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Dynamic vibration absorbers – optimization and design

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Summary. The main aim of the paper is improved dynamic vibration absorbers (DVA) design, taking into account the complex rotating machines dynamics. It is often impossible to balance the rotating elements so as to reduce the vibration to an acceptable level. The paper considers the provision of DVA or number of such DVA. Such originally designed DVA reduces vibration selectively in the maximum mode of vibration without introducing vibration in other modes. The result is achieved at a far lower expense than would be required to replace the concrete and steel foundation with a more massive one. By installing DVA, one can minimize excitation virtually at the source. In order to be more effective, a vibration absorption system should react in all the frequency domains. The present absorber also has as an advantage that it can be constructed so that it has a wide-range vibration absorption property. This construction allows for an easy connection of the above rotor equipment.

In order to determine the optimal parameters of DVA, the complete modeling of the rotating machine dynamics is obvious. The two degrees of freedom model are totally inadequate to accurately calculate the vibration frequencies of the construction and therefore, for a sufficiently accurate determination of its dimensional characteristics so as to determine such frequencies. It is therefore necessary in practice to dimension the construction through more complex modeling. In particular, concentrated mass and rigidity calculation methods may be adopted based on an even more accurate theoretical determination.

The numerical schemes (NS) row is considered for the complex vibration-excited constructions. Methods of decomposition and the NS synthesis are considered on the basis of new methods of modal synthesis. Complex NS are obtained of discretely-continua type, which enables in the adaptive mode to calculate tension not only in the continuum elements, but in the places of the most tension concentration in joints. Traditional design methodology, based on discontinuous models of structures and machines is not effective for high frequency vibration. The present research develops a modern prediction and control methodology, based on complex continuum theory and application of special frequency characteristics of structures. Complex continuum theory allows to take into consideration system anisotropy, supporting structure strain effect on equipment motions and to determine some new effects that are not described by ordinary mechanics of the continuum theory. The absorbers in accordance with this paper may be applied not only to electric machines ore aeronautic structures, but also to any other type of vibration-excited structure, such as cars, chisel installation, optical, magneto-optical disks, washing machines, refrigerators, vacuum cleaners, etc.

Key words: rotating machines, discreet-continuum theory, modal analysis, dynamic vibration absorber, optimal design, electrical plants, gas- and oil pipelines, aerospace technique, chisel installation, cars, magneto-optical disks, home appliances.

INTRODUCTION

Currently, the majority of the machinery in Ukraine (and in the countries of former East block), namely electric power stations, gas-and-oil facilities as well as our heat and water supplies are close to breakdown. Any moment a disastrous destruction of gas-compressor plants, turbo generators and pumps due to vibration is possible, since the equipment has been operating for a very long time and not enough money is available to replace it. Even on new blocks of atomic power stations, in particular on the newly introduced block Khmelnitskiy nuclear station, there were difficulties at start-up with raised level of vibration.

The most effective way to solve this problem is to apply optimally designed dynamic absorber or a set of such absorbers. It is desirable to eliminate unwanted vibration in many applications. On of the most visible applications is transportation. In automobiles, aircraft, and watercraft vibration can cause irritation and even motion sickness to the occupants, and can also cause accelerated wear and mechanical fatigue to the vehicle hardware. Other applications include manufacturing equipment, where imbalances of rotating machinery can cause eccentric rotation, which can degrade surface finishing and machining tolerances. As the speeds of electronic information storage devices, such as compact discs, DVDs, and hard drives increase, the effect of imbalance on the rotating discs becomes increasingly important. Still it is possible to give another dozen of examples of machines and constructions where DVA application is expedient.

Tuned vibration absorbers, also known as dynamic vibration absorbers, were first invented in 1909 (Den Hartog, 1956). Since then they have become widely used in aircrafts, skyscrapers, large watercrafts, telephone poles, and many other areas. This type of absorber is often called a passive tuned vibration absorber, as it does not require any external power source to perform the absorption.

Tuned vibration absorbers (TVAs) have often been used in many devices like these, but the static nature of the devices has often limited their effectiveness. TVAs consist of a mass attached by an elastic element to the primary vibrating mass. The mass and spring are chosen such that they will resonate at a particular frequency (the tuned frequency). Ideally, the impedance of the tuned frequency of the primary mass goes to infinity with the addition of the TVA, meaning that the primary mass will not vibrate at that frequency regardless of the excitation applied

THE ANALYSIS OF RECENT RESEARCH AND PUBLICATIONS

DVA is found to be an efficient, reliable and low-cost suppression device for vibrations caused by harmonic or narrow-band excitations. Bases of nonlinear mechanical processes are considered in [1-5]. Theoretical bases of DVA are expounded in [6-10]. The specific of application of DVA is considered in [11-16]. In [17,18] it is considered DVA with shock mass. In [19-21] particle of DVA. In [22-25] the specific of application of DVA is considered.

Most leading textbooks on mechanical vibrations discuss the basic equations of DVA, whereas some extended ones, e.g. the numerical schemes (NS) row for the complex vibration-excited construction and methods of decomposition or the NS synthesis are considered in our paper on the basis of new methods of modal synthesis [26-31].

In Ukraine, despite the big need for such investigations, only few scientists are working in this field, including the authors of the present paper. The patents on inventions of different types of DVA have been studied for the last 5 years. It has allowed us to define the direction of the future technologies, their novelty and the possibility to acquire new patents in Ukraine. Let us consider the typical developments and their weak points

As an example, we shall consider DVA that represents a console beam with a weight on the free end. A set of metal plates is used as a weight. Such construction is proposed by Moscow laboratory of vibration at MEI. Earlier such design was offered in US Patent 4150888.

However, such type of DVA has very little effect on all but the closest frequencies to the tuned frequency. Thus, DVAs are generally only effective if the principle frequency of excitation is well known and constant. As the excitation frequency drifts from the tuned frequency of the absorber, the attenuation rapidly decreases. The presence of the DVA can also create new resonance peaks in the primary mass's frequency response. If excitation frequency drifts significantly, the primary mass will resonate causing amplification of the excitation beyond what would naturally occur if the absorber were not present. Damping may be added to the absorber, and its presence can decrease this resonance caused by the absorber, but its presence also defeats the operation of the absorber, by preventing it from completely eliminating the tuned frequency.

Though tens and hundreds scientific centres in the world are engaged in questions of dynamics of complex machines with rotary elements modelling, much less in questions of modelling of such machines with DVA and their optimum designing are engaged among them:

1. DYNAMIC ABSORBERS FOR SOLVING RESONANCE PROBLEMS, Randy Fox Senior Staff Instructor, Entek IRD International Corp. Houston, TX. J. R. Hodgkins, P. Avitabile, University of Massachusetts, Lowell. They investigate DVA attached to a horizontal centrifugal pump in a major petroleum refinery. (The irony of the situation is that the authors of this development apparently are not familiar with works Den-Hartog and Timoshenko [3,5], who has in details explained the theory of DVA. Taking into account the main points as parasitic resonant peaks, influence of damping in many respects would have helped the authors, furthermore, for the adequate modelling of dynamic processes and designing of DVA it is desirable to investigate a complicated system: namely DVA pump

together with taking into account the flexibility of rotor, pliability of bearings, cases, influence of eccentricities etc. Such a situation is characteristic for many designs with DVA, when for an exposition of complicated dynamic processes primitive two-mass calculated schemes are used, and frequently the graphs are drawn that are simply parodying real processes.

2. V.C. Ho, A.M. Veprik and V.I. Babitsky [25] have presented an original theoretical and experimental approach to designing Wideband DVA. It would be interesting in the given class of problems of protection against vibration of electronic baseplates to investigate their dynamics on the basis of specified theories of layered plate's fluctuation. It, first, would allow to receive the fastest way of designing DVA and, second, to study more full dynamics of such systems. We investigated such problems for a long time in connection with studying mechanical vibrations in chemical sources of current for former All-Union Research Accumulator Institute (St. Petersburg), and for panels of solar batteries for satellites (DO "Southern", Dnepropetrovsk). It is possible also to recollect Showdown's works on plates with DVA.

3. In space designs, it is important to reduce vibration loadings, which may exceed static ones in several times. Many processes and elements may be a source of vibration. Application of dampers here may be ineffective. DVA designs, which are applied in the European space paper, are known: SPACE SOLUTIONS FOR NOISE AND VIBRATION CONTROL M. LATHUILIERE, A. PELLERIN, C. TRENY, METRAVIB RDS, 200 chemin des Ormeaux 69760 Limonest – France. METRAVIB RDS provide active and passive devices to solve vibration, shock and acoustic problems on structures. These solutions are often structural damping by means of DVA.

4. The vibration problem also limits development of optical disk drives that operate disks at constant linear velocity. The DVA can be easily assembled and may effectively absorb vibration of optical disk drives so that the pickup head may easily read information on the optical disk. Samsung, Seagate and other leading companies imply such inventions.

5. Many leading automobile manufacturers have utilized DVA for many years to control vibration annoying and potentially damaging resonance problems. Since automobile must operate over a wide range of operating speeds, with numerous variable exciting force frequencies from an engine, transmission, drive shaft, axles and other rotating components, it would be nearly impossible to avoid resonance problems. For example, DVA is provided for suspension, which improves vehicle roll behaviour, and sound minimization. (Opel, Volkswagen).

However, the majority of these practical DVA applications are based on inadequate mathematical models for complex structures designs and ineffective designing of passive type DVA. The presence of such DVA can also create new resonance peaks in the primary mass's frequency response. If excitation frequency drifts significantly, the primary mass will resonate causing amplification of the excitation beyond what would naturally occur if the absorber were not present. Our passive DVA designs have no these disadvantage.

Other means of enhancing the performance of the absorber have fallen into two categories: active DVA and adaptive tuned DVA. Active DVA typically uses a force actuator to alter the motion of the absorber mass to enhance its attenuation capability, and sometimes the actuator replaces the spring element entirely. This typically requires large amounts of external power for the actuator and sophisticated controlling hardware.

Adaptive tuned vibration absorbers, sometimes called adaptive-passive tuned vibration absorbers, use the same principles as passive tuned absorbers, however, they employ a mechanism that is capable of altering the absorber's properties on the fly that allow it to vary the tuned frequency of the absorber. However, their mechanical schemes are based on ordinary passive DVA schemes and at insufficient regulation have the same lacks. The following examples of these DVA can be given [17-21].

For the optimum designing system of DVA – Rotating machine it is necessary to construct and investigate a mathematical model of complex structure. Among scientific and technical centres in: Ukraine: Lviv Polytechnic National University, Institute of problems of durability (Kiev), Kiev National Polytechnic University, National transport university (Kiev) can be named.

In the world, it is possible to name many centres where these problems are being developed. However, researchers of complex dynamic processes of interaction between DVA and complex rotating machines are considerably less known.

OBJECTIVES

The basic purpose of this paper is the development of mathematical models of complicated machines and buildings in view of their interaction with the system of DVA, optimized on its vibration-absorption properties.

THE MAIN RESULTS OF THE RESEARCH

The problem of vibration fields modelling of complicated designs deformation and strain is considered for the purports of dynamic absorption. The problem is solved on the basis of four modified methods of modal synthesis (Fig.1).

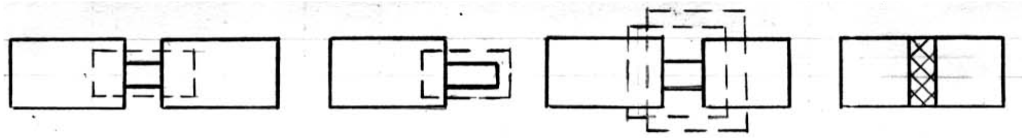

Fig. 1. Four modified methods of modal synthesis

The basis of these methods is in the solution of a set of equations in a normal form at the minimum application of matrix operations. The essence of the first method consists in reviewing knots of junctions as compact elastic elements for which inertial properties are taken into account without reviewing their strain, and massively connected parts - as deformable elements, their inertia being taken into account on the basis of modal expansion. The second one consists in a choice of a base extended element and alignment of the elements joined to it with the help of a special choice of their coordinate functions. The third - in the combined choice of a system of coordinate functions for a knot of junction, which after matrix transformations without application of global operations allows receiving solving set of equations of a dynamic equilibrium in a normal form. The fourth - in reviewing junctions as elastic extended elements, which are modelled by the specified equations in view of shift, compression and modifications of the form during a strain. The reduced common numerical schemes of dynamics of designs are considered further on certain engineering plants. The third scheme is illustrated with calculation of vibration processes in a turbo generator. The fourth point is – with calculation of rubber-metal shock-absorbers. The specified relations of deformation in stratified, inhomogeneous constructions of anisotropic designs made of aggregates are considered. According to the cinematic hypotheses the set of equations of dynamic equilibrium for slanting, inhomogeneous stratified elements is obtained. Conditions of contact between areas in the plan are obtained on the foundation also of cinematic hypotheses. Such approach is coordinated with a common method of constructing theories for thin-walled elements for solving cinematic hypotheses. It has allowed determining fields of strains in such junctions, not being limited to engineering hypotheses. A line of problems for rubber elements is considered also. The method of cinematic and force hypotheses solves a problem of rubber element deformation between rigid curvilinear holders about a priori unknown area of contact. Formulas for the definition of the rigidity of such knot are obtained. The important engineering problem of curving of a plate in an elastic holder has considered also. For a solution of these problems the complex of programs (Fig. 2) is developed.

In the given paper, we apply the technique that has been used for a long time. It consists in the construction of few-parametrical numerical schemes, based on the discretely-continua models of machines. In our opinion, modern packages of applied programs - ANSYS, NASTRAN, COSMOS etc. are good rather for scientific calculations, than for engineering. On the basis of these packages, it is difficult to allocate principal and most influential parameters of the machine that determine its long durability, dynamic characteristics, a mass, adaptability to manufacture and eventually, cost and competitiveness. Therefore we follow the paths of development of modified few parametrical methods of modal synthesis. In addition, multiple nonlinear elements (dampers of variable rigidity, elements of variable dry friction etc.) contain few-parametrical models. Interaction with a liquid (for example in tanks of the agricultural machines), also on the foundation of the few-parametrical models is considered.

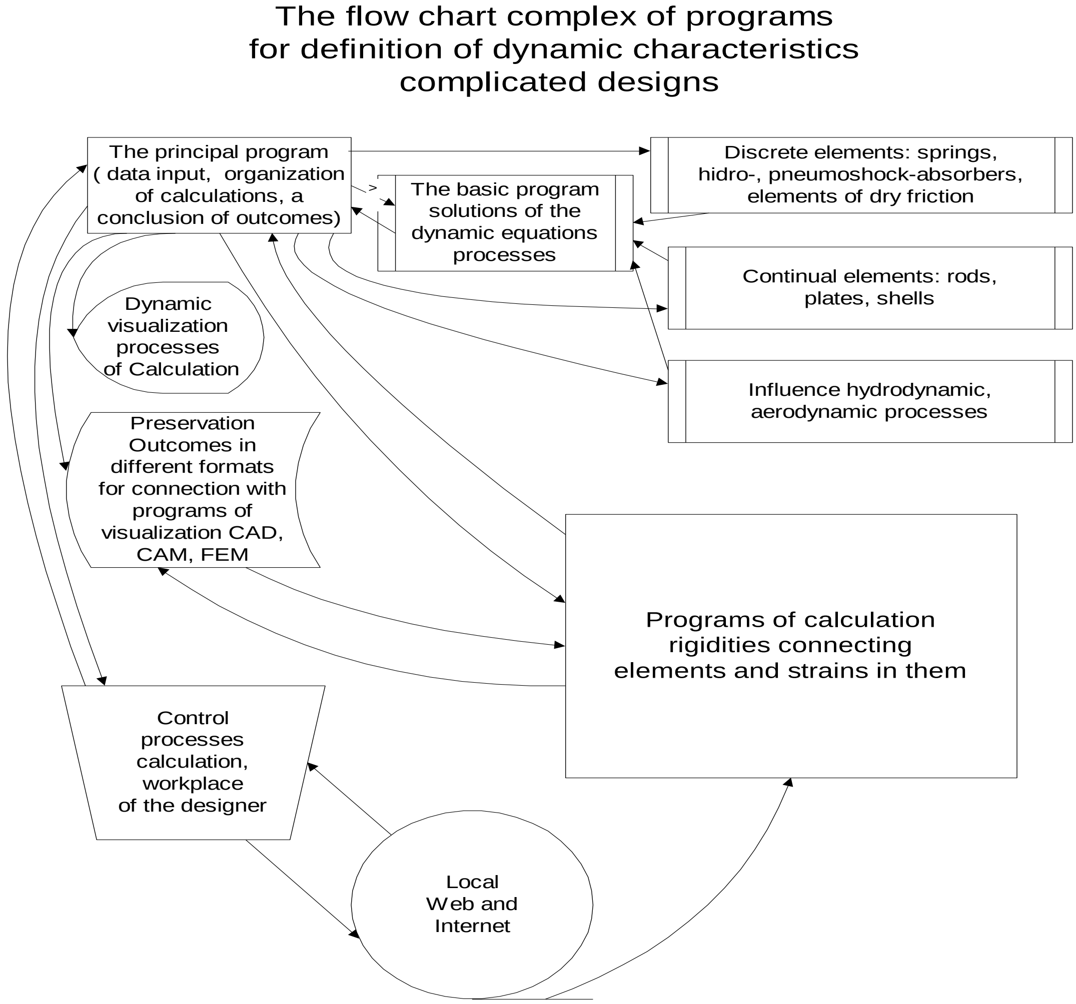

Fig. 2. The flow chart program complex

Obtained few-parametrical models are easily analyzed considering methods of nonlinear programming, mainly the modified methods of a deformable polyhedron and genetic algorithms.

At the stage of program realization of algorithms, we mainly use the method of direct imitation of dynamic processes that allows taking into account many nonlinear effects which are inevitably inherent to any machine or design, and especially, to rotating machines. To solve the obtained systems of the ordinary differential equations modified Gear method and new bilateral numeral methods of first and second order of accuracy [32] for the solution of Cauchy problem is used. It allows investigating random conditions of oscillations of object being under research.

Impact and particle dampers have been extensively studied and investigated at lust time. This is due to the fact that they are simple in design and easy to implement [17-21]. The row of DVA with the impact masses and particle type are also developed. It is considered in [29-31] and in a number of patents.

CONCLUSIONS

1. The paper deals with the methods of calculation and optimization of different types of dynamic vibration absorbers for constructions and building vibrations reduction. The discrete-continuum models with the attached discrete elements are grounded.
2. The algorithms for vibration reduction are received. The main aim of this research is the different type dynamic vibration absorbers investigation and optimization. A technique is developed to give the optimal dynamic vibration absorbers for the elimination of excessive vibration in harmonic and impact-forced mechanical system.
3. In order to determine the optimal parameters of DVA, the complete modeling of dynamics of devices should be performed. A few parameters of numerical schemes of vibration analysis are under discussion.
4. Investigations of the elastic influence on the damping properties of the basic construction and dynamic vibration absorbers are carried out. On this basis, the algorithms for vibration reduction are elaborated.

5. Finally, the obtained results develop the genetic algorithms for optimal design searching by discrete-continuum DVA's system – base system modeling. The new vibro-absorbing elements are proposed.

REFERENCES

1. **Inman D.J. 1996.** Engineering Vibration. Prentice Hall, Englewood Cliffs.
2. **Snowdon J.C. 1968.** Vibration and Shock in Damped Mechanical Systems. Wiley, New York.
3. **Timoshenko S. 1955.** Vibration Problems in Engineering, third ed. Van Nostrand Company, New York.
4. **Ormondroyd J. and. Den Hartog D.B. 1928.** The theory of the dynamic vibration absorber. Trans. Am. Soc. Mech. Engr. 50, A9–A22.
5. **Den Hartog D.B. 1956.** Mechanical Vibrations, fourth ed. McGraw-Hill, New York.
6. **Bishop R.E.D. and Welbourn D.B. 1952.** The problem of the dynamic vibration absorber, Engineering. 174 , 796.
7. **Warburton G.B. 1957.** On the theory of the acceleration damper, J. Appl. Mech. 24, 322–324.
8. **Hunt. J.B. 1979.** Dynamic Vibration Absorbers. Mechanical Engineering Publications, London.
9. **Snowdonv J.C.** Platelike dynamic vibration absorber, J. Engng. Ind., ASME paper No. 74-WA/DE-15.
10. **Korenev B.G. and Reznikov L.M. 1993.** Dynamic Vibration Absorbers: Theory and Technical Applications. Wiley, UK. J.S.
11. **Aida T., Aso T., Nakamoto K. and Kawazoe K. 1998.** Vibration control of shallow shell structures using shell-type dynamic vibration absorber, J. Sound Vibration. 218, 245–267.
12. **Kolovsky M.Z. 1999.** Nonlinear Dynamics of Active and Passive Systems of Vibration Protection, Springer Verlag, Berlin.
13. **Kauderer H. 1958.** Nichtlineare Mechanik, Springer Verlag, Berlin.
14. **Pipes L.A. 1953.** Analysis of a nonlinear dynamic vibration absorber, J. Appl. Mech. 20, 515–518.
15. **Roberson R.E. 1952.** Synthesis of a nonlinear vibration absorber, J. Franklin Inst. 254, 105–120.
16. **Ibrahim R.A. 2008.** Recent advances in nonlinear passive vibration isolators. Journal of Sound and Vibration . 314, 371–452.
17. **Park J., Wang S. and Crocker M.J. 2009.** Mass loaded resonance of a single unit impact damper caused by impacts and the resulting kinetic energy influx. Journal of Sound and Vibration. 323, 877–895.
18. **Saeki M. 2005.** Analytical study of multi-particle damping. Journal of Sound and Vibration. 281,1133–1144.
19. **Marhadi K.S. and Kinra V.K. 2005.** Particle impact damping: effect of mass ratio, material, and shape. Journal of Sound and Vibration. 283, 433–448.
20. **Shah B.M., Pillet D., Bai Xian-Ming, Keer L.M., Jane-Wang Q. and Snurr R.Q. 2009.** Construction and characterization of a particle-based thrust damping system. Journal of Sound and Vibration. 326, 489–502.
21. **Inoue Masanobu, Yokomichi Isao and Hipaki Koju. 2014.** Design of Particle/GranulesDamper for Vertical Vibration with Approximate Analysis Journal of System Design and Dynamics, Vol. 7, No. 4, 367–467.
22. **Diveiev B. 2003.** Rotating machine dynamics with application of variation-analytical methods for rotors calculation. Proceedings of the XI Polish – Ukrainian Conference on "CAD in Machinery Design – Implementation and Education Problems". – Warsaw, June , 7–17.
23. **Kernytskyy I., Diveyev B., Pankevych B. and Kernytskyy N. 2006.** Application of variation-analytical methods for rotating machine dynamics with absorber. Electronic Journal of Polish Agricultural Universities, Civil Engineering. Volume 9, Issue 4. Available Online http://www.ejpau.media.pl/
24. **Stocko Z., Diveyev B. and Topilnyckyj V. 2007.** Discrete-continuous methods application for rotating machine-absorber interaction analysis. Journal of Achievements in Materials and Manufacturing Engineering. Vol. 20, ISS. 1-2, January-February, 387–390.
25. **Ho V.C., Veprik A.M. and Babitsky V.I. 2003.** Ruggedizing printed circuit boards using a wideband dynamic absorber. Shock and Vibration. 10, 195–210.
26. **Nishimura H., Yoshida K. and Shimogo T. 1990.** Optimal Active Dynamic Vibration Absorber for Multi-Degree-of-Freedom Systems, JSME International Journal, Vol. 33, No. 4, Ser. III, 87–99.
27. **Hollkamp J. J. and Starchville T. F. 1994.** A Self-Tuning Piezoelectric Vibration Absorber. Journal of Intelligent Material Systems and Structures. Vol. 5, July. No. 6, 65–76.

28. **Von Flotow A., Beard A. and Bailey D. 1994.** Many devices use various methods to alter the stiffness of the spring element. Adaptive Tuned Vibration Absorbers: Tuning Laws, Tracking Agility, Sizing, and Physical Implementations. Noise-Con 94, 437–454.

29. **Diveyev B., Vikovych I, Dorosh I. and Kernytskyy I. 2012.** Different type vibration absorbers design for beam-like structures. Proceeding of ICSV19 Vilnius, Lithuania,Vol. 2 (Electronic edition).

30. **Cherchyk H., Diveyev B., Martyn V. and Sava R. 2014.** Parameters identification of particle vibration absorber for rotating machines. Proceeding of ICSV21, Beijing, China. (Electronic edition).

31. **Diveyev B., Vikovych I., Martyn. V. and Dorosh I. 2015.** Optimization of the impact and particle vibration absorbers. Proceeding of ICSV22, Florence, Italy. Vol. 2 (Electronic edition).

32. **Pelekh Ya. M., Mentynskyi S. M. and Pelekh R. Ya. 2016.** Nonlinear numerical methods for the solution of initial value problem for ordinary differential equation // Scientific Bulletin Mukachevo State University: Journal of Scientific articles ISSUE 20 (15), 65–75.