Interaction between projectile driving band and forcing cone of weapon barrel

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Abstract: - The paper deals with engraving process of the projectile into the barrel forcing cone in course of ramming. Static model of interaction problem (contact problem) between the projectile driving band and forcing cone are established and solved by using simulation software ANSYS Workbench ver.13.0 with Finite Element Method. Research results as deformations and stress fields, and the reaction forces etc. are shown in form of graphs and charts. Material model of the problem is performed with copper alloy (Cu)material, containing 99.99% Cu. Calculations and simulations are performed on two types of weapons, self-propelled howitzer 152 mm mod 77 (SPH 77) with high explosive projectile (HE), and 125 mm T72 tank cannon with shaped charge penetration projectile (HEAT). Reaction forces of the interaction process have been verified by experiments performing on two above-mentioned system of weapon. The calculation results are compatible with the experimental results with reasonable deviations from 5% at SPH 77 and 7% at T72 tank cannon. Research results are also background to upgrade the state of the art knowledge to the Czech Defense Standard (COS) regarding to ramming device of artillery weapons and tank cannons.

Key-Words: - Driving band, Forcing cone, Loading simulation, Cartridge ramming, Contact problem, Interaction problem, Finite element method, Projectile ramming

1 Introduction
As it is known, combat efficiency of heavy guns, especially of self-propelled howitzers and tanks depend on many tactical and technical characteristics. Two of them are the rapidity of fire and the safety of projectile ramming in unstable motion of combat vehicles on the battlefield. High rapidity of fire gives the self-propelled howitzers and tanks abilities to increase chances to destroy opposing forces weapons in the battlefield, see [1], and [11].

After projectile feeding, see [2], the next operation is projectile ramming. The ramming is the operation substantially limiting the rapidity of fire. Very important factor is the safety of projectile ramming, when vehicles move in the battlefield with high speed and in bad road conditions or the towed howitzer fires with the higher elevation angle. The interface configuration between barrel and projectile shall ensure that no projectile falls back out of its seating at any angle of elevation, see [3], [4]. To reduce the risk of fall back to a minimum, the quality of ramming device is established in course of technical inspections according to the standard [7] where the force necessary to extract projectile from the barrel is determined as one of the main procedures. State of the projectile in the barrel at the end of ramming is depicted in Fig. 1.

![Fig.1 Projectile position in the barrel after ramming](image)

Fig.1 Projectile position in the barrel after ramming 1- forcing cone; 2- driving band; 3- projectile; 4- barrel; 5- guiding part; $F_{Ram}$- ramming force

Firstly, the ramming force $F_{Ram}$ ensures holding the projectile in the cartridge chamber when the driving band is engraved into the barrel forcing cone to prevent fall back risk of the projectile from the cartridge chamber. Secondly, ramming process creates deformation field between driving band and forcing cone to seal area between projectile and guiding part of barrel, to ensure that propellant gases do not overcome this space when firing.
Thirdly, accurate position of the projectile in cartridge chamber ensures to optimize movement of projectile in barrel when firing to decrease vibrations of projectile and the barrel wear.

However, the determination of ramming force is very difficult by the standard calculation because it is not only the complicated nonlinear plastic-elastic problem but it also depends on many factors whose identification is uneasy. The standard [7] deals with the determination of the force, which is necessary to hold the projectile in the barrel after ramming. Nevertheless, this standard only defines the opposite force determined in course of the projectile extraction during technical inspections.

The force calculation by means of the simple model was published in [6] but the results have not been proved by measuring and not satisfy requirements in researching ramming process.

The ramming velocities at the end of the operation obtained from measuring in year 2009 achieved 0.77 m/s. The engraving force \( F_{ENG} \) is from 17 to 19 kN for 152 mm HE projectile according to the experiments published in [4], and [5].

In the following section, we will simulate ramming process by using software ANSYS.

### 2 Overview of finite element method

Let us assume a static structural problem of plastic-elastic theory consists of 15 unknowns, including movements \( u, v, w \), deformations \( \varepsilon_x, \varepsilon_y, \varepsilon_z, \gamma_{xy}, \gamma_{yz}, \gamma_{zx} \), stress of considered element in the model \( \sigma_x, \sigma_y, \sigma_z, \tau_{xy}, \tau_{yz}, \tau_{zx} \), see [8]. Let us consider the following simple problem in Fig. 2:

\[
\frac{d^2 u}{dx^2} + \frac{\rho g}{E} = 0; \quad u(0) = 0; \quad \frac{du}{dx} \bigg|_{x=L} = 0. \quad (1)
\]

In order to solve the problem by using of the finite element method the Lagrange’s variation principle will be considered [8], and [9]. The principle is: between all of movement functions which conserve continuation of a body and satisfy its geometry boundary conditions so total potential energy of the body is stationary value. It is possible to demonstrate that this total stationary energy exists and is a minimal function. The principle is presented as follows:

\[
\Pi = W - P \rightarrow \text{min}, \quad (2)
\]

where \( \Pi \) is total potential energy of the body (Lagrange’s potential), \( P \) is tenseness energy of body, \( W \) is potential of exterior load.

Task of the finite element method is solved in [8] whose roots \( u(x) \) is selected in form:

\[
u(x) = \sum_{i=1}^{N} a_i w_i(x),
\]

where the \( a_i \) weight coefficients will be found so they satisfy minimum conditions of expression (2), which is also a function of \( u(x) \).

Fig.2 presents principle of finite element method. The original geometry is divided on simple regions, called "elements" which are connected continuously by "nodes" which locate on the boundary between elements. Using selected basic functions at nodes, for example, \( N_i(x) \) and \( N_j(x) \), will be established unknown function, for example, deformation function \( u(x) \) at any element “e”.

**Fig.2 Scheme of simple problem**

The beam in Fig.2 is loaded by individual mass at the orientation of the beam axis. Its properties are section \( S \), length \( L \), Young’s modulus \( E \), beam material density \( \rho \), and gravity acceleration \( g \).

We get the simple equation from the equation system of plastic-elastic theory applied on above problem and after expanding and considering the boundary conditions:

\[
\begin{align*}
\sigma & = \frac{E}{1+\nu} \varepsilon, \\
\end{align*}
\]

**Fig.3 Approximated root \( u(x) \) of any element “e”**

Fig. 3 shows schema for establishing approx. root \( u(x) \) of any element “e”, basic functions can be selected so we have best approx. root with reasonable deviation. We receive function \( u(x) \) (deformation) of element “e” as follows:
\[ u(x) = N_i(x)u_i + N_j(x)u_j. \]  

Similarly, we can establish the approximated deformation function for every element of the beam using the basic function \( N \) (it also called shape function) and deformation \( u \) (unknown weighted coefficients) at nodes. After that the total deformation of the beam is found as sum of the deformation functions of every element of the beam. For example, the deformations function of the beam which consists of three elements:

\[ u(x) = N_1^1(x)u_1 + N_2(x)u_2 + N_3^2(x)u_3 + N_4(x)u_4 \]  

The total potential energy of any element “e” is calculated as \( \Pi^e \) and the total potential energy of the beam is sum of all elements energy.

\[ \Pi = \Pi(u(x)) = \sum \Pi^e \]  

By substituting \( u(x) \) in (4) into (5) we get the total potential energy as follows:

\[ \Pi(u_1, u_2, u_3, u_4) = \Pi^1(u_1, u_2) + \Pi^2(u_2, u_3) + \Pi^3(u_3, u_4) \]  

Function \( u(x) \) in (4) will be root of the problem presented in (1) if it satisfies the minimal condition of expression in (2). The minimal condition is presented as follows:

\[ \frac{\partial \Pi(u_1, u_2, u_3, u_4)}{\partial u_i} = 0; \quad i = 1, 2, 3, 4 \]  

By solving of the equation system (7) we get roots which are deformation at every node, other root such as deformation velocities, stress, reaction force etc. will be induced from this root.

### 3 Interaction between projectile driving band and forcing cone

The geometry model of the interaction problem between projectile driving band and forcing cone in form of 2D model has been worked out using of an axisymmetric type of element for barrel, projectile, and driving band. This selection has advantages that number of elements and degrees of freedom will be small, and it is quite easy to solve this mathematical model by personal computers.

Calculation of ramming process is performed for 152 mm rifled barrel with HE projectile in SPH 77 and for 125 mm smooth bore barrel in T72 with HEAT projectile. Driving band has been assumed copper alloy 99.99% Cu, see [9], [10], [11], [12], and [13].

Finite element model (FEM- mesh) of the problem is shown in Fig. 4. In Fig. 4.a there is the model, especially meshing status of the model consisting of the barrel, projectile, and driving band. Movement orientation of the projectile is from the left to the right when it is rammed and from right to left when the projectile is extracted from the barrels during technical inspections and tests see Fig.4.a, and Fig.4b.

![Fig.4a FEM of SPH 77](image)

![Fig.4b FEM of T72](image)

![Fig.4c Detail of SPH 77](image)

![Fig.4d Detail of T72](image)

Details in Fig.4c and Fig.4d are: 1 - forcing cone; 2 - driving band; 3 - projectile.

### 4 Simulation results

The simulation results are presented onward. It is necessary to note that the calculations are made in static model. The dynamics of the problem will be follow next time.

Firstly, the equivalent stress (von Mises theory) for both studied cases are depicted in Fig. 5a (SPH 77) and in Fig. 5b (T72). The maximal equivalent stress during ramming and extraction processes are given in Fig. 6a and Fig. 6b. These stresses are calculated automatically by ANSYS software from the pressure fields of the contact among the projectile, its driving band and the barrel forcing cone. In the engraving process of driving band into forcing cone, the stress field
changes continuously with time and the position between driving band and forcing cone.

![Fig.5a Equivalent stress in case of SPH 77](image)

Due to the longer forcing cone with smaller angle with respect to the barrel axis the stress is greater in T72 barrel than in the SPH 77 barrel.

![Fig.5a Equivalent stress in case of T72](image)

The most deformed part of these three bodies is the driving band. Its plastic deformations are represented in Fig. 7a and Fig. 7b.

![Fig.6a Maximal equivalent stress in SPH 77 system](image)

![Fig.6b Maximal equivalent stress in T72 system](image)

![Fig.7a Equivalent plastic deformation of SPH 77 projectile driving band](image)

![Fig.7b Equivalent plastic deformation of 125 mm projectile driving band](image)
Reaction forces during of engraving and extracting processes with copper alloy driving band are presented in Fig. 8a and Fig. 8b.

Characteristics of projectile ramming process into the forcing cone and projectile extraction process from the forcing cone with different materials are presented in Table 1. The extraction forces have been calculated with deviations from 5% at SPH77 and 7% at T72 tank cannon, see [4], [5].

Table 1 Characteristics of ramming and extracting processes SPH 77 barrel and 125 mm T72 barrel

<table>
<thead>
<tr>
<th>Driving band material</th>
<th>( \rho )</th>
<th>( E )</th>
<th>( \nu )</th>
<th>( \sigma )</th>
<th>( \tau )</th>
</tr>
</thead>
<tbody>
<tr>
<td>Cu</td>
<td>8940</td>
<td>1.12 \times 10^{11}</td>
<td>0.35</td>
<td>70 \times 10^{6}</td>
<td>1 \times 10^{6}</td>
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<table>
<thead>
<tr>
<th>Driving band material</th>
<th>( D_1 )</th>
<th>( D_2 )</th>
<th>( f )</th>
</tr>
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<tbody>
<tr>
<td>SPH 77</td>
<td>T72</td>
<td>SPH 77</td>
<td>T72</td>
</tr>
<tr>
<td>Cu</td>
<td>155.6</td>
<td>125.2</td>
<td>155.9</td>
</tr>
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<table>
<thead>
<tr>
<th>Driving band material</th>
<th>( \sigma_N )</th>
<th>( \sigma_E )</th>
</tr>
</thead>
<tbody>
<tr>
<td>SPH 77</td>
<td>T72</td>
<td>T72</td>
</tr>
<tr>
<td>Cu</td>
<td>2.31 \times 10^{8}</td>
<td>4.19 \times 10^{8}</td>
</tr>
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</table>

<table>
<thead>
<tr>
<th>Driving band material</th>
<th>( \varepsilon_{Dx} )</th>
<th>( \varepsilon_E )</th>
</tr>
</thead>
<tbody>
<tr>
<td>SPH 77</td>
<td>T72</td>
<td>T72</td>
</tr>
<tr>
<td>Cu</td>
<td>4.2 \times 10^{-4}</td>
<td>4.72 \times 10^{-4}</td>
</tr>
</tbody>
</table>

In the Table 1 there are the characteristics:
- \( \rho \) - density (kg.m^{-3});
- \( E \) - Young’s modulus (Pa),
- \( \nu \) - Poisson’s ratio;
- \( \sigma \) - Yield strength (Pa),
- \( \tau \) - Tangent modulus (Pa);
- \( f \) - Friction coefficient,
- \( D_1 \) - Diameter of forcing cone end point (mm),
- \( D_2 \) - Diameter of driving band (mm),
- \( \sigma_N \), \( \sigma_E \) - Max. normal and equivalent stress (Pa),
- \( \varepsilon_{Dx} \) - Max. directional deformation at \( x \) axis (m),
- \( \varepsilon_E \) - Max. equivalent plastic strain (mm/mm),
- \( F_{Ram} \), \( F_{Ext} \) - Max. ramming and extraction force (N).

5 Evaluation and conclusion

Results of the ramming and extraction projectiles problem from artillery barrels are very variable such as are the equivalent stress field, directional stress field, the normal stress field, the directional deformation field, the equivalent plastic strain of driving band, the reaction force of the ramming process and the extracting process as well. These results depend on input data remarkably, such as dimensions of forcing cone and driving band, physical-mechanical characteristics of materials.

The calculations show that in case when dimensions of forcing cone are minimal and dimensions of the driving band are maximal the ramming and extraction forces can reach to 81 kN. It is approximately 4.5 times bigger than in case when dimensions of the forcing cone and the driving band belong to design tolerances in the technical drawings. With these cases, calculating results are compatible with experiment results in [4], [5]. On the other hand, ramming and extraction forces depend on wearing degree of forcing cone diameter remarkably (at point between forcing cone and leading part). Therefore the special measuring arrangement has been designed, see Fig. 9.
Fig. 9 Measurement device of extraction force
The example of the location on the weapon is represented in Fig. 10.

Fig. 9 Measurement device placed on the weapon
The measuring device enables to determine the force/displacement and force/time history during extractions of the projectiles from the barrels. In the future, it will be necessary to measure some characteristics of ramming and extracting process, such as are ramming force, stress and deformation of forcing cone and driving band.

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References: