УДК 624.21

CHOICE OF OPMIMUM MATERIAL FOR STRENGTHENING THE ENTRANCE EDGES OF STEAM TURBINE ROTOR BLADES

D. Gluschcova, Prof., D. Sc. (Eng.), E. Grinchenko, Eng., Ph. D. (Eng.), Ye. Voronova, Assoc. Prof.,
Kharkov National Automobile and Highway University

Abstract. *The influence of electrode material on the state of surfacing layer of steam turbine rotor blades is researched. The strengthened layer was produced by electro-spark alloying, using alloy T15K6 and steel 15Х11МФШ. The microstructure, microhardness and thickness of surfacing layers were investigated. The advantages of steel 15Х11МФШ for strengthening the entrance edges of steam turbine rotor blades which makes it possible to simultaneously harden both the entrance edges of rotor blades as well as increase their erosive resistance are grounded.*

Key words: *electro-spark alloying, electrode, surfacing layer, microstructure, microhardness, strengthening.*

ВЫБОР ОПТИМАЛЬНОГО МАТЕРИАЛА ДЛЯ УПРОЧНЕНИЯ ВХОДНЫХ КРОМОК РАБОЧИХ ЛОПАТОК ПАРОВЫХ ТУРБИН

Д.Б. Глушкова, проф., д.т.н., Е.Д. Гринченко, инж., Е.М. Воронова, доц.,
Харьковский национальный автомобильно-дорожный университет

Аннотация. *Исследовано влияние материала электрода на состояние наплавленного слоя рабочих лопаток паровых турбин. Упрочненный слой формировался электроискровым легированием сплавом Т15К6 и сталью 15Х11МФШ. Исследовались микроструктура, микротвердость и толщина наплавленного слоя. Обоснованы преимущества стали 15Х11МФШ для упрочнения входных кромок рабочих лопаток паровых турбин.*

Ключевые слова: *электроискровое легирование, электрод, наплавленный слой, микроструктура, микротвердость, упрочнение.*

ВИБІР ОПТИМАЛЬНОГО МАТЕРІАЛУ ДЛЯ ЗМІЦНЕННЯ ВХІДНИХ КРАЙОК РОБОЧИХ ЛОПАТОК ПАРОВИХ ТУРБІН

Д.Б. Глушкова, проф., д.т.н., О.Д. Грінченко, інж., Є.М. Воронова, доц.,
Харківський національний автомобільно-дорожній університет

Анотація. *Досліджено вплив матеріалу електрода на стан наплавленого шару робочих лопаток парових турбін. Зміцнений шар формувався електроіскровим легуванням сплавом Т15К6 і сталлю 15Х11МФШ. Досліджувались мікроструктура, мікротвердість і товщина наплавленого шару. Обґрунтовано переваги сталі 15Х11МФШ для зміцнення вхідних крайок робочих лопаток парових турбін.*

Ключові слова: *електроіскрове легування, електрод, наплавлений шар, мікроструктура, мікротвердість, зміцнення.*

Introduction

Rotor blades of steam turbines determine the serviceability of the turbine. Their working conditions require high hardness of leading edges. Further, erosion damage reduces their resistance. To increase the service life of the blades the leading edges areexposed to such processing methods like hardening by high frequency currents and applicationof the widely used alloy T15K6 based on carbides *Ti* and *W* asareinforcing electrode. The binder for this alloy is Co.

Analysis of publications

However, the mode of operation of the blades is such that requires increased resistance to shockerosion, lack of adverse influence of coating formation parameters on mechanical properties, high corrosion properties.

Application of the above methods has limitations. Thus, using the high-frequency current makes it difficult to technically temper the radius blend from the blade airfoil portion to the bookshelf bandage and use of the widely applied alloy T15K6as a reinforcing electrode is limited due to the presence of cobalt - an element that as a result of activation forms long-lived isotopes, which reduce the erosion resistance of blades.

In connection with the above, the objective of the present work was to develop a method that would enable to simultaneously reinforce the leading edges of the blades and reduce their erosion resistance.

Purpose and problem statement

In the given paper there were tested two materials to be used as an electrode: alloy T5K16 and steel 15H11MFSH.

The electric spark method is based on the phenomenon of electric erosion of materials under spark discharge in a gaseous medium, the polar erosion product transport on the layer of modified structure and alloy. As a result of electrical breakdown of the interelectrode gap there occurs a spark, in which the flow of electrons leads to local heating of the electrode (anode) [1]. On the surface of the cathode under the influence of high thermal loads there is carried out mixing of both the cathode and the anode material that promotes the formation of proper adhesion between the substrate and the formed layer. Figure 1 shows the general scheme of the electrosparkalloying [ESA].

The composition of the doped layer may differ significantly from the composition of the raw materials. It is caused by the specifics of the ESA impact, which consists in the ultra-high heating and cooling rates, the contact of surfaces to each other and with the surrounding elements of the environment under pulse exposure to high temperatures and pressures.

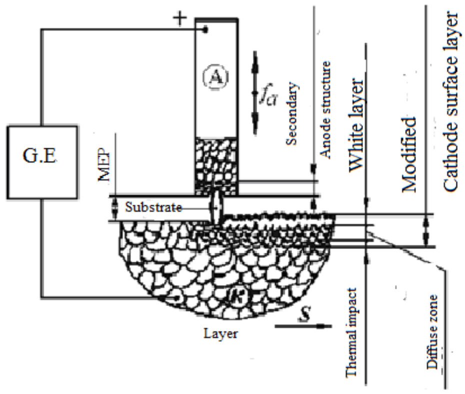


Fig. 1. General scheme of the electrospark alloying

The study was conducted, using samplesof steel 15H11MFSH that was thermally treated to obtain the hardness of 285NV with removing the decarburized layer to the depth of 1mm along the hardening plane. Workson strengthening the samples were carried out, using electrospark equipment EIL8A.

Results of investigation and their discussion

The microstructure of the base metal of specimens presents sorbitol with retaining orientation along martensitic planes. The structure of the samplesis of different uniformity, the structure contains grains of different etchability, and the size of the needles corresponds to 7–8 points (Fig. 2).

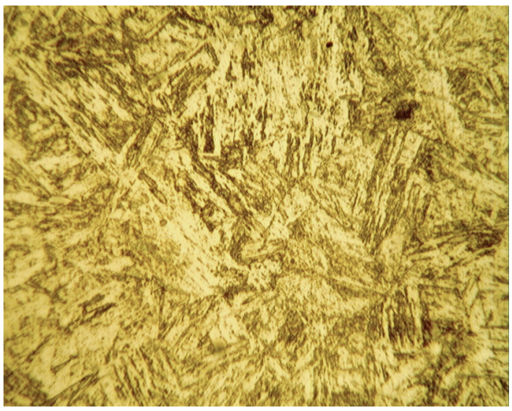


Fig. 2. Microstructure of the main sample metal

Control of the hardened surface is carried out by visual inspection with a magnifying glass with × 3, × 10 power.

On the surface of the samples after hardening by both alloy T15K6and steel 15H11MFSH defects such as cracks were not revealed. Fig. 3 shows the appearance of the surface hardened by alloy T15K6. The layer is homogeneous, fine-grained and in some places there can be found small sizecraters.

Fig. 3. The appearance of the sample surface strengthened by alloyT15K6

Fig. 4 shows the appearance of the surface hardened by steel 15H11MFSH. The layeris homogeneous, fine-grained, has small craters in small quantities.

Fig. 4. The appearance of the sample surface, hardened by steel 15H11MFSH

To assess the quality of adhesion of doped layers with the substrate, the samples after hardening were tested according to the following scheme:
– samples №№1,2 were tested for bending at an angle of 90 °, using a mandrel R = 20 mm;
– samples №№ 3,4 were tested for bending at an angle of 70 °, using a mandrel R = 40 mm.

The test results are shown in Table 1.

When viewingthebends the peel of the hardened layer from the base metal was not detected. Measurement of the thickness of the hardened layer was carried out in sections, manufactured according to the cross-sectional plane of the sample.

Table 1 Bending Test Results

Sample brand	Material	Test results	Notes
1	T15K6	No ruination	in the place of bending detected tears
2	15Х11МФШ	No ruination	in the place of bending detected tears
3	15Х11МФШ	No ruination	in the place of bending no detected tears
4	15Х11МФШ	No ruination	in the place of bending no detected tears

The surface hardened layer is characterized by heterogeneity of the layer thickness, but the average value of the thickness in case of hardening by alloy T15K6 and steel 15H11MFSH is virtually identical (Fig. 5).

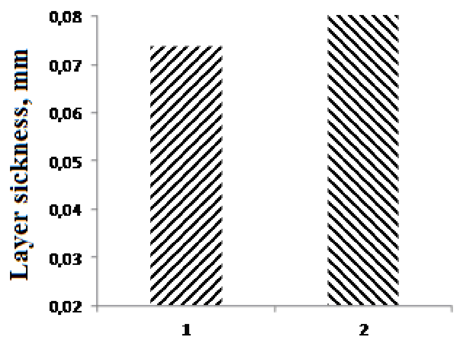


Fig. 5. Histograms of the mean values of the hardened layer thickness: 1 – by alloy T15K6; 2 – by steel steel 15H11MFSH

Study of the microstructure of the deposited layer showed that the structure is homogeneous, almost no etchability. In some places there were detected individual pores. When surfacing by steel 15H11MFSH, the layer structure is of mainly dendritic structure. In the surface layer of the base metal under high temperatures there was observed the formation of the light etchabilityzone formed by diffusion of the electrode material into the sample depth, and the dark-etchability zone of under alloying. In some places there were detected pores.

Fig. 6 shows histograms of microhardness measurement in the zone «hardened layer - base metal» of the samples under study.

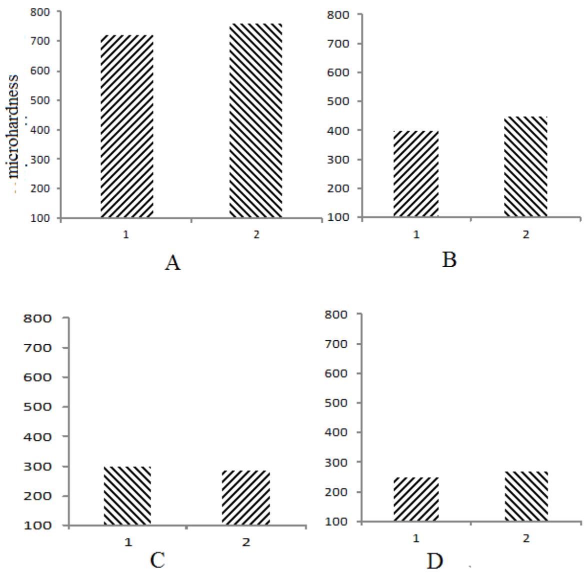


Fig. 6. Histograms of microhardness measurement in samples hardened by alloy T15K6 (1) and steel 15H11MFSH (2): a – deposited layer; b – transition (diffusion) zone; c – HAZ (~ 0,05 mm from the border); d – HAZ (~ 0,1 mm from the border)

As it follows from the above histograms, in all areas the micro-hardness at hardening by alloy T15K6and steel 15H11MFSH is practically identical.

Conclusions

When there was performed visual inspection and carried out metallographic analysis of samples reinforced by the electrospark method, using the equipment EIL 8A by electrodes made of steel 15H11MFSH and hard alloyT15K6 cracks were not revealed.

When conducting the bending test, none of the samples, hardened by both the solid alloy T15K6 and steel 15H11MFSH, failed.

On examination of the bends, the peel of the hardened layer from the base metal was not detected.

The average thickness of the surface layer hardened by both alloy T15K6 and steel 15H11MFSH was virtually identical.

The microhardness of the deposited layer, the transition zone, HAZ at different distances from the border when using both the hardened alloy T15K6 and steel 15H11MFSH do not practically differ.

6. Based on these studies it is recommended to replace the applied reinforcing electrode made of alloy T15K6 and steel 15H11MFSH to increase the hardness of the leading edges of steam engine rotor blades.

References

1. Мулин Ю.И. Электроискровое легирование рабочих поверхностей инструментов и деталей машин электродными материалами, полученными из минерального сырья / Ю.И. Мулин, А.Д. Верхотуров. – Владивосток: Дальнаука, 1999. – 110 с.
2. Современные методы упрочнения поверхностей деталей машин. Аналитический обзор // Научно-технический прогресс в машиностроении / под ред. К.В. Фролова. – 2009. – Вып. 9.1. – С. 205–271.
3. Костецкий Б.И. Трение, смазка и износ в машинах / Б.И. Костецкий. – К.: Техника, 2000. – 396 с.
4. Большаков В.И. Об эффективности защитных покрытий деталей газотурбинных двигателей / В.И. Большаков, В.И. Харченко, В.Н. Журавель // Перспективные задачи современной науки: сб. науч. тр. – 2002. – С. 109–116.
5. Большаков В.И. Етапи ідентифікації параметричних технологій та шляхи їхреалзації / В.И. Большаков, В.М. Волчук, Ю.У. Дубров // Вісник НАН України, – 2013. – № 8. – С. 66–72.

References

1. Mulin Yu.I., Verhoturov A.D. *Elektroiskrovoe legirovanie rabochih poverhnostey instrumentov i detaley mashin elektrodnymi materialami, poluchennymi iz mineralnogo syrya*. [Electro-spark doping of working surfaces of tools and machine parts by electrode materials obtained from minerals. Vladivostok, Dalnauka Publ., 1999. – 110 p.
2. *Sovremennye metody uprochneniya poverhnostey detaley mashin. Analiticheskiy ob-*

zor [Modern methods of hardening the surfaces of machine parts, Analytical review] *Nauchno-tehnicheskiy progress v mashinostroenii / pod red. K.V. Frolova*. [Scientific and technological progress in engineering, ed. K. Frolov]. 2009. Vol. 9.1. pp. 205–271.
3. Kostetskiy B.I. *Trenie, smazka i iznos v mashinah*. Friction, lubrication and wear in machines. Kyiv, Engineering Publ., 2000. 396 p.
4. Bolshakov V.I., Harchenko V.I., Zhuravel V.N. *Ob effektivnosti zaschitnyh pokrytiy detaley gazoturbinnyh dvigateley*. [On the effectiveness of protective coatings of gas turbine engine part], *Perspektivnye zadachi sovremennoy nauki*: *sb. nauch. tr.* [Advanced problems of modern science: Collection of scientific works]. 2002. pp. 109–116.
5. Bolshakov V.I., Volchuk V.M., Dubrov Yu. U. *Etapy identifikatsiy parametrichnyh tehnologiy ta shliahy yih realizatsiy* [Stages of parametric technology identification and ways of their identification]. Visnyk NAN Ukrayiny, 2013. Vol. 8. pp. 66–72.

Рецензент: В.Г. Солодов, профессор, д.т.н. ХНАДУ.

УДК 62-932:62.532

СТРУКТУРНА МОДЕЛЬ СИСТЕМИ УПРАВЛІННЯ РОБОЧИМ ПРОЦЕСОМ ЕКСКАВАТОРА

О.Г. Гурко, доц., к.т.н.,
Харківський національний автомобільно-дорожній університет

Анотація. *На основі аналізу завдань, які повинні вирішуватися системою автоматичного управління робочим процесом гідравлічного екскаватора, запропоновано ієрархічну структуру системи управління. Виконано декомпозицію процесу управління, що дозволило розробити структурну модель, яка відображає особливості багаторівневої територіально-розподіленої системи управління робочим процесом екскаватора.*

Ключові слова: *екскаватор, земляні роботи, автоматизація, структура, планування завдань.*

СТРУКТУРНАЯ МОДЕЛЬ СИСТЕМЫ УПРАВЛЕНИЯ РАБОЧИМ ПРОЦЕССОМ ЭКСКАВАТОРА

А.Г. Гурко, доц., к.т.н.,
Харьковский национальный автомобильно-дорожный университет

Аннотация. *На основе анализа задач, которые должны решаться системой автоматического управления рабочим процессом гидравлического экскаватора, предложена иерархическая структура системы управления. Выполнена декомпозиция процесса управления, что позволило разработать структурную модель, отражающую особенности многоуровневой территориально-распределённой системы управления рабочим процессом экскаватора.*

Ключевые слова: *экскаватор, земляные работы, автоматизация, структура, планирование задач.*

A STRUCTURAL MODEL OF AN EXCAVATOR WORKFLOW CONTROL SYSTEM

A. Gurko, Assoc. Prof., Ph. D. (Eng.),
Kharkov National Automobile and Highway University

Abstract. *Earthwork improving is connected with excavators automation. In this paper, on the basis of the analysis of problems that a hydraulic excavator control system have to solve, the hierarchical structure of a control system have been proposed. The decomposition of the control process had been executed that allowed to develop the structural model which reflects the characteristics of a multilevel space-distributed control system of an excavator workflow.*

Key words: *earthwork, automated excavator, intelligent excavation system, structure, task planning.*

Вступ

На сьогодні гідравлічні екскаватори (ГЕ) є найбільш поширеними машинами для земляних робіт. ГЕ використовуються як на великих будівельних майданчиках (наприклад, при будівництві доріг, дамб та ін.), так і в обмежених міських умовах (при спорудженні траншей, котлованів тощо). При цьому вимоги до якості, швидкості та економічності виконання цими машинами робіт постійно підвищуються. До недавнього часу вирішення проблеми підвищення ефективності виконання земляних робіт здійснювалося в основному традиційними методами – за рахунок удосконалення конструкцій вузлів та механізмів ГЕ, у тому числі й за рахунок підвищення їх універсальності шляхом розширення номенклатури та складності робочого обладнання, збільшення числа його ступенів вільності. Це ускладнює роботу машиніста, вимагає від нього дуже високої кваліфікації. Крім того, ускладнення ГЕ веде до того, що людина вже не в змозі реалізувати всі можливості машини, а неправильні дії машиніста можуть призвести до нештатних режимів роботи і нещасних випадків. У зв'язку з цим виникло протиріччя між стрімким розвитком теорії та практики екскаваторобудування і фізіологічними можливостями людини-машиніста. Подолання цього протиріччя можливе лише за рахунок впровадження системи автоматичного управління робочим процесом гідравлічного екскаватора (САУРПГЕ), що допомагатиме машиністу при проведенні земляних робіт або повністю візьме на себе функції управління ГЕ.

Аналіз публікацій

Питанню автоматизації робочих процесів машин для земляних робіт і, зокрема, екскаваторів присвячено чимало публікацій. Значна їх кількість спрямована на автоматизацію кар'єрних екскаваторів [1–2]. Наприкінці 70-х – на початку 80-х років XX ст. з'являються роботи з автоматизації ГЕ. Так, в 1983 р. захищено дисертацію [3], в якій доведено, що системи управління екскаваторами з електромеханічними приводами не можуть бути застосовані для управління ГЕ, та розроблено систему управління операцією копання для екскаватора ЕО-4121А. Як елементну базу системи управління використано гідравлічні логічні елементи. Впровадження системи дозволило збільшити продуктивність екскаватора на 14 %, зменшити питому витрату палива на 15 %, а також знизити навантаження на машиніста за рахунок зменшення числа перемикань механізмів під час копання.

У цьому ж році в США компанією «Southern California Gas Company» було розпочато програму «The Robot Excavator (REX) Development Program», метою якої було створення роботизованої екскаваторної системи для робіт по заміні газотранспортних комунікацій [4]. З того часу основна тенденція розвитку САУРПГЕ пов'язана з їх роботизацією і, відповідно, застосуванням відповідних методів робототехніки з урахуванням особливостей робочого процесу ГЕ [5–12].

Наукові дослідження в області автоматизації робочих процесів екскаваторів привели до серійного виробництва САУРПГЕ. При цьому досить широко застосовуються системи, засновані на використанні GPS/ГЛОНАСС і лазерних технологій. Як правило, ці системи використовують бортові ЕОМ з людино-машинним інтерфейсом, який в реальному часі відображає інформацію про стан машини і конфігурацію робочого обладнання. На основі отриманої інформації машиніст має можливість коригувати свої дії з метою підвищення якості копання [13–15]. Вказані системи продаються окремо та можуть бути встановлені на вже існуючу машину, хоча провідні виробники екскаваторів, такі як Caterpillar, Volvo, Komatsu, вже використовують їх як штатне обладнання. Найбільш відомими марками сучасних систем такого типу є TOPCON, Leica, Trimble, TF – Technologies A/S та ін., що випускають продукцію зі схожими можливостями. Однак переважна більшість наведених систем є індикаторними або напівавтоматичними, основною ланкою в яких залишається машиніст.

Результати вказаних вище та інших робіт зробили суттєвий внесок у вирішення проблеми автоматизації робочого процесу ГЕ. Проте аналіз цих досліджень показує, що активній автоматизації підлягають, в основному, транспортні операції та технологічний контур стеження за заданою траєкторією копання; при цьому розробка САУРПГЕ проводиться без використання системного підходу, а питанням комплексної автоматизації всього екскаватора приділяється недостатньо уваги.

Мета і постановка завдання

Метою цього дослідження є підвищення ефективності робочого процесу ГЕ за рахунок розробки структурної моделі САУРПГЕ.

Для досягнення вказаної мети необхідно виділити основні окремі завдання САУРПГЕ; зробити вибір та обґрунтування структурної моделі САУРПГЕ.

Структурний аналіз САУРПГЕ

Розширено управління робочим процесом ГЕ вимагає вирішення трьох завдань: планування земляних робіт, управління переміщенням ГЕ і його маніпулятора та вимірювання значень параметрів, необхідних для досягнення цілей управління. Відповідно САУРПГЕ повинна здійснювати зазначені функції. Звідси найбільш доцільною є ієрархічна структура САУРПГЕ (рис. 1), де, за аналогією зі структурою АСУ ТП, нижнім рівнем є рівень датчиків, що вимірюють необхідні параметри процесу, і виконавчих механізмів, що впливають на ці параметри. На цьому рівні також здійснюється узгодження сигналів датчиків зі входами управляючих пристроїв, а управляючих сигналів – з виконавчими механізмами, забезпечується зв'язок між підсистемами та надання інформації машиністові.

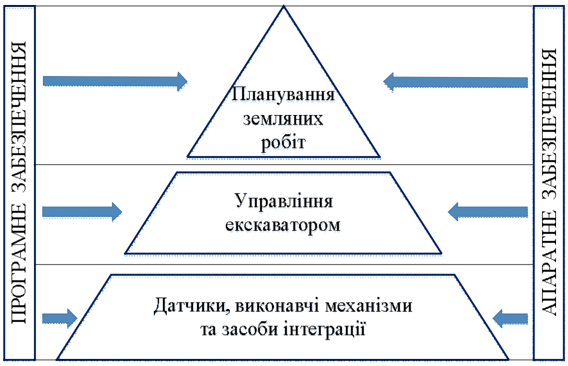


Рис. 1. Ієрархічна структура системи управління екскаваційними робо́тами

Середній рівень – рівень обчислювальних та управляючих пристроїв, що здійснюють збір та обробку даних, які поступають з датчиків, та формують управляючі впливи на виконавчі механізми.

Верхній рівень – це рівень проектного офісу. На цьому рівні здійснюється планування земляних робіт та контроль їх виконання.

Складність проблеми автоматизації робочого процесу ГЕ приводить до необхідності виконання її декомпозиції, тобто подання кожної із зазначених задач у виді множини взаємопов'язаних підзадач. При цьому потрібно виконати структурний синтез САУРПГЕ, тобто визначити оптимальну або найбільш раціональну її структуру.

Аналіз системних особливостей процесу управління земляними роботами дозволяє дійти висновку про територіально-розподілений характер САУРПГЕ. Розглянемо роботу системи більш детально (рис. 2). На першому рівні – рівні планування земляних робіт – здійснюється розбиття будівельного майданчика на зони і формування планів та / або нарядів на виконання робіт для кожної зони. Для цього будується 3D модель будівельного майданчика, а у подальшому також виконується вимірювання поточної робочої зони ГЕ, відзначаючи її зміну. Ці роботи проводяться лазерними та супутниковими комплексами. Потім, на підставі отриманої моделі місцевості, а також даних про параметри потрібного земляного спорудження, здійснюється планування робіт ГЕ (або групи ГЕ). План робіт включає визначення найбільш раціональних траєкторій кромки ковша ГЕ при розробці забою. Ці траєкторії враховують особливості кожної машини і можуть бути розраховані, наприклад, з метою мінімізації витрат енергії на копання [16]. Планування робіт також передбачає знаходження оптимальних місць розташування ГЕ під час виконання робіт. Детальне планування рухів безпосередньо ГЕ є метою не цього, а наступного рівня ієрархії (рис. 2).

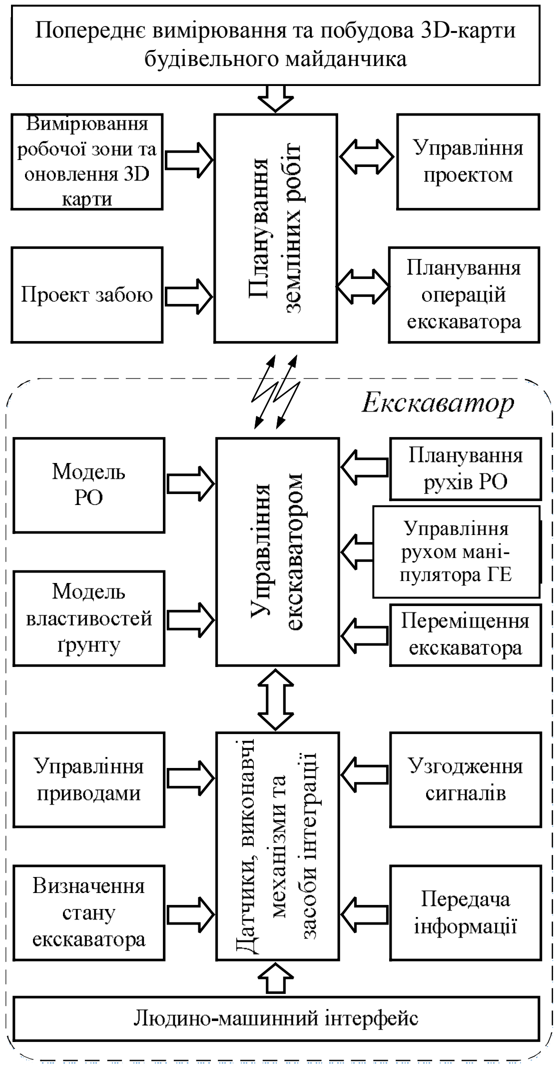


Рис. 2. Структурна модель системи управління екскаватором

Інші важливі завдання, що виконуються на рівні планування земляних робіт, включають планування переміщень екскаватора між ділянками, перевірку якості виконання екскаваційних робіт, а також створення бази даних про хід робіт, що виконуються, силами інформаційної системи управління проектами [17, 18].

Таким чином, основною метою планування завдань є визначення оптимальної послідовності дій ГЕ для досягнення глобальних цілей, таких як мінімізація часу виконання робіт з максимально ефективним використанням парку машин.

Сформований план робіт передається машині по бездротових лініях зв'язку.

Другим рівнем є «Управління екскаватором», на якому забезпечується управління окремим ГЕ. Апаратура цього рівня одержує план робіт, що розроблений на попередньому рівні ієрархії, на підставі якого розраховує закони зміни узагальнених координат маніпулятора ГЕ, що забезпечують проходження кромки ківша за заданими траєкторіями, виробляє управляючі впливи на виконавчі пристрої маніпулятора ГЕ на підставі інформації про поточний стан робочого обладнання та оточуючого середовища, а також з урахуванням прогнозу сил взаємодії ківша з ґрунтом.

Ефективна робота САУРПГЕ не може бути реалізована без системи навігації ГЕ, метою якої є визначення маршруту руху ГЕ від однієї точки робочого майданчика до іншого як у процесі розробки одного забою, так і для переміщення до нового місця копання [19, 20].

Третій рівень ієрархії «Датчики, виконавчі механізми та засоби інтеграції» відповідає за роботу виконавчих елементів та іншого апаратного забезпечення. Розробка електрогідравлічних клапанів для електронного управління ланками маніпулятора ГЕ екскаватора є одним з основних напрямів розвитку даного рівня. Крім того, на цьому рівні здійснюється інтеграція й управління кожним із вказаних рівнів системи.

Висновки

Таким чином, проблема автоматизації робочого процесу ГЕ є вкрай складною й такою, що важко формалізується. Вона потребує одночасної розробки та використання законів управління гідроприводом маніпулятора ГЕ, більш досконалих приводів для здійснення руху ланок маніпулятора, технічних засобів для визначення координат стану робочого обладнання та навантажень на ньому, синтезу бортової мережі машини, системи мобільного офісу тощо.

У роботі одержано розв'язок задачі структурного синтезу САУРПГЕ, що є необхідним для розробки ефективної повністю автономної системи управління виконанням земляних робіт. Раціональною є ієрархічна структура САУРПГЕ, на нижньому рівні якої розміщені датчики, виконавчі механізми і засоби інтеграції; середній рівень – рівень пристроїв управління екскаватором, а верхній рівень – це рівень планування земляних робіт й контролю за їх виконанням.

Виконано декомпозицію завдань, що вирішуються на кожній з підсистем САУРПГЕ, що дозволяє визначити найменш розроблені структурні елементи системи. Для рівня управління екскаватором ними є підсистеми планування рухів маніпулятора екскаватора та реалізації цих рухів в умовах невизначеності, а також переміщення самого ГЕ від однієї точки місцевості до іншої. Розробка вказаних підсистем є метою подальших досліджень.

Література

1. Остриров В.Н. Разработка и исследования системы оптимального управления процессом подъёма ковша экскаватора-драглайна на выгрузку: автореф. дис. на соискание уч. степени канд. техн. наук: спец. 05.13.07 «Автоматизация технологических процессов и производств» / В.Н. Остриров. – М., 1980. – 20 с.
2. Певзнер Л.Д. Алгоритмический и структурный синтез автоматизированного управления шагающим экскаваторомдраглайном: автореф. дис. на соискание уч. степени д-ра техн. наук: спец. 05.13.07 «Автоматизация технологических процессов и производств» / Л.Д. Певзнер. – М., 1987. – 31 с.
3. Акинфиев А.А. Создание системы управления операцией копания для одноковшового гидравлического экскаватора с целью повышения эффективности его

работы: дис. канд. техн. наук: 05.05.04 / А.А. Акинфиев. – М., 1983. – 236 с.
4. Haynes J. N. A. Robotics-Shaping the future of pipeline excavation for repair / J. N. Haynes, A. Briggs // Proceeding of 17th World Gas Conference, Washington, DC, USA. – 1988.
5. Vaha P.K. Excavator dynamics and effect of soil on digging / P.K. Vaha, A.J. Koivo, M.J. Skibiniewski // Proceedings of the International Symposium on Automation and Robotics in Construction. – 1991. – P. 297–306.
6. Bradley D. A. Artificial intelligence in the control and operation of construction plant – the autonomous robot excavator / D. A. Bradley, D. W. Seward, J. E. Mann, M. R. Goodwin // Automation in construction. – 1993. – Vol. 2, №. 3. – P. 217–228.
7. Yu H. Review of Modelling and Remote Control for Excavators / H. Yu, Y. Liu, M.S. Hasan // International Journal of Advanced Mechatronic Systems. – 2010. – Vol. 2, № 1. – P. 68–80.
8. Gu J. Improved control of intelligent excavator using proportional-integral-plus gain scheduling // J. Gu, D. J. Seward // Journal of Central South University. – 2012. – Vol. 19, № 2. – P. 384–392.
9. Дуданов И.В. Автоматизация исполнительных систем гидравлического экскаватора: автореф. дис. на соискание уч. степени канд. техн. наук: спец. 05.13.07 «Автоматизация и управление технологическими процессами и производстами (промышленность)» / И.В. Дуданов. – Самара, 2008. – 16 с.
10. Choi J. Tracking Control of Hydraulic Excavator Using Time Varying Sliding Mode Controller with Fuzzy System / J. Choi // Advanced Science Letters. – 2012. – Vol. 15, № 1. – P. 78–82.
11. Kim Y. B. Dynamically optimal trajectories for earthmoving excavators / Y.B Kim, J. Ha, H. Kang, et al // Automation in Construction. – 2013. – Vol. 35. – P. 568–578.
12. Gurko A. Trajectory Tracking Control of an Excavator Arm Using Guaranteed Cost Control / A. Gurko, O. Sergiyenko, J.I. Nieto Hipólito, et al // Lecture Notes in Electrical Engineering. – 2016. – Vol. 383. – P. 177–196.
13. Калабин Е. Системы управления для экскаваторов 3Dxi / Е. Калабин – [Электронный ресурс]. – Режим доступа: http://prin.ru/articles/135/.
14. Trimble GCS600 Grade Control System For Excavators – [Electronic source]. – Access mode: http://www.sitech-wc.ca/Products_and_Solutions/ Machine_Control_ Systems/Trimble_GCS600_Excavators. aspx.
15. Leica iCON excavate iXE3. Advanced 3D Excavator Control System [Electronic source]. – Access mode: http://www.leica-geosystems.com/en/Leica-iCON-excavate-iXE3_81272.htm
16. Михирев П.А. Основы теории ковшовых автоматизированных рабочих органов / П. А. Михирев. – Новосибирск: Наука, 1986. – 166 с.
17. Lee S. Development of a heuristics-based task planning system for intelligent excavating system / S. Lee, J. Kim, J. Park, J. Seo, Y. Kim // Proceeding of 26th ISARC, Texas, USA. – 2009. – P. 307–316.
18. Seo J. Task planner design for an automated excavation system / J. Seo, S. Lee, J. Kim, S. K. Kim // Automation in Construction. – 2011. – Vol. 20, № 7. – P. 954–966.
19. Kim S.K. Intelligent navigation strategies for an automated earthwork system / S.K. Kim, J. Seo, J. S. Russell // Automation in Construction. – 2012. – Vol. 21. – P. 132–147.
20. Schmidt D. Construction site navigation for the autonomous excavator Thor / D. Schmidt, K. Berns // Proceeding of 6th International Conference on Automation, Robotics and Applications. – 2015. – P. 90–97.

References

1. Ostrirov V.N. *Razrabotka i issledovaniya sistemy optimal'nogo upravleniya protsessom pod’ema kovsha ekskavatora-draglaina na vygruzku*. Avtoref. dis. na soiskanie uch. stepeni kand. tekhn. nauk: spets. 05.13.07 «Avtomatizatsija tehnologicheskih protcessov i proizvodstv» [Development and investigation of an optimal control system of dragline-excavator bucket lifting process for unloading]. Moscow, 1980. 20 p.
2. Pevzner L.D. *Algoritmicheskii i strukturnyi sintez avtomatizirovannogo upravleniya shagayushchim ekskavatorom-draglainom*. Avtoref. dis. na soiskanie uch. stepeni d-ra tekhn. nauk: spets. 05.13.07 «Avtomatizatsija tehnologicheskih protces-sov i proizvodstv» [Algorithmic and structural synthesis of automated system of walking excavator-dragline]. Moscow, 1987. 31 p.
3. Akinfiev A.A. *Sozdanie sistemy upravleniya operatsiei kopaniya dlya odnokovshovogo*

gidravlicheskogo ekskavatora s tsel'yu povysheniya effektivnosti ego raboty. Dis. kand. tekhn. nauk: 05.05.04 [Design of a digging operation control system for a single-bucket hydraulic excavator with the purpose of improving its performance]. Moscow, 1983. 236 p.
4. Haynes J. N., Briggs A. Robotics-Shaping the future of pipeline excavation for repair. Proceeding of 17th World Gas Conference, Washington, DC, USA. 1988.
5. Vaha P.K., Koivo A.J., Skibiniewski M.J. Excavator dynamics and effect of soil on digging. Proceedings of the International Symposium on Automation and Robotics in Construction. 1991. pp. 297–306.
6. Bradley D.A., Seward D.W., Mann J.E., Goodwin M.R. Artificial intelligence in the control and operation of construction plant – the autonomous robot excavator. Automation in construction. 1993, Vol. 2, no. 3. pp. 217–228.
7. Yu H., Liu Y., Hasan M.S. Review of Modelling and Remote Control for Excavators. International Journal of Advanced Mechatronic Systems. 2010. Vol. 2, no. 1. pp. 68–80.
8. Gu J., Seward D. J. Improved control of intelligent excavator using proportional-integral-plus gain scheduling. Journal of Central South University. 2012. Vol. 19, no. 2. pp. 384–392.
9. Dudanov I.V. *Avtomatizatsiya ispolnitel'nykh sistem gidravlicheskogo ekskavatora*. Avtoref. dis. na soiskanie uch. stepeni kand. tekhn. nauk: spets. 05.13.06 «Automatizatsija i upravlenie tehnologichskimi processami i proizvodstvami (promyshlennost)» [Automation of actuating systems of hydraulic excavator]. Samara, 2008. 16 p.
10. Choi J. Tracking Control of Hydraulic Excavator Using Time Varying Sliding Mode Controller with Fuzzy System. Advanced Science Letters. 2012. Vol. 15, no. 1. pp. 78–82.
11. Kim Y.B., Ha J., Kang H. Dynamically optimal trajectories for earthmoving excavators. Automation in Construction. 2013. Vol. 35. pp. 568–578.
12. Gurko A., Sergiyenko O., Nieto J.I. Hipólito. Trajectory Tracking Control of an Excavator Arm Using Guaranteed Cost Control. Lecture Notes in Electrical Engineering. 2016. Vol. 383. pp. 177–196.
13. Kalabin E. *Sistemy upravleniya dlya ekskavatorov Topcon 3Dxi* [Control systems for excavators Topcon 3Dxi]. Avialable at: http://prin.ru/articles/135/.
14. Trimble GCS600 Grade Control System For Excavators. Avialable at: http: // www. sitech-wc.ca/ Products_and_Solutions/ Machine_Control_ Systems/ Trimble_ GCS600_ Excavators.aspx.
15. Leica iCON excavate iXE3. Advanced 3D Excavator Control System. Avialable at: http://www.leica-geosystems.com/en/Leica-iCON-excavate-iXE3_81272.htm.
16. Mikhirev P.A. *Osnovy teorii kovshovykh avtomatizirovannykh rabochikh organov* [Basic theory of automated bucket working tools]. Novosibirsk, Nauka Publ., 1986. 166 p.
17. Lee S., Kim J., Park J., Seo J., Y. Kim. Development of a heuristics-based task planning system for intelligent excavating system. Proceeding of 26th ISARC, Texas, USA. 2009. pp. 307–316.
18. Seo J., Lee S., Kim J., Kim S. K. Task planner design for an automated excavation system. Automation in Construction. 2011. Vol. 20, no. 7. pp. 954–966.
19. Kim S.K., Seo J., Russell J.S. Intelligent navigation strategies for an automated earthwork system. Automation in Construction. 2012. Vol. 21. pp. 132–147.
20. Schmidt D., Berns K. Construction site navigation for the autonomous excavator Thor. Proceeding of 6th International Conference on Automation, Robotics and Applications. 2015. pp. 90–97.

Рецензент: Є.С. Венцель, професор, д.т.н., ХНАДУ.

УДК 621,521

VERIFICATION OF FLUID FLOW CALCULATIONS IN VORTEX CHAMBER SUPERCHARGERS

A. Rogovyi, Assoc. Prof., Ph. D. (Eng.),
Kharkov National Automobile and Higway University

Abstract. *On the basis of numerical modeling (URANS) by means of specialized program complexes verification of fluid flow calculation in vortex chamber superchargers was carried out. It is determined that it is better to apply a model of incompressible liquid for calculations with the turbulence model, considering the streamline curvature and system rotation (SST curvature correction).*

Key words: *vortex chamber supercharger, numerical calculation, suction discharge, streamline curvature, correction, turbulence model.*

ВЕРИФИКАЦИЯ РАСЧЕТОВ ТЕЧЕНИЙ В ВИХРЕКАМЕРНЫХ НАГНЕТАТЕЛЯХ

А.С. Роговой, доц., к.т.н.,
Харьковский национальный автомобильно-дорожный университет

Аннотация. *Путем сравнения с экспериментальными данными проведена верификация математического моделирования течения в вихрекамерных нагнетателях на основе использования специализированных программных продуктов. Получено, что для расчетов лучше применять модель несжимаемой жидкости с моделью турбулентности, учитывающей кривизну линий тока и вращение потока.*

Ключевые слова: *вихрекамерный нагнетатель, численные расчеты, расход всасывания, кривизна линий тока, поправка, модель турбулентности.*

ВЕРИФІКАЦІЯ РОЗРАХУНКІВ ТЕЧІЇ У ВИХОРОКАМЕРНИХ НАГНІТАЧАХ

А.С. Роговий, доц., к.т.н.,
Харківський національний автомобільно-дорожній університет

Анотація. *Шляхом порівняння з експериментальними даними проведено верифікацію математичного моделювання течії у вихорокамерних нагнітачах на основі використання спеціалізованих програмних продуктів. Отримано, що для розрахунків краще застосовувати модель нестисливої рідини з моделлю турбулентності, що враховує кривизну ліній струму й обертання потоку.*

Ключові слова: *вихорокамерний нагнітач, числові розрахунки, витрата всмоктування, кривизна ліній струму, поправка, модель турбулентності.*

Introduction

Today, the methods of numerical calculations of various fluid and gas flows have become widespread. However, despite significant growth of computer power, turbulent flows calculation, remains one of the challenges of fluid dynamics computation [1]. Though, recently its application has increased in the methods of direct numerical simulation (DNS) and large eddy simulation (LES), their wide practical application is observed in hydroaerodynamics problems solution, for today is practically not possible, owing to extreme computing work content [2, 3].

Therefore, at calculations of difficult flows, it is necessary to use the semiempirical methods basing on Reynolds averaged Navier-Stockes equations. Semiempirical models and their updates exists much enough and, unfortunately, for today, there is no universal model of this kind, besides there is a pessimistic appraisal of that the look-alike universal model will be hardly constructed [3]. Therefore, at study of such and such pneumatic and hydraulic units and application packages of CFD, first of all, it is necessary to effect verification of used models for selection approaching turbulence model with the minimum errors from experimental data.

Analysis of publications

In many industries, working conditions are difficult and use of pumps and compressors vane and displacement types leads to the raised expenses for equipments replacement and a manufacture stop owing to the raised deterioration of mobile working bodies and sealing [4]. Besides, influence of vibration, temperatures, presence of abrasive particles and liquids chemical aggression reduce efficiency and worsen performance data of the superchargers used in such service conditions [5].

It is possible to reduce the working costs by using of more reliable and durable superchargers which the superchargers concerning the fluidics are: jet pumps [6], vortex injectors [7] and vortex chamber superchargers [8]. Jet devices possess high indicators of reliability and durability owing to absence of mobile working parts, are widely used in difficult service conditions, but have low enough indicators of efficiency which do not exceeding 30 % [9].

It is possible to improve power efficiency indicators , using more perfect ways of energy transfer in designing of jet devices which are developed vortex chamber superchargers [10]. Owing to a combination in their work not only energy transfer by means of a turbulent exchange, but also action of centrifugal force it is possible to raise efficiency, especially at pumping dry substances [11]. These superchargers concerning the fluidics, possess high indicators of reliability and durability, thanks to absence of mobile parts [12].

The first mentions about vortex chamber superchargers have appeared in publications [13, 14], i.e. they yet have no wide spreading in the industry and large-scale researches including by means of computing methods, practically it was not spent. Thus, actual there is a problem of model turbulence selection for simulation of fluid flows in vortex chamber superchargers (VCS) for maintenance of the minimum calculation errors and parameters prediction of a fluid flow.

Features of working process in VCS, first of all, are connected with hydrodynamic features of the swirled flows, such as vacuum presence on an axis of a rotating flow and excess pressure upon peripheries [15]. Hence, turbulence model selection for fluid flow calculation in VCS demands from model of the adequate description and effects prediction of the swirled flows [16]. For today, many researches concerning a choice of turbulence models for various devices at which there are confined vortex: cyclones [17], vortex valves [18], vortex pipes and vortex injectors. In the majority of the works devoted to the description of fluid flows in vortex devices authors come to a conclusion that the most suitable from the computing duration and by criterion of a calculations error minimality the simulation model on a basis Reynolds averaged Navier-Stockes equations with use SST turbulence model with rotation-curvature correction [19]. Comparison of fluid flows simulation results in vortex chamber superchargers with use of various turbulence models and their updatings for today was not spent.

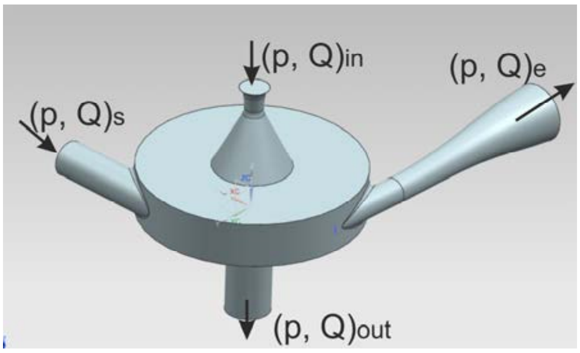
Analysis and problem statement

The aim of the work is verification of fluid flows mathematical modelling in vortex chamber supercharger on the basis of the numerical decision of the Reynolds averaged Navier-Stockes equations by means use specialised program complexes.

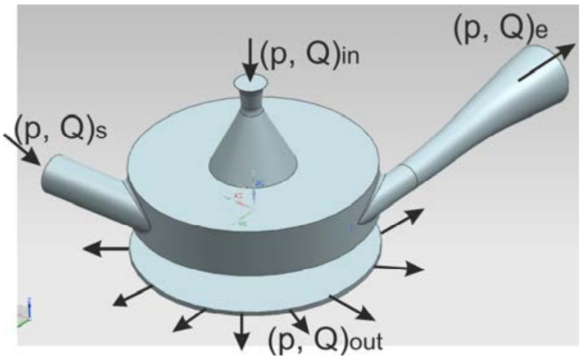
Materials and methods

Verification of numerical researches was made by comparison of experiment results with CFD simulations in program complex OpenFoam [20]. Comparison was made on integrated parameters such as the fluid flow rate on an input channel of the device (the supply fluid flow rate), the fluid flow rate on an exit from the device and the fluid flow rate which is pumped over VCS. Comparison on integrated parameters is dictated by that essential nonstationarity fluid flow and vortex core precession in the chamber leads to that fluid flow kinematic characteristics in the device change the values, therefore to measure them and to make comparison difficult enough. Except a quantitative estimation of integrated parameters calculation errors, qualitative comparison of fluid flow patterns for what experimental sample VCS has been executed with transparent face covers was made. The estimation of calculation errors is executed for two designs VCS: with and without radial diffuser, installed in the exit axial channel (fig. 1).

Experimental setup for physical research included vortex chamber supercharger, blower, receiver and measuring equipment. Pressure in channels was measured by manometers, ambient temperature – mercury thermometers, the fluid flow rates in channels – flowmeters. Air was the working and pumped over medium in experimental researches.



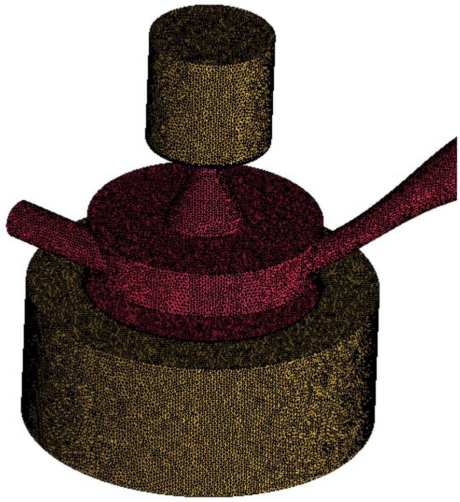
a



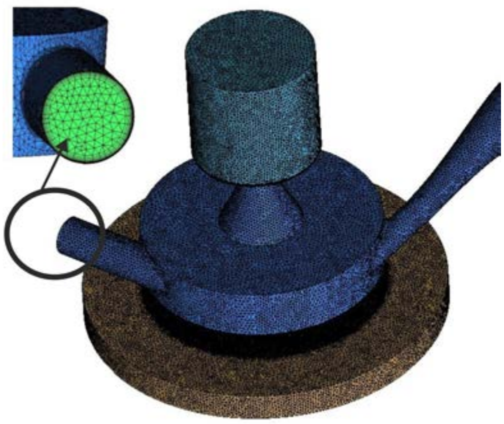
b

Fig. 1. 3D VCS model: a – VCS without radial diffuser; VCS with radial diffuser, installed in the exit axial channel

On the basis of the article analysis [2, 3, 16–19], devoted CFD modeling of the swirled flows in various devices one can draw a conclusion that to the best on calculation time and accuracy of a kinematic parameters prediction is SST turbulence model with rotation-curvature correction [19]. Application of more perfect models DNS, LES, and also hybrid demands considerable time expenses and high-efficiency computer systems [2] that complicates carrying out of a great number of calculations by optimization of a superchargers flowing part. Therefore in the given work it was used SST turbulence model and its correction for definition of the most suitable to flow simulation in VCS. Following calculation models were compared: coarse NCF – calculation of an incompressible liquid on a coarse mesh, coarse NCF-CC – an incompressible liquid on a coarse mesh taking into account the rotation-curvature correction, coarse CF – a compressed liquid on a coarse mesh, coarse CF-CC – a compressed liquid on a coarse mesh with curvature cirrection, NCF – an incompressible liquid, NCF-CC – an incompressible liquid with curvature correction, CF – a compressed liquid, CF-CC – a compressed liquid with curvature correction.



a



b

Fig. 2. Design mesh of VCS: a – without radial diffuser; b – with radial diffuser

The mesh (fig. 2) consisted of 7 million elements for simulation VCS with radial diffuser and 4,5 million elements for VCS without radial diffuser, and has been constructed so that to provide parameter Y+<2. The greater number of elements for the device with diffuser is caused by reduction of the elements size in diffuser owing to small width of the channel (fig. 2, b). The choice of elements number has been dictated by comparison of calculation results on more coarse meshes and meshes with a great number of elements (a 15 million order). As a result of calculations it has been received that use of meshes with number of elements more than 7 million not rationally, owing to absence of the big differences in errors, but considerable computing expenses.

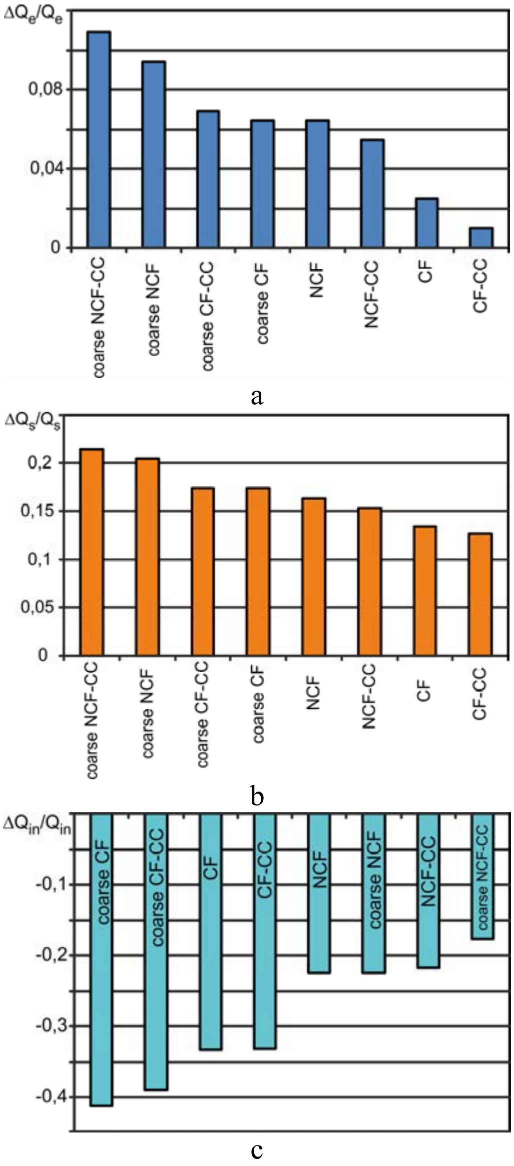


Fig. 3. Errors of fluid flow calculation results in VCS with the radial diffuser: a – fluid flow rate in the exit channel; b – fluid flow rate in the supply channel; c – fluid flow rate sucked in the device

At the task of boundary conditions of axial exits and vortex chamber entries that in the swirled flows pressure is distributed on stream radius was considered. Therefore the rated operating conditions has been increased and exit boundary conditions on new boundary face where pressure is almost equal to zero are set and does not change on radius [12].

The main results of the research

On fig. 3 results of fluid flow calculations comparison in vortex chamber superchargers with the radial diffuser and the integrated parameters gained experimentally are resulted.

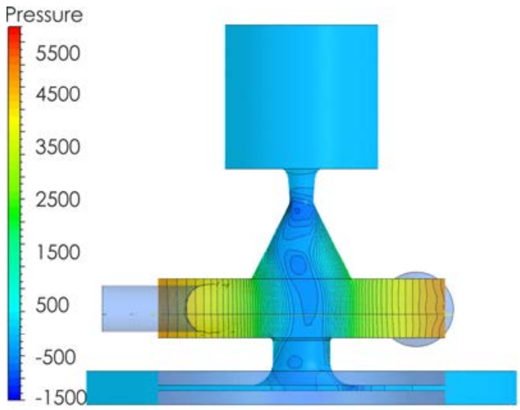
As it is possible to see from fig. 3, models taking into account curvature of streamlines and rotation have the least errors. At application of these models of compressible liquid calculation of the fluid flow rate on an exit and the supply channels makes an order of 10 %. Incompressible liquid models give for these two fluid flow rates an error on 2–3 % the big. Differently with the sucking fluid flow rate in the device. Here, models of incompressible liquid calculation taking into account the rotation-curvature correction have the minimum error, and this error considerable – exceeds 15 %. The calculations spent for compressible fluid led to increase in a calculation error of the sucking fluid flow rate in the device which made more than 30 %. From what it is possible to draw a leading-out that it is better to apply model of incompressible liquid to calculations VCS with model of the turbulence considering curvature of streamlines and rotation.

The difference in calculation errors of sucking fluid flow rate originates owing to different magnitude of vacuum on an apparatus axis (fig. 4). On fig. 4 pressure patterns in VCS with and without the radial diffuser are resulted. Owing to nonstationarity fluid flow in VCS, and also a vortex core precession [21] calculations were spent in transient statement. At comparison of the patterns resulted on fig. 4 it is possible to notice that the greatest vacuum on an axis is gained at calculation of model NCF-CC shown on fig. 4, b that explains the least sucking fluid flow rate error. Thus, than more precisely the model predicts vacuum on an axis, especially exact there are calculations as a whole.

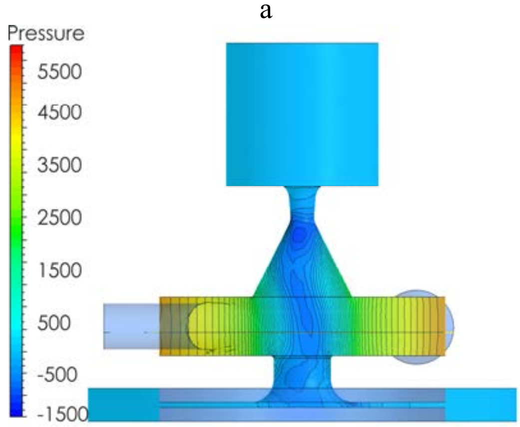
Besides, correct prediction of vacuum on an axis is necessary for the further calculations of gas bubble behaviour getting in the vortex chamber [22].

Rotation-curvature correction use allows to compute the sucking fluid flow rate on 5–15 % more precisely, in connection with more exact calculation of vacuum magnitude on an axis.

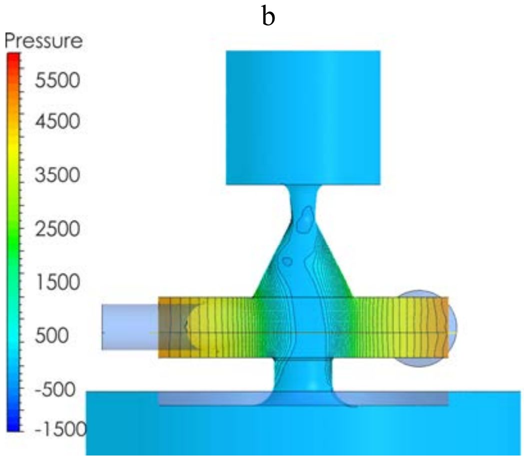
On fig. 5 results of fluid flow pattern calculations in vortex chamber superchargers without the radial diffuser and the integrated parameters gained experimentally are resulted.



a



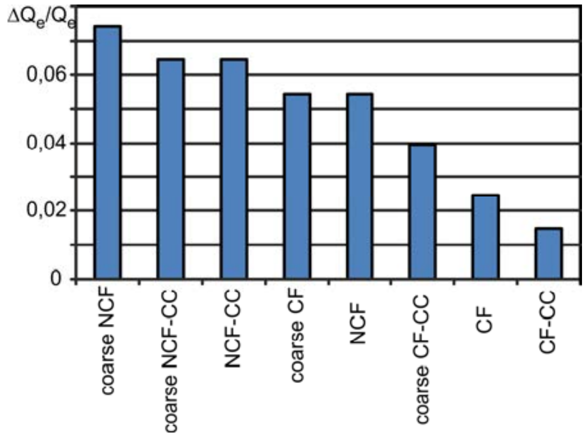
b



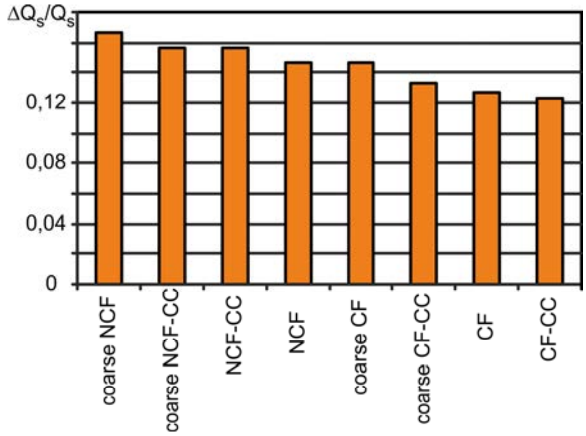
c

Fig. 4. Design distributions of pressure: VCS with diffuser, model CF-CC; VCS with diffuser, model NCF-CC; VCS without diffuser, model NCF-CC

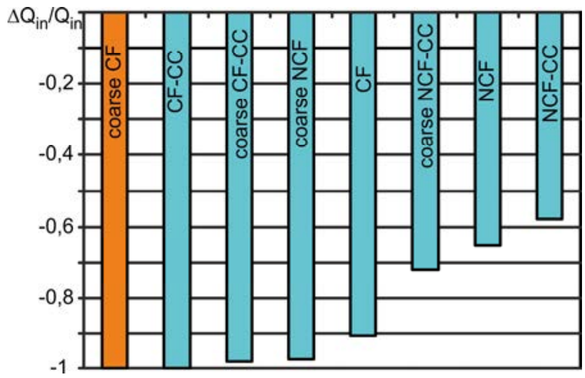
As it is possible to see from fig. 5, models taking into account streamline curvature have the least error. Unlike calculations VCS with radial diffuser, here not all models could predict presence sucking fluid flow rate in the device. So, for example, calculation on model coarse CF, has led to the negative sucking fluid flow rate, i.e. to ejection of a working flow from the vortex chamber outside.



a



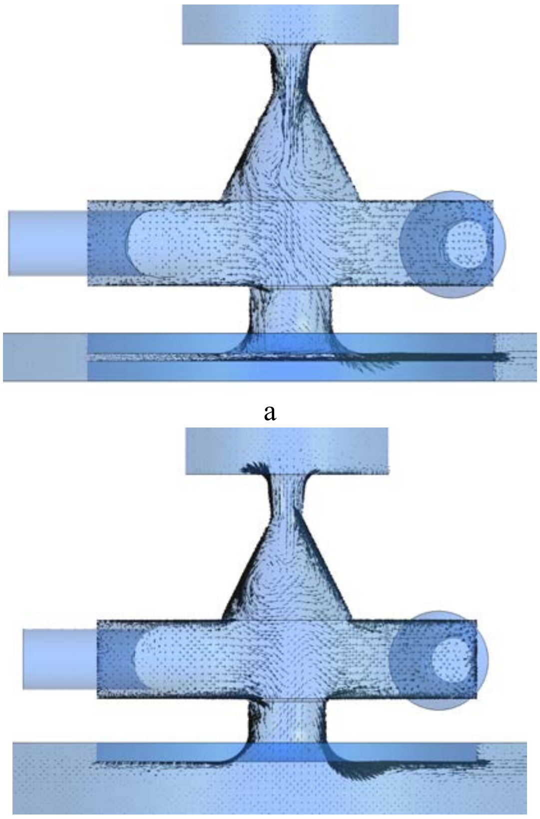
b



c

Fig. 5. Errors of fluid flow calculation results in VCS without the radial diffuser: a – fluid flow rate in the exit channel; b – fluid flow rate in the supply channel; c – fluid flow rate sucked in the device

At calculations VCS without the radial diffuser, simulation inaccuracy of the sucked fluid flow rate more than inaccuracy for VCS with the diffuser. For such construction of a supercharger crucial there is use of model with rotation-curvature correction since the sucking fluid flow rate, for many calculation models, on order surpasses the sucking fluid flow rate calculated without curvature correction.



a

b

Fig. 6. Profiles of velocity in VCS: a – with the radial diffuser; b – without a radial diffuser

Originating negative the sucking fluid flow rate is well visible on fig. 6 where the field of velocity vectors in VCS is demonstrated. The sucking fluid flow rate discharge rate in supercharger formed on a axial of the vortex chamber, thrown out flow of the device – on periphery of the axial channel of an input that it is possible to see on fig. 6, b.

Conclusions

On the basis of the numerical decision of the Reynolds averaged Navier-Stockes equations by means use specialised program complexes verification of mathematical modeling of a fluid flow in vortex chamber supercharger is effected.

It is better to apply model of incompressible fluid to calculations VCS with turbulence model to the effects of streamline curvature and system rotation.

Use of rotation-curvature correction allows to simulate the fluid flow rate of suction on 5–15 % more precisely, owing to more exact calculation of vacuum magnitude on axis.

At calculations VCS without a radial diffuser, simulation inaccuracy of the sucking fluid flow rate more than inaccuracy for VCS with a diffuser. For such construction of a supercharger crucial there is use of model with rotation-curvature correction since the sucking fluid flow rate, for many calculation models, on order surpasses the sucking fluid flow rate calculated without curvature correction.

References

1. Приходько А.А. Компьютерные технологии в аэрогидродинамике и тепломассообмене / А.А. Приходько. – К.: Наукова думка, 2003. – 380 с.
2. Солодов В.Г. Современное состояние проблемы моделирования крупномасштабной турбулентности / В.Г. Солодов // Вісник НТУ«ХПІ». Серія: Гідравлічні машини та гідроагрегати. – 2016. – № 20 (1192). – С. 108–115.
3. Гарбарук А.В. Моделирование турбулентности в расчетах сложных течений: учебное пособие / А.В. Гарбарук, М.Х. Стрелец, М.Л. Шур. – С.Пб.: Изд-во Политехн. ун-та, 2012. – 88 с.
4. Syomin D. Vortex mechanical devices in control systems of fluid mediums / D. Syomin, V. Pavljuchenko, Y. Maltsev, A. Rogovoy, D. Dmitrienko // Teka Komisji Motoryzacji i Energetyki Rolnictwa. – 2010. – Vol. 10. – P. 440–445.
5. Баранов Ю.Д. Обоснование параметров и режимов работы систем гидротранспорта горных предприятий / Ю.Д. Баранов, Б.А. Блюсс, Е.В. Семененко, В.Д. Шурыгин. – Днепропетровск: Новая идеология, 2006. – 416 с.
6. Соколов Е.Я. Струйные аппараты / Е.Я. Соколов, Н.М. Зингер. – 3-е изд., перераб. – М.: Энергоатомиздат, 1989. – 352 с.
7. Суслов А.Д. Вихревые аппараты / А.Д. Суслов, С.В. Иванов, А.В. Мураш-

кин, Ю.В. Чижиков. – М.: Машиностроение, 1985. – 256 с.
8. Syomin D. Features of a working process and characteristics of irrotational centrifugal pumps / D. Syomin, A. Rogovyi // Procedia Engineering. – 2012. – Vol. 39. – P. 231–237. doi: 10.1016/ j.proeng. 2012.07.029.
9. Роговой А.С. Энергетическая эффективность пневмотранспортних установок / А.С. Роговой // Вісник СНУ ім. В. Даля. – 2016. – №1 (225). – С. 189–196.
10. Syomin D. Power characteristics of superchargers with vortex work chamber / D. Syomin, A. Rogovoy // Teka Komisji Motoryzacji i Energetyki Rolnictwa. – 2010. – Vol. 19. – P. 232–240.
11. Сёмин Д.А. Экспериментальные исследования характеристик струйно-вихревого насоса / Д.А. Сёмин, А.С. Роговой // Вісник СумДУ. – 2005. – 12(84). – С. 64–70.
12. Роговий А.С. Удосконалювання енергетичних характеристик струминних нагнітачів: дис... канд. техн. наук: 05.05.17 / А.С. Роговий. – Луганськ, 2007. – 193 с.
13. Beck Jeffrey L. Vortex injection method and apparatus. U.S. Patent No. 4,449,862. 22 May 1984.
14. Деклараційний патент на корисну модель 9805, МПК B65G53/30 Струминний насос. Сьомін Д.О., Роговий А.С.; заявник і патентовласник СНУ імені В. Даля. – № u200503142. заявл. 05.04.2005; опубл. 17.10.2005, Бюл. №10. – 3 с.
15. Сьомін Д.О. Вплив умов входу середовища, що перекачується, на енергетичні характеристики вихрекамерних насосів / Д.О. Сьомін, А.С. Роговий // Вісник Національного технічного університету «ХПІ»: зб. наук. пр. Серія: Гідравлічні машини та гідроагрегати. – 2015. – № 3 (1112). – С. 130–136.
16. Shur M. L. Turbulence modeling in rotating and curved channels: assessing the Spalart-Shur correction / M. L. Shur // AIAA journal. – 2000. – Vol. 38, №. 5. – P. 784–792.
17. Alahmadi Y.H. Modified shear stress transport model with curvature correction for the prediction of swirling flow in a cyclone separator / Y.H. Alahmadi, A.F. Nowakowski // Chemical Engineering Science. – 2016. – Vol. 147. – P. 150–165.
18. Yin J. Large eddy simulation of unsteady flow in vortex diode / J. Yin, L. Jiao, L. Wang // Nuclear Engineering and Design. – 2010. – Vol. 240, no. 5. – P. 970–974.
19. Smirnov P.E. Sensitization of the SST turbulence model to rotation and curvature by applying the Spalart–Shur correction term / P.E. Smirnov, F.R. Menter // Journal of Turbomachinery. – 2009. – Vol. 131, no. 4. – P. 041010.
20. Jasak H. OpenFOAM: open source CFD in research and industry / H. Jasak // International Journal of Naval Architecture and Ocean Engineering. – 2009. – 1(2). – P. 89–94.
21. Oberleithner K. Formation and flame-induced suppression of the precessing vortex core in a swirl combustor: experiments and linear stability analysis / K. Oberleithner, M. Stöhr, S.H. Im, C.M. Arndt & A.M. Steinberg // Combustion and Flame. – 2015. – Vol. 162, no. 8. – P. 3100–3114.
22. Syomin D. Mathematical simulation of gas bubble moving in central region of the short vortex chamber / D. Syomin, A. Rogovyi // TEKA. Commission of motorization and energetics in agriculture. Lublin-Lugansk. XB. TEKA Kom. Mot. Energ. Roln. – OL PAN. – 2012. – Vol. 12, no. 4. – P. 279–284.

References

1. Prikhod'ko A. A. *Komp'yuternye tekhnologii v aerogidrodinamike i teplomassoobmene* [Computer technologies in aerohydrodynamics and heat- mass exchange], Kiev, Naukova dumka Publ., 2003, 380 p.
2. Solodov V.G. *Sovremennoe sostoyanie problemy modelirovaniya krupnomasshtabnoi turbulentnosti* [Current state of the problem of large-scale turbulence modeling]. *Visnyk NTU «KhPI». Seriya: Hidravlichni mashyny ta hidroahrehaty.* 2016. no. 20 (1192), pp. 108–115.
3. Garbaruk A.V., Strelec M.H., Shur M.L. *Modelirovanie turbulentnosti v raschetah slozhnyh techenij* [Turbulence modelling in calculations of difficult flows], Saint Petersburg, Politehn. University Publ., 2012. 88 p.
4. Syomin D., Pavljuchenko V., Maltsev Y., Rogovoy A., Dmitrienko D. Vortex mechanical devices in control systems of fluid mediums. *Teka Komisji Motoryzacji i Energetyki Rolnictwa.* 2010. Vol. 10. pp. 440–445.

5. Baranov Yu. D., Blyuss B.A., Semenenko E.V., Shurygin V.D. *Obosnovanie parametrov i rezhimov raboty sistem gidrotransporta gornykh predpriyatii* [Substantiation of parameters and environment of hydraulic handling systems of mining companies]. Dnepropetrovsk, Novaya ideologiya Publ., 2006, 416 p.
6. Sokolov E.Ja., Zinger N.M. *Strujnye apparaty* [Jet apparatus]. 3rd ed., Moscow, Jenergoatomizdat Publ., 1989, 352 p.
7. Suslov A.D., Ivanov S.V., Murashkin A.V., Chizhikov Ju.V. *Vihrevye apparaty* [Vortex Apparatus]. Moscow, Mashinostroenie Publ., 1985, 256 p.
8. Syomin D., Rogovyi A. Features of a working process and characteristics of irrotational centrifugal pumps. Procedia Engineering, 2012, vol. 39, pp. 231–237. doi:10.1016/j.proeng.2012.07.029.
9. Rogovoi A.S. *Energeticheskaya effektivnost' pnevmotransportnikh ustanovok.* [Energy efficiency of pneumatic installations]. *Visnyk SNU im. V. Dalya.* 2016, no. 1(225), pp. 189–196.
10. Syomin D., Rogovoy A. Power characteristics of superchargers with vortex work chamber. TEKA. Commission of motorization and power industry in agriculture. 2010, no. 19., pp. 232–240.
11. Syomin D.O., Rogovyi A.S. *Eksperimentalnye issledovanija kharakeristic stujnovihrevogo nasosa* [Experimental investigations of the characteristics of vortex-fluid pumps], Visnik SumDU, 2005, no 12 (84), pp. 64–70.
12. Rogovyi A.S. *Udoskonalyuvannya enerhetychnykh kharakterystyk strumynnykh nahnitachiv*. Diss, kand. tekhn. nauk [Perfecting of the power characteristics of inkjet superchargers]. Lugansk, 2007, 193 p.
13. Beck, Jeffrey L. Vortex injection method and apparatus. Patent U.S. no. 4,449,862. 22 May 1984.
14. Syomin D.O., Rogovyi A.S. *Strumynnyy nasos* [Jet pump]. Patent Ukraine, no. u200503142, 2005.
15. Syomin D.O., Rogovyi A.S. *Vplyv umov vkhodu seredovyshcha, shcho perekachuyet'sya, na enerhetychni kharakterystyky vykhrekamernykh nasosiv* [Influence of transfer mediums input conditions on power characteristics of the vortex chamber pumps]. *Visnyk NTU«KhPI». Seriya: Hidravlichni mashyny ta hidroahrehaty*, 2015, no. 3 (1112), pp. 130–136.
16. Shur M. L., Strelets M. K., Travin A. K., & Spalart P. R. Turbulence modeling in rotating and curved channels: assessing the Spalart-Shur correction. *AIAA journal*, 2000, Vol. 38, no. 5, pp. 784–792.
17. Alahmadi Y.H., Nowakowski A.F. Modified shear stress transport model with curvature correction for the prediction of swirling flow in a cyclone separator. *Chemical Engineering Science*, 2016, Vol. 147, pp. 150–165.
18. Yin J., Jiao L., Wang L. Large eddy simulation of unsteady flow in vortex diode. *Nuclear Engineering and Design*, 2010, Vol. 240, no. 5, pp. 970–974.
19. Smirnov P. E., Menter F. R. Sensitization of the SST turbulence model to rotation and curvature by applying the Spalart–Shur correction term. *Journal of Turbomachinery,* 2009, Vol. 131, no. 4. doi: 10.1115/1.3070573
20. Jasak H. OpenFOAM: open source CFD in research and industry. *International Journal of Naval Architecture and Ocean Engineering*, 2009, no. 1(2), pp. 89–94.
21. Oberleithner K., Stöhr M., Im S.H., Arndt C. M. & Steinberg A. M. Formation and flame-induced suppression of the precessing vortex core in a swirl combustor: experiments and linear stability analysis. *Combustion and Flame*, 2015, Vol. 162, no. 8, pp. 3100–3114.
22. Syomin D., Rogovyi A. Mathematical simulation of gas bubble moving in central region of the short vortex chamber. *TEKA. Commission of motorization and energetics in agriculture.* 2012, Vol. 12, no. 4, pp. 279–284.

Рецензент: П.Н. Андренко, профессор, д.т.н., НТУ «ХПИ».

УДК 625.76.08 : 517.938

ANALYSIS OF MOTOR-GRADER LOADING ON THE BASIS OF FRACTAL DIMENSION

A. Polyarus, Prof., D. Sc. (Eng.), R. Paschenko, Prof., D. Sc. (Eng), E. Poliakov, Ph. D. (Eng.), A. Lebedinskyi, student, Kharkov National Automobile and Highway University

Abstract. *The possibility of fractal dimension application for analysis of grader loading modes is considered in the article. The fractal dimensions of experimental dependences of load on the coupling pin of the grader at its different working conditions are calculated. It is determined that the magnitude of the fractal dimension allows to estimate the highest and lowest load of the grader.*

Key words: *fractal dimension, grader blade, load mode.*

АНАЛІЗ НАВАНТАЖЕННЯ АВТОГРЕЙДЕРА НА ОСНОВІ ФРАКТАЛЬНОЇ РОЗМІРНОСТІ

О.В. Полярус, проф., д.т.н., Р.Е. Пащенко, проф., д.т.н., Є.О. Поляков, к.т.н., А.В. Лебединський, студент,
Харківський національний автомобільно-дорожній університет

Анотація. *Розглянуто можливість використання фрактальної розмірності для аналізу режимів навантаження автогрейдера. Розраховано фрактальні розмірності експериментальних залежностей навантаження на шворні автогрейдера за різних робочих умов автогрейдера. Показано, що величина фрактальної розмірності дозволяє оцінити найбільші та найменші навантаження автогрейдера.*

Ключові слова: *фрактальна розмірність, відвал автогрейдера, режим навантаження.*

АНАЛИЗ НАГРУЗКИ АВТОГРЕЙДЕРА НА ОСНОВЕ ФРАКТАЛЬНОЙ РАЗМЕРНОСТИ

А.В. Полярус, проф., д.т.н., Р.Э. Пащенко, проф., д.т.н., Е.А. Поляков, к.т.н., А.В. Лебединский, студент,
Харьковский национальный автомобильно-дорожный университет

Аннотация. *Рассмотрена возможность использования фрактальной размерности для анализа режимов нагрузки автогрейдера. Рассчитаны фрактальные размерности экспериментальных зависимостей нагрузки на шкворне при различных условиях работы автогрейдера. Показано, что величина фрактальной размерности позволяет оценить наибольшие и наименьшие нагрузки автогрейдера.*

Ключевые слова: *фрактальная размерность, отвал автогрейдера, режим нагрузки.*

Introduction

The main working body of any motor-grader (grader) is a fully steerable blade with knives mounted at an angle to its longitudinal axis. Depending on the soil structure the grader blade is under the influence of alternating dynamic loads. In its turn the impact of loads can lead to the machine failures due to the appearance of cracks in metal structures. In addition to the soil structure, the load blade grader is also influenced by the parameters of its work, such as grader blade deflection, turning the angle of the blade, rotation frequency of the motor shaft, etc. The influence of these parameters can be investigated by using the vibration diagnostic methods. Various parameters of the machine lead to different modes of dynamic loads. The level and nature of the vibrations acting on the machine construction due to alternating loads can be estimated by analyzing the time-stress dependences on the grader coupling pin.

Analysis of publications

Investigation of earth-moving machines parameters is carried out to predict the possible malfunctions of their work. In this case various vibration diagnostics methods are used and some of them are presented in [1]. The research results presented in [2] show that it is possible to estimate the dynamic forces that act on the metal structure by using the results of dynamics of earth-moving machines analysis with a sharp increase in the resistance of movement. To carry out such an analysis it is necessary to obtain the original data about loads on the elements of machines received in different modes of their operation. In [3] the possibility of the phase portraits application for classification of grader load modes is considered. On the basis of experimental data describing the loads on the grader coupling pin the phase portraits for different modes of its work were constructed.

Purpose and problem statement

It is expediently the determination of grader load modes to carry out on the basis of experimentally obtained signal implementations which describe the load on the grader coupling pin. In this case, different load modes lead to various forms of recorded signals. The value of the fractal dimension (FD) [4] can be the characteristic of signal shape and the contrast of these shapes due to the loading on the grader coupling pin, in turn, leads to different values of FD. Therefore, the analysis of the possibility of FD using for estimation of load on grader coupling pin is of practical importance.

The aim of the article is estimation of fractal dimension usability for determination of grader load mode changes.

Obtaining the experimental data

An essential element of grader is a ball coupling pin through which the tractive forces from the drive wheels to the grader blade during performing of work operations are transmited. Therefore, it is advisable to estimate the loads acting on this element of grader. For this purpose, the field measuring experiments were carried out on the test area of our university. As a research facility the grader DZK-251 produced at Kriukov's Railway Car Building Plant was used. The procedure of experimental research conducting and measuring system that was used at the same time are described in [5]. Measurements of stress on the grader pin were performed with using of strain gauge transducers (sensors).

Changing parameters of grader during the experiments are shown in Fig. 1.

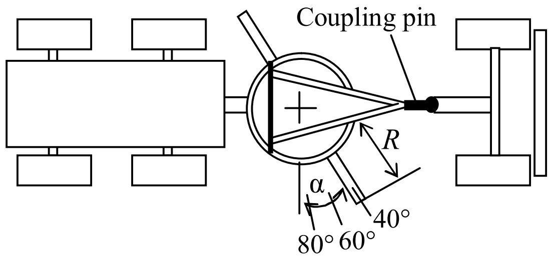


Fig. 1. Changing parameters of motor-grader

In Fig. 1 the letter α means grader blade rotation angle and R – grader blade deflection. The figure also shows the point of load measurement on the pin.

During the experiments the influence of various positions of the grader blade at the stress arising on the grader pin was evaluated. The grader parameters were changed as follows: grader blade deflection (R) was equal to 0 m, 0,7 m and 1,4 m; blade angle of rotation (α) – 40°, 60° and 80°; number of the motor shaft revolutions (f) – 900 rev/min, 1100 rev/min and 1300 rev/min. Indications of the sensors in the form of digital data from the measurement system were recorded in the permanent memory of the computer. In [3], all the time dependences of the stress (σ, MPa) on the grader pin from variable parameters outlined above were presented and detailed analysis of these temporary implementations was carried out. In this paper, for example, we'll adduce only temporary dependences of stresses (Fig. 2): on the grader pin (in voltage) when grader blade deflection (R) is respectively equal to 0 m (a), 0,7 m (b) and 1,4 m (c) for fixed values of the blade angle (α = 80°) and the number of the motor shaft revolutions (f = 1300 rev/min). The nature of time realizations for the other variable parameters is similar to those which are shown in Fig. 2.

In the figures on the ordinate Y the stress on the grader pin (σ, MPa) is represented and on the horizontal axis I – execution time of work operations. In this case the value of I = 1 × 104 corresponds to the time t = 10 s. The time t = 10 s corresponds to the value of $I = 1{\times}10^4$.

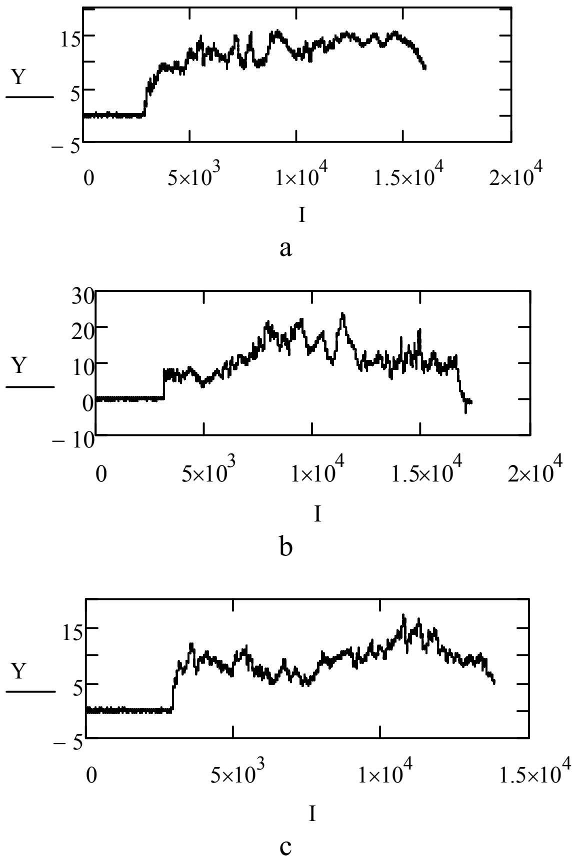


Fig. 2. The time realizations of stress signals on grader pin at R = 0 m (a), R = 0,7 m (b), R = 1,4 m (c)

Fig. 2 shows that at the first 2–3 seconds grader was working without interaction with the ground and the load on the pin was minimal. The stress on the pin was increasing abruptly during the interaction of the blade and the ground. Later the load signal on the pin was changed irregularly and had an indented character. It should be noted that the indented nature of the stress changing is practically independent of the grader blade deflection.

Thus, from the analysis of the stresses on the pin it can be concluded if there was a load on the motor grader or it was working without load. However, over the time realizations of stresses it is practically impossible to determine under what parameters of grader the load is maximal since the maximum value of the signal amplitude is almost the same and the nature of its changes is indented at any parameters of grader.

As well as in the study of stresses at the grader blade deflection during fulfillment of working operations at different turning angle of the blade the stress amplitude on the pin is increased and has an indented character when the blade interacts with a soil.

The character of time realizations of the stress on the grader pin at different frequency of the motor shaft which is given in [3] doesn’t differ from the character of time realizations that has been shown in the Fig. 2. Initially, the work operations are done with no load and then the stresses on the pin are increasing abruptly and have an irregular character.

Thus it can be easily defined the time of motor-grader loading occurrence with the help of time realizations of loadings (by an abrupt increase in the stress amplitude), however, it is hardly to determine the relationship between motor-grader loading conditions and its parameters as a part of the stress time realizations of working operations. For analysis of signal forms that describe the load on the grader pin we’ll use the value of the fractal dimension.

Calculation of fractal dimension

In practice, the dimension of Hausdorf- Besicovitch D [6] is often used for estimation of fractal characteristics of various structures

$$D = \lim_{\varepsilon \to 0} \frac{\log N(\varepsilon)}{\log\left(\frac{1}{\varepsilon}\right)},$$

where $N(\varepsilon)$ – number of covering elements; ε – the side length of covering element.

All existing FD calculation methods include the calculation of volume, area or length of the fractal shape and its changes during scaling.

The method of the fractal dimension determining with using of signals covering by squares comprises the following steps [7].

1. Some value of ε is defined, the time domain of the source data existence is divided into squares with a side ε and the number of squares that covered all the known points (Fig. 3) are calculated. As a result, one value $N(\varepsilon)$ is obtained.

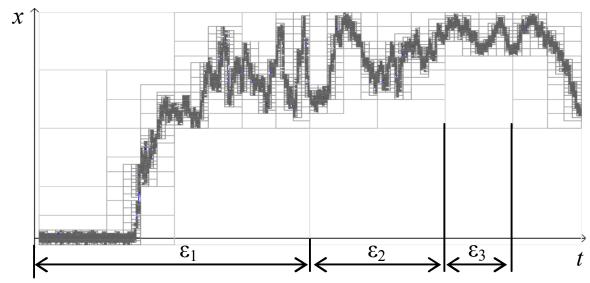


Fig. 3. Example of arrangement of the original sample covering

2. Assume that the calculations of $N(\varepsilon)$ were performed for different lengths of the side ε (at Fig. 3 these values are ε_1, $\varepsilon_2 = \varepsilon_1/2$, $\varepsilon_3 = \varepsilon_1/4$). As it follows from the definition of FD [4], for small values of ε the number of the covering elements $N(\varepsilon)$ should be equal to $\sim\varepsilon^{-D}$ and in this case $\log N(\varepsilon) = -D\cdot\log \varepsilon$. Now, with using of the obtained data the dependence $\log N(\varepsilon)$ versus $\log\left(\frac{1}{\varepsilon}\right)$ (Fig. 4) is plotted.

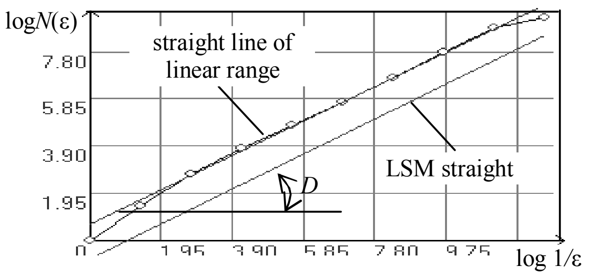


Fig. 4. The determination of Hausdorf-Besicovitch dimension with the use of covering method

3. The FD estimating is reduced to the search of «the most linear» area of the relationship between $\log N(\varepsilon)$ and $\log 1/\varepsilon$; construction of the linear approximation of the form $\log N(\varepsilon) = -b\cdot\log\varepsilon + C$ in this area, for example, by using of least squares method (LSM) [8]; FD estimation by evaluation of LSM line slope.

It should be noted that the choice of the most linear area in this algorithm is a difficult thing to formalize. Approximation of a linear part of the plot with using of LSM does not always produce reliable results. The straight line plotted on the basis of linear approximation for 10 points (LSM straight line) is shown in Fig. 4. In addition, another straight line is depicted at the same figure according to the selected 7 points when choosing the linear range of the plot (straight line area). It can be seen that the slopes of the lines do not differ significantly, however, FD is calculated more precisely when choosing linear range. Thus it needs to calculate the slope of approximating line using linear range of plot of $\log N(\varepsilon)$ as a function of $\log 1/\varepsilon$.

Analysis of grade loads using the fractal dimension

Let's consider the possibility of FD using for the analysis of grader load. For this purpose, we calculate the values of the FD for the time realizations of stresses on the pin at different grader parameters (grader blade deflection, grader blade rotation angle, frequency of the motor shaft). The fractal dimension was calculated by using of the method described above.

The FD (D) dependences on the grader blade rotation angle with blade deflection $R = 0$ m and frequency of the motor shaft: $f = 900$ rev/min (solid line), $f = 1100$ rev/min (dotted line) and $f = 1300$ rev/min (dash-and-dot line) are shown in Fig. 5.

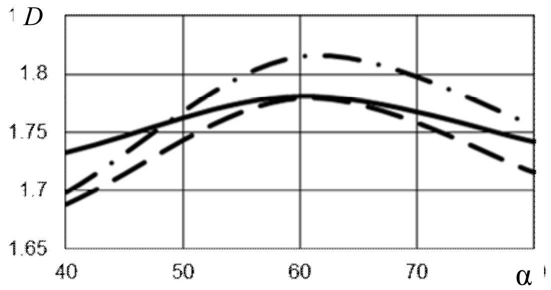


Fig. 5. FD dependences on the grader blade deflection at $R = 0$ m

As it can be seen from the curves in Fig. 5, the maximum value of FD and consequently the greatest unevenness of time stress realizations occurs when the grader blade rotation angle value is equal to $\alpha = 60°$. Moreover, the greatest FD value is fixed when the frequency of the motor shaft reaches the value of $f = 1300$ rev/min. The minimum values of FD for three curves are observed at $\alpha = 40°$. However, in contrast to the greatest FD values among the maximum values, the greatest FD values among the minimum values were fixed at $f = 900$ rev/min. Also, as it follows from Fig. 5, increasing of the grader blade deflection more than 60° leads to the FD decrease.

Fig. 6 shows that similar behavior is observed when the grader blade deflection value is $R = 1{,}4$ m. However, the highest values are obtained at two angles of blade deflection $\alpha = 40°$

and α = 60°, but not at a single one as it was previously.

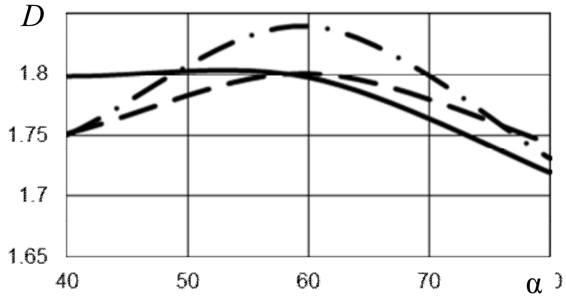


Fig. 6. FD dependences on grader blade deflection at R = 1,4 m

However, the FD behavior differs from the cases examined above when grader blade deflection value is R = 0,7 m (Fig. 7).

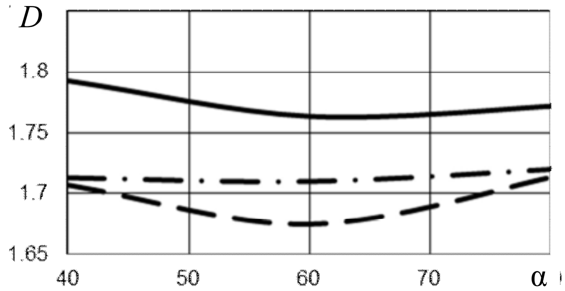


Fig. 7. FD dependences on grader blade deflection at R = 0,7 m

Fig. 7 shows that FD is less dependent on the grader blade deflection and at frequency of the motor shaft f = 1300 rev/min it practically doesn't change. The maximum values of FD were obtained when f = 900 rev/min, but the minimum values correspond to α = 60°. Also the minimum values of FD were calculated at f = 1100 rev/min.

Thus the analysis of FD values for various grader blade deflections showed that when grader work operations are in progress the maximum loads (unevenness of initial signal) are observed at α = 60°, f = 1300 rev/min and R = 1,4 m, and minimum – at f = 1100 rev/min and R = 0,7 m. Let's consider the FD of time stress realizations on the pin for other grader parameters.

The FD dependences on grader blade deflection at blade rotation angle α = 40° and frequency of the motor shaft f = 900 rev/min (solid line), f = 1100 rev/min (dotted line) and f = 1300 rev/min (dash-and-dot line) are presented in Fig. 8. Fig. 8 shows that with the increasing of grader blade deflection the FD value also increases due to greater unevenness of measured signals. The maximum FD values and consequently the big loads were fixed at f = 900 rev/min.

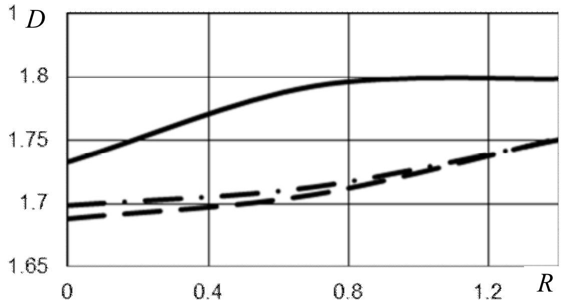


Fig. 8. FD dependences on grader blade deflection at α = 40°

Meanwhile, when the grader blade rotation angle is α = 60°, the FD minimum value occurs at grader blade deflection value R = 0,7 m for any frequency of the motor shaft value (Fig. 9) and FD minimum value – at f = 1100 rev/min. Thus, the minimum loads of grader will be received when its blade deflection R = 0,7 m and f = 1100 rev/min.

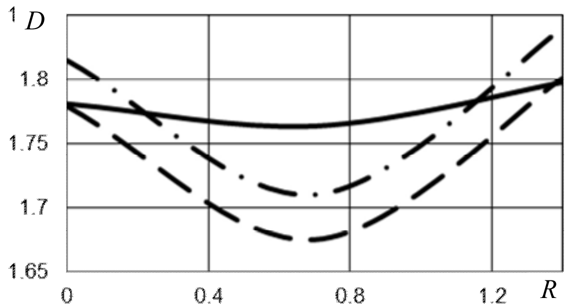


Fig. 9. FD dependences on grader blade deflection at α = 60°

It should be noted that similar FD behavior is held at f = 1100 rev/min and f = 1300 rev/min (Fig. 10) when α = 80° but at f = 900 rev/min the maximum value of FD was received at R = 0,7 m.

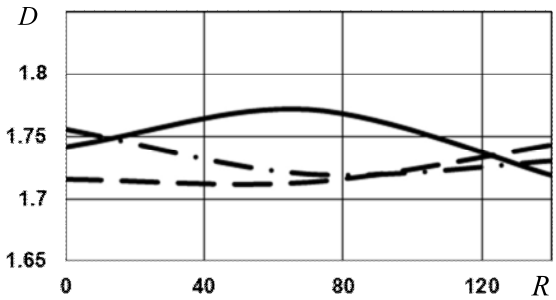


Fig. 10. FD dependences on grader blade deflection at α = 80°

Thus analysis of dependences of FD values on grader blade deflection has revealed that the maximum FD values were achieved at *f* = 900 rev/min for the all grader blade rotation angles and the minimum value at α = 60°, *f* = 1100 rev/min and *R* = 0,7 m.

The dependences of FD values on the frequency of the motor shaft at grader blade rotation angle at α = 40° and grader blade deflection at *R* = 0 m (solid line), *R* = 0,7 m (dotted line) и *R* = 1,4 m (dash-and-dot line) are illustrated in Fig. 11.

As it follows from Fig. 11, the greatest values of FD are observed at *f* = 900 rev/min and the minimum values – at *f* = 1100 rev/min for any grader blade deflection value. The minimum FD value was fixed at *f* = 1100 rev/min and *R* = 0 m.

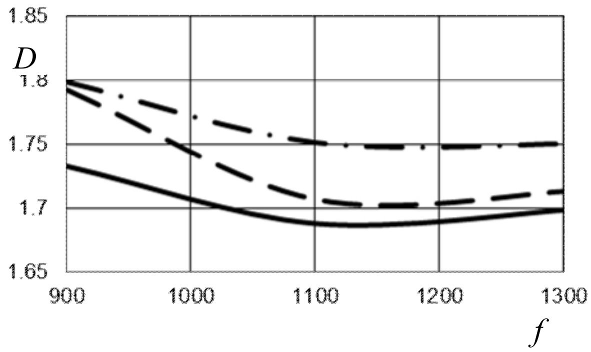


Fig. 11. FD dependences on frequency of the motor shaft at α = 40°

Fig. 12 shows that at grader blade rotation angle α = 60° the minimum FD values also occur at *f* = 1100 rev/min, however the maximum FD values were not fixed at *f* = 900 rev/min but they were detected at *f* = 1300 rev/min, *R* = 0 m and *R* = 1,4 m.

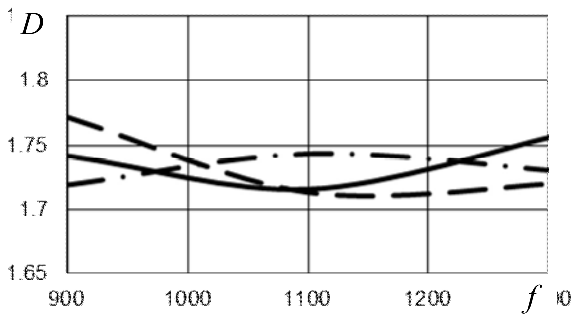


Fig. 12. FD dependences on frequency of the motor shaft at α = 60°

The significant differences in FD values behavior occur at grader blade rotation angle value α = 80° (Fig. 13). In this case the spread of FD values is insignificant, i. e. it depends less on frequency of the motor shaft. In addition, it's difficult to estimate the maximum and minimum FD values at various frequency of the motor shaft and grader blade deflection values, because practically they don't differ.

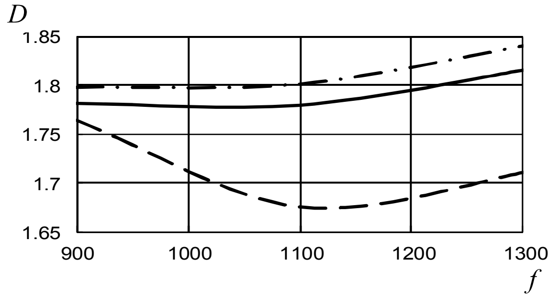


Fig. 13. FD dependences on frequency of the motor shaft at α = 80°

Thus analysis of dependence of FD values on frequency of the motor shaft showed that the minimum FD values can be seen at *f* = 1100 rev/min, α = 60°, *R* = 0,7 m, and the maximum values – at *f* = 900 rev/min, α = 40° and *R* = 1,4 m and also at *f* = 1300 rev/min, α = 60° and *R* = 1,4 m.

Conclusions

The calculations of FD can be used for numerical estimation of irregularities of signals received from the sensors mounted on motor-grader pin.

Analysis of FD of experimental stress signals measured on motor-grader pin showed that its value depends on grader parameters when work operations are in progress.

Analysis of dependences of FD values on blade rotation angle, grader blade deflection and frequency of the motor shaft showed that when work operations are in progress the minimum loadings occur at α = 60°, *R* = 0,7 m и *f* = 1100 rev/min and the maximum loadings – at α = 40°, α = 60°, *R* = 1,4 m и *f* = 900 rev/min.

During further research it is advisable to consider the possibility of the fractal dimension using for the analysis of phase portraits of stress signals on motor-grader pin.

In the further work it is necessary to assess the possibility of using of the work results to improve the methods for determining grader operating modes used in current normative documents.

References

1. Барков А.В. Мониторинг и диагностика роторных машин по вибрации / А.В. Барков, Н.А. Баркова, А.Ю. Азовцев. – С.Пб.: Изд. СПМТУ, 2000. – 158 с.
2. Холодов А.М. Динамика землеройно-транспортных машин при резком возрастании сопротивлений / А. М. Холодов // Труды ХАДИ. – 1960. – Вып. 22. – С. 71–81.
3. Определение режима нагрузки автогрейдера с использованием фазовых портретов / А.В. Полярус, Р.Э. Пащенко, Е.А. Поляков, Я.С. Бровко // Строительство, материаловедение, машиностроение: сб. науч. тр. – 2013. – Вип. 72. – С. 160–169.
4. Пащенко Р.Э. Основы теории формирования фрактальных сигналов / Р.Э. Пащенко. – Харьков: ХООО «НЭО "Эко-Перспектива"», 2005. – 296 с.
5. Шевченко В.А. Экспериментальная оценка влияния положения грейдерного отвала на нагрузки, действующие в основной раме автогрейдера ДЗК-251 / В.А. Шевченко, А.А. Резников, В.В. Крецул // Вестник ХНАДУ: сб. науч. тр. – 2010. – Вып. 49. – С. 62–66.
6. Федер Е. Фракталы / Е. Федер. – М.: Мир, 1991. – 254 с.
7. Малинецкий Г.Г. Современные проблемы нелинейной динамики / Г.Г. Малинецкий, А.Б. Потапов. – М.: Едиториал УРСС, 2002. – 360 с.
8. Львовский Е.Н. Статистические методы построения эмпирических формул / Е.Н. Львовский. – М.: Высшая школа, 1988. – 240 с.

References

1. Barkov A.V., Barkova N.A., Azovtsev A.Yu. *Monitoring i diagnostika rotornykh mashin po vibratsii* [Monitoring and diagnostics of rotor machines by vibration], Sankt-Peterburg, SPMTU Publ., 2000. 158 p.
2. Kholodov A.M. *Dinamika zemleroyno-transportnykh mashin pri rezkom vozrastanii soprotivleniy* [Dynamics of earth-moving – transport machines with a sharp increase of resistance]. Trudy KhADI. 1960. Vol. 22. pp. 71–81.
3. Polyarus A.V., Pashchenko R.E., Polyakov Ye.A., Brovko Ya.S. *Opredelenie rezhima nagruzki avtogreydera s ispolzovaniem fazovykh portretov* [Determination of load regime of grader with using of the phase portraits]. Stroitelstvo, materialovedenie, mashinostroenie: sbornik nauchnykh trudov. 2013. Vol. 72. pp. 160–169.
4. Pashchenko R.E. *Osnovy teorii formirovaniya fraktalnykh signalov* [Fundamentals of the theory of fractal signal creating]. Kharkov, KhOOO «NEO "EkoPerspektiva"» Publ., 2005. 296 p.
5. Shevchenko V.A., Reznikov A.A., Kretsul V.V. *Eksperimentalnaya otsenka vliyaniya polozheniya greydernogo otvala na nagruzki, deystvuyushchie v osnovnoy rame avtogreydera DZK-251* [Experimental estimation of grader blade position acting in main frame of autograder]. Vestnik KhNADU. 2010. Vol. 49. pp. 62–66.
6. Feder Ye. Fraktaly [Fractals]. Moscow, Mir Publ., 1991. 254 p.
7. Malinetskiy G.G., Potapov A.B. *Sovremennye problemy nelineynoy dinamiki* [The modern problems of nonlinear dynamics]. Moscow, Yeditorial URSS Publ., 2002. 360 p.
8. Lvovskiy Ye.N. *Statisticheskie metody postroeniya empiricheskikh formul* [The statistical methods of empirical formulas building]. Moscow, Vysshaya shkola Publ., 1988. 240 p.

Рецензент: В.А. Шевченко, доцент, к.т.н., ХНАДУ.

УДК 534.1

RESONANT OSCILLATIONS OF A ROTOR ON AXIALLY PRELOADED BALL BEARINGS UNDER THE JOINT ACTION OF UNBALANCE AND VIBRATION OF SUPPORTS

S. Filipkovskyi, Assoc. Prof., Ph. D (Eng.), R. Makovyey, Asst. Prof., Kharkov National Automobile and Highway University

Abstract. *The model of nonlinear vibrations of the rotor supported by axial preload angular ball-bearings was developed. The frequency response of the system is obtained by the continuation method at joint action of unbalance and vibration of supports. Analysis showed that vibrations occurred not only at fundamental resonant frequencies but also at frequencies less than the resonant ones in integer times. The character of periodical decisions is investigated.*

Key words: *rotor, angular ball bearing, nonlinear vibrations, resonance.*

РЕЗОНАНСНЫЕ КОЛЕБАНИЯ РОТОРА НА ШАРИКОПОДШИПНИКАХ С ОСЕВЫМ НАТЯГОМ ПРИ СОВМЕСТНОМ ДЕЙСТВИИ ДИСБАЛАНСА И ВИБРАЦИИ ОПОР

С.В. Филипковский, доц., к.т.н., Р.Г. Маковей, ст. преп., Харьковский национальный автомобильно-дорожный университет

Аннотация. *Получена модель нелинейных колебаний ротора на радиально-упорных шарикоподшипниках с предварительным осевым натягом. Методом продолжения по параметру получена амплитудно-частотная характеристика системы при совместном действии дисбаланса и вибрации опор. Анализ показал, что колебания возникают не только на основных резонансных частотах, но и на частотах меньше резонансных в целое число раз.*

Ключевые слова: *ротор, радиально-упорный шарикоподшипник, нелинейные колебания, резонанс.*

РЕЗОНАНСНІ КОЛИВАННЯ РОТОРА НА ШАРИКОПІДШИПНИКАХ З ОСЬОВИМ НАТЯГОМ ПРИ СПІЛЬНІЙ ДІЇ ДИСБАЛАНСУ І ВІБРАЦІЇ ОПОР

С.В. Філіпковський, доц., к.т.н., Р.Г. Маковєй, ст. викл., Харківський національний автомобільно-дорожній університет

Анотація. *Отримано модель нелінійних коливань ротора на радіально-упорних шарикопідшипниках з попереднім осьовим натягом. Методом продовження по параметру отримано амплітудно-частотну характеристику системи при спільній дії дисбалансу і вібрації опор. Аналіз показав, що коливання виникають не тільки на основних резонансних частотах, але і на частотах менше резонансних в ціле число разів.*

Ключові слова: *ротор, радіально-упорний шарикопідшипник, нелінійні коливання, резонанс.*

Introduction

Many devices of special vehicles, for example gyroscopic instruments, fans, centrifugal compressors operate under conditions of vibration, which propagates through the machine structure, even in the presence of vibration isolation. The rotors of these units must be protected from impacts that may occur as a result of opening and closing the clearances between the rolling balls and races of bearings under transverse rotor vibration. These rotors are mounted on angular contact ball bearings with axial preload.

Analysis of publications

The equations for determining the non-linear stiffness of preloaded bearings are derived in [1], however, for the carried out in this article research they are linearized. In the article [2], numerically and experimentally there were investigated the transverse vibrations of the preloaded angular-contact ball bearing rotor caused by an unbalance of the disc as well as show their dependence on the nonlinear contact forces. In article [3] there was studied the parametric instability of the shaft with ball bearings under the influence of a variable axial force.

In article [4] there were explored the free oscillations of the preloaded angular-contact ball bearing rotor as well as derived the backbone curves and nonlinear normal modes of oscillations at different angles between the line of action of the contact force and the bearing axis. In work [5], there was analyzed the nonlinear model of ball bearings, obtained on the basis of the formulas given in article [1] and defined the limits of applicability of this model.

Purpose and problem statement

Effect of supports vibration on forced oscillations of the rotor is not investigated so far. The solution to this problem is urgent, since in the nonlinear rotor systems there often occur super- and sub-resonance oscillations. The aim of this study is to investigate the resonant oscillations occurring in the preloaded angular-contact ball bearing rotor caused by the simultaneous action of the unbalance and vibration of supports.

Design model

The rotor is a shaft with a disk fixed eccentrically relative to supports (Fig. 1). Designation and conditions of machines operation, in which they use axial preloaded ball bearings, are such that the co-relation of the length and diameter of the shaft determine the stiffness of the shaft in the order of magnitude more than the rigidity of bearings. Therefore, the shaft is considered to be a non-deformable body, the rotor center of mass is considered to be concentrated in the center of the disc, and the degrees of freedom are the spindles movement relative to the outer bearing rings.

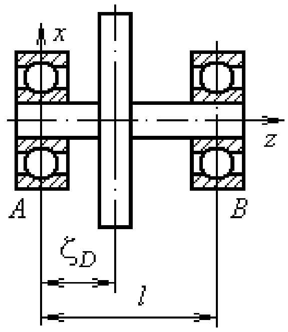


Fig. 1. Rotor design

The components of elastic bearing reactions along the coordinate axes were derived in work [1]. One can consider them to be the components of the vector function $\mathbf{K}(\mathbf{X})$, where $\mathbf{X}$ is the vector of generalized coordinates.

Equations of rotor oscillations

The length of the shaft will be denoted l; movement of the shaft center line in the directions of the coordinate axes u_x, u_y are as follows:

$$u_x(\zeta,t)=x_1(t)\frac{l-\zeta}{l}+x_2(t)\frac{\zeta}{l},$$
$$u_y(\zeta,t)=y_1(t)\frac{l-\zeta}{l}+y_2(t)\frac{\zeta}{l}, \qquad (1)$$

where ζ is the coordinate of the shaft cross-section along the axis z, $x_1(t)$, $x_2(t)$, $y_1(t)$, $y_2(t)$ are generalized coordinates describing the radial movement of spindles; t is a time. The inner rings of ball bearings produce both radial and axial oscillations relative to the outer rings. Let's note that the movement is insufficient compared with the length of the shaft. Then, the longitudinal oscillations of the rotor along the coordinate axis z can be described by a generalized coordinate $u_z=z(t)$.

To generate the equations of motion, one can use the Lagrange equations. Under our assumptions, the expression of the kinetic energy of the shaft T_B as a function of generalized coordinates will be as follows

$$\mathrm{T}_B=\frac{\rho I}{2l}(\dot{y}_1-\dot{y}_2)^2+\frac{\rho I}{2l}(\dot{x}_2-\dot{x}_1)^2+\rho Il\Omega^2-$$
$$-\frac{2\rho I\Omega}{l}(\dot{x}_2-\dot{x}_1)(y_1-y_2)+\frac{\rho Sl}{6}\times \qquad (2)$$
$$\times\left(\dot{x}_1^2+\dot{x}_1\dot{x}_2+\dot{x}_2^2+\dot{y}_1^2+\dot{y}_1\dot{y}_2+\dot{y}_2^2\right)+\frac{\rho Sl}{2}\dot{z}^2,$$

where ρ is density of the shaft material, I and S are the second moment of area and the area of the shaft, respectively; Ω is an angular speed of the rotor. The kinetic energy of the disk T_D as a function of generalized coordinates will be

$$
\begin{aligned}
T_D = &\frac{I_1}{2}\left(\frac{\dot{y}_1 - \dot{y}_2}{l}\right)^2 + \frac{I_1}{2}\left(\frac{\dot{x}_2 - \dot{x}_1}{l}\right)^2 + \\
&+ \frac{I_0}{2}\Omega^2 + I_0\Omega\frac{(\dot{y}_1 - \dot{y}_2)(x_2 - x_1)}{l^2} + \\
&+ \frac{m_0}{2}\left[\dot{x}_1\left(1 - \frac{\zeta_D}{l}\right) + \dot{x}_2\frac{\zeta_D}{l}\right]^2 + \\
&+ \frac{m_0}{2}\left[\dot{y}_1\left(1 - \frac{\zeta_D}{l}\right) + \dot{y}_2\frac{\zeta_D}{l}\right]^2 + \frac{m_0}{2}\dot{z}^2,
\end{aligned}
\quad (3)
$$

where I_1 and I_0 are the diametrical and polar moments of inertia of a disk, respectively, m_0 is the mass of the disk, ζ_D is the disk coordinate along the axis z.

From the assumption that the shaft is non-deformable, it follows that the potential energy of system deformation is represented only by the energy of deformation of bearings $\Pi = \Pi_\Pi(x_1, y_1, x_2, y_2, z)$. Derivatives of the potential energy on generalized coordinates are components of the vector function $\mathbf{K}(\mathbf{X})$.

Damping is due to bearings lubrication, usually it is determined on the basis of experiments and described by the model of viscous friction [6, 7]. In this case, the Rayleigh dissipation function Φ has the form

$$\Phi = \frac{C}{2}\left(\dot{x}_1^2 + \dot{y}_1^2 + \dot{x}_2^2 + \dot{y}_2^2 + \dot{z}^2\right), \quad (4)$$

where C is the damping factor.

Using expressions (1), (2), (3) and (4), one can obtain the equation of oscillations in the matrix form

$$\mathbf{M}\cdot\ddot{\mathbf{X}} + \mathbf{G}\cdot\dot{\mathbf{X}} + \mathbf{C}\cdot\dot{\mathbf{X}} + \mathbf{K}(\mathbf{X}) = \mathbf{Q}(t), \quad (5)$$

where $\mathbf{M}$ is the mass matrix, $\mathbf{G}$ is the gyroscopic matrix, $\mathbf{C}$ is the damping matrix, $\mathbf{Q}(t)$ is the right-hand part vector.

Oscillations are excited by the combined effect of the disk unbalance forces and the vibration of supports, therefore

$$\mathbf{Q}(t) = \mathbf{Q}_D(\Omega, t) + \mathbf{Q}_\Pi(\omega, t), \quad (6)$$

where $\mathbf{Q}_D(\Omega, t)$ is the vector of forces due to unbalance of the disk, $\mathbf{Q}_\Pi(\omega, t)$ – the vector of kinematic excitation of oscillations, ω is the angular frequency of vibration of supports. The first vector is obtained by differentiating (3). Its components have the following form

$$
\begin{aligned}
Q_D(\Omega,t)_1 &= m_0 a\Omega^2\left(1 - \frac{\zeta_D}{l}\right)\cos\Omega t, \\
Q_D(\Omega,t)_2 &= m_0 a\Omega^2\left(1 - \frac{\zeta_D}{l}\right)\sin\Omega t, \\
Q_D(\Omega,t)_3 &= m_0 a\Omega^2\frac{\zeta_D}{l}\cos\Omega t, \\
Q_D(\Omega,t)_4 &= m_0 a\Omega^2\frac{\zeta_D}{l}\sin\Omega t, \\
Q_D(\Omega,t)_5 &= 0.
\end{aligned}
$$

The second vector in (6) should be written as follows [8]

$$\mathbf{Q}_\Pi(\omega, t) = \mathbf{M}\cdot\mathbf{A}_\Pi(\omega, t),$$

where $\mathbf{M}$ is the mass matrix, $\mathbf{A}_\Pi(\omega, t)$ is the vector of vibration acceleration of supports,

$$
\begin{aligned}
&\mathbf{A}_\Pi(\omega t) = \\
&= \begin{bmatrix} A_{\Pi x1} & A_{\Pi y1} & A_{\Pi x2} & A_{\Pi y2} & A_{\Pi z}\end{bmatrix}^{\mathrm{T}}\sin\omega t,
\end{aligned}
$$

where $A_{\Pi x1}, ..., A_{\Pi z}$ are the vibration acceleration amplitude.

Numerical analysis of forced vibrations

To study the periodic solutions of the equation (5), we'll build a frequency response of peak-to-peak displacements caused by frequency ω. Frequency Ω is considered to be fixed. Let's define the dimensionless parameters as follows: $x_A = x_1/z_0$, $y_A = y_1/z_0$, $x_B = x_2/z_0$, $y_B = y_2/z_0$, $z_A = z/z_0$, $\overline{\omega} = \omega/\omega_1$, $\overline{\Omega} = \Omega/\omega_1$, $\tau = t\cdot\omega_1$, where z_0 is an axial displacements of the inner ring of the bearing with respect to the outer ring due to the action of the preload force, ω_1 is the fundamental resonant frequency of the linearized system. In this work, analysis of the solutions of equation (5) is formed, using the continuation method that was proposed in work [9] and improved in work [10] in the study of nonlinear rotor vibrations caused by unbalance.

Oscillations of the non-deformable rotor with one disk $l = 0{,}5$ m, $\zeta_D = 0{,}125$ m, the shaft diameter $d = 0{,}025$ m, $m_0 = 10$ kg, $I_1 = 0{,}1$ kg·m^2, $I_0 = 0{,}2$ kg·m^2 that rotates on angular contact bearings of average series as per GOST Standard 831-75 are considered. The bearing parameters are as follows: the radius of the outer race $R_2 = 27{,}5167$ mm; $\alpha = 15°$; the radius of the inner race $R_1 = 16{,}000$ mm; the radius of the cross section of races $R_K = 5{,}930$ mm; the ball diameter $d_B = 11{,}510$ mm; the number of balls $N_B = 7$; the modulus of elasticity $E = 2{,}1 \cdot 10^{11}$ Pa; Poisson ratio $\mu = 0{,}3$.

At joint action of the unbalance and vibration of supports the basic resonant oscillations occur in form when the shaft spindles are located at one side of the bearing axis and move in a circle in the shaft rotation direction. The frequency of these oscillations corresponds to the third frequency of free oscillations and is further defined by $\overline{\omega}_3$. Besides this resonance there appear resonances of other forms of rotor oscillations as well as super-resonance oscillations. Fig. 2 shows the frequency response of the coordinate y_B due to parameter $\overline{\omega}$.

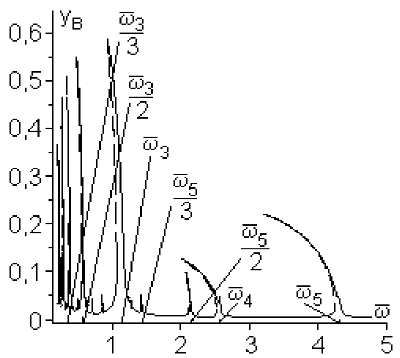


Fig. 2. Frequency response y_B due to $\overline{\omega}$

The resonance peaks $\overline{\omega}_5$, $\overline{\omega}_5/2$ and $\overline{\omega}_5/3$ correspond to the modes when the shaft spindles are located on opposite sides of the axis of symmetry of bearings and during oscillations move towards the rotor rotation. Super-resonance frequencies $\overline{\omega}_5/2$ and $\overline{\omega}_5/3$ refer to resonance frequency $\overline{\omega}_5$ as integers – 1/2 and 1/3.

Resonance $\overline{\omega}_4$ corresponds to the form when the shaft spindles are located on opposite sides of the axis of symmetry of bearings and during oscillations move oppositely the shaft rotation. The frequencies $\overline{\omega}_4$ and $\overline{\omega}_5$ represent the fourth and fifth frequency of free oscillations. In Fig. 2 resonances with lower frequencies corresponding to these modes are noticeable. Their peak-to-peak displacements are small and the frequencies are also treated as integers.

The resonant peak $\overline{\omega}_3$ has the highest magnitude. In the region of low frequencies in Fig. 2 there can be seen super-resonances $\overline{\omega}_3/2$ and $\overline{\omega}_3/3$ as well as not marked in the figure $\overline{\omega}_3/4$ and $\overline{\omega}_3/5$, which correspond to the modes when the shaft spindles are located at one side of the axis of symmetry of bearings and during oscillations move towards the shaft rotation. Their frequencies refer to the frequency of the fundamental resonance as 1/2, 1/3, 1/4, 1/5.

Analysis of resonant oscillation modes

The orbit s of the centers of shaft spindles on the main resonances of all modes are close to a circles as shown in Fig. 3 for the mode corresponding to $\overline{\omega}_3$ ($\overline{\omega} = 0{,}9858$).

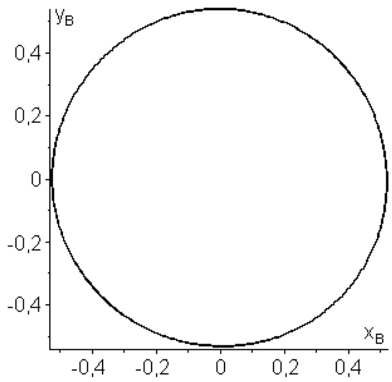


Fig. 3. The orbit of the spindle B, $\overline{\omega} = 0{,}9858$

For super-resonance frequency during each cycle of oscillation the shaft spindle describes as many loops close to the circumference as many times the frequency is lower than the fundamental frequency for this mode, as it is shown in Fig. 4 for $\overline{\omega}_3/2$ and in Fig. 5 for $\overline{\omega}_3/3$.

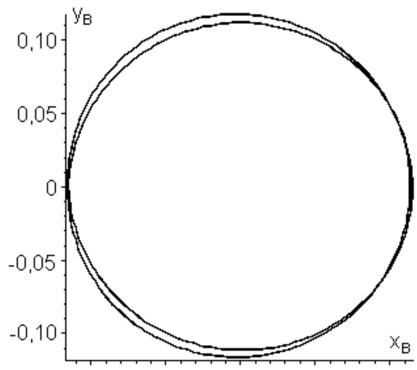


Fig. 4. The orbit of the spindle B, $\overline{\omega} = 0{,}5619$

Between the peaks with big peak-to-peak displacements in Fig. 2 there can be clearly seen the peaks with relatively small displacements and frequencies relating to the resonance frequency $\overline{\omega}_4$ and $\overline{\omega}_5$ as integers. As a result of the superposition of oscillations according to several modes, here the orbits of the centers of spindles are more complex, as it is shown in Fig. 6 for the frequency $\overline{\omega} = 0{,}5098$.

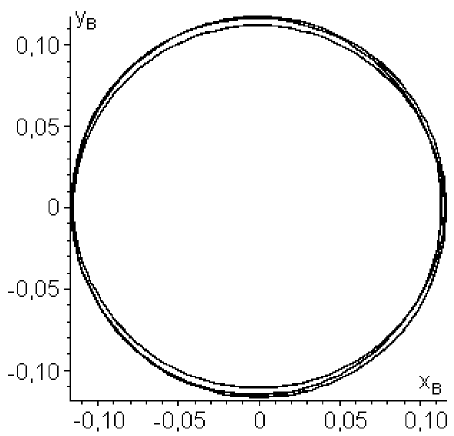


Fig. 5. The orbit of the spindle B, $\overline{\omega} = 0{,}3746$

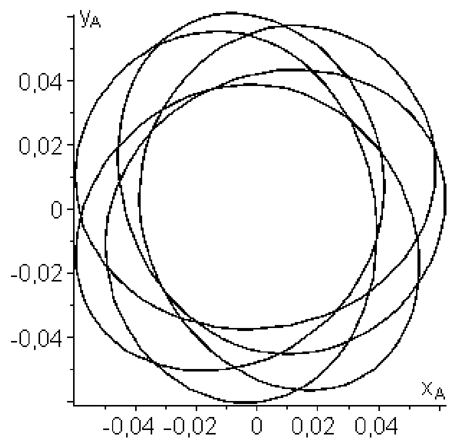


Fig. 6. The orbit of the spindle A, $\overline{\omega} = 0{,}5098$

In this mode, there occurs the superposition of oscillations according to the modes of resonances $\overline{\omega}_3$ and $\overline{\omega}_5$.

Conclusions

Analysis of the nonlinear preloaded angular-contact ball bearings rotor dynamics has shown that at joint action of unbalance and vibration of supports there are excited several forms of rotor oscillations. All frequency responses are soft. In this case, besides the main resonance oscillations there occur super-resonance oscillations at frequencies lower than the resonant ones in an integer number of times.

Resonances corresponding to the modes, when the shaft spindles are located on one side of the symmetry axis of bearings have the largest amplitude, and the resonances corresponding to the modes, when the shaft spindles are located on opposite sides of the axis of symmetry of bearings and during oscillating move oppositely to the shaft rotation – the lowest amplitude.

This system behavior is explained by the complexity of disturbances due to the fact that the rotor rotation frequency is within the range of vibration frequencies of supports. The superposition of these disturbing vibrations leads to the fact that in the disturbing load there can be observed beats that cause super-resonant oscillations.

References

1. Новиков Л. З. Определение собственных частот колебаний электродвигателя, связанных с нелинейной упругостью подшипников / Л. З. Новиков // Изв. АН СССР. Механика и машиностроение. – 1961. – № 6. – С. 84–91.
2. Bai C. Subharmonic resonance of a symmetric ball bearing – rotor system / C. Bai, H. Zhang, Q. Xu // International Journal of Non-Linear Mechanics. – 2013. – Vol. 50. P. 1–10.
3. Young T. H. Dynamic stability of rotor-bearing systems subjected to random axial forces / T. H. Young, T. N. Shiau, Z. H. Kuo // Journal of Sound and Vibration. – 2007. – Vol. 305(3). – P. 467–480.
4. Филипковский С.В. Свободные нелинейные колебания многодисковых роторов на шарикоподшипниках / С.В. Филипковский, К.В. Аврамов // Проблемы прочности. – 2013. – № 3. – С. 86–96.
5. Филипковский С. В. Модель радиально-упорного шарикоподшипника для анализа нелинейных вибраций ротора / С. В. Филипковский // Автомобильный транспорт: сб. науч. тр. – 2015. – Вып. 37. – С. 135–142.
6. Бальмонт В.Б. О колебаниях момента сопротивления вращению шарикоподшипника / В.Б. Бальмонт, Б О. Дубовецкий, А.М. Авдеев, Г.В. Селезнёв // Машиноведение. – 1988. – № 3. – С. 73–81.

7. Позняк Э.Л. Маятниковые колебания несимметричного жесткого ротора в подшипниках с зазорами / Э.Л. Позняк, Т.Н. Гладышева, В.Б. Ковалев // Проблемы машиностроения и надежности машин. – 1990. – № 4.– С. 33–40.
8. Тимошенко С. П. Колебания в инженерном деле / С. П. Тимошенко, Д. Х. Янг, У. Уивер. – М.: Машиностроение, 1985. – 472 с.
9. Беломытцев А. С. Алгоритм решения нелинейной краевой задачи для обыкновенных дифференциальных уравнений в области многозначности / А. С. Беломытцев, В. Н. Карабан // Журнал вычислительной математики и математической физики. – 1986. – № 7. – С. 1099–1102.
10. Филипковский С. В. Нелинейные колебания ротора на радиально-упорных шарикоподшипниках / С. В. Филипковский, А. С. Беломытцев // Вестник ХНАДУ: сб. науч. тр. – 2014.– Вып. 64. – С. 66–73.

References

1. Novikov L. Z. *Opredelenie sobstvennyh chastot kolebanij elektrodvigatelja, svjazannyh s nelinejnoj uprugost'ju podshipnikov* [Determination of the natural frequencies of the motor oscillations associated with the nonlinear elasticity of bearings]. Izv. AN SSSR. Mehanika i mashinostroenie. 1961. no. 6. pp. 84–91.
2. Bai C., Zhang H., Xu Q. Subharmonic resonance of a symmetric ball bearing – rotor system. International Journal of Non-Linear Mechanics, 2013. Vol. 50. pp. 1–10.
3. Young T. H., Shiau T. N., Kuo Z. H. Dynamic stability of rotor-bearing systems subjected to random axial forces. Journal of Sound and Vibration, 2007, Vol. 305(3), pp. 467–480.
4. Filipkovskij S.V., Avramov K.V. *Svobodnye nelinejnye kolebanija mnogodiskovyh rotorov na sharikopodshipnikah* [Nonlinear Free Vibrations of Multi-Disk Rotors on Ball Bearings]. Problemy prochnosti, 2013, no. 3. pp. 86–96.
5. Filipkovskiy S.V. *Model radialno-upornogo sharikopodshipnika dlya analiza nelineynyih vibratsiy rotora* [The model of angular ball-bearing for analysis of nonlinear vibrations of rotor]. Avtomobilnyiy transport, 2015, Vol. 37. pp. 135–142.
6. Bal'mont V.B., Duboveckij B.O., Avdeev A.M., Seleznjov G.V. *O kolebanijah momenta soprotivlenija vrashheniju sharikopodshipnika* [About oscillations of the moment of resistance to rotation of the ball bearing]. Mashinovedenie, 1988, no. 3. pp. 73–81.
7. Poznjak E.L. Gladysheva T.N., Kovalev V.B. *Majatnikovye kolebanija nesimmetrichnogo zhestkogo rotora v podshipnikah s zazorami* [Pendulum oscillations of asymmetric rigid rotor in bearings with clearances]. Problemy mashinostroenija i nadezhnosti mashin, 1990, no. 4. pp. 33–40.
8. Timoshenko S.P., Jang D. H., Uiver U. *Kolebanija v inzhenernom dele* [Vibration Problems in Engineering]. Moskow, Mashinostroenie Publ., 1985. 472 p.
9. Belomytcev A.S., Karaban V.N. *Algoritm reshenija nelinejnoj kraevoj zadachi dlja obyknovennyh differencial'nyh uravnenij v oblasti mnogoznachnosti* [An algorithm for solving nonlinear boundary value problems for ordinary differential equations in the ambiguity area]. Zhurnal vychislitel'noj matematiki i matematicheskoj fiziki. 1986. no. 7. pp. 1099–1102.
10. Filipkovskij S.V., Belomytcev A.S. *Nelinejnye kolebanija rotora na radial'no-upornyh sharikopodshipnikah* [Nonlinear vibration of a rotor on the angular contact ball bearings]. Vestnik Harkovskogo nacionalnogo avtomobil'no-dorozhnogo un-ta. 2014. Vol. 64. pp. 66–73.

Рецензент: М. А. Подригало, профессор, д.т.н., ХНАДУ.

УДК 378.2(092)

MYKOLA HOVORUSHCHENKO (1924–2011): BIOGRAPHY OF A SCIENTIST

A. Domanovskyi, Assist. Prof., Ukrainian Studies Department,
Kharkov National Automobile and Highway University

Abstract. *The article is devoted to the biography of Mykola Yakovych Hovoruschenko (1924–2011). M. Hovorushchenko is known for his role in the development of the first modern stations of automated diagnostics. Under the direction of M. Hovorushchenko there were trained approximately 8500 engineers, more than 80 patents for inventions were obtained.*

Key words: *Mykola Yakovych Hovoruschenko (1924–2011), biography, scientist, contribution to science.*

МИКОЛА ГОВОРУЩЕНКО (1924–2011): БІОГРАФІЯ НАУКОВЦЯ

А.М. Домановський, доц.,
Харківський національний автомобільно-дорожній університет

Анотація. *Статтю присвячено огляду біографії та наукового доробку Миколи Яковича Говорущенка (1924–2011) – професора, доктора технічних наук, заслуженого діяча науки.*

Ключові слова: *Микола Якович Говорущенко (1924–2011), біографія, науковець, науковий внесок.*

НИКОЛАЙ ГОВОРУЩЕНКО (1924–2011): БИОГРАФИЯ УЧЕНОГО

А.Н. Домановский, доц.,
Харьковский национальный автомобильно-дорожный университет

Аннотация. *Статья посвящена обзору биографии и научного вклада Николая Яковлевича Говорущенко (1924–2011) – профессора, доктора технических наук, заслуженного деятеля науки.*

Ключевые слова: *Николай Яковлевич Говорущенко (1924–2011), биография, ученый, научный вклад.*

The Honored Worker of Science, Doctor of Technical Sciences, Professor Mykola Yakovych Hovoruschenko [1–8] was born on May 24, 1924 in the village of Klynove, Borisov District, Belgorod Region. He graduated from high school in the city of Belgorod in 1940 and entered Kharkiv Automobile and Highway Institute (KhADI). His studies did not last long before they were interrupted the war. Hovorushchenko began his military service in the 381st Reserve Regiment in Pugachev, in the rank of sergeant. After working for several months as a teacher in the preparatory section of the regiment, the young man volunteered for the front, where he served his way from an ordinary infantryman to a regimental staff officer. From February 1943 to November 1945 he fought on the 3rd and 4th Ukrainian fronts, took part in the liberation of Kharkiv, served as a topographer, military interpreter, head of secret record keeping, and Komsomol organizer at the regimental headquarters.

In September 1945, Hovorushchenko was demobilized as a student who had finished his 1st year of university studies. He returned home in November of that year [1]. From 1946 to 1950 Hovorushchenko was a student at KhADI. As an honors student, he was offered an opportunity to pursue a postgraduate degree. In

1954, at the Moscow Automobile and Road Institute (MADI), he successfully defended his candidate thesis on evaluating the fuel efficiency of automobiles on uneven roads. In the same year Hovorushchenko became chair of the Department of Motor Vehicle Operation and Maintenance at KhADI.

At that time he already began exploring issues related to the diagnostics of motor vehicles for improving their maintenance and repair. He led the development of modern automated diagnostic stations for cars and trucks, as well as the construction and deployment of several dozen sets of diagnostic equipment in different cities of Ukraine and the former USSR. The 1950s were a particularly fruitful period in the scientific and educational work of the department. In cooperation with the Department of Road Construction and Maintenance, research was begun on the interaction between the motor vehicle and the road. An automated lab for handling complex research problems in various road and traffic conditions, unique for that time, was created [5, p. 4; 9, p. 9].

From 1957 to 1959, Hovorushchenko served as Dean of the Faculty of Distance Education, and from 1962 to 1964 – Dean of the Faculty of Motor Vehicles, KhADI's leading faculty. In 1965 he defended his doctoral thesis on «The Theoretical Basis for Operational Calculations of Vehicle Movement on Roads with Varying Degree of Evenness» [9, p. 6]. Gradually, a strong scientific school emerged under the leadership of Mykola Yakovych, numerous theses were defended under his direction.

In 1965, on the orders of the Minister of Automotive Transportation of the Ukrainian SSR, a sector research lab for the basic problems of motor vehicle operation and maintenance was set up at KhADI, where critical issues of diagnostic theory and the theoretical foundations for methods and regimes of preventive maintenance and repair of motor vehicles were explored. Hovorushchenko became the head of the lab. In 1973, by then a renowned and accomplished scientist, he was sent to Mongolia as a UNESCO expert to organize an institute of technology there.

Hovorushchenko is known for his important role in the development of the first modern stations of automated diagnostics, begun in 1965. In 1970, the work prepared by the Department of Motor Vehicle Operation and Maintenance, entitled «The Development and Application of Methods and Means for Diagnosing the Maintenance Condition of the Motor Vehicle Stock» was a contender for the Ukrainian SSR State Award for Science and Technology [4].

Under the direction of Mykola Hovorushchenko, the department held the first Union-wide science and technology conference on the diagnosis and prognosis of the condition of motor vehicle stock in September 1967, which formulated the theoretical foundations of diagnosis and the basic principles of a new approach to the preventive maintenance and repair of motor vehicles on the basis of reliable diagnostic information [4, p. 21].

In 1970–1971, at the Department of Motor Vehicle Operation and Maintenance, in the sector lab of Ukraine’s Ministry of Automotive Transportation, the first experimental model of a mobile station for diagnosing the maintenance condition of cars (PDS-I) was designed and produced under Hovorushchenko’s direction. The KhADI model PDS-I was designed for determining the technical condition of privately owned vehicles by traffic control authorities and for annual mandatory maintenance inspections. The station consisted of a special diagnostic trailer and a truck. The equipment in the trailer allowed to diagnose all the basic systems and aggregates of a vehicle based on 60 parameters. Two operators were able to process up to 50 cars per shift. The data of the express-diagnosis were recorded on a tape that was later deciphered by the operators. The driver received a completed diagnosis card with a statement on the maintenance condition of the vehicle based on 72 parameters and recommendations for fixing the defects [10]. Another advantage of the station was its complete autonomy and independence from external power supply. It could be set up in 30 minutes.

From May 23 to June 6, 1973 PDS-IV was demonstrated at the international exhibition Autoservice-73 in Moscow, attended by representatives from 25 countries. The KhADI model was recognized as the first in the Soviet Union and widely considered one of the most interesting items at the exhibition. The diagnosis station received praise from foreign representatives (USA, Japan, the Federal Republic of Germany, et al.). Versatility, portability, high degree of automation, small size and cost were named as the main advantages of the model [11, p. 61; 12–28].

On May 24, 1974 the State Committee on Science and Technology of the USSR and the Council of Ministers of Ukraine resolved to create in KhADI the only research lab for the problems of the diagnosis of motor vehicle maintenance condition in the Soviet Union. Since 1974, more than 30 models of diagnostic equipment were developed at the lab, more than 60 patents for research in the field of diagnostics were received. Much of this work was done under the direction of Hovorushchenko.

Hovorushchenko authored more than 300 works, including over 40 monographs, textbooks, and manuals. He received more than 50 patents. One of his first works was the collective monograph Operational Characteristics of Highways (Moscow, Autotransizdat 1961) [29], summarizing research on the performance of motor vehicles in various road and traffic conditions. Continued research in this direction was presented in Professor Hovorushchenko's textbook Basic Theory of Automobile Operation and Maintenance (1971) [30; 9, p. 9; 11, p. 53].

Mykola Yakovych published numerous works on the diagnosis of motor vehicles. His first monograph on this subject came out right before the opening of the first Union-wide conference on «The Foundations of Motor Vehicle Maintenance Diagnosis» (M. Y. Hovorushchenko, A. V. Gogayzel, B. I. Klimetz, 1967). The results of further research on this subject in our country and abroad are summarized in Professor Hovorushchenko's monographs Diagnosis of the Technical Condition of Automobiles (Moscow, 1970) [31] and Automobile Diagnosis: Today and Tomorrow (1976) [32]. In 1984, he published a textbook on the Operation and Maintenance of Motor Vehicles [33; 5, p. 6–7].

Beginning in 1982, Mykola Yakovych was involved in the development of the Comprehensive Program of Scientific and Technological Progress in the Area of Transportation in Ukraine to 2005. For 13 years (1977–1990) he was a member of the Expert Council on Transportation at the Higher Attestation Commission of the USSR [2]. In the mid-1980's the Department of Motor Vehicle Operation and Maintenance under his direction continued to develop highly useful scientific projects, the results of which were actively implemented by the industry.

Financial hardship that prevailed in the 1990 s caused serious difficulties in conducting applied research in motor vehicle diagnosis at the department. The department's faculty under Hovorusjcjenko's direction had to focus their work on theoretical issues related to the design and operation of transportation systems and machines.

In 1993, Professor Hovorushchenko became Academician of the Transportation Academy of Ukraine and Academician of the Academy of Transportation of Russian Federation. In the late 1990s the department performed the first series of studies on the theory of motor vehicle operation and transportation systems engineering and published (in collaboration with the Department of Motor Vehicles) the monograph Transportation Systems Engineering, as well as works on the Economic Cybernetics of Transportation and Technological Cybernetics of Transportation [34; 35; 36].

In October 1997, the department under the direction of Professor Hovorushchenko hosted a conference on the protection of air quality from harmful vehicle emissions. In 1998 the department organized a nation-wide conference on systems engineering in road transportation, which coincided with the 65th anniversary of the department. The department also received accreditation certificates for their fuel and operation materials research and analysis lab, as well as for the mobile diagnosis station [4, p. 22].

In the year 2000, the Department of Motor Vehicle Operation and Maintenance was renamed Department of Systems Engineering and Diagnosis of Transportation Machinery. In recent years, the faculty have been actively working on the theoretical principles of systems engineering and on the technological and economic cybernetics of transportation.

The Department of Systems Engineering and Diagnosis of Transportation Machinery under the direction of Mykola Hovorushchenko become one of the leading departments of the present-day KhNADU. It includes 13 teaching and research laboratories, including the Problem Lab for the Diagnosis and Prognosis of the Technical Condition of Motor Vehicles and two laboratories accredited by the State Standard Commission of Ukraine. The Problem Lab for the Diagnosis and Prognosis of the Technical Condition of Motor Vehicles developed and helped to implement approximately 30 models of diagnostic equipment, constructed diagnostic stations, and formulated the foundations of a new approach to maintaining motor vehicles in good technical condition, based on a monitoring system of diagnostics.

The department, and Professor Hovorushchenko in particular, have trained approximately 8500 engineers, 61 candidates of science and 9 PhDs. More than 80 patents for inventions have been received, more than 40 monographs, textbooks, and manuals have been published. The department's faculty developed a comprehensive program of scientific and technological progress in the sphere of transportation and its socio-economic results for the period from 1985 to 2005.

To his last days, Mykola Hovorushchenko continued working in the field he loved, the field that shaped his life [37–41]. He made an important contribution to the science of motor vehicles in Ukraine and Eastern Europe [8].

References

1. Власенко В. Я прожив важке, але цікаве життя... / В. Власенко // Автодорожник. – 5 травня 2005 р. – С. 5.
2. Коваль Е.А. Говорущенко Николай Яковлевич / Е. А. Коваль // Видатні педагоги вищої школи м. Харкова / В.І. Астахова, К.В. Астахова, А. О. Гайков та ін. – Харків: Книжкова фабрика «Глобус», 1998. – С. 195–196.
3. Говорущенко Н. Я. // Украинская советская энциклопедия. – К.: Полиграфкнига, 1980. – Т. 12. – С. 604.
4. Кафедре «Системотехника и диагностика транспортных машин» – 70 лет / сост. В. Н. Варфоломеев; под общ. ред. Н. Я. Говорущенко. – Харьков: ХНАДУ СДТМ, 2004. – 27 с.
5. Варфоломеев В. Н. Говорущенко Николай Яковлевич (к 80-летию со дня рождения и 55-летию научно-педагогической и общественной деятельности) / В. Н. Варфоломеев. – Харьков: ХНАДУ, 2004. – 16 с.
6. Ковальов В.І. Кафедрі експлуатації автомобільного транспорту – 65 років / В.І. Ковальов // Автодорожник. – 22 січня 1999 р. (п'ятниця). – № 1(1432). – С. 2, 3.
7. Домановський А.М. Микола Якович Говорущенко – життя в науці / А. М. Домановський. – Х.: ХНАДУ, 2010. – 44 с.
8. Домановський А. Пам'яті заслуженого керівника кафедри ТЕСА / А. Домановський, О. Васильківська, О. Журавльов // Автодорожник. – 2011. – № 5 (1569). – С. 8–9.
9. Ковальов В. ХАДІ – одна сім'я: збірник історичних нарисів / В. Ковальов. – Харків: ХДАДТУ, 2000. – 194 с.
10. Мостославский В. Диагностика на колесах / В. Мостославський // За рулем. – 1973. – № 6. – С. 7.
11. Ковальов В. І. ХНАДУ: історичний нарис / В.І. Ковальов. – Харків: ХНАДУ, 2007. – 104 с.
12. Техника и наука. Научная организация труда. – 1973. – № 7. – С. 8.
13. Правда. – 13 апреля 1973 г.
14. Известия. – 24 мая 1973 г.
15. Радянська Україна. – 23 травня 1973 р.
16. Сельская жизнь. – 29 мая 1973 г.
17. Соціалістична Харківщина. – 3 червня 1973 р.
18. Красное Знамя. – 6 июня 1973 г.
19. Строительная газета. – 23 мая 1973 г.
20. Neues Leben. – 13 Juni 1973.
21. Огонек. – № 24. – Июнь 1973.
22. Автодорожник Украины. – № 3. – 1973.
23. Ленінська зміна. – 9 червня 1973 р.
24. Автодорожник. – 16 червня 1973.
25. Автодорожник. – 23 червня 1973.
26. Изобретатель и рационализатор. – № 8. – 1973.
27. Автомобильный транспорт. – 8 августа 1973 г.
28. Информационный сборник. Торгово-промышленная палата СССР. – М., 1975.
29. Бируля А. К. Эксплуатационные качества автомобильных дорог / А. К. Бируля, Н. Я. Говорущенко, Д. В. Ермакович. – М.: Автотрансиздат, 1961. – 134 с.
30. Говорущенко Н.Я. Основы теории эксплуатации автомобилей / Н.Я. Говорущенко. – К.: Вища школа, 1971. – 231 с.
31. Говорущенко Н.Я. Диагностика технического состояния автомобилей / Н.Я. Говорущенко. – М.: Транспорт, 1970. – 135 с.
32. Говорущенко Н. Я. Диагностика автомобиля сегодня и завтра / Н. Я. Говору-

щенко. – Харьков: Вища школа, 1976. – 48 с.
33. Говорущенко Н. Я. Техническая эксплуатация автомобилей / Н.Я. Говорущенко. – Харьков: Вища школа, 1984. – 312 с.
34. Говорущенко Н. Я. Системотехника транспорта / Н. Я. Говорущенко, А. Н. Туренко. – Харьков: РИО ХГАДТУ, 1999. – 468 с.
35. Говорущенко Н. Я. Экономическая кибернетика транспорта / Н. Я. Говорущенко, В.Н. Варфоломеев. – Х.: ХГАДТУ, 2000. – 218 с.
36. Говорущенко Н. Я. Техническая кибернетика транспорта / Н. Я. Говорущенко, В.Н. Варфоломеев. – Х.: ХГАДТУ, 2001. – 268 с.
37. Говорущенко Н. Я. Системотехника проектирования транспортных машин / Н. Я. Говорущенко, А. Н. Туренко. – Харьков: ХНАДУ, 2002. – 166 с.
38. Говорущенко Н. Я. Автомобиль… Каким он будет завтра? / Н.Я. Говорущенко. – Х.: ХНАДУ, 2003. – 54 с.
39. Говорущенко Н. Я. Обеспечение безопасности движения на автомобильном транспорте / Н.Я. Говорущенко, В.П. Волков, И. К. Шаша. – Х.: ХНАДУ, 2006. – 360 с.
40. Говорущенко М. Я. Проектне забезпечення формування виробничо-технічної бази підприємств автомобільного транспорту: навч. посіб. / М. Я. Говорущенко, В.М. Варфоломєєв, В.П. Волков, Н.А. Волошина. – Х.: ХНАДУ, 2007. – 116 с.
41. Говорущенко Н.Я. Роликовые стенды для проверки тормозных и тяговых свойств автомобилей / Н.Я. Говорущенко. – Х.: ХНАДУ, 2009. – 344 с.

Referens

1. Vlasenko V. *Ya prozhyv vazhke, ale tsikave zhyttya...* [I have lived a difficult but interesting life...]. *Avtodorozhnyk*, 2005, May 5, p. 5.
2. Koval' E. A. *Govorushchenko Nikolay Yakovlevych Vydatni pedahohy vyshchoyi shkoly m. Kharkova* [Outstanding teachers of high school. Kharkov]. Kharkiv, 1998, pp. 195–196. (In Russ.)
3. *Govorushhenko N. Ja. Ukrainskaja sovetskaja jenciklopedija* [Ukrainian Soviet Encyclopedia]. Kiev, Polihrafkniha Publ., 1980, Vol. 12, p. 604.
4. *Kafedre «Sistemotehnika i diagnostika transportnyh mashin» – 70 let. Ed. V.Ya. Hovorushenko* [Department of «System engineering and diagnostics of transport vehicles» – 70 years. Ed. V.Ya. Hovorushenko]. Kharkov, 2004. 27 p.
5. Varfolomeev V. N. *Govorushhenko Nikolaj Jakovlevich (k 80-letiju so dnja rozhdenija i 55-letiju nauchno pedagogicheskoj i obshhestvennoj dejatel'nosti)* [Govorushchenko Nikolai Jakovlevich (on the 80th anniversary and the 55th anniversary of scientific-pedagogical and social activities)]. Kharkov, 2004. 16 p.
6. Kovalov V. I. *Kafedri ekspluatatsiyi avtomobilnoho transportu – 65 rokiv* [The Department of Road Transport – 65 years]. *Avtodorozhnyk,* January 22, 1999, p. 2–3.
7. Domanovsky A. M. *Mykola Yakovych Hovorushchenko – zhyttya v nautsi* [Mykola Yakovych Hovoruschenko – life in science]. Kharkiv, 2010. 44 p.
8. Domanovsky A., Vasyl'kivs'ka O., Zhurav-l'ov O. *Pam'yati zasluzhenoho kerivnyka kafedry* [Memory of the honored head of the department]. *Avtodorozhnyk,* 2011, no. 5 (1569), pp. 8–9.
9. Koval'ov V. *KhADI – odna sim'ya: zbirnyk istorychnykh narysiv* [HADI – one family: a collection of historical essays]. Kharkiv, 2000. 194 p.
10. Mostoslavskij V. *Diagnostika na kolesah* [Diagnosis on Wheels]. *Za rulem*. 1973, no. 6, p. 7.
11. Koval'ov V. *KhNADU: istorychnyy narys* [KhNADU: historical review]. Kharkiv, 2007. 104 p.
12. *Tehnika i nauka. Nauchnaja organizacija truda* [Technology and science. The scientific organization of labor]. 1973, no. 7, p. 8.
13. *Pravda* [True]. 1973. April 13. (In Russ.)
14. *Izvestija* [News]. 1973. May 24. (In Russ.)
15. *Radyans'ka Ukrayina* [Soviet Ukraine]. 1973. May 23. (In Ukr.)
16. *Sel'skaja zhizn'* [Rural life]. 1973. May 29,
17. *Sotsialistychna Kharkivshchyna* [Socialist Kharkiv]. 1973. June 3. (In Ukr.)
18. *Krasnoe Znamja* [Red flag]. 1973. June 6. (In Russ.)
19. *Stroitel'naja gazeta* [Building newspaper]. 1973. May 23. (In Russ.)
20. Neues Leben. 1973. June 13.
21. *Ogonek* [Flame]. June 1973, no. 24. (In Russ.)

22. *Avtodorozhnik Ukrainy* [Road service worker of Ukraine]. 1973, no. 3. (In Russ.)
23. *Lenins'ka zmina* [Lenin change]. 1973. June 9. (In Ukr.)
24. *Avtodorozhnyk* [Road service worker]. 1973. June 16. (In Ukr.)
25. *Avtodorozhnyk* [Road service worker]. 1973. June 23. (In Ukr.)
26. Izobretatel' i racionalizator [Inventor and innovator]. 1973, no. 8. (In Russ.)
27. *Avtomobil'nyj transport* [Automobile transport]. 1973. August 8. (In Russ.)
28. *Informacionnyj sbornik. Torgovo-promyshlennaja palata SSSR* [Information Collection. Chamber of Commerce of the USSR]. Moscow, 1975.
29. Birulja A. K., Govorushhenko N. Ja., Ermakovich D. V. *Jekspluatacionnye kachestva avtomobil'nyh dorog* [The Performance of Highways]. Moscow, Avtotransizdat Publ., 1961. 134 p.
30. Govorushhenko N. Ja. *Osnovy teorii jekspluatacii avtomobilej* [Fundamentals of the Theory of Operation of Cars]. Kiev, Vyscha shkola Publ., 1971. 231 p.
31. Govorushhenko N. Ja. *Diagnostika tehnicheskogo sostojanija avtomobilej* [Diagnostics of Technical Condition of Vehicles]. Moscow, Transport Publ., 1970. 135 p.
32. Govorushhenko N. Ja. *Diagnostika avtomobilja segodnja i zavtra* [Diagnostics of the Car Today and Tomorrow]. Kharkov, Vyscha shkola Publ., 1976. 48 p.
33. Govorushhenko N. Ja. *Tehnicheskaja jekspluatacija avtomobilej* [Technical Operation of Vehicles]. Kharkov, Vyscha shkola Publ., 1984. 312 p.
34. Govorushhenko N. Ja., Turenko A. N. *Sistemotehnika transporta* [Transport Systems Engineering]. Kharkov, Vyscha shkola Publ., 1999. 468 p.
35. Govorushhenko N. Ja., Varfolomeev V. N. *Jekonomicheskaja kibernetika transporta* [Economic Cybernetics of Transport]. Kharkov, KhGADTU Publ., 2000. 218 p.
36. Govorushhenko N. Ja., Varfolomeev V. N. *Tehnicheskaja kibernetika transporta* [Technical Cybernetics of Transport]. Kharkov KhGADTU Publ., 2001. 268 p.
37. Govorushhenko N. Ja., Turenko A. N. *Sistemotehnika proektirovanija transportnyh mashin* [System Engineering Design of Transport Machines]. Kharkov, KhNADU Publ., 2002. 166 p.
38. Govorushhenko N. Ja. *Avtomobil'... Kakim on budet zavtra?* [Car ... What Will It be Tomorrow?]. Kharkov, KhNADU Publ., 2003. 54 p.
39. Govorushhenko N. Ja., Volkov V. P., Shasha I. K. *Obespechenie bezopasnosti dvizhenija na avtomobil'nom transporte* [Ensuring Safety in Road Transport]. Kharkov, KhNADU Publ., 2006. 361 p.
40. Hovorushchenko M.Ya., Varfolomyeyev V.M., Volkov V.P., Voloshyna N. A. *Proektne zabezpechennya formuvannya vyrobnycho-tekhnichnoyi bazy pidpryyemstv avtomobil'noho transportu* [Project Providing of Industrial-technical Base of Enterprises of Motor Transport]. Kharkiv, KhNADU Publ., 2007. 116 p.
41. Govorushhenko N. Ja. *Rolikovye stendy dlja proverki tormoznyh i tjagovyh svojstv avtomobilej* [Roller Stands for Testing of Brake and Traction Properties of Cars]. Kharkov, KhNADU Publ., 2009. 344 p.

Рецензент: В.П. Волков, профессор, д.т.н., ХНАДУ.