6]. For specific models of vehicles, it is possible to find in the NHTSA reports the speed of crash during tests, the mass of cars as well as depth of deformation in six evenly disposed points of the deformation area obtained. These reports are accessible in [7]. It is the database including information about crash tests and contains

the vehicles ranked according to manufacturer, type, model and year of production. In enclosed tables of included are values of the speed answering the zero permanent deformation and knowing both mass of the vehicle and length of the deformation area, it is possible to calculate:

$$b_0 = V_0 \quad (16)$$

$$b_1 = \frac{V_{zd} - V_0}{C_{sr}} \quad (17)$$

The next step is calculate the coefficients:

$$A = \frac{mb_0b_1}{L} \quad (18)$$

$$B = \frac{mb_1^2}{L} \quad (19)$$

In the database included are also the photos of the tested vehicles that can be useful for estimation of speed of crash.

4. Conclusions

Basing on the presented dependences and methods it is possible for the expert to accurately estimate the energy absorbed during permanent deformation of the vehicle.

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DETECTION OF THE FATIGUE OPERATING LIFE OF THE WELDED JOINT OF AN ALUMINIUM ALMGSi07 ALLOY AT UNIAXIAL TORSION STRESS OF THE SPECIMENS

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Abstract:

The paper deals with an experimental measurement of the fatigue operating life of a specimen from the aluminium AlMgSi07 alloy on a built testing equipment, into which the welded joint was implemented. An importance of the measurement is in a comparison of achieved results with the same material, which was not subjected to welding. The result of the research is verification of the influence of welding on the fatigue operating life of the concrete material at torsion stress.

KEYWORDS: MATERIAL, FATIGUE, STRESS, ALLOY

1. Introduction

If the material is exposed to the action of the variable forces, respectively to the strain of the supercritical size, after some time there is a fracture, which is the result of the complicated processes in the structure of the material. The maximal stress value is so low that the material endures its static effect without any damage. Gradual eroding has a cumulative character and is based on creation and spreading of cohesion failures. From the time of observation of the fatigue failure, like a basic fatigue characteristics is used a Wöhler's curve, which shows dependence between stress amplitude of simple symmetric cyclic loading and the number of cycles to the fracture. Wohler's curve is not a suitable material characteristic for the field of low cyclic fatigue. In the first place, it is because of big plastic strains, which leads to the significant difference between values of contractual stress σ_a determined on the basis of the initial cross-section, and of the real stress in the sample, in the second place, because of small slope of the curve in the studied area. From the practical point of view of fatigue strength calculations is more appropriate characteristics in the field of low cycle fatigue dependence of the amplitude of the total strain γ_{xy} from the number of cycles to the fracture N_f , which is called Manson – Coffin's curve, fig.1.

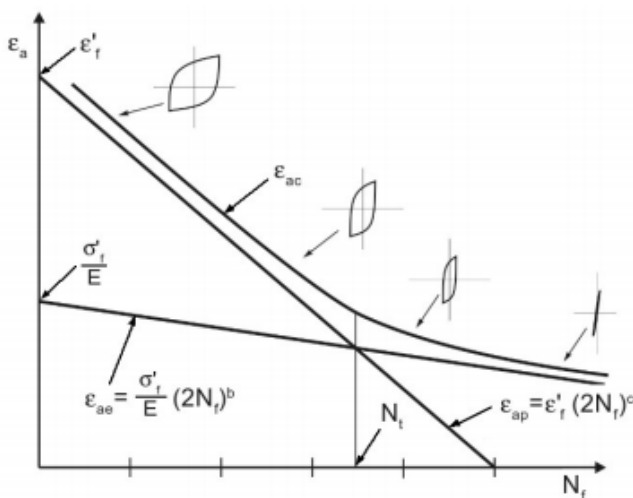


Fig. 1: Manson – Coffin's curve

2. The experimental material

The expansive development of the materials helps to the fact that constantly it is more used materials on the other basis than the iron. The example is replacement of the heavy constructions by the lighter ones, from the materials with lower density. It is for instance aluminium and its alloys, which have very favourable ratio of the strength to density and from the terms of the output, aluminium is between non-iron metals on the top in the world. Because of this reason a commercial aluminium AlMgSi07 alloy with a normalised chemical composition was chosen like an experimental material for the carrying out of the fatigue testing. Testing material was supplied in the shape of rod of circular cross section with a diameter 10mm and length 1520mm. Then it was divided on a bend saw BOMAR 1300 and achieved measure of semi-finished product for the production of the testing sample with the length 150mm. The geometry of the sample is precisely defined (fig. 2) and results from the constructional solution to the fatigue testing equipment, in which the testing will be carried out. Because of this it was conveniently used a semi-finished product with a diameter of the 10mm when it was not needed a further treatment of the diameter of the sample.

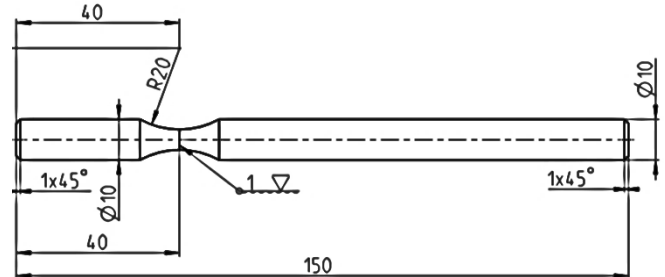


Fig. 2: The geometry of the testing sample for the fatigue testing status

3. Testing equipment

The experimental detection of the fatigue characteristics of the aluminium AlMgSi07 alloy was performed at a workplace for measurement of the fatigue on the fatigue testing equipment (fig. 3). Considering the character of the testing equipment, the loading was realized in the controlled amplitude of the strain with the zero medium component of the strain.

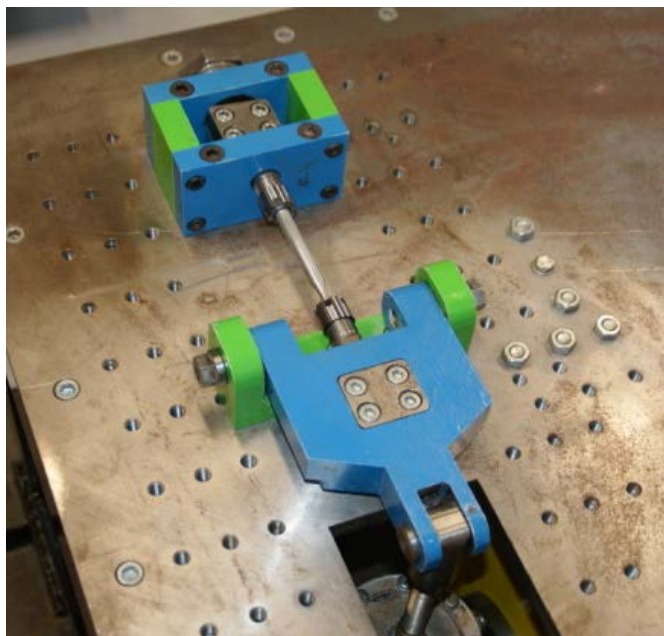


Fig. 3: Made fatigue testing equipment

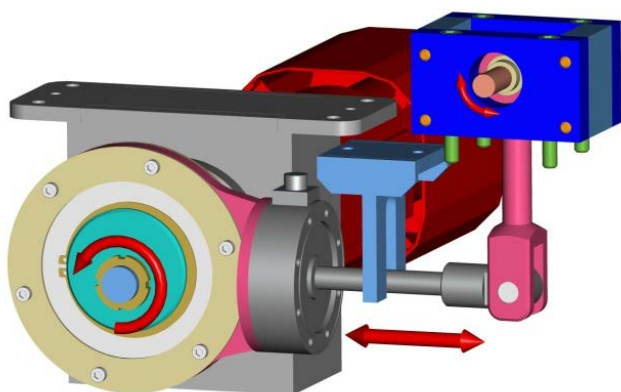


Fig. 4: A model of stress mechanism for the cyclic torsion stress of the sample

Loading mechanism for the cyclic torsion stress of the sample (fig. 4) is driven by synchronous servomotor, whose speed is regulated by the frequency changer up to 100 Hz. The rotational movement of the servomotor is transmitted by the excenter through the force transducer on the crank, which is acting linear motion.

4. Measurement of the fatigue life

Input characteristics of the material were experimentally detected from the tension diagram of the welded material AlMgSi07 (fig. 5).

To be possible to immediately express the size of the amplitude of the relative strain γ_{ac} for the each stress level, it was needed to create calibration dependences (fig. 6, 7) of the relative strain from the number of excenter teeth, whose change of the relative position by rotation, size of the displacement is set. The values of the strain were gained by the FEM analyses at all stress levels in the place of concentration of the stresses by the finite element method.

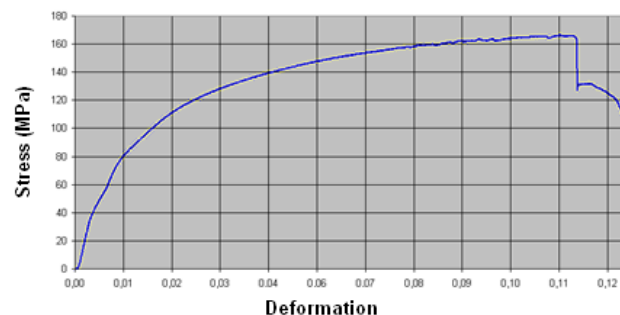


Fig. 5: Diagram of the static tensile test performed on the welded material AlMgSi07

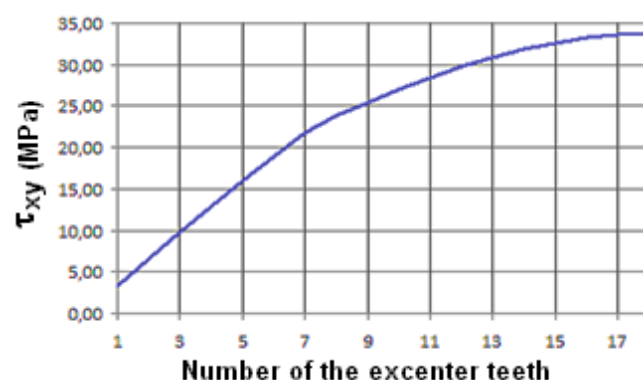


Fig. 6: Values of the stress at the torsion stress of the welded samples, $e = 4$ mm, material AlMgSi07

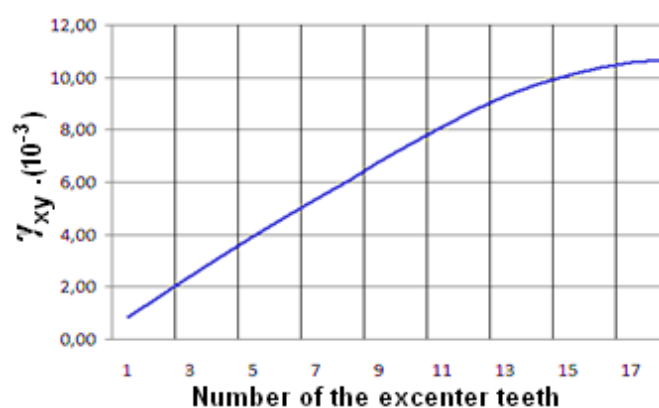


Fig. 7: Values of the strain at the torsion stress of the welded samples, $e = 4$ mm, material AlMgSi07

By the analysis of the state of stress, it was proved the veracity of the assumption that the largest stress at the fatigue testing will be in the area of sample contraction (fig. 8). The analysis was performed in the computing program ADINA. After making of the strain dependence of the testing sample, it was realised the measurement of the fatigue life of the material on the welded testing samples, which were cyclic torsion stressed. Size of the load was changing.

Measuring numbers of cycles for each loading were processed into Manson-Coffin's curve $\gamma_{ac} - \log N_f$ (fig. 9) for uniaxial torsion stress.

From the graph, it is liquet that with gradual decreasing of the amplitude of the strain there is an increase of cycles to fracture. Behind the limit of 105 cycles we already talk about high cycle fatigue. Above this value another criterions are used for comparison with an experiment than at low cycle fatigue. Object of research is field of extremely stressed weld joints, so low cycle fatigue.

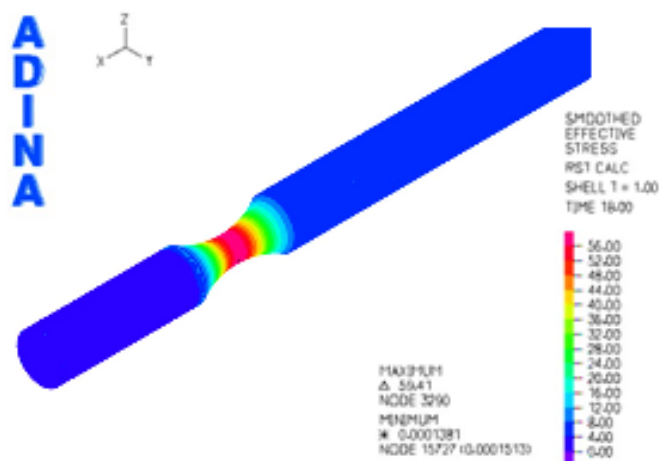


Fig. 8: Distribution of effective stress by HMM theory in the sample from material AlMgSi07, which was torsion stressed, at maximal strain

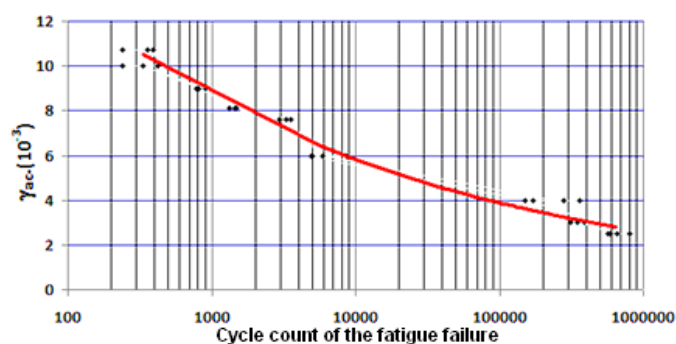


Fig. 9: Manson-Coffin's curve of the welded material AlMgSi07 stressed by cyclic torsion

5. Comparison of results

Because of verification of accuracy of the measurement of the testing equipment it is appropriate to perform a comparison of measuring results with fatigue criterions. Such a comparison is possible with usage of program fatigue Calculator. Fatigue Calculator is program from the internet page Fatigue, where it is possible to easy and quickly makes an estimation of the fatigue life. By this programme we can find out a number of cycles, at which a fatigue fracture happen at a low cycle and also at a high cycle fatigue. Because of the fact, that we have available a possibility to compare the experimentally gained results with the computing, we choose the same conditions of the method of loading of the testing rod – sinusoidal cyclic stress. Because of simply comparison, all found numbers of cycles to the fracture at individual loading level were processed into a graph (fig. 10). Values of the stresses, respectively strain in the required units are inserted to the programme from the calculated tensor and it is for τ_{xx} . It is worked with the symmetric frequency of the loading cycles 30 Hz. After starting a calculation of the life for the field of the low cycle fatigue, Fatigue Calculator displays calculated values of the number of cycles to the fracture with different models of damage, and they are Nf Fatemi-Socie (F-S), Nf SWT, Nf Brown-Miller (B – M).

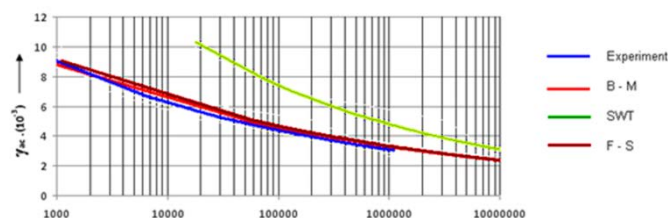


Fig. 10: Comparison of Manson-Coffin's curves of chosen fatigue criterions for the valuation of low cycle uniaxial fatigue and from experimentally achieved values at torsion stress of samples

6. Conclusion

In the paper, there is described an issue of the measurement of the low cycle uniaxial fatigue of the welded material AlMgSi07 on the built testing equipment in the torsion. Gained results will be served for the comparison with the results from the experiments for the same material without the usage of welding technology.

Acknowledgements

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A RESEARCH ABOUT COEFFICIENT AND MOMENT OF FRICTION DURING INTERACTION OF DEPOSITED LAYERS OBTAINED THROUGH INCREASED WIRE ELECTRODE VIBRATING FREQUENCY

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Abstract: The article presenting a results about dynamics of moment and coefficient of friction change is established by experimental research of applied layers upon steel parts through vibrating gas metal arc overlaying process with increased wire electrode vibrating frequency. It was established that increasing of the vibrating frequency at rate of 150 Hz leading to significantly shorter time for interaction of both surfaces to obtain a process of established wear.

Keywords: running-in, coefficient of friction, iron and steel parts; agricultural machinery; overlaying

Introduction

The interaction of frictional surfaces appears to be one of the most important stages of the process of reconditioning causing a significant influence to the durability of the automotive and agricultural equipment. Basically, it is a non-stable process of alteration of tribological characteristics, geometrical parameters, physical and chemical properties of the applied layers during friction (Garkunov, 2001). One of the main tribological characteristics is the moment of friction because of its intensive alteration during friction and it is an important criteria at point of process's evaluation.

The aim of the research is an establishing of alteration of the tribological characteristics – moment and coefficient of friction obtained during process of friction of applied layers within vibrating gas arc overlaying process.

Exposition

A frictional pair “roller-sector” with an appropriate geometrical and physical parameters is accepted as a physical model of the real frictional pair “shaft-bearing”. The parameters of the frictional pair are determined by statistical analysis about reconditioning of worn parts from automotive and agricultural equipment (Tonchev and Stanev, 1979).

The material of frictional pair is accepted as main variable of the process of interaction during sliding friction as there were three different frictional rollers – a roller made of steel grade 45 without applied layer; a roller made of steel grade 45 with applied layer of 30HGSA through wire electrode's vibrating frequency at rate of 47,6 Hz and rolled made of steel grade 45 with applied layer of 30HGSA through wire electrode's vibrating frequency at rate of 150 Hz. The material of specimens is chosen in relation to the most used materials for details of agricultural and automotive equipment – steel grade 45 and lead bronze BO30.

The treated rollers of Steel 45 has a deposited layers with 30HGSA wire electrode with diameters of 1,6 mm within vibrating frequency of 150Hz quenched by conducted current at high frequency. The overlaying process is done through ENTON 60 overlaying apparatus and modified overlaying apparatus. The working regime parameters were accepted as follow – wire electrode's vibrating frequency at rate of 150 Hz; amplitude at rate of 1,7 mm; voltage of 20V; welding current of 180A, welding speed of 2,51 m/min; wire electrode feeding rate of 2,0 m/min; welding pace rate of 3,25 mm; wire electrode outlet rate of 15 mm; shielding gas rate of 10 l/min [TODOROV and NIKOLOV, 2012].

The standard and overlayed rollers were mechanically and thermally treated. They were quenched and retorted to hardness of 55±2 HRC (HV5 = 6100±400 MPa) and grinded to final rate. The rollers are with diameter of 50+0,02 mm; misalignment and unperpendicularity not more than 0,01 mm; width of 12+0,05 mm;

thickness of the applied layer of 0,5 mm measured on the radius; roughness Ra = 0,28...0,32 µm for overlayed with DUR 500 roller and Ra = 0,55...0,58 µm for overlayed with 30 HGSA roller. The weight of the rollers is varying between 160 and 170g.

The sectors are made of steel with applied anti-frictional layer of leaden bronze BO-30 hardened at rate of HV5 = 490±50 MPa and thickness of 0,1...0,5 mm measured along the radius. The length of the frictional area of the sector is 20 mm with a width of 10+0,05 mm. The surface area of contact is 2 cm² and the weight of the sectors is varying between 15 and 20 g. The outer surface of the sectors is grinded to diameter of 70+0,1 mm as well as the inner surface - to diameter of 50+0,06 mm. The roughness of the sectors is measured at rate of Ra = 2,0...2,8 µm. The radial distance between the roller and sector is at rate of 0,04...0,05 mm.

The moment of friction is accepted as main criteria for evaluation of the processes of interaction and wear as three of its components are established during experiments – a moment of friction at the end of loading; a moment of friction during interaction of the frictional surfaces and moment of friction at the end of the experiments. A coefficient of friction as secondary criteria for evaluation is accepted.

The examination is taken on a standing machine SMC-2 equipped with an appropriate systems and devices to ensure working conditions similar to the operational. A cooling oil camera with a capacity of 150 ml is used to control the oil temperature at rate of 40°C that is typical for “cold” interaction of new or reconditioned engines as well as system for oil mixing and magnetic cleaning of wear products. The examination is taken on “roller-sector” scheme within following working regime: roller turnovers cycle rate of 540 min⁻¹; relative sliding velocity at rate of 85 m/min; loading of the frictional pair at rate of 1 MPa/min; pressing force of sectors to the rollers at rate of 100 daN; pressure at rate of 5 MPa and tribological characteristic PV at rate of 425 MPa..m/min.

The alteration of the moment of friction is controlled and recorded continuously by inductive sensor placed between spindle and reducer. The signal is being transformed by controlling plate that measuring the difference in the voltage value at both coils of the sensor. The registration and recording of the signal is done by multifunctional device NI-USB 6210 connected to personal computer. The results of the experiments are being recorder in *.xls document as its further calculation is done through Microsoft Office Excel.. The moment of friction is being recorded continuously and presented graphically and digitally on the screen.

The results of the experimental research are presented graphically at figures 1 to 4 which showing the alteration of the moment and coefficient of friction during process of interaction and wear of the frictional surfaces. The interaction of frictional surfaces is characterized by moment of friction at the end of loading as its value determines the ability of the surface against excessive wear and blocking. The alteration of this parameter is shown at fig.1.

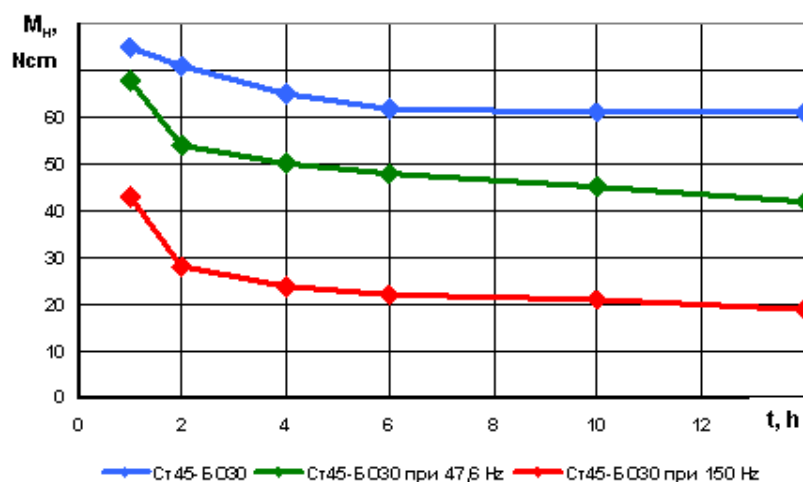


Figure 1 Dynamics of alteration of the moment of friction at the end of loading during interaction of standard and overlaid frictional pairs within sliding friction conditions.

As it is shown at fig.1, the initial alteration of the moment of friction has a steep character during the first two hours of the examination for both overlaid pairs explained by lower roughness of their surfaces compared to the standard one. Also, the moment of friction reduces its value almost double that is criteria about fluent process of interaction and established wear of the frictional surfaces. The overlaid pairs are characterized by lower values of the moment of friction which is a sign about better interaction with the material of the sector. The lowest value of the moment for the pair

overlaid at rate of 150 Hz showing that increasing of wire electrode's vibrating frequency decreases the moment of friction nearly 2,5 times. Therefore, the coefficient of friction decreasing as well as possibility of excessive wear on the frictional surfaces. The moment of active interaction is basically a difference between values of both – moment of friction at the end of loading M_H and moment of friction at the end of examination M_K . Its alteration is shown at fig.2.

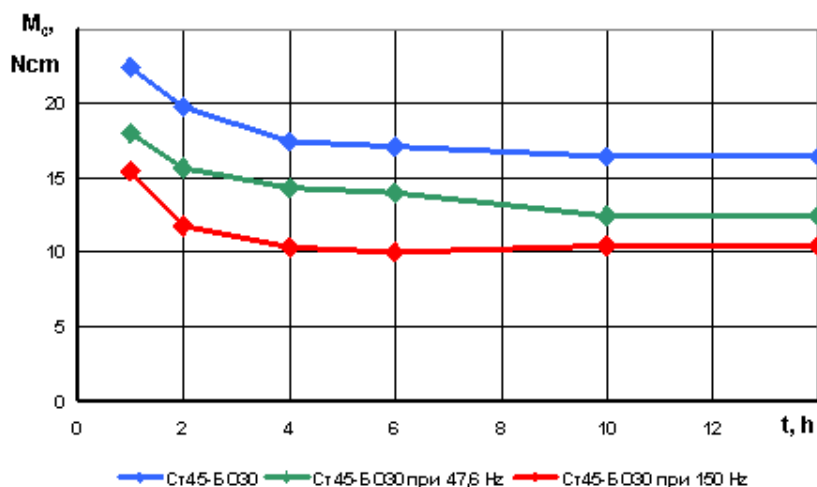


Figure 2 Dynamics of alteration of moment of active interaction during interaction of standard and overlaid frictional pairs within sliding friction conditions.

As it is shown at fig.2 all of the pairs has similar character of alteration as it is most intensive within the first few hours. The standard pair has higher alteration of the moment of friction during interaction in comparison to both overlaid pairs as its interaction takes around 10 hrs. Obviously, the process of interaction of the

pair overlaid with wire electrode vibrating frequency at rate of 150 Hz is done within first 4 hrs followed by process of established wear judging by insignificant alteration of the moment of active interaction from 4-th till 14-th hours – at rates of 10,2 to 10,5 Nm.

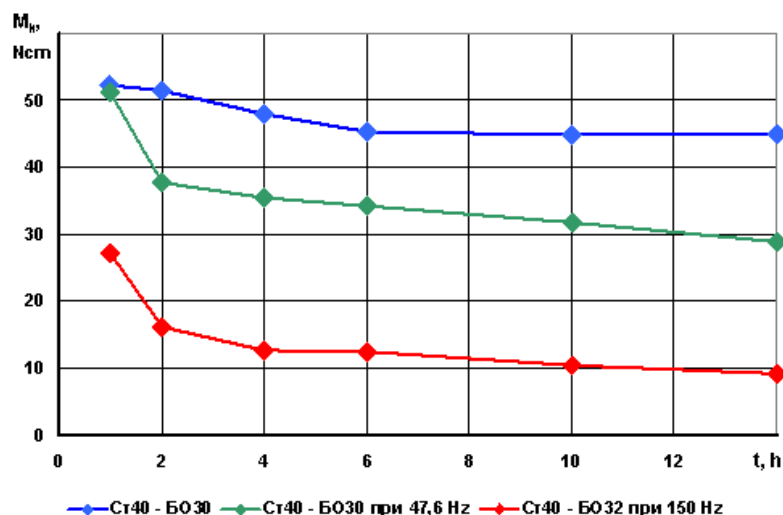


Figure 3 Dynamics of alteration of moment of friction at the end of examination during interaction of standard and overlaid frictional pairs within sliding friction conditions.

The alteration of the moment of friction at the end of the examination M_k is shown at fig.3 as there is a trend to its decreasing for all 3 pairs. As it is shown, the moment of friction for the pair overlaid at a rate of 150 Hz decreases steeply within the first 4 hrs followed by insignificant alteration to the end of examination at rates of 12,3 to 9,8 Nm. The lower values of the moment of friction at the end of examination compared to the ones for the moment of

friction at the end of loading could be considered as criteria about the process of interaction because of its fluent transition into process of established wear.

The coefficient of friction f during interaction was accepted as secondary criteria for process evaluation. Its value is determined through the following formula:

$$f = \frac{M_c}{N \cdot r},$$

M_c – moment of friction during interaction, Ncm; N – applied loading, N; r – radius of the roller, cm

The values of the coefficients of friction during interaction are presented at table 1.

parameters	frictional pairs	duration of the experiments, hrs					
		1	2	4	6	10	14
M_c , N.cm	St.45-BO30 standard	22.8	20	17.7	17.3	16.8	16
	St.45-BO30 at 47,6 Hz	18.1	15.5	14.3	14	12.5	12.5
	St.45-BO30 at 150 Hz	15.4	11.8	10.2	9.9	10.1	10.1
f_c	St.45-BO30 standard	0.009	0.008	0.007	0.006	0.006	0.006
	St.45-BO30 at 47,6 Hz	0.007	0.006	0.005	0.005	0.005	0.005
	St.45-BO30 at 150 Hz	0.006	0.004	0.004	0.003	0.004	0.004

Conclusion

1. The applied layers obtained through vibrating gas arc overlaying process with wire electrode vibrating frequency at a rate of 150 Hz passing into process of established wear according to insignificant alteration of the moment of active interaction from 4th till 14th hours at rates of 10,2 to 10,5 Nm.

2. The pair overlaid at the mentioned rate of 150 Hz is characterized by almost 2 times lower value of the moment of friction at the end of loading in comparison to both – standard and overlaid at a rate of 47,6 Hz pairs.

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INTEGRATED TECHNOLOGY AND MULTIMODULE TRACKED VEHICLES UNIT FOR WETLANDS PROTECTION

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Abstract

Integrated technology and new generation of multimodule tracked vehicles unit was designed by PIMR- Industrial Institute of Agricultural Engineering in Poznan within the research project WND-POIG.01.03.01-00-164/09. The Tracked Vehicles Unit (TVU) was built as a part of cooperation with Polish firm Hydromega. TVU is dedicated to work on wetlands and should have positive impact on ecology and better environment protection thanks to use of biodegradable oils and lubricants, diesel engine with Stage III emission standard, innovative tracks which have different internal and external speeds during turns. A single-phase technology is designed for harvesting/collecting and transporting biomass (reeds, grass, bushes, small trees) on wetlands, especially these that are located in National Parks and Nature 2000 areas. The TVU is designed to work with different tool modules for cutting, shredding and transporting collected biomass on wetlands. Amphibian type of the TVU's hulls allows for work in inlands water (rivers, lakes, canals).

Keywords: VEHICLE, TRACKED, UNIT, BIOMASS, MOWING, TRANSPORT, WETLAND, ENVIROMENT, PROTECTION

1. Introduction

PIMR - Industrial Institute of Agricultural Engineering in Poznan, Poland realizes R&D project No. WND-POIG.01.03.01-00-164/09 on implementation of the technologies for works related to stopping unwanted (grass, reeds, bushes) greenery succession on wetlands and restoring the breeding areas of endangered species of birds [1].

Existing technology in Poland

The farmers had outdated and worn out vehicles and machines, which are potentially and very often the real cause of decreasing the environment's protection – mainly, because crawler-type vehicles – snow grooming ratracs - are poorly adapted by farmers or operators for harvesting biomass in summer months. Overheating of the ratrac's engine with protecting cover lifted (Fig. 1) is very dangerous; a few times the biomass dust and residues of oil and grease ignited a fire resulting in destruction of the engine and oil spillage from broken hydraulic hoses. To our knowledge, the ratracs had mineral oil rather than biodegradable oil, the spills have not been recovered and remediation actions have not been taken. One cc of mineral oil can spoil up to 5000 cc of clean water and it will be in ground or water for years [2].

Devastation of peat's top layer that was made by sharp edges of grousers of the tracks in the field is very dangerous for vegetation and environment, especially on wetlands located in National Parks in Poland and Europe as well (Fig. 2) .



Fig. 1. Ratrac adapted by farmer to work on wetlands, blue color cover is open - for better cooling of the engine¹ □



Fig. 2. Top peat layer devastation made by steel tracks of snow grooming vehicles¹

2. Virtual models of tracked vehicles unit

The virtual models of Tracked Vehicle Unit (TVU) prototype with track modules that are powered by 6 hydraulic engines and independently steered was designed in PIMR. The TVU is designed

to work with different tool modules for cutting, shredding and transporting collected biomass on wetlands (Fig. 3-5)

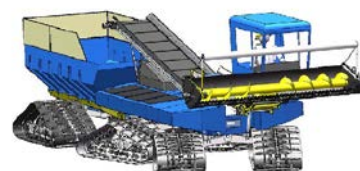


Fig. 3. Virtual model of the Tracked Vehicle Unit for cutting grass, reeds¹



Fig. 4. Virtual model of the Tracked Vehicle Unit for cutting reed, underwater plants on lakes and channels¹



Fig. 5. Virtual model of the Tracked Vehicle Unit for moving , collecting and transporting biomass (grass and reeds)¹

During turns the delta shape tracks (fig. 6) in tractor unit have ability to lean out in different direction (fig. 7). In addition, developed by small medium size firm Hydromega [4], hydraulic tracks' steering system will control speed during making such turns and/or turning.

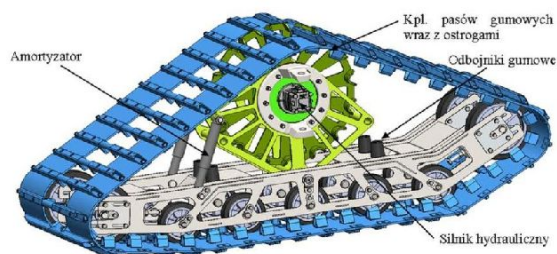


Fig. 6. Delta shape track implemented in TVU [3]

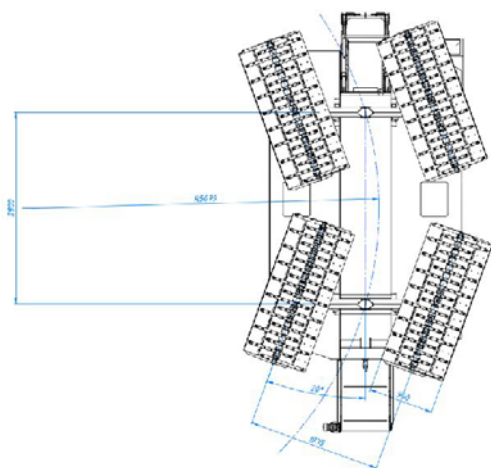


Fig. 7. Delta tracks' steering system should allow to lean out in different direction the front pair and rear pair of the tracks [3]

3. Prototype of tracked vehicles unit

In 2013 year the PIMR-BE Research Group designed prototype of the TVU that is dedicated for harvesting and collecting the biomass and transporting it to temporary storages. The prototype unit was built together with Hydromega [4].

The main purpose is that innovative tracked vehicle unit should be able to move on boggy, muddy terrains and, in addition, that can work on lakes, small rivers, and channels. Caterpillar vehicles, like snow groomers (ratracs) and tracked tractors, can drive on boggy terrains, harvest and transport biomass, but when there is a few centimeters of water on the surface of the ground, they can get stuck, if one of tracks will sink into a hidden meliorative ditch or hollow.

An amphibian type unit can minimize threats - survive in such situations and continue harvesting work on wetlands.



Fig. 8. TVU ready to preliminary field test, on the top of mast is mounted Racelogic's HD video camera. Presented version of tractor and trailer is an amphibian unit that should be able to move through water (lakes, channels) (October 2013)¹



Fig. 9. TVU removes small tree (October 2013)¹



Fig. 10. TVU mowing grass during preliminary field tests (October 2013)¹



Fig. 11. Effects of mowing grass. Trails without damages of the top ground layer (October 2013)¹



Fig. 12. Effects of TVU's cutting and collecting biomass on wetland - no visible damages of top layer of peat¹

The Racelogic's systems [5] (Video Vbox equipped with two HD video cameras and Vbox3i) were used to collect data during fieldwork and video equipment for recording images from cameras placed: in the cabin, on the top of the telescopic mast (fig. 8) and on the Unmanned Aerial Vehicle (UAV), that was rebuilt as KoBE in PIMR and adapted to take-off and land on fields or lakes [6].



Fig. 13. KoBE during take-off on the Bablino lake¹



Fig. 14. No visible ground deformation during TVU turning (picture made by video camera mounted on KoBE)¹



Fig. 15. TVU's tractor during preliminary lakes tests (picture made by KoBE's video camera)¹



Fig. 16. TVU during winter tests (February 2014)¹

3. PIMR's integrated technology for removing biomass on protected wetlands

Developed by PIMR integrated technology for removing unwanted greenery from protected wetlands includes following operations:

1. mowing grass and reeds ,
2. cutting reeds and underwater plants,
3. removing small trees and bushes
4. collecting biomass
5. transporting biomass to temporary storages and/or special prepared places in the field
6. Transporting biomass to the warehouses located outside protected areas

Operations 1-5 need to be checked in different locations and weather condition during 2014-2015 years. Operation no. 6 is alternative technology of transporting biomass to the warehouses located outside protected wetlands that was invented by PIMR and it already got several Polish and European patents [7, 8]. This biomass train technology is practically ready for market implementation, especially on mires and peat land of the protected wetlands.

New innovative technology is based on special steel adapters that are used to form biomass train and transport it, by rolling bales on the ground, behind towing vehicles e.g. agricultural tractor with wheels (fig. 17) or tracks, the tracked tractor of TVU (fig 18) or medium size trucks and pickups equipped with delta track modules or 4x4 wheels (fig. 19).



Fig 17. Biomass train towed near Notec River, Byszowice village [9]

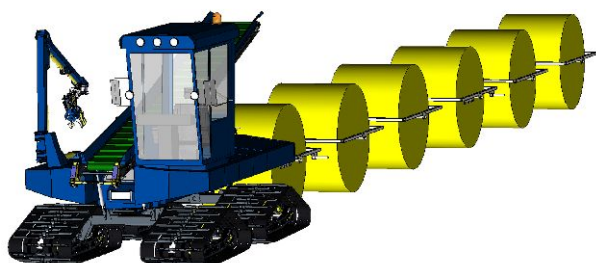


Fig. 18. TVU's biomass train - tractor towing bales¹[10 , 11]



Fig. 19. Mitsubishi L200 research truck with bale (dia. 1,8 m) [9]

PIMR bought Kubota M9960 agricultural tractor in which wheels were replaced by Soucy Track's delta type modules. Tracked Kubota tractor will be used in research field tests to tow biomass bales (fig. 20).



Fig. 20. Kubota M9960 tractor with delta tracks module made by Soucy Track, Canada¹ [12]

4. Conclusions

1. The preliminary results indicate a good drive ability of new tracks units, practically, with no negative impact on the ground as in the case of machines built on the snow grooming vehicle's (ratrack) chassis.
2. Diameter of turn using only front pair of tracks is around 34m, but when two pairs are actively helping to turn - it is around 9-10m.
3. The Tracked Vehicle Unit ran easily on the grass, as well as on peat land or snow.
4. More advanced field tests of mowing, cutting, shredding and transporting biomass as well as of the TVU's impact on peat land will be run during 2014 and 2015 years.
5. Developed and patented by PIMR technology of special adapters and methods of forming biomass train is practically ready for demonstrator type projects and for market implementation.

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RELIABILITY EVALUATION AND COMPARATIVE ANALYSIS OF ENGINES FOR LIGHT-DUTY VEHICLES WITH ELECTRONIC FUEL CONTROL SYSTEMS USING PARETO APPROACH

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Abstract: Over the past several years, reliability has become an increasingly important topic when it comes to continuous improvement. Higher engine reliability reduces repair costs, and contributes to increased performance. In addition, it increases safety and reduces potentially serious failures. This paper presents an evaluation and comparative analysis of the reliability of two different types of electronic fuel control systems - Common rail and Epic Lucas using the Pareto method. Their most common failures and repairs costs are identified and present in diagrams

Keywords: RELIABILITY, FAILURES, REPAIRS, PARETO CHART, FUEL SYSTEMS

1. Introduction

The limitation of the natural resources, increased demands for environmental protection and the complexity of the system design of the modern cars require high reliability during their operation. Successful solution of solving these tasks can be achieved by taking management decisions, based on information obtained from statistical data for the failures occurred, the repair expenses and the unexpected breakdown of the vehicles.

The statistical data can be used to determine the most common failures of the structure, probable causes and ways to minimize them. This kind of tasks can be successfully solved using the Pareto approach, one of the best methods for quality management.

In 1897 Italian economist Vilfredo Pareto introduced a formula describing the variability of distributions of wealth. This idea in 1907 was graphically illustrated by a chart from the American economist Max Lorenz. The two scientists showed that most often the largest part of income or wealth does not belong to the large number of people. The famous American expert on quality management Joseph Juran used this method. He applied the Pareto principle to quality issues. This allowed to divide the factors affecting the quality of "the vital few and the trivial many". Generally it turned out that a large number of defects and the associated losses come from a relatively small number of reasons. J. Zhuang named this approach the Pareto Analysis.

2. Materials and methods

Pareto chart is a vertical bar graph in which values are plotted in decreasing order of relative frequency from left to right. Pareto charts are extremely useful for analyzing what problems need attention first because the taller bars on the chart, which represent frequency, clearly illustrate which variables have the greatest cumulative effect on a given system.

There are seven steps to identifying the important causes using Pareto Analysis:

1. Form a table listing the causes and their frequency as a percentage.
2. Arrange the rows in the decreasing order of importance of the causes.
3. Add a cumulative percentage column to the table.
4. Plot with causes on x-axis and cumulative percentage on y-axis.
5. Join the above points to form a curve.
6. Plot a bar graph with causes on x-axis and percent frequency on y-axis.

7. Draw a line at 80% on y-axis parallel to x-axis. The point of intersection with the curve on x-axis separates the important causes on the left and less important causes on the right.

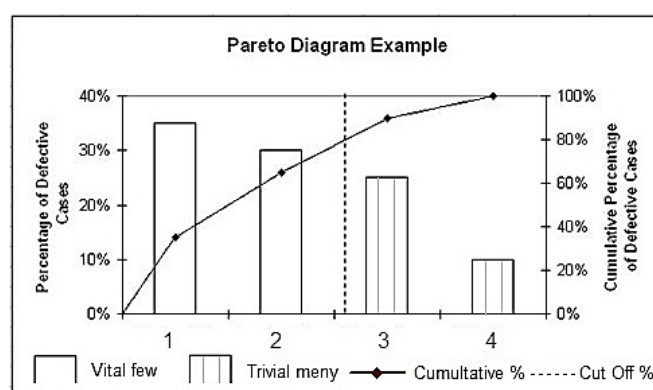


Fig1. Pareto diagram

The aim of the study is to identify and compare the most typical failures in two modern vehicles in terms of the cost for preventing them. The investigated vehicles are Renault Kangoo and Fiat Doblo. Both vehicles models belong to the same class - light type pickup. Renault Kangoo has EPIC-Lucas electronic fuel control system, and Fiat Doblo - Bosch Common rail fuel system. The vehicles were driven under the same conditions and loads.

Pareto principle was used as a scientific tool to analyze the repair costs for various types of failures. C_{sp} is indicator which determine the average specific costs (materials and labor) for distance covered per time unit.. The vehicles were driven under the same conditions and loads.

$$C_{sp.} = \sum_j^N \frac{C_{ij}}{L_j}$$

Where C_{ij} is the cost for removing the i^{th} type of failure of the j^{th} vehicle;

L – the mileage of j^{th} vehicle covered during the study

3. Experimental results

The analyses was based on statistical data from maintenance department reports. Failures of different engine systems for 55 Renault Kangoo and 60 Fiat Doblo light-duty vehicles were examined. The total vehicle distance travelled for Renault Kangoo is $\sum L_j = 6586000$ km and 5043000 km for Fiat Doblo, the average distance is 119700 km and 84000 km respectively. For Renault engines 293 failures were registered and divided into 10 groups, and for Fiat Doblo - 488 failures, divided in 13 groups.

For each group of failures, C_{sp} costs and their relative share in percentage $C\%$ were determined using formula (1) (Table 1 and Table 2).

Fiat Doblo

Tabl. 1

N ^o	Type of failure (repair)	C_{sp} lv/1000 km	$C\%$
1	Engine repair	0.94	10.30
2	Camshaft chain replacement	0.84	9.20
3	Starter motor repair	0.54	5.91
4	Water pump replacement	0.35	3.83
5	Rubber engine mount replacement	0.28	3.07
6	Turbocharger repair	0.25	2.74
7	Alternator belt replacement	0.15	1.64
8	Engine head repair	0.13	1.42
9	Flowmeter replacement	0.13	1.42
10	High pressure pump repair	0.09	0.99
11	Alternator belt replacement	0.10	1.10
12	Thermostat replacement	0.05	0.55
13	EGR valve replacement	0.02	0.22

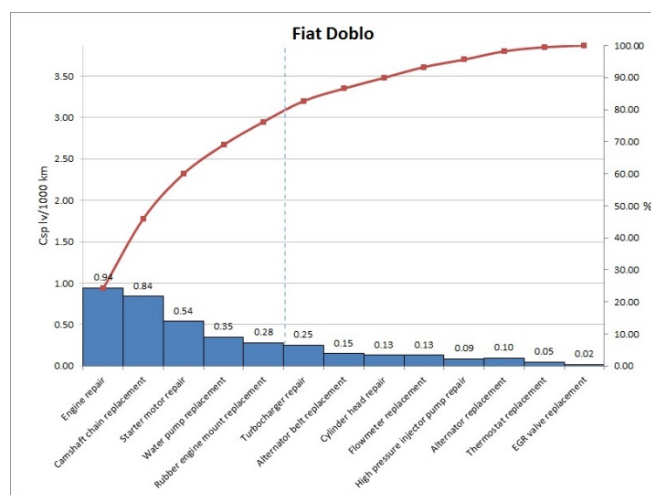


Fig. 3. Pareto chart for Fiat Doblo engines

The classification of the typical failures requires the cumulative growth curve for the costs $C\%$ to be greater than the calculated average. The average value of the costs that were necessary for maintenance and prevention of engine failures for Renault Kangoo was:

$$C_{\%} = \frac{100}{10} = 10.00\%$$

The average value of the costs that were necessary for maintenance and prevention of engine failures for Fiat Doblo was:

$$C_{\%} = \frac{100}{13} = 7.69\%$$

The diagram on fig. 2 indicates that Renault Kangoo most common failures and repairs includes are the following:

- high pressure injector pump repair
- starter motor repair
- rubber engine mount replacement
- engine repair.

More detailed analysis demonstrated that the most frequently observed reason for the repair of injection pump is the occurrence of failures in rotor position sensor. This is a structural problem for EPIC-Lucas fuel-injection pumps, which practically cannot be eliminated.

The diagram on fig. 3 shows that Fiat Doblo typical failures and repairs include:

- engine repair
- camshaft chain replacement
- starter motor repair
- water pump replacement
- rubber engine mount replacement

Further in the research, it was noticed that most common reason for the repair of engines is breaking the camshaft chain and the emerging consequences of that failure. The most frequent cause is the low reliability of the timing chain. Therefore, it is recommended to change the timing chain before the schedule time provided in the instructions for maintenance and repair of this model.

Starter motor repairs, observed in both of the studied vehicles, can be associated with the frequent start-stop cycle conditions occurring during normal service in urban areas.

3. Conclusion

The study of the failed parts is a good tool that can help determine the most unreliable components and systems as well as the reasons for their frequent repairs. It also aids the study of design

Renault Kangoo

Tabl. 2

N ^o	Type of failure (repair)	C_{sp} lv/1000 km	$C\%$
1	High pressure pump repair	2.41	27.54
2	Starter motor repair	1.23	14.06
3	Rubber engine mount replacement	0.80	9.14
4	Engine repair	0.54	6.17
5	Engine head repair	0.48	5.49
6	Alternator belt replacement	0.33	3.77
7	Water pump replacement	0.29	3.31
8	Thermostat replacement	0.22	2.51
9	Alternator repair	0.11	1.26
10	EGR valve repair	0.02	0.23

To analyze the operating reliability of engines for the investigated vehicles based on data from Table 1 and Table 2, bar and Pareto charts were built.

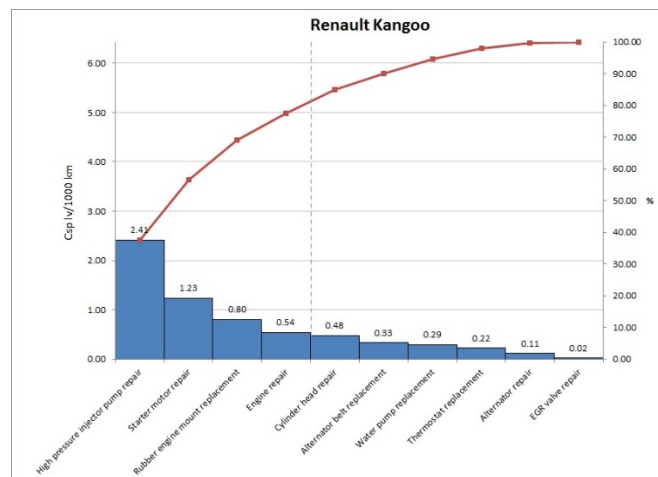


Fig. 2. Pareto chart for Renault Kangoo engines

sufficiency and reliability of vehicle systems. In addition, the results of the survey indicated that many product failures occur at moderately high miles but prior to expected wear out, which is due to conditions occurring in the normal service.

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