Design of a Striker Mechanism for a Sniper Rifle

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Abstract

In his work, the author designs a striker mechanism for a 20x102 mm calibre sniper rifle. The author conducts analyses of dynamic quantities of the striker mechanism and effects of mechanical strokes in the striker mechanism. An analysis of individual striker components load was carried out by the finite element method. The design was made by CAD system.

Key words: sniper rifle, striker mechanism, finite element method, critical force, hammer energy

1. Introduction

Nowadays, ground and special forces have been equipped with up-to-date large-calibre sniper rifles. Their high combat value lies in their capability of destroying targets of great importance while not having to deploy own forces, and thus avoiding casualties. Striker mechanism and its design have a considerable effect on the rifle qualities. In the present paper, I design a striker mechanism for a 20 x 102 mm calibre sniper rifle and analyse effects of mechanical strokes inside the striker mechanism.

2. Cartridge

The present striker mechanism is designed for a sniper rifle using 20 x 102 mm ammunition. Actuation energy of the cartridge cap and cartridge dimensions are essential in working out the design.

Fig. 2.1 20 x 102 mm cartridge

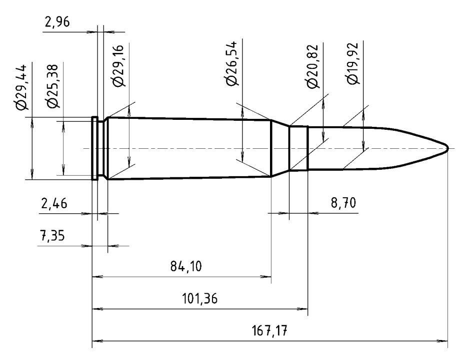


Fig. 2.2 Fundamental dimensions of 20 x 102 mm cartridge

3. Analysis of Design Options

3.1 Rifle concept

I have designed a striker mechanism for a repeater sniper rifle with a rotating retractable breech using 20 x 102 mm ammunition. Moreover, I have outlined the weapon frame together with the breech and its carrier since all the constituents of the striker mechanism are closely interrelated and interdependent.

3.2 Options to design a striker mechanism

Firing mechanisms serve to actuate the powder charge [6]. There are the following types of actuation:

- electric,
- mechanic,
- combined.

Since electric actuation reliability is not as high as mechanic actuation reliability, I have opted for the latter. Mechanic actuation is rendered by striker mechanisms. Striker mechanisms with their own striker springs are appropriate for a sniper rifle with a rotating breech. Striker mechanisms can be as follows:

- mechanisms with a rotating hammer,
- mechanisms with a parallel hammer (hammer can either be firmly fixed to the striker or they can be separate).

I have selected a striker mechanism with its own striker spring with a parallel hammer, which together with the firing pin constitute one entity – striker mechanism with a hammer (*Fig. 3.1*). I have chosen this option of design for its simplicity and reliability.

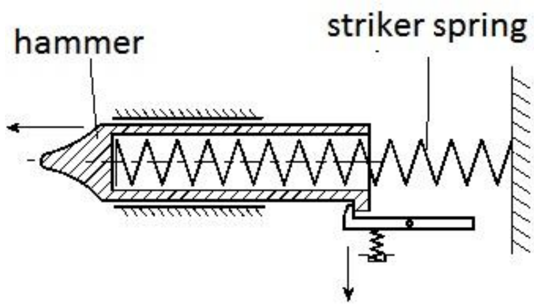


Fig. 3.1 Striker mechanism with a hammer

3.3 Options to design a trigger mechanism

Trigger mechanism is to hold securely the striker mechanism in stretched position, to release or stretch it [6]. Trigger mechanism in sniper rifles affects greatly the fire accuracy in terms of fire delay, which is mainly seen during the fire at moving targets [7]. Trigger mechanisms can perform the function of:

- pulling,
- releasing:
  - a) with a functional grip,
  - b) with a support.

I have selected the trigger release mechanism with a support because compared with the former, it does not affect the firing accuracy so much. Accuracy, however, is a must in sniper rifles. A release trigger mechanism with a support is an easy and reliable structure.

4. Striker Mechanism

4.1 Technical description

Striker mechanism is illustrated in *Fig. 4.1*. The striker is aligned with the breech body, representing the striker mechanism frame. The striker coil is being pressed between a notch located on the hammer and the insert screwed tightly to the breech.

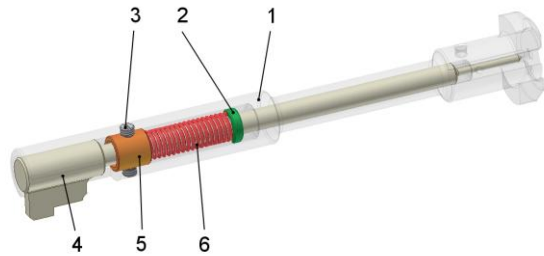


Fig. 4.1 Striker mechanism

1 – breech, 2 – stop, 3 – screw, 4 – hammer, 5 – insert, 6 – striker spring

After firing, the hammer moves owing to the force by the striker spring. First, the firing pin strikes the primer and after completing the course of 1.4 mm, the striker hits the stop, which is the insert located inside the breech. The stop located on the insert was chosen for operational reasons. The striker stop could also be placed on the breech split or the striker could directly stop in the breech front section, where the striker extends to the striker pin. Yet, the two options mentioned could cause operational problems if the bearing surfaces were polluted by oil and dust, which would change the path traveled by the striker after hitting the primer.

4.2 Calculation of the striker spring force

The ignition energy of the primer is fundamental to calculate the striker spring force. For I could not identify the exact value of the primer ignition energy used in the 20 x 102 mm cartridge, then following [5] and being aware of several ignition energy values of the smaller caliber ammunition primers, I have selected E_{ini} = 0,54 *J*.

- required hammer energy (taking into account the reliable actuation):

$$E_{ú} = 1{,}5.E_{ini} = 1{,}5.0{,}54 = 0{,}81J \tag{4.1}$$

- striker spring energy required for the primer actuation:

$$E_{BP} = E_{ú} = 0{,}81J \tag{4.2}$$

- striker spring mean force required for the primer actuation:

$$F_{STR} = \frac{E_{ú}}{l_h} = \frac{0{,}81}{0{,}006} = 135N \tag{4.3}$$

where l_h is the working stroke of the striker spring

- striker spring force while taking into account the action by the trigger mechanism:

I select $F_{BP} = 145N$

4.3 Design of the striker spring

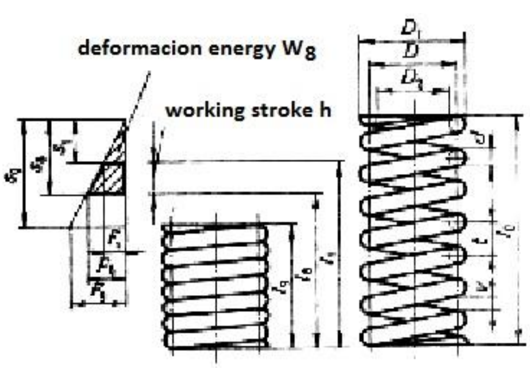


Fig. 4.2 Characteristics of a compression cylindrical spring

d - wire diameter *(mm)*
D - mean diameter of the spring *(mm)*
D_1 - outside diameter of the spring *(mm)*

D_2 - inside diameter of the spring *(mm)*
h - working stroke *(mm)*
t - spacing of active threads in a free state *(mm)*
v - allowance between active threads in a free state *(mm)*
s_x - deformation (compression) of the spring *(mm)*
l_x - length of the spring *(mm)*
F_x - working force exerted by the spring *(N)*

Indexes denoting the state of the spring:

0 - free state (spring is not loaded)
1 - pre-stressed state (the lowest work load)
8 - fully loaded state (the highest work load)
9 - boundary state (compression of the spring until touching the threads)

Selected parameters:

- type of the spring: cylindrical compressive spring
- spring material: 14 260
- allowed boundary stress in torsion: $\tau_{Dm} = 1100\ MPa$
- shear modulus: $G = 8{,}04.10^4\ MPa$
- mean diameter of the spring: $D = 15\ mm$
- wire diameter: $d = 2\ mm$
- number of stop threads: $n_z = 2$
- number of machined threads: $n_0 = 1$
- toughness of the spring: $c = 3\ N.mm^{-1}$

Calculation

- Force operating range of the spring:

$$F_P = c.l_h = 3.6 = 18N \qquad (4.4)$$

- winding ratio:

$$i = \frac{D}{d} = \frac{15}{2} = 7{,}5 \qquad (4.5)$$

- Correction factor of torsional stress:

$$K = \frac{i+0{,}2}{i-1} = \frac{7{,}5+0{,}2}{7{,}5-1} = 1{,}185 \qquad (4.6)$$

- initial spring force (the lowest workload):

$$F_1 = F_{BP} - \frac{F_P}{2} = 145 - \frac{18}{2} = 136N \qquad (4.7)$$

- spring tension at the lowest workload:

$$\tau_1 = \frac{8.F_1.D.K}{\pi.d^3} = \frac{8.136.15.1{,}185}{\pi.2^3} = 769{,}482\ MPa \qquad (4.8)$$

- final spring force (the highest workload):

$$F_8 = F_{BP} + \frac{F_P}{2} = 145 + \frac{18}{2} = 154N \qquad (4.9)$$

- spring stress at maximum workload:

$$\tau_8 = \frac{8.F_8.D.K}{\pi.d^3} = \frac{8.154.15.1{,}185}{\pi.2^3} = 871{,}326\ MPa \qquad (4.10)$$

- number of active threads:

$$n = \frac{G.d^4}{8.D^3.c} = \frac{8{,}04.10^4.2^4}{8.15^3.3} = 15{,}881 \approx 16 \qquad (4.11)$$

- total number of threads:

$$z = n + n_z = 16 + 2 = 18 \qquad (4.12)$$

- spring compression at the lowest workload:

$$s_1 = \frac{F_1}{c} = \frac{136}{3} = 45{,}333\ mm \qquad (4.13)$$

- spring compression at the highest workload:

$$s_8 = \frac{F_8}{c} = \frac{154}{3} = 51{,}333\ mm \qquad (4.14)$$

- spring force in a boundary state:

$$F_9 = \frac{\tau_{Dm}.\pi.d^3}{8.D.K} = \frac{1100.\pi.2^3}{8.15.1{,}185} = 194{,}416 N \qquad (4.15)$$

- spring length in a boundary state:

$$l_9 = (z + 1 - z_0).d = (18 + 1 - 1).2 = 36 mm \qquad (4.16)$$

- spring compression in a boundary state:

$$s_9 = \frac{F_9}{c} = \frac{194{,}416}{3} = 64{,}805\ mm \qquad (4.17)$$

- spring length in a free state:

$$l_0 = l_9 + s_9 = 36 + 64{,}805 = 100{,}805 mm \qquad (4.18)$$

- spring length in a pre-stressed state:

$$l_1 = l_0 - s_1 = 100{,}805 - 45{,}333 = 55{,}472 mm \qquad (4.19)$$

- length of a fully loaded spring:

$$l_8 = l_0 - s_8 = 100{,}805 - 51{,}333 = 49{,}472 mm \qquad (4.20)$$

- allowance between active threads in a free state:

$$v_0 = \frac{s_9}{n} = \frac{64{,}805}{16} = 4{,}05 mm \qquad (4.21)$$

- allowance between active threads at the maximum workload:

$$v_8 = \frac{s_9 - s_8}{n} = \frac{64{,}805 - 51{,}333}{16} = 0{,}842 mm \qquad (4.22)$$

- minimum allowance between active threads:

$$v_{\min} = \frac{d.i}{50} = \frac{2.7{,}5}{50} = 0{,}3 mm \qquad (4.23)$$

- condition check $v_8 \geq v_{min}$:

$0{,}842\ mm \geq 0{,}3\ mm$ - condition is met

it is not necessary to check the critical speed of the spring compression as manpower will be applied to compress the spring.

4.4 Check of the firing pin energy for a primer actuation

Weights: hammer m_U = 0,327 *kg*
stop m_Z = 0,006 *kg*
striker spring m_{BP} = 0,021 *kg*

Striker spring: $c = 3\ N.mm^{-1}$
$F_1 = 136\ N$
$F_8 = 154\ N$

Hammer course: $s = 6\ mm$

Calculation:

- mass of the hammer along with its moving elements (striking mass):

$$m_{UC} = m_U + m_Z + \frac{1}{3} \cdot m_{BP} = \\ = 0{,}327 + 0{,}006 + \frac{1}{3} \cdot 0{,}021 = 0{,}34\,\text{kg} \quad (4.24)$$

- acceleration of the striking mass:

$$a_{UC} = \frac{F_1 + F_8}{2.m_{UC}} = \frac{136 + 154}{2.0{,}34} = 426{,}471\, m.s^{-2} \quad (4.25)$$

- duration of the striking mass movement:

$$t_{UC} = \sqrt{\frac{2.s}{a_{UC}}} = \sqrt{\frac{2.0{,}006}{426{,}471}} = 0{,}0053\,s \quad (4.26)$$

- impact speed of the striking mass:

$$v_{UC} = a_{UC}.t_{UC} = 426{,}471.0{,}0053 = 2{,}26\,\text{m.s}^{-1} \quad (4.27)$$

- impact energy of the striking mass:

$$E_{UC} = \frac{1}{2} \cdot m_{UC} \cdot v_{UC}{}^2 = \frac{1}{2} \cdot 0{,}34 \cdot 2{,}26^2 = 0{,}868\,J \quad (4.28)$$

- required initiation (actuation) energy of the hammer and its associated elements:

$$E_{ú} = 0{,}81\,J$$
$$E_{UC} \geq 0{,}81\,J$$

4.5 Checking on the firing pin strut

The firing pin is tightly connected with the hammer positioned coaxially with the breech body while crossing its center. Upon actuation, the striker spring force ($F_1 = 136\ N$) is applied on the firing pin in the direction of the firing pin axis. The firing pin strikes the primer placed inside the shell casing and the primer gets deformed plastically. As this process is dynamic, the force exerted on the firing pin will probably be stronger than F_1. It is not easy to express F_1 exactly analytically as the primer gets plastically deformed in the course of the actuation.

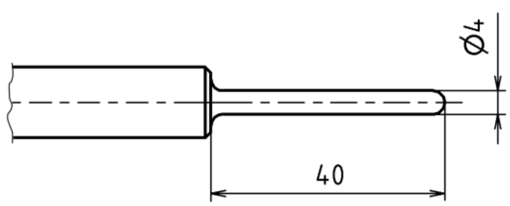


Fig. 4.3 Firing pin

Calculation of the critical force

- critical force F_{KR} is calculated according to the following formula:

$$F_{KR} = \frac{\pi^2.E.J}{l_0^2} \quad (4.29)$$

where:

E – material Young's modulus *(MPa)*,
J – moment of inertia of the cross- section *(mm^4)*,
l_0 – reduced length *(mm)*.

- moment of inertia of the cross-section:

$$J = \frac{\pi . d^4}{64} = \frac{\pi . 4^4}{64} = 12{,}566\, mm^4 \qquad (4.30)$$

d – diameter of the firing pin *(mm)*

- reduced length (the case when the rod is fixed in one end and pivoted in the other):

$$l_0 = \frac{l}{\sqrt{2}} = \frac{40}{\sqrt{2}} = 28{,}284\, mm \qquad (4.31)$$

l –length of the firing pin *(mm)*

- critical force will be:

$$F_{KR} = \frac{\pi^2 . E . J}{l_0^2} = \frac{\pi^2 . 2{,}1.10^5 . 12{,}566}{28{,}284^2} = 32556{,}255\, N \qquad (4.32)$$

$$F_1 < F_{KR}$$
$$136\, N < 32556{,}255\, N$$

The force F_1 exerted on the firing pin is below the critical force value, is much lower than the critical force value, thus pressure will act more on the firing pin.

4.6 Design of the firing pin fuse

4.6.1 Technical description of the firing pin fuse

The firing pin fuse secures the weapon against unintended firing. Its main part is the cylinder with a groove. Turning the cylinder around its axis provides for locking and unlocking of the firing pin.

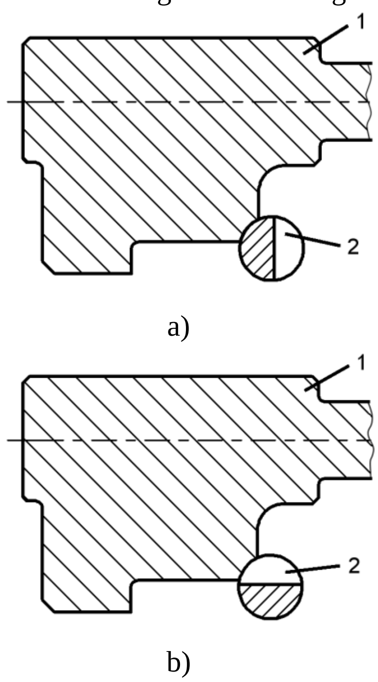
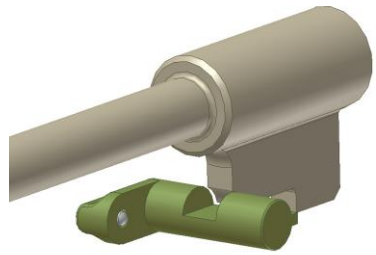


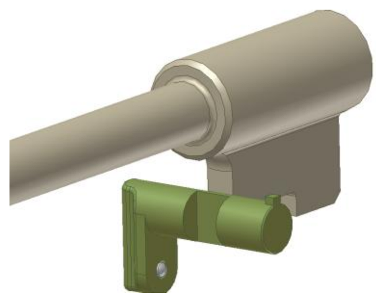
Fig. 4.4 Principle of operation of the firing pin fuse

1 – hammer, 2 – hammer fuse

The principle of operation of the hammer is illustrated in *Fig. 4.4*. *Fig. 4.4a* shows the hammer fuse positioned to block the hammer movement. *Fig. 4.4b* shows the hammer fuse twisted by 90˚ to allow movement. The position of the hammer and hammer fuse is shown in *Fig. 4.5*.



a)



b)

Fig. 4.5 Hammer fuse and hammer

There is an opening in the control lever of the hammer fuse *(Fig. 4.6)* in which the pressure spring and pressure rod are inserted. These components keep the striker fuse in the selected position. The pressure rod is pressed by the pressure spring towards the external part of the breech carrier (integral part of the weapon frame). There are two holes in the breech carrier to fit the pressure rod head. Thus, the hammer fuse is secured against spontaneous turns.

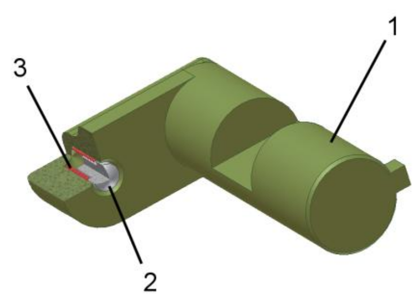


Fig. 4.6 Partial cross-section of the hammer fuse

1 – hammer fuse body, 2 – pressure rod, 3 – pressure spring

4.6.2 Design of the hammer fuse spring

Selected parameters:

- type of the spring: cylindrical compressive spring
- spring material: 14 260
- allowed limit stress in torsion: $\tau_{Dm} = 1100\ MPa$
- shear modulus: $G = 8{,}04.10^4\ MPa$
- mean average of the spring: $D = 2{,}4\ mm$
- wire diameter: $d = 0{,}4\ mm$
- number of stop threads: $n_z = 2$
- number of machined threads: $n_0 = 1$
- toughness of the spring: $c = 3\ N.mm^{-1}$

Calculation:

- winding ratio:

$$i = \frac{D}{d} = \frac{2,4}{0,4} = 6 \qquad (4.33)$$

- correction factor of torsional stress:

$$K = \frac{i + 0,2}{i - 1} = \frac{6 + 0,2}{6 - 1} = 1,24 \qquad (4.34)$$

- spring force in a boundary state:

$$F_9 = \frac{\tau_{Dm}.\pi.d^3}{8.D.K} = \frac{1100.\pi.0,4^3}{8.2,4.1,24} = 9,29\ N \qquad (4.35)$$

- number of active threads:

$$n = \frac{G.d^4}{8.D^3.c} = \frac{8,04.10^4.0,4^4}{8.2,4^3.3} = 6,2 \qquad (4.36)$$

- spring compression in a boundary state:

$$s_9 = \frac{F_9}{c} = \frac{9,29}{3} = 3,097\ mm \qquad (4.37)$$

- total number of threads:

$$z = n + n_z = 6,2 + 2 = 8,2 \qquad (4.38)$$

- spring length in a boundary state:

$$l_9 = (z + 1 - z_0).d = (8,2 + 1 - 1).0,4 = 3,28\ mm \qquad (4.39)$$

- spring length in a free state:

$$l_0 = l_9 + s_9 = 3,28 + 3,097 = 6,377\ mm \qquad (4.40)$$

4.7 Analysis of mechanical strokes in the striker mechanism

Given the proposed structure of the striker mechanism, analyzing the effects of mechanic strokes inside the striker mechanism, does not mean to examine dynamics between the components engaged, but to examine contact stresses in the place of stroke.

Upon starting the striker mechanism, the hammer is released and accelerated by the striker spring force. First, the hammer firing pin hits the primer and the primer undergoes plastic deformation. Next, the hammer travels 1,4 mm and hits the stop which is the insert located in the weapon breech. This brings about elastic deformation of the hammer and insert. When analyzing stresses generated during these two hits in the striker mechanism, we should base our assumption on the fact that some kinetic energy is converted into deformation work used for plastic deformation of the primer and some energy will be converted to stress energy of the hammer and insert.

It would be very complicated to provide an accurate analytical analysis, so I decided to examine the effects of mechanical hits in the striker mechanism by MKP technique addressed in the following sub-section.

4.8 Load analysis of the striker mechanism components by means of MKP

4.8.1 Static analysis of the hammer load in stressed state

To analyze the effects of the hammer load in a stressed state, I applied MKP technique. I applied *ANSYS* software to conduct all MKP analyses. In a stressed state, the striker spring force is exerted through the stop on the hammer. The stop is located in the breech and gripped to a grip. Boundary conditions are determined by the load and the way the hammer is mounted. Results of the analysis are illustrated in *Fig. 4.7* and *Fig. 4.8*.

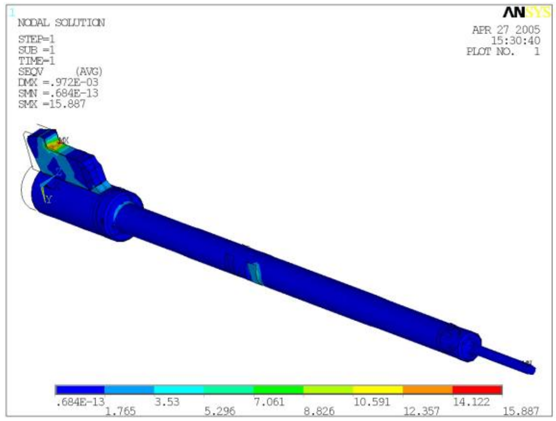


Fig. 4.7 Hammer stresses in a stressed state of the rifle

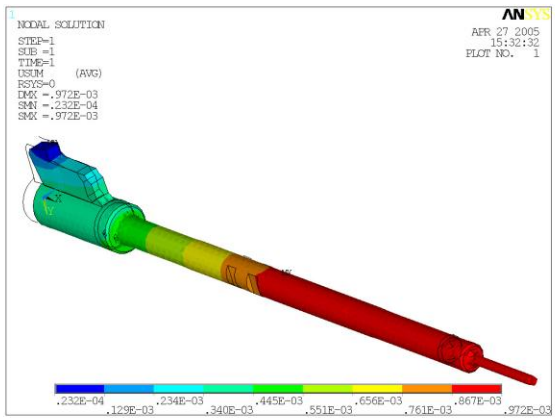


Fig. 4.8 Hammer deformations in a stressed state of the rifle

4.8.2 Analysis of mechanical stroke effects in the course of striker mechanism operation

To analyze the effects of mechanical strokes in the striker mechanism, I applied MKP and *ANSYS* software. To address dynamic processes, *ANSYS* software has a special *LS-DYNA* program. However, it is very complicated, time-consuming and excellent knowledge of mechanics and experience in working with *LS-DYNA* are needed to make an accurate model for the purposes of a dynamic analysis. That is why I transformed the dynamic task into the static task by increasing the values of the forces exerted. The analysis of the effects of strokes in the striker mechanism is divided into two steps. First, I addressed hits of the firing pin and the primer. Then, I addressed hits of the hammer and insert because this is the sequence in which the striker mechanism works. Results of the analysis are shown in *Fig. 4.9, Fig. 4.10, Fig. 4.11* and *Fig. 4.12.* In the course of the striker mechanism operation, the highest stress generated on the primer is approximately of 194,6 MPa. Results obtained by means of MKP technique cannot be considered absolutely accurate as they depend on many factors (for instance crosslinking density, selection of the element type, definition of boundary conditions, etc.).

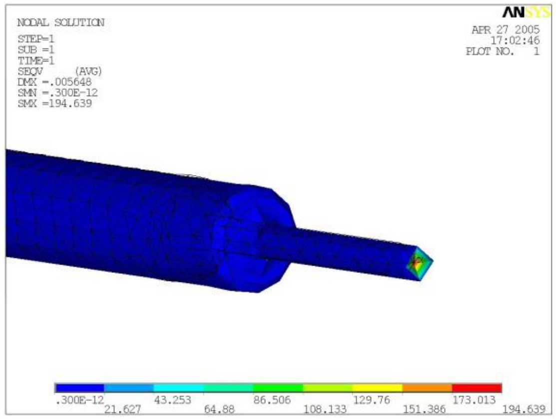


Fig. 4.9 Stresses inside the firing pin upon hitting the primer

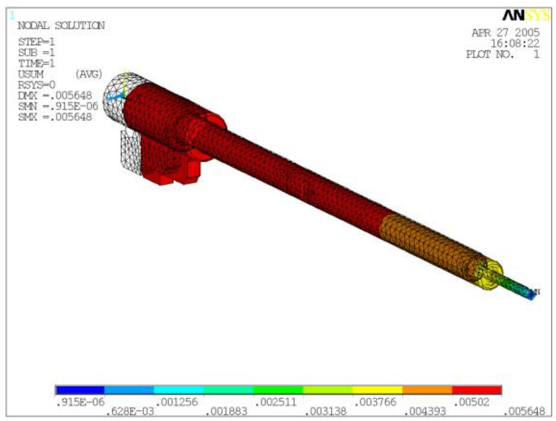


Fig. 4.10 Striker deformations upon hitting the primer

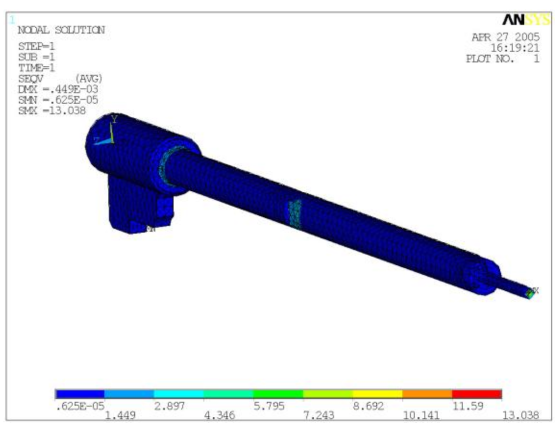


Fig. 4.11 Stresses inside the striker upon hitting the insert

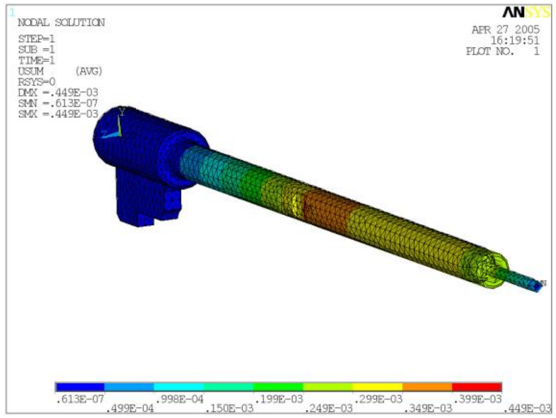


Fig. 4.12 Striker deformations upon hitting the insert

5. Conclusion

My work resulted in designing a striker mechanism for a 20 x 102 mm caliber sniper rifle. Simplicity of the design option and its applicability under real conditions were of utmost importance. When analyzing the effects of mechanical strokes in the striker mechanism, I employed the finite element method and used CAD Autodesk Inventor in the designing phase of my work. The use of cutting-edge technologies in the phase of design substantially increases the work efficiency.

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