ABSTRACT AND REFERENCES

APPLIED MECHANICS

DOI: 10.15587/1729-4061.2020.200020

**CHARACTERISTICS OF DEFORMATION PATTERN AND ENERGY ABSORPTION IN HONEYCOMB FILLER CRASH BOX DUE TO FRONTAL LOAD AND OBLIQUE LOAD TEST (p. 6-11)**

**Moch Agus Choiron**
Brawijaya University, Malang, Indonesia
**ORCID**: http://orcid.org/0000-0002-4052-4832

A crash box design is developed to enhance the crash box's abilities to absorb crash energy. Previous research has developed the crash box by adding filler material. Adding the filling material to the crash box will increase energy absorption. Aluminum honeycomb has a combination of lightweight mass and an ability to absorb crash energy. The addition of filler material to the crash box will also reduce the possibility of global bending in the crash box. The method of study is a computer simulation using ANSYS Academic software ver 18.1. This research used circular, square and hexagonal cross-section variations, which reached the same cross-sectional area design. Geometry model for the crash box and honeycomb filler is defined as crash box thickness (tc) 1.6 mm, honeycomb filler thickness (t) 0.5 mm for single layer and 1 mm for double layer and crash box length (l) 120 mm. The materials used were AA6063-T6 for crash boxes and AA3003 for honeycomb fillers. The test model consisted of two types, namely frontal load and oblique load test. The impactor velocity (v) is set to 15 m/s. The impactor and the fixed support are modeled as a rigid body, while the crash box is assumed as an elastic body. Observations were done by using the characteristics of deformation pattern and the absorption amount of produced energy due to the given loading model. Based on the deformation pattern results, it can be found that in the crash box model with square and hexagon honeycomb filler, the occurred deformation pattern was concertina, while the crash box with circular honeycomb filler was the mixed mode in the frontal load test. Regarding the oblique loads, the crash box remains to collapse the global bending on all models. Simulation results with the frontal load test model found that the crash box with circle-shaped honeycomb has the highest energy absorption while the crash box with hexagonal honeycomb filler has the highest Specific Energy Absorption (SEA). In the oblique load test, it was found that the crash box with hexagonal honeycomb filler has the highest energy absorption and SEA. By comparing the hexagonal crash box model with and without honeycomb filler, it is noted that the hexagonal crash box with honeycomb filler has higher Crash Force Efficiency

**Keywords**: crash box, honeycomb filler, frontal load and oblique load test

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DOI: 10.15587/1729-4061.2020.183319

**ANALYSIS OF FREE OSCILLATIONS OF CIRCULAR PLATES WITH VARIABLE THICKNESS BASED ON THE SYMMETRY METHOD (p. 12-18)**

**Kirill Trapezon**
National Technical University of Ukraine
"Igor Sikorsky Kyiv Polytechnic Institute", Kyiv, Ukraine
**ORCID**: http://orcid.org/0000-0001-5873-9519

**Alexandr Trapezon**
G. S. Pisarenko Institute for Problems of Strength of the National Academy of Sciences of Ukraine, Kyiv, Ukraine
**ORCID**: http://orcid.org/0000-0002-8567-9854

Based on symmetry and factorization methods, a general analytical solution to the fourth-order differential equation has been derived for a problem on the free axisymmetric oscillations of a circular plate of variable thickness. The law of the thickness change is the concave parabola $h=H_0(1-\mu\rho)^2$, where $\mu$ is a constant coefficient that determines the degree of plate concaveness. The solution has been given by the Bessel functions of zero and the first order of the actual and imaginary argument. A circular ring plate has been considered whose inner contour is rigidly fixed and whose outer edge is free, for three values of the $\mu$ coefficient. We have determined the first three natural values for the problem (frequency numbers) and their natural functions (oscillation shapes). It has been shown that the natural frequencies of the first three shapes of oscillations decrease, with the increase in concaveness (increase in $\mu$), to varying degrees, determined by the number of the frequency number $\lambda_i$ (i=1, 2, 3). At $\mu$=1.21417 and $\mu$=1.39127, the frequencies decrease, compared to the case of $\mu$=0.5985, by (1;1.3) %, (17.6;24) %, (22.85;30.35) %, respectively, for $\lambda_1$, $\lambda_2$, $\lambda_3$. One can see a significant drop in frequency on the higher shapes of oscillations ($\lambda_2$, $\lambda_3$) and a slight drop in the basic shape ($\lambda_1$). We have established the values and coordinates of extreme deflections (antinodes of oscillations) and the indicative coordinates of the nodal cross-sections. The reported numerical parameters, along with the frequency indicators, are a means of identifying the oscillational properties of a plate when it is studied in practice. We have built the graphic dependences for radial $\sigma_r$ and tangential $\sigma_\theta$ cyclical stresses at the basic shape for each of the three variants of the concaveness of a parabolic plate. It has been established that the increase in the ratio of edge thickness, that is, concaveness, leads to an increase in $\sigma_r$ in the cross-sections outside the end constraint. These stresses, which operate far from the free edge, for example at the end constraint or the area of the maximum $\sigma_\theta$, are greater than $\sigma_\theta$ in varying degrees. Because of this, these stresses pose a major threat in terms of the cyclical strength of the plate when $\sigma_r$ reaches destructive values. We have pointed to the possibility to provide, by increasing the concaveness of the parabolic plate, the optimal ratio between the value of $\sigma_r$ at the end constraint and $\sigma_r$ operating away from the fastening. This ratio, approximately equal to 1, is ensured at $\mu$=1.39127 considered in the current work.

**Keywords**: circular plate, variable thickness, symmetry method, natural number, stressed-strained state.

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DOI: 10.15587/1729-4061.2020.198661

## DISCOVERING A PATTERN IN THE FREE VIBRATIONS OF GENTLY SLOPING SHELLS OF DIFFERENT GEOMETRY IN THE CLASSIC AND REFINED STATEMENTS (p. 19-25)

**Oleksandr Grigorenko**
S. P. Timoshenko Institute of Mechanics of the National Academy of Sciences of Ukraine, Kyiv, Ukraine
**ORCID**: http://orcid.org/0000-0002-9154-8906

**Oleksandr Parkhomenko**
Mykolaiv V. O. Sukhomlynskyi National University, Mykolaiv, Ukraine
**ORCID**: http://orcid.org/0000-0002-7940-7414

**Valentyna Darmosiuk**
Mykolaiv V. O. Sukhomlynskyi National University, Mykolaiv, Ukraine
**ORCID**: http://orcid.org/0000-0003-3275-8249

**Larisa Vasil'eva**
Mykolaiv V. O. Sukhomlynskyi National University, Mykolaiv, Ukraine
**ORCID**: http://orcid.org/0000-0002-9046-6941

The paper reports an effective numerical procedure to solve problems on the free oscillations of isotropic gently sloping shells using a spline-approximation method of unknown functions along one of the coordinate directions. By applying the proposed procedure, we have examined the resonance frequencies of the oscillations of cylindrical shells and shells of double curvature both in a square and rectangular plan. The calculations were conducted and compared based on two theories: classic (by Kirchhoff-Love) and refined (by Timoshenko-Mindlin). We have established the dependence of natural oscillation frequencies on the ratio of shell thickness and their dimensions in the plan. It has been revealed that the frequencies of free oscillations of gently sloping shells, computed in the refined statement, have lower values than the corresponding frequencies calculated in the classic statement. With the increasing thickness of the shells, the difference in the values of corresponding frequencies increases. The calculations results were compared with the frequencies computed analytically by expanding the unknown functions into a Fourier series. The comparison has allowed us to determine the optimal scope of application of each theory. It has been established that the frequencies of free vibrations of thin gently sloping shells should be computed in a classic statement. The calculation of non-thin shell frequencies (at a ratio of the thickness to the smallest size in the plan of $h/a \geq 0.05$) at any geometric parameters of the shells should be performed in the refined statement. Our results have confirmed the theoretical assumptions about the importance of considering the turning angles, first, of a rectilinear element, caused by transverse offsets, in calculating the natural oscillation frequencies of the non-thin shells. The versatility and high accuracy of the spline approximation method have been confirmed.

**Keywords**: free oscillations, gently sloping shells, classic theory by Kirchhoff-Love, refined theory by Timoshenko-Mindlin.

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DOI: 10.15587/1729-4061.2020.198451

## ANALYSIS OF CONDITIONS OF EFFECTIVE CRACK DETECTION IN SIMPLY SUPPORTED ROD BASED ON CHANGE OF DAMPING (p. 26-32)

**Anatoliy Bovsunovsky**
National University of Food Technologies, Kyiv, Ukraine
**ORCID**: http://orcid.org/0000-0001-9562-0250

**Aleksandr Litvinenko**
National University of Food Technologies, Kyiv, Ukraine
**ORCID**: http://orcid.org/0000-0001-8975-2265

**Sergey Kadomsky**
National University of Food Technologies, Kyiv, Ukraine
**ORCID**: http://orcid.org/0000-0003-4420-7612

**Yiuriy Boyko**
National University of Food Technologies, Kyiv, Ukraine
**ORCID**: http://orcid.org/0000-0002-8972-7446

Existing experimental studies of the sensitivity of damping characteristics to the presence of cracks in structural elements are contradictory. Some studies declare high damping sensitivity but others conclude that a change in the dissipative ability of the structure is not enough for reliable crack diagnostics. This difference may be brought about by the influence of many factors on damping efficiency in relation to crack detection. To predict a possible change in the damping characteristic taking into consideration these factors, an experimental-analytical procedure based on the approaches of fracture mechanics was developed. This procedure has made it possible to identify conditions for reliable detection of an edge crack in a rod on two supports under transverse and longitudinal vibrations. It has been shown that the sensitivity of the damping characteristic to the presence of damage is inversely proportional to the damping level of an undamaged structure. Damping change is effective for diagnosing damages in relatively rigid structures. In this case, the stress level in the damaged area must be high enough so that the crack periodically opens or is constantly open. Based on the analysis of the study results, a condition was formulated that can help engineers easily determine the effectiveness of the damping characteristic for crack diagnostics. The damping characteristic is effective if the ratio of the energy dissipated in the crack to the two-fold potential energy of the structure deformation exceeds the product of the vibration damping characteristic of an undamaged structure by the relative error in its determination.

**Keywords**: logarithmic decrement of vibrations, rod on two supports, edge crack, vibration-based diagnostics of damage.

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**DOI: 10.15587/1729-4061.2020.198660**

**DEFINING PATTERNS IN THE LONGITUDINAL LOAD ON A TRAIN EQUIPPED WITH THE NEW CONCEPTUAL COUPLERS (p. 33-40)**


**Oleksij Fomin**
State University of Infrastructure and Technologies, Kyiv, Ukraine
**ORCID**: http://orcid.org/0000-0003-2387-9946

**Alyona Lovska**
Ukrainian State University of Railway Transport, Kharkiv, Ukraine
**ORCID**: http://orcid.org/0000-0002-8604-1764

**Oleksandr Kovtun**
Danube Institute of the National University "Odessa Maritime Academy", Izmail, Ukraine
**ORCID**: http://orcid.org/0000-0002-6531-2561

**Volodymyr Nerubatskyi**
Ukrainian State University of Railway Transport, Kharkiv, Ukraine
**ORCID**: http://orcid.org/0000-0002-4309-601X



The longitudinal-dynamic load on a railroad train has been studied at its steady motion along the track of a homogeneous profile. A value of the longitudinal loading that a train is exposed to has been established. The calculations were carried out for a train consisting of 40 similar semi-wagons. The magnitude of the longitudinal loading, in this case, is taken to equal 1.2 MN. It is important to note that when increasing the motion speed, as well as the weight of a train, the magnitude of the longitudinal load may exceed the specified value. This contributes to the additional loading on the bearing structures of cars on the train and can cause damage to them. In addition, significant longitudinal-dynamic loads contribute to disrupting the motion stability of cars in the train.

In order to reduce the longitudinal-dynamic efforts in the train under operating modes, including braking, it has been proposed to use, instead of a standard automatic coupling device, a conceptual coupler. In this case, the impact's kinetic energy is damped by transforming it into the work of a viscous resistance force. This resistance is created by moving a viscous liquid through the throttle holes of the piston based on the principle of hydraulic damper operation.

To substantiate the use of a conceptual coupler, the calculation has been performed based on a method for determining the strength of the coupling device through the imaginary separation of a train into two parts.

Taking into consideration a coefficient of the viscous resistance, which is created by the conceptual coupler, the acceleration experienced by a train reached about 0.8 m/s$^2$. In other words, the use of a conceptual coupler makes it possible to reduce the longitudinal loading on a train by almost 30 % compared with the standard scheme of interaction between a locomotive and cars.

The rod of the conceptual coupler has been estimated for strength. It has been established that the maximum equivalent stresses do not exceed permissible limits.

The proposed measures would contribute to the reduction of a dynamic load on a railroad train under the loading modes of operation. The implementation of a given concept could also contribute to bringing down the damage to railroad stock in exploitation.


**Keywords**: railroad train, longitudinal dynamics, dynamic loading, conceptual coupler, modeling of dynamics.

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**DOI: 10.15587/1729-4061.2020.199486**

**A NUMERICAL STUDY OF PERFORMANCE OF THE SMALL-SIZE UAV PUSHING TANDEM PROPELLER WITH JOINED BLADES (p. 40-48)**

**Mykola Kulyk**
National Aviation University, Kyiv, Ukraine
**ORCID**: http://orcid.org/0000-0003-2149-4006

**Fedir Kirchu**
National Aviation University, Kyiv, Ukraine
**ORCID**: http://orcid.org/0000-0001-8437-4402

**Hussein Hanesh**
National Aviation University, Kyiv, Ukraine
**ORCID**: http://orcid.org/0000-0002-8995-7866

A new shape of the tandem blade with an improved arrangement of profiles, in relation to the known propellers, in which the profiles are located similar to those in the tandem wing of an airplane is developed. A new arrangement of profiles along the height of the blade is proposed. The basis for the design was the location of the profiles according to the type of tandem blade rows of compressors and fans. Such an approach made it possible to eliminate the aerodynamic shadowing of the blades and increase the aerodynamic loading. To join the blades in the final part, a spiral tip connector is used, which allowed to significantly reduce the secondary end losses by preventing the formation of the tip vortex.

To study the characteristics of tandem propellers and the structure of gas-dynamic flows around them, a computational model of the propeller in a periodic formulation was developed, which significantly reduced the calculation time. The simulation was carried out in the ANSYS CFX software package, which implements an algorithm for solving non-stationary Reynolds averaged Navier-Stokes equations closed by the SST turbulence model. As a result of the simulation, the characteristics of the tandem propeller were obtained, which confirmed the correctness of the chosen approach for the design of the tandem blade. The efficiency of the developed propeller reaches 75 % in the design mode, which is a very good indicator for small propellers operating at low Reynolds numbers. For comparison, the efficiency of classic propellers with similar geometric characteristics is in the range of 50–60 %. When using the tandem propeller with joined blades as a pusher propulsion, a decrease in its thrust by 3–4 % was observed, which is due to the formation of a vacuum zone in the hub part and in the spinner area.

**Keywords**: propeller, tip vortex, Moebius band, box propeller, tandem propeller.

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**DOI: 10.15587/1729-4061.2020.200998**

## DETERMINATION OF THE RATIONAL NUMBER OF BLADES OF THE CENTRIFUGAL WHEEL OF A SUBMERSIBLE PUMP (p. 49-58)

**Madina Isametova**
Satbayev University, Almaty, Republic of Kazakhstan
**ORCID**: http://orcid.org/0000-0003-4630-271X

**Dimitar Karaivanov**
University of Chemical Technology and Metallurgy, Sofia, Bulgaria
**ORCID**: http://orcid.org/0000-0002-1709-677X

**Rollan Nussipali**
Satbayev University, Almaty, Republic of Kazakhstan
**ORCID**: http://orcid.org/0000-0001-6965-0667

The CAD/CAE/CAM method of end-to-end design of the impeller of a seven-stage submersible pump ODDESSEzentralasien –UPP 13-7/6 used for pumping sulfuric acid in hydrometallurgy is presented.

The studies are conducted in order to increase the efficiency of the pump manufactured at the KARLSKRONA LC AB LLP plant (Kazakhstan). Computer calculations of the centrifugal wheel with 8 and 9 blades for strength were carried out in the NASTRAN top-level CAE system. The influence of the number of centrifugal wheel blades on the level of stresses arising in the sections of the blades of the cover and main centrifugal wheel discs is determined. The maximum stress in the sections of the wheel with 8 blades reached 319 MPa and the wheel with 9 blades 199 MPa. The influence of the number of blades on the dynamic characteristics of the rotor shaft is examined. To do this, design mechanical and computer schemes of dynamic calculation are simulated to determine the amplitude-frequency characteristics of the rotor shaft. The harmonics amplitudes at frequencies caused by liquid pulsation at the blade frequency of 400 Hz and 450 Hz reached $1 \cdot 10^{-4}$ m and $8 \cdot 10^{-4}$ m, respectively. Based on the results of computer modeling of static and dynamic problems, a model of the impeller of a centrifugal multistage pump with a rational number of 8 double curvature blades is developed. The choice of the number of blades meets the criterion of wheel strength and the dynamic criterion of the shaft-wheel system.

For the production of the prototype wheel, an analysis of the process parameters of 3D printing in terms of surface roughness of finished products is carried out. Based on the analysis, stereolithography the technology is chosen and centrifugal wheels are printed for further bench hydrodynamic tests in a plant. The studies based on CAD/CAE/CAM computer modeling allow reducing the time and costs of developing a rational wheel geometry that meets the criterion of both the strength of the wheel itself and the criterion of vibration activity of the rotor shaft.

**Keywords**: submersible pump, centrifugal wheel, computer-aided design systems, amplitude-frequency response.

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**DOI: 10.15587/1729-4061.2020.200428**

**INDENTIFYING THE CONDITIONS FOR THE OCCURRENCE OF STATIC SELF-BALANCING FOR AN ASSYMETRIC ROTOR ON TWO ISOTROPIC ELASTIC SUPPORTS (p. 59-66)**

**Irina Filimonikhina**
Central Ukrainian National Technical University, Kropyvnytskyi, Ukraine
**ORCID**: http://orcid.org/0000-0002-1384-6027

**Viktor Deikun**
Central Ukrainian National Technical University, Kropyvnytskyi, Ukraine
**ORCID**: http://orcid.org/0000-0001-5420-5878

**Iryna Ienina**
Flight Academy of the National Aviation University, Kropivnitsky, Ukraine
**ORCID**: http://orcid.org/0000-0002-2122-7808

**Mareks Mezitis**
Riga Technical University, Riga, Latvia
**ORCID**: http://orcid.org/0000-0003-0269-7297

**Vladimir Pirogov**
Central Ukrainian National Technical University, Kropyvnytskyi, Ukraine
**ORCID**: http://orcid.org/0000-0002-5843-4552

**Guntis Strautmanis**
Riga Technical University, Riga, Latvia
**ORCID**: http://orcid.org/0000-0001-8405-939X

**Sergij Yakimenko**
Central Ukrainian National Technical University, Kropyvnytskyi, Ukraine
**ORCID**: http://orcid.org/0000-0002-5759-9603

This paper reports the established conditions for static self-balancing for the case of an asymmetric rotor on two isotropic elastic supports, balanced by a passive automatic balancer of any type. In general, the plane of static imbalance does not coincide with the plane of an automatic balancer.

The energy method has been used under the assumption that the mass of an automatic balancer's loads is much smaller than the mass of the rotor.

It has been established that the static balancing of the rotor by an automatic balancer of any type is possible in the following cases:

– a long rotor when the rotor rotates at speeds between the first and second and above the third characteristic velocities;

– a spherical rotor when the rotor rotates at speeds between the first and second characteristic velocities;

– a short rotor at speeds exceeding a certain characteristic velocity provided that the automatic balancer is close to the center of the rotor mass.

The rotor asymmetry increases the number of resonant speeds but the number of regions where the self-balancing is occurred does not change.

The imbalance of the rotor and its location do not affect the characteristic rotation speeds of the rotor. An automatic balancer in the range of rotor rotation velocities that ensure the self-balancing tends to maximally reduce the deviation of its center from the rotor rotation axis. When the rotation velocity of a long or spherical rotor approaches the second characteristic speed, the automatic balancer's capacity ceases to provide for the complete elimination of the automatic balancer's axis deviation from the rotor's rotation axis.

The result obtained summarizes the findings derived earlier when using the empirical criterion for the occurrence of self-balancing. The energy method, in contrast to the empirical method, has made it possible to estimate the residual deviation of the rotor's longitudinal axis from the rotation axis. That allows the estimation of the reserve or the calculation of the automatic balancer's balancing capacity.

The type of automatic balancers is not taken into consideration in such studies. Therefore, the results obtained are suitable for automatic balancers of any type, and the method itself is suitable for constructing a general theory of passive self-balancing (applicable for automatic balancers of any type).

**Keywords**: rotor, isotropic support, automatic balancer, stationary motion, motion stability, steady motion equation.

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**DOI: 10.15587/1729-4061.2020.198501**

## SOLUTION OF THE SYSTEM OF GAS-DYNAMIC EQUATIONS FOR THE PROCESSES OF INTERACTION OF VIBRATORS WITH THE AIR (p. 67-73)

**Roman Antoshchenkov**
Kharkiv Petro Vasylenko National Technical University of Agriculture, Kharkiv, Ukraine
**ORCID**: http://orcid.org/0000-0003-0769-7464

**Anton Nikiforov**
Kharkiv Petro Vasylenko National Technical University of Agriculture, Kharkiv, Ukraine
**ORCID**: http://orcid.org/0000-0001-7788-8878

**Ivan Galych**
Kharkiv Petro Vasylenko National Technical University of Agriculture, Kharkiv, Ukraine
**ORCID**: http://orcid.org/0000-0002-9137-036X

**Victor Tolstolutskyi**
State-Owned Enterprise Kharkiv Machine Building Design Bureau named A. A. Morozov, Kharkiv, Ukraine
**ORCID**: http://orcid.org/0000-0002-8609-7055

**Vitalina Antoshchenkova**
Kharkiv Petro Vasylenko National Technical University of Agriculture, Kharkiv, Ukraine
**ORCID**: http://orcid.org/0000-0002-3963-6263

**Sergey Diundik**
National Academy of National Guard of Ukraine, Kharkiv, Ukraine
**ORCID**: http://orcid.org/0000-0003-3558-0028

The modern practice of using vibratory machines involving small seeds of low weight faces such an undesirable phenomenon as the effect exerted on the kinematics of vibrational movement of particles of fractions of the seed mixture by the aerodynamic forces and momenta. The periodic movement of air relative to the working planes of a vibratory machine arises due to fluctuations in the packets of these planes, which form flat aerodynamic channels. Consequently, the issues of studying the processes of interaction between the working bodies of vibratory machines and the air environment, aimed to justify their structural improvements, appear relevant. Existing mathematical models, which assess the parameters of air movement relative to the working planes of vibratory machines, produce only a generalized pattern and are flat. This paper proposes a statement, as well as an estimated finite difference scheme, of solving a three-dimensional boundary value problem on calculating the field of velocities and pressures in the region of air, located between two parallel synchronously oscillating planes. The problem employs a system of differential equations to describe the flow of the perfect gas. The finite difference scheme has been solved by a sweep method.

Using the sweep method to solve these kinds of problems makes it possible to ensure the convergence and stability of estimation schemes, regardless of the step and other parameters of the grid applied. A variant of the calculation has been given, which demonstrated the feasibility of the proposed method for the assigned boundary conditions and parameters of the vibrational mode of machine operation. It has been established that in the working space enclosed between two oscillating planes there are both vertical (transverse) and horizontal (longitudinal) components of air velocity, which change over time.

**Keywords**: gas dynamics, system pf differential equations, boundary value problem, grid method, tridiagonal matrix algorithm, velocity field.

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