Design process of executive elements in stroke mechanisms based on rammer's deformations and stress modeling

M S Chepchurov¹, V P Voronenko² and A V Chirkov³

¹Department of "Mechanical equipment and machine building technology", Belgorod State Technological University named after V.G. Shukhov, Kostyukov St., 46, Belgorod, 308012, Russia
²Department of Technological Engineering, Moscow State Technological University “STANKIN” (“MSUT "STANKIN"”), Russia, Moscow, GSP-4, Vadkovsky lane, 1, 127994
³Department of machine building technology Technical University of Liberec, Studentská 1402/2, 461 17 Liberec 1, Czech Republic, EUROPE

E-mail: avtpost@mail.ru

Abstract. The article deals with the strength calculation issues of products by example of rammer under short-term loads in hammer drill. Based on finite element analysis’ results of the rammer geometrical model, it is proposed to select a material with a strength limit significantly exceeding the allowable residual stresses inside the rammer. The harmonic analysis allows us to estimate the rammer’s strength. The dynamic coefficient may be calculated by the use of Vibran 2.0 device (amplitude and signal frequency) and by calculation results of the maximum displacement. The FreeCAD software package was used for this. It allows to create internal scripts to automatically search for the optimal design parameters. A method of determining mechanism’s static load and its dynamic coefficient under periodic impact loads has been developed. Static loads calculations and harmonic analysis according to the developed method allow to determine structural strength of the product, to select the rammer’s material and to find the dynamic coefficient (which takes into account the stroke loads in calculations). The proposed calculation method may be used not only to solve the describing problem, but also for a whole class of problems related to stroke loading, where such parameters as application time of force or duration time of force’s impulse are important.

1. Introduction

Designing process for machine components with rod elements under stroke loads is associated with additional time for choosing the material of rammer and its geometric form. It is not enough to perform only static strength calculations and endurance analysis for such elements. Because of this fact most of constructers have to order additional researches based on standard documents’ requirements [7] with the use of special equipment [3], which demands additional costs. The result of such tests is not always sufficient for considering the right decision and leads to repeating of experiments. The whole cost of designing process and experimental work is shifted to the cost of the end product. That decreases the competitiveness of designed product and delays the mass production start-up date.
grated CAD / CAE automation systems for designing procedures with the FEM module may significantly reduce the costs of experiments and perform a great number of iterations in a short period of time [16]. In this case, the experiment itself moves to the virtual reality. Integration of the various auto-designing modules requires several things to organize in order to make a right decision in project lifecycle of product: to set automatic information exchange between all of such modules and to set process of outputting results at least. In this case, the designer or constructor should have a huge base of reference knowledge in order to choose correctly all of the physic and mechanical parameters for such part of hammer tool as a rammer.

Static problems may be solved by CAE applications rather simply, but dynamic tasks require additional qualification from a designer (theoretical physics, mathematic methods, material resistance etc.), as fully presented in [11], that may be problematic for a general constructor or designer. The authors propose a simplified, but effective way to solve the problem related to the stroke loads in axial direction for round rod part. This method also includes a process of selection rammer’s material by the use of the licensed software GPL2.

2. Materials and methods

2.1. Materials

The authors faced with problems of material and geometric parameters, when developed a special device (for hammering dowels). The device consists of a rammer installed in a support, which is carrying by a spring fixed on the rammer with the help of stopper. The 3D model is presented in Fig.1a, and the really manufactured device in Fig.1b.

![Figure 1. Device for hammering dowels in a hammer drill: a- 3D model; b - photo of the real prototype.](image-url)
Figure 2. Hammer and loading scheme: a) the shape of the hammer’s rammer, b) the scheme of loads.

The aim of calculations was to determine the best shape of the rammer’s work zone (or rammer’s impacting end) in accordance with the sketch shown in Fig.1a and selected axial loading scheme (Fig.2b), as the paper [18] recommends.

The purpose of researches is to determine rammer’s diameter $D$, length $H$, which are in a certain range from $D_{\text{min}}$ to $D_{\text{max}}$ and from $H_{\text{min}}$ to $H_{\text{max}}$ respectively. It is necessary to determine such values, which will not exceed limits of maximum stresses for selected material under short-term action of the $F$ force. The general scheme of loads (Fig.2b) contains the force $F$, applied to the upper end of the rammer (rod), which has deformation $\delta_1$. Its value depends on many factors: the applied force $F$, the cross-section of the rod and the material’s elastic modulus.

The source of the force $F$ is the hammer drill’s mechanism (mechanical unit) of periodic rammer’s displacements, i.e. it sets a certain frequency of a short-term applied force. In this case, the stroke energy, transmitted to the striker, expends on heating the elements and partly returns back to the executive element of the hammer drill’s mechanism. Instrument’s manufacturers claim different values of stroke energy, depending on the power of device. Meanwhile electro pneumatic rotary hammers provide maximum stroke energy [8] up to 20 $J$. The strokes frequency from a heavy hammer drill reaches $1000 \ldots 2000$ strokes / min, which corresponds to oscillation frequency of the executive element $17 \ldots 35$ $Hz$.

Measurement results of rammer’s vibrations, obtained by Vibran 2.0, were processed with the use of its special software. The software window with the results of harmonic analysis is presented on Fig.3.

The information about stroke frequency and types of hammer drill’s movements in open sources is rather contradictory, so the authors made their own experiments and measured hammer drill frequency characteristics with Vibran 2.0 device. Measurements were based on the recommendations [15]. According to Fig.3, the movements of the executive element are periodic, but its movement over the period is irregular, because the graph of displacement is not sinusoidal. Therefore, we can’t take such form of movement for further modeling as periodic movements of rammer [14]. The analysis of amplitude-frequency characteristic (Fig.4) shows maximum displacements $\approx 0.9$ $mm$ at a frequency of 29.9 $Hz$, but the signal shows another additional harmonics at frequencies from 51.8 $Hz$ to 209.3 $Hz$. The movement amplitude of the executive component at a maximum frequency of 209.3 $Hz$ is approximately 13% of the amplitude of the carrier frequency. Consequently, apart from calculations for static strength, it is necessary to perform a frequency analysis and to define critical values of mechanism’s own frequencies. That also means to define whether the real investigated oscillations reach at least one of mechanism’s own resonant frequencies.
Rammer as an ordinary tool undergoes deformation. These deformations may exceed the maximum allowable value for selected material in some moment. Consequently, the material is selected according to the allowable yield strength. There are characteristics of some materials in Table.1, it also presents another parameter of materials - stroke viscosity or KCU (the stroke energy value at which the material will be destroyed). As it can be seen from Table.1: these values, as well as the yield strength, depend not only on the chemical composition, but also on the chosen heat treatment method. A material with a high yield strength may have a lower stroke viscosity. The table was compiled taking into account the requirements to loaded elements of mechanism with strength and elasticity properties, for which the authors used paper [19].

Figure 3. Results of the harmonic analysis on the basis of “Vibran 2.0” data.

Figure 4. Amplitude-frequency characteristic of movements of the rammer.
Table 1. Steel properties.

| Steel grade DIN(EN) | Heat treatment: annealing, normalization | Heat treatment: quenching with oil cooling |
|---------------------|----------------------------------------|------------------------------------------|
|                     | Young's modulus, MPa | Strength's limit, MPa | Stroke strength, J | Young's modulus, MPa | Strength's limit, MPa | Stroke strength, J |
| C 20, Ck 20         | 210                    | 420                      | 2.2                | -                    | -                    | -                    |
| C 45, Ck 45         | 209                    | 610                      | 1.2                | 203.6                | 1200                 | 0.9                  |
| 42Cr4               | 210                    | 630                      | 1.6                | 204                  | 850                  | 1.2                  |
| 20MnCr5G           | 208                    | 700                      | 2.2                | 203.6                | 1000                 | 2.0                  |

2.2. Methods

The choice of CAE – application implies the use of an appropriate theoretical knowledge for solving the problem. Since the authors assume to use FEM analysis for simulation processes (rod under stroke loading), it is necessary to consider the recommendations for choosing equations of mathematical physics, for example [12].

During stroke process, the load is dynamic, and static strength calculations should be done by taking into account previously calculated dynamic coefficient.

Various sources recommend to use the same dynamic calculations methods as for static ones: \( F_d = F_{st} \cdot k \); \( \delta_{max} = \delta_{max} \cdot k \leq [\delta] \), where \( k \) is the dynamic coefficient.

Let’s consider, that stroke causes deformation of the elastic rod in the longitudinal direction. All of the energy is directed only to its compression. Its cross section in that case should be chosen in such a way as to eliminate bending in the transverse direction [2]. In the case of significant values of applied force, the rammer should be checked for the critical limiting values, according to one of the already known methods, for example:

\[
F_{cr} = \frac{\pi^2 \cdot E \cdot I_{mnt}}{L^2}
\]

where \( E, \text{ MPa} \) — Young’s modulus; \( I, \text{ mm}^2 \) — axial inertia moment of the rod in the transverse direction; \( L, \text{ m} \) — the length of the rod or the distance between the fixed rod’s ends.

Another one thing should be noted. Authors used the distance between the supports as the distance from the hammer drill’s chuck end to the rod’s end in calculations of rammer. Actually, the critical force in form of production \( E \cdot I \) may be considered as the rod bending stiffness. Calculations for such an element as a rammer may include just the previously selected values of the Young’s modulus without further stability check. But deflection of the rod under the longitudinally applied force should be calculated in accordance with the recommendations [12]. For selected scheme of loading, for example, on Fig.5, you can use the differential equation:

\[
\frac{d^2 \gamma}{dx^2} = \frac{M}{E \cdot I}
\]

where \( M = -F \cdot \gamma, \text{ N} \cdot \text{m} \) is the bending moment under the action of \( F \) force; \( x \) is the coordinate of maximum point of deflection, m.

\[
T_0 = U_L + U_r + U_t
\]

where \( U_L \) is the deformation energy of the rod, fixed according to the scheme on Fig.6; \( U_r = \frac{m \cdot v^2}{2} \), \( J \) is the energy required to return the power tool working element – striker (flying piston) with mass \( m, \text{ kg} \); \( U_t \) is the energy spent to heat the colliding parts, which can be neglected in case of elastic system.
The mass of a striker (flying piston) may be from 0.05 kg to 0.8 kg, depending on the power of hammer drill. For example, for a chosen one, the weight of the striker was 0.07 kg, the frequency of the vibrations was ≈30 Hz, the maximum movement was 30 mm, i.e. \( \frac{1}{30} \) movements were done for a one period, or taking into account the frequency \( \frac{1}{60} \) s, the time of striker’s movement, or its speed, \( h \approx 0.03 \cdot 60 = 1.8 \text{ m/s} \). whence, the energy spent on striker’s return is:

\[
A_T = \frac{0.07 \cdot 1.3^2}{2} = 0.2268 \text{ J}.
\]

With the declared stroke energy of 2 J, \( A_T \) means about 10% of this energy, thus

\[
T_0 - A_T = U_L \Rightarrow 0.9 \cdot T_0 = U_L
\]

Let’s calculate the deformation potential energy:

\[
U_c = \frac{1}{2} \cdot F \cdot \delta_c,
\]

J, where \( \delta_c \) is the value of static deformation, mm.

Reaction of system C to the action of force \( F \): \( \delta_d = R_d \cdot c \cdot \text{mm} \), where \( c \) is the proportionality coefficient, according to Hooke’s law. Thus, it may be introduced as \( U_c = \frac{c}{2} \cdot \delta_c \cdot J \), a and

\[
U_d = \frac{1}{2} \cdot R_d \cdot \delta_d = \frac{F}{2 \cdot \delta_c} \cdot \delta_d^2, \text{ J}.
\]

If we substitute the result expressions in the original expression (3), we will have:

\[
0.9 \cdot T_0 = \frac{F}{2 \cdot \delta_c} \cdot \delta_d^2, \text{ J}, \text{ given that according to Hooke's law}
\]

\[
c = \frac{F}{\delta_c}
\]

(6)

Thus:

\[
0.8 \cdot T_0 - \frac{F}{2 \cdot \delta_c} \cdot \delta_d^2 = 0,
\]

(7)

that gives us several unknown variables: \( F, \delta_c, \delta_d \), which must be determined on the basis of calculations and results of experiments.

We will try to use similar calculations for some kind of situation with free fall (in our case, a striker of a power tool).

The load of mass \( m \) falls from the height \( H \), so the work is \( A = m \cdot g \cdot H \cdot \text{J} \), i.e. it can be written as

\[
0.9 \cdot T = mgH, \text{ whence } H = 0.9 \cdot T_0 / mgH.
\]
In the dynamic coefficient calculations from various sources, it is \( H \) value that allows us to calculate the dynamic coefficient:

\[
K_d = 1 + \sqrt{\frac{2H}{\delta_c}};
\]

\[
K_d = 1 + \sqrt{\frac{m \ g \ T_0}{\delta_c}} = \sqrt{\frac{0.9 \ T_0}{m \ g}} \delta_c
\]

Let’s equate the expressions of stroke energy: energy of speed and energy of height:

\[
\frac{m \ v^2}{2} = m \cdot g \cdot H \quad \text{or} \quad H = \frac{g \cdot v^2}{2} \quad (10) \quad \text{m}.
\]

Or the final dynamic coefficient:

\[
K_d = \sqrt{\frac{g \cdot v^2}{\delta_c}}
\]

It remains to calculate the force applied to the rod’s end used in finite element calculations and modeling.

According to the «Vibran 2.0» data, the maximum movement amplitude of executive element will be \( 0.9 \cdot 10^{-3} \) meters, then the work force is \( F = \frac{v^2}{A_{max}} \cdot H \), where \( A_{max} \) is the maximum movement amplitude or path, and \( T_0 = 0.97 \cdot T \), J, as described above. Let’s define:

\[
F = \frac{0.97 \cdot 3.09 \cdot 10^{-3}}{0.9 \cdot 10^{-3}} \approx 3100, \quad \text{N}.
\]

After creating the FEM analysis, we assign border conditions and loads. The border conditions are constant, but the force, applied to the rod’s end, may vary depending on both factors: amplitude of movements and impulse energy of power tool as it is in our case (in general case - only energy of impulse).

The authors used FreeCAD as a software package for designing and calculating [1] with a GLP2 license, which complies the requirements about free software, for example [9]. The parametric modeling system of product’s geometry allows to the use calculated data from the electronic table of the same project, as shown in Fig.7.

The spreadsheet and the window of project are shown in Fig.6. All the details of creation processes, modeling process and FEM analyze in FreeCAD with the help of CalculiX calculator [4] were done in accordance with the recommendations outlined in paper [13].

**Figure. 7.** Window of the electronic table and project.
The obtained calculation results should be analyzed for normal and maximum displacements in the model [17] and also should be determined the value of these displacements and deformations. According to the Vibran 2.0 data, the oscillatory process is not strictly sinusoidal, i.e. there are another additional harmonics, the frequency of which may coincide with the rammer oscillation frequency. For this, we will perform a mechanism’s harmonic analysis by the use of CalculiX. The analysis result (Fig.7b) showed that in the range from 0 to 1000 Hz there are only two resonance frequencies (379.61 and 379.728 Hz), which exceeds the limit values of the harmonics. Consequently, dangerous damage to the construction is not indicated. In this case, the maximum (by von Mises) stress is only 82.37 MPa, that is less than with a static load only. The resonant frequencies are also recorded into the table for further analysis. At the same time, there are not specific resonant frequencies in the cells of the table (Fig.6), but there are parameters of the finite element model. That makes it possible to automate part of the calculations.

3. Results and discussion
The finite element analysis results make it possible to select a rammer’s material with an acceptable strength’s limit using the table.1 of this paper.

The dynamic coefficient may be obtained with the help of calculation results based on the maximum displacements and Vibran 2.0 data (amplitude and frequency of signal) [12]. The calculated dynamic coefficient, according to expression (9) is 15.69 (for this case).

The FreeCAD package allows to create internal scripts for the automatic design calculations. This option is very useful for automatic searching of optimal design parameters [5]. The authors used Netgen 3.0 as a grid generator, where the grid changes automatically.

Automatic calculations should be organized with the direct access to project parameters from script: in order to make possible to change diameters, length, loads, to select the supporting surface. When all the initial parameters are set, the launch of the calculator will be performed also automatically. The results obtained by the FEM are processed by the FreeCAD integrated post-processor.

Thus, the authors obtained results allowing to determine the rational geometric form for the rammer. For example, if we change the type of support of rammer’s end, the stresses arising into it will be also changed. Fig.8 shows the restriction scheme of rammer’s end movement (along two surfaces) (Fig.8,a) and the results of equivalent stresses calculations (by von Mises) (Fig.8,b).

According to the calculations, the maximum equivalent von Mises stress for the two-surface limiting scheme is 325 MPa, which allows us to assign any material, for example, C20 without heat treatment, according to the data from Table 1.

During tests of the device [10] with a rammer from this material, the authors noticed the deformation of rammer’s end surface (face of rammer), because at some moments the rammer relied only on this surface. Calculations were performed again, but with a different scheme for limiting movements, as shown in Fig.8,a.
The results of the von Mises equivalent stress calculations (Fig. 8, b) show that the maximum stress value may reach 572 MPa. As the width of the end surface’s ring decreases, they will increase. So, to avoid catastrophic consequences for all the device, the authors chose the material C45 DIN17200, which have a strength’s limit (after quenching with oil cooling) - 1200 MPa. The successful application of the authors’ device may be found at https://cloud.mail.ru/public/2HBk%2F4q6iDpMmd.

Special steels with corresponding properties were described in "Research and Development - 2016" (Springer, Cham) and may also be assigned as a material for an stroke part, which allows to achieve mass production with minimum cost for such products.

4. Summary
Method for calculating the whole system’s static load and its dynamic coefficient under the periodic stroke action has been developed. It makes possible to determine the magnitude of the maximum static deformation stress with the use of a finite element analysis. The technique can be used for calculating other displacements and stresses in the rods with periodic axial loads.

All the calculations (static loading and harmonic analysis) allow us to determine the mechanism’s strength, to select the material for the blank (workpiece), to determine the dynamic coefficient for the rammer. The authors also proved the efficiency of using software packages for parametric modeling with the GLP2 license. There is a free program code in this software, that allows to use automatic searching for the best solution.

The proposed calculating method for both: the dynamic coefficient and the applied force magnitude is based on empirical calculations of stroke mechanisms. That makes possible to use it not only for solving the problem described in this article, but also for a whole class of problems with stroke loads, where exist properties, like time of force’s application or duration of force’s impulse.

5. References
[1] FreeCAD, URL: https://www.freecadweb.org, Accessed 20 August 2019
[2] Kruszka L, Vorobiov Y S, Ovcharova N Y 2014 FEM Analysis of Cylindrical Structural Elements under Local Shock Loading Applied Mechanics and Materials 566 499-504 https://doi.org/10.4028/www.scientific.net/AMM.566.499
[3] Atapin V G, Rodionov A I, Rykov A A, Titorenko V P, Yur'ev G S, Ivanov Y A 2009 Stands and devices for material tests and the specialist of products on impact resistance and shock stability Scientific bulletin of Novosibirsk State Technical University 3(36) 87-98
[4] Wittig K 2018 CalculiX USER’S MANUAL - CalculiX GraphiX, Version 2.14 May 29 159 URL: http://www.dhondt.de/cgx_2.14.pdf
[5] Falck D, Collette, B 2012 Solid Modeling with the power of Python, Freecad [How-To] Packt

Figure 8. The calculation of the equivalent von Mises stresses for two-surfaces limitation: a - scheme for assigning the constraints; b - calculation results.
[6] Adhikari S, Murmu T, McCarthy M A 2013 Dynamic finite element analysis of axially vibrating nonlocal rods. *Finite Elements in Analysis and Design* 63 42-50 doi: 10.1016/j.finel.2012.08.001

[7] GOST R 51371-99 2000 Mechanical environment stability test methods for machines, instruments and technical products. Test for influence of shocks *Moscow: Gosstandart Rossii: Izdvo standartov*

[8] Chen H, Sun Y 2017 Development and application of reliability test platform for high-speed punch machine clutch brake system. *J Mech Sci Technol* 31 53–61 doi.org/10.1007/s12206-016-1207-1

[9] Luca A, Francesco L, Paolo S 2016 Can Open-Source 3D Mechanical CAD Systems Effectively Support University Courses *International Journal of Engineering Education* 32 3(A) 1313-1324

[10] Zhukov E M, Chepchurov M S 2014 The device for driving in of expansion bolt shields *Patent RF 142819*

[11] Polocoşer T, Kasal B, Li X J 2017 Design of Experiment and Pitfalls of Low-Velocity Pendulum Impact Testing. *J. dynamic behavior mater* 3 436-460 doi.org/10.1007/s40870-017-0123-5

[12] Nasdala L 2010 FEM-Formelsammlung Statik und Dynamik *Springer-Verlag* 230

[13] Bernd K 2007 FEM – Grundlagen und Anwendungen der Finite-Elemente-Methode im Maschinen und Fahrzeugbau *Springer-Verlag* 498

[14] Yang Y 2009 Understanding of Vibration Stress Relief with Computation Modeling *J. of Materi Eng and Perform* 18 856–862 https://doi.org/10.1007/s11665-008-9310-9

[15] Buzdugan G, Mihăilescu E, Radeş M 1986 Instrumentation for vibration measurement. In: Vibration measurement. *Mechanics: Dynamical Systems Springer Dordrecht* 8 https://doi.org/10.1007/978-94-017-3645-9_5

[16] Louhichi B, Abenhaim G N, Tahan A 2015 CAD/CAE integration: updating the CAD model after a FEM analysis *Int J Adv Manuf Technol* 76 391-400 doi.org/10.1007/s00170-014-6248-y

[17] Madenci E, Guven I 2006 The finite element method and applications in engineering using *Ansys® Springer US* 686

[18] Rajendran R, Sai K P, Basu S 2009 Axial impact studies on steel tubes and zircaloy rod *Exp Tech* 33(1) 17-22 doi: 10.1111/j.1747-1567.2008.00363.x

[19] Yamada Y 2007 Materials for Springs *Springer-Verlag Berlin Heidelberg* 377

[20] Lomaeva T V, Lukin L L, Maslov L N, Shavrin O I, Skvortsov A N 2016 Properties of Structural Steels with Nanoscale Substructure In: Anisimov K et al. (eds) Proceedings of the Scientific-Practical Conference "Research and Development - " *Springer Cham* 385-396

[21] Teterina I, Avdeeva A 2019 The determination of the structural parameters of mechanical engineering products by the finite element method *Bulletin of BSTU named after V G Shukhov* 2 149-155 doi: 10.12737/article_5c73fc382e33e6.05913951

Acknowledgements
This work was realized under the support of the President Scholarship; in the framework of the Program of flagship university development on the base of the Belgorod State Technological University named after V G Shukhov, using equipment of High Technology Center at BSTU named after V G Shukhov.