The Influence of the Temperature Change on the Force in the Hydraulic Brake of the Artillery System

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Abstract: Hydraulic brake is an important part of the artillery system because reliable amortization of energy of the recoil parts depends on its functioning. Therefore, it is important to estimate time-dependent pressure field inside the brake which depends on many factors, such as geometry of the brake parts, velocity of recoil parts and physical characteristics of the oil that is used in the brake. During recoil and counterrecoil process of the recoil parts of the artillery system, heating of the oil and the other parts of the hydraulic brake occurs. In this paper the influence of the temperature change of the hydraulic brake parts on the recoil force is analyzed. Since the hydraulic brake assembly is axisymmetric, the problem is considered to be two-dimensional. Differential equations that describe flow field in the brake are solved numerically using CFD software. A comprehensive comparison between numerical and experimental results has been performed and good agreement is observed.

Keywords: Artillery system, Hydraulic brake, Temperature change, Mathematical model, Numerical simulation.

1. Introduction

During the firing process, high pressure is developed in the gun barrel [1] due to the combustion of gunpowder and cause the creation of high-intensity forces in a short period of time [2-3]. Today, most of the battle platforms require fire power. High fire power produces the large recoil force that is transmitted to the weapon system. Attention should be paid to the maximum permissible recoil length to avoid possible damage on the weapon system. These two opposite conditions can be provided with adequate recoil system design as a hydraulic brake and recuperator. The temperature interval of the weapon system usage is from $-30°\mathrm{C}$to $+50°\mathrm{C}$, and the physical characteristics of the oil vary. The numerical analysis [4-7] aims to get threshold values of the force intensity which appears in the hydraulic brake, velocity and recoil length at the temperature interval limits. The movement of the piston in the cylinder during recoil causes the fluid heating [8], which must be taken into account when shooting at the upper extreme temperature of the gun and also during the design process of the replenisher. The experiment [10] was carried out by placing strain gauges on the outer surface of the hydraulic brake piston rod, which made it possible to measure the stress condition. By measuring the stress condition, it was determined the pressure which acts on the piston. All phenomena which can not be measured by the experimental method will be comprehensively obtained by numerical method. Since the values of the measured quantities are consistent with the results of the numerical simulation it was concluded that the other phenomena obtained by simulation correspond to realistic conditions. In this paper, a comparative analysis of the results obtained by the experimental method and numerical simulation was performed.

2. Working principle and main parts of the hydraulic brake

The hydraulic brake, as a part of the recoil system, performs the functions of a complex process.
According to its purpose and the complexity of the processes that take place, a mathematical model was developed as well as numerical analysis. The hydraulic brake provides resistances during recoil and counter-recoil process. In the first phase, the recoil process takes place, after the recoil process is completed, the second phase is the counter-recoil process. The main parts of the hydraulic brake are a cylinder, piston rod,

throttling bar, replenisher and valve which are shown in figure 1. The characteristic volumes and the channels are also indicated in figure 1, for a better explanation of fluid flow in the cylinder. When the first phase occurs (recoil), the piston rod starts to move which provide fluid flow from volume 1 to volume 2 trough the channels 1. Flow area which is located between volume 1 and volume 2 is defined by the gap size between the throttling ring and the throttling bar. The throttling bar cross-section area is variable along the axis of symmetry which causes the change of the flow area that depends on recoil distance during movement. One part of fluid from channels 1 continues to flow through the gap between throttling bar and throttling ring and the other part of the fluid also continue flow through the channels 2 filling the volume 3. When the piston rod starts to move in the opposite direction (counter-recoil), the volume 3 decreases, thus providing the motion of the fluid from the volume 3 into the volume 2 through the grooves on the inside surface of the throttling bar. The valve which is located on the rear part of the throttling bar is closed, forced by the spring and pressure inside the volume 3. During the counter-recoil process of the piston rod into the starting position, fluid was forced to flow from the volume 2 into volume 1 by the channels 1.

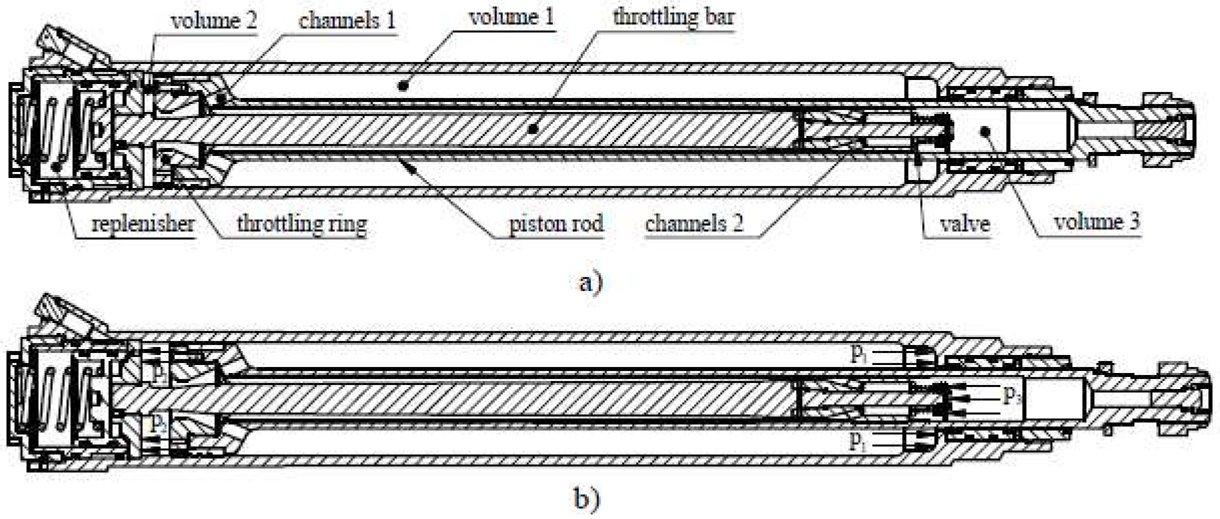


Figure 1. Hydraulic brake: a) main parts of the hydraulic brake
b) pressures in different volumes of the hydraulic brake

3. Analytical model

Differential equation of the piston movement can be written in the next form:

$$m_r \frac{dV}{dt} = F_p - F_{mb} - F_r - F_{hb} + m_r g sin\varphi - F_{fr} \tag{1}$$

where m_r and V denotes mass and velocity of the recoil parts, respectively; F_p-gunpowder pressure force;F_{mb}-muzzle brake force; F_r-recuperator force; F_{hb}-hydraulic brake force; φ-elevation angle; F_{fr}-friction force between the gun barrel and the cradle, the pistons of the hydraulic brake and the recuperator and the rubber seal, etc;

In figures 2, 3 and 4 are showncharacteristic curves of the combustion products pressure force, recuperator force and the muzzle brake force change during time, respectively.

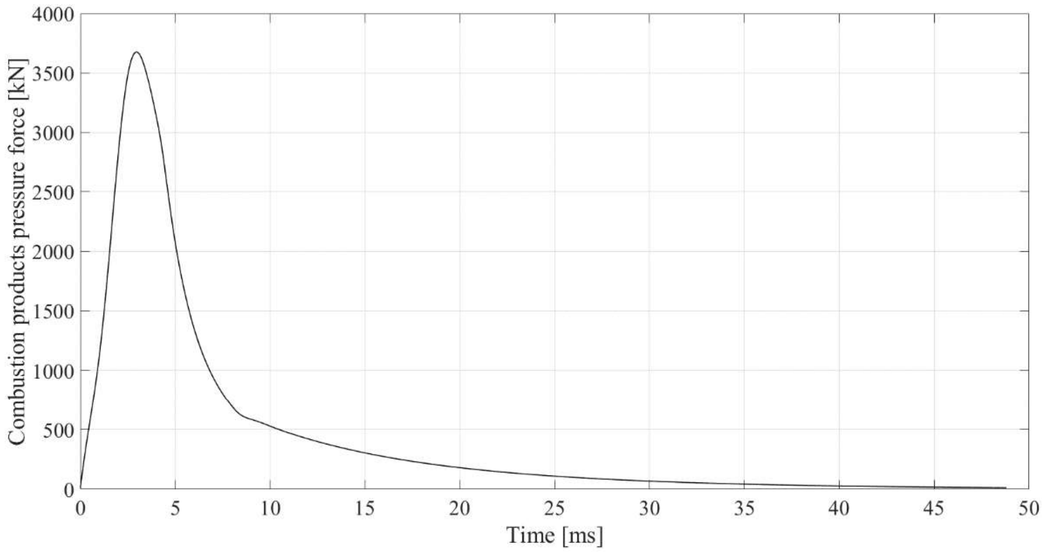


Figure 2. Combustion products pressure force

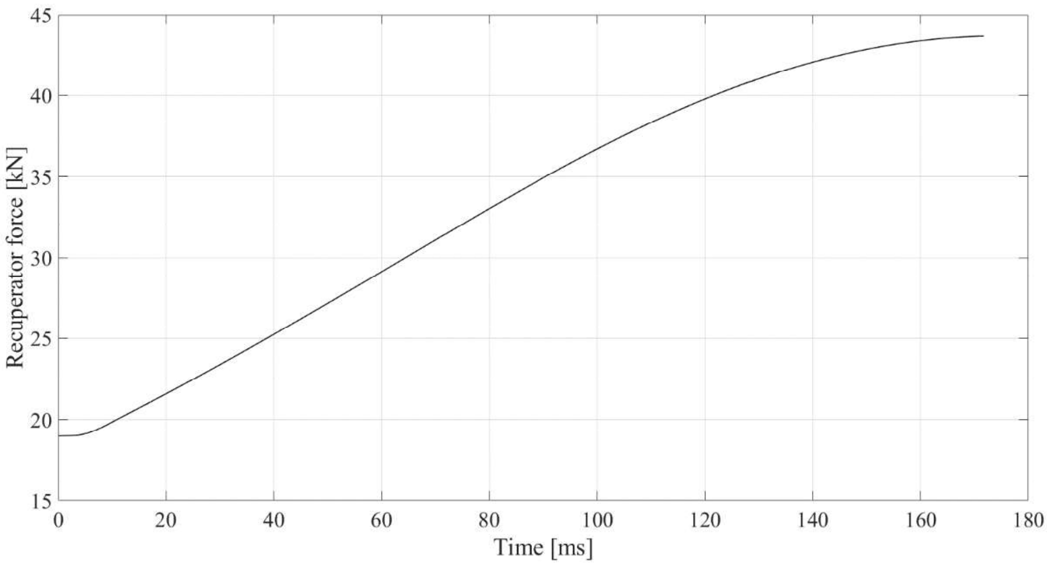


Figure 3. Recuperator force

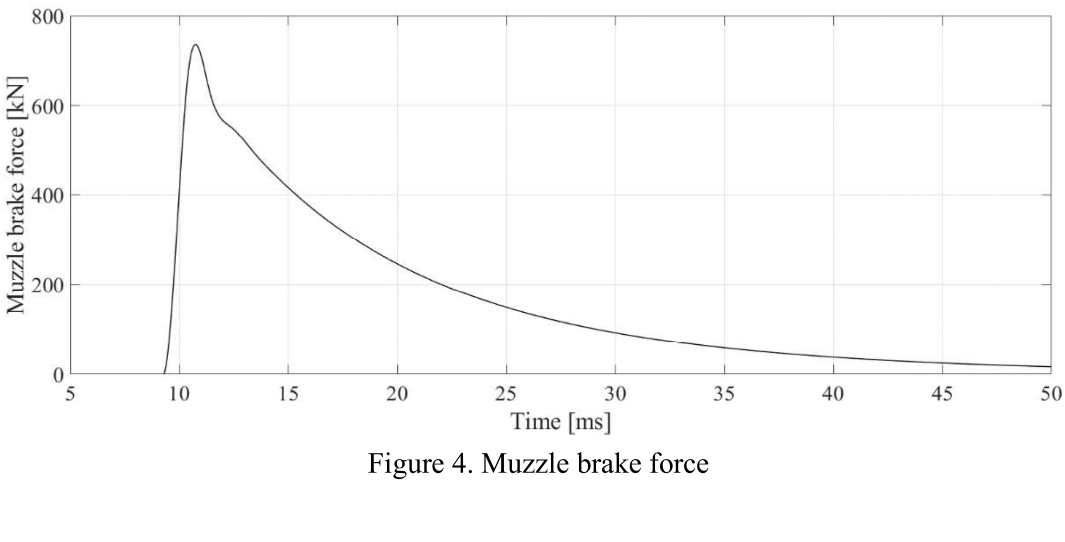


Figure 4. Muzzle brake force

Hydraulic brake force depends on the pressure values in the volumes 1, 2 and 3 according to figure 1.a:

$$F_{hb} = \int_{A_1} p_1 \vec{n} dA - \int_{A_2} p_2 \vec{n} dA - \int_{A_3} p_3 \vec{n} dA \quad (2)$$

where p_1, p_2 and p_3 represent the pressures in the volumes 1, 2 and 3 (figure 1.a) and A_1, A_2 and A_3 are the surface areas on which pressures acts, respectively (figure 1.b).

During the recoil process, pressures p_1 and p_3 can be determined using Bernoulli's and continuity equation [2]:

$$p_1 = \frac{\rho V^2}{2} \frac{(A_1 - A_3)^2}{A_{12}^2} \frac{\alpha_{12}}{\varepsilon_{12}^2 \beta_{12}^3} \quad (3)$$

$$p_3 = \frac{\rho V^2}{2} \left[\frac{(A_1 - A_3)^2}{A_{12}^2} \frac{\alpha_{12}}{\varepsilon_{12}^2 \beta_{12}^3} - \frac{A_3^2}{A_{13}^2} \frac{\alpha_{13}}{\varepsilon_{13}^2 \beta_{13}^3} \right] \quad (4)$$

whereρ denotes density of the fluid; A_{12}-surface area between volumes 1 and 2; A_{13}-surface area at the entrance of the volume 3; α,β-energy and momentum correction factor, respectively; ε-discharge coefficent;

Pressure p_2 can be neglected because during the piston movement out of the cylinder, cavitation occurs in the volume 2.

During recoil and counterrecoil process, heating of the some parts of the hydraulic brake ocurs. This is caused by the friction between metal parts and rubber seals, viscous forces in the oil and by dynamic pressures in some parts of the brake. The temperature rise after one firing cycle can be expressed as [2]:

$$\Delta T = \frac{\Delta Q_{hb}}{c_m m_m + c_{oil} m_{oil}} \quad (5)$$

where ΔQ_{hb} denote heat generated during one firing cycle; c_m and m_m-specific heat and mass of the metal parts of the hydraulic brake, respectively; c_{oil} and m_{oil}-specific heat and mass of the hydraulic oil, respectively; Between two firings, hydraulic brake is cooling due to radiation and natural convection. If the gun fire N projectile in $\Delta\tau$ seconds, the temperature of the hydraulic brake can be calculated using next expresion [2]:

$$T_N = T_s + (T_0 - T_s) \cdot exp\left(\frac{-\alpha_i A_o}{C_{hb}} N\Delta\tau\right) + \Delta T \sum_{i=1}^{N} exp\left(\frac{-\alpha_i A_o}{C_{hb}} N\Delta\tau\right) \quad (6)$$

whereT_N denotes the temperature of the hydraulic brake (metal parts and oil) after N firings during $\Delta\tau$ time period, T_s-temperature of the surrounding air and α_i is combined radiation and covection heat transfer coefficient [2]:

$$\alpha_i = \varepsilon\sigma(T + T_s)\left(T^2 + T_s{}^2\right) + \frac{\lambda}{d} C(G_r \cdot P_r)^n \quad (7)$$

whereε denotes emissivity coefficient, σ-Stephen-Boltzmann constant, λ-thermal conductivity coefficient of air, d-diameter of the hydraulic brake cylinder, G_r, P_r-Grashof and Prandtl number, C, n-constants that depends on geometry and flow regime

4. Numerical model

Beside experimental analysis, the numerical simulation was performed to investigate pressure change in the selected period. In the numerical model, it was used the multi-phase model which permits cavitation. For viscous model is setup model with standard wall functions. The fluid material inside the hydraulic brake is STEOL where all needed physical properties are given in Table 1 [3]. Other main parts of the hydraulic brake were made of steel. The important note is that the complete process during the numerical simulation was performed into the closed system without special requirements for boundary conditions. Based on mass flow through the flow areas by the channels 1 and 2, it was modified flow space into hydraulic brake into a 2d model based on the 3d model. The gap area between the throttling ring and the throttling bar was chosen as the referent flow area in the initial position. After determination of referent value, for flow properties calculation in 2D, it was calculated third referent dimension used by FLUENT CFD software. As mentioned previously, it was used to appropriately determine all other dimensions of 2D model hydraulic brake in aim to be the same mass flow as a 3D model. Domain for numerical analysis is shown in figure 5. The fluid flow within the hydraulic brake is conditioned by the motion of the piston rod relative to the cylinder, thus it is necessary to dynamically analyze the complete process in a numerical model. The motion law of the piston rod is defined by the UDF [4] file which is compiled in SDOF solver [4] for dynamic mesh purpose [6,7]. To create the UDF file based on the mathematical model (section 3), the polynomial form of the pressure force on the breech block, the recuperator, and the muzzle brake force is taken into account. The weight force component was not considered because the analysis was under zero elevation.

Table 1. Physical characteristics of the hydraulic oil (STEOL)

Density $[kg/m^3]$	Thermal expansion coefficient $[K^{-1}]$	Dynamic viscosity $[kg/m \cdot s]$						Specific heat $[J/kg \cdot K]$	Thermal conductivity $[W/m \cdot K]$
		$-30℃$	$-20℃$	$0℃$	$20℃$	$50℃$	$80℃$		
1100	$65 \cdot 10^{-5}$	0.389	0.144	0.029	0.011	0.004	0.0019	2930	0.24

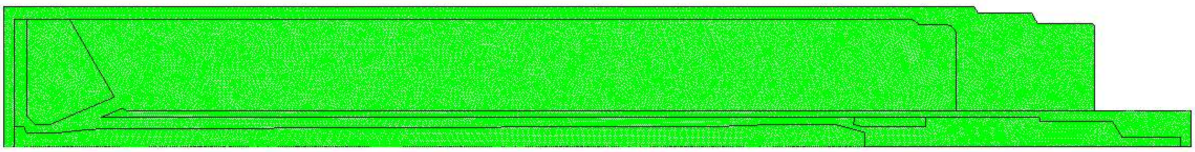


Figure 5. Hydraulic brake model in 2D

5. Experimental and numerical results-comparison and analysis

The experimental results of stress measurements in the hydraulic brake piston rod show that the highest pressures occurs around 10 ms. For this reason, all the numerical results presented in this section are analysed up to 20 ms. In the first 20 ms of relative motion of the piston and cylinder, the influence of temperature change on the physical characteristics of the fluid can be neglected. The recoil length was measured by the inductive bar and from that results it can be calculated the velocity of the recoil parts. In the figure 6 are shown the experimental and numerical results of the recoil velocity.

Pressure distribution and velocity magnitude of the flow field are shown only for temperature T=288K. For extreme temperatures all phenomena are about the same as on the temperature T=288K, so they will be not shown in this paper. But, in the further analysis it will be given recoil velocity vs. time on the extreme temperatures.

Oscillation of the velocity that is obtained by experiment are caused by vibrations of the inductive bar during recoil process. From figure 6. it can be seen that there are acceptable disagreement in the numerical and experimental results, which means that the numerical result of the hydraulic brake force in the observed period of time is acceptable (all other forces that acts on the recoil parts are known). According to that, all others proceses that ocurs in the hydraulic brake can be described using numerical results.

In figure 7. is shown change of the average total temperature in time. Average total temperature of the oil rises up to approximately 13 ms which is caused by great velocity of the fluid at some points and viscous heating. After 13 ms, there is total temperature drop because cavitation occurs at the points with maximum velocities.

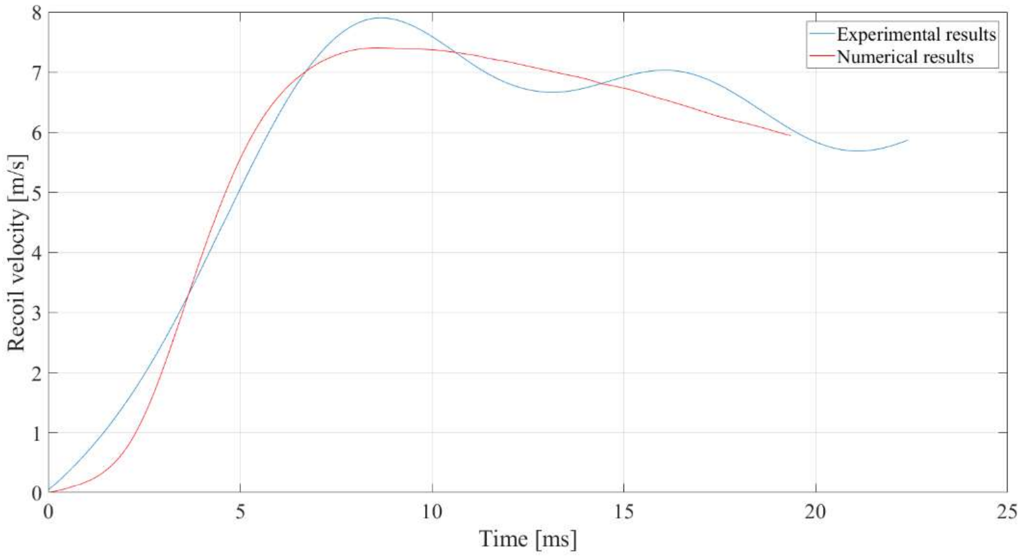


Figure 6. Recoil velocity vs.time

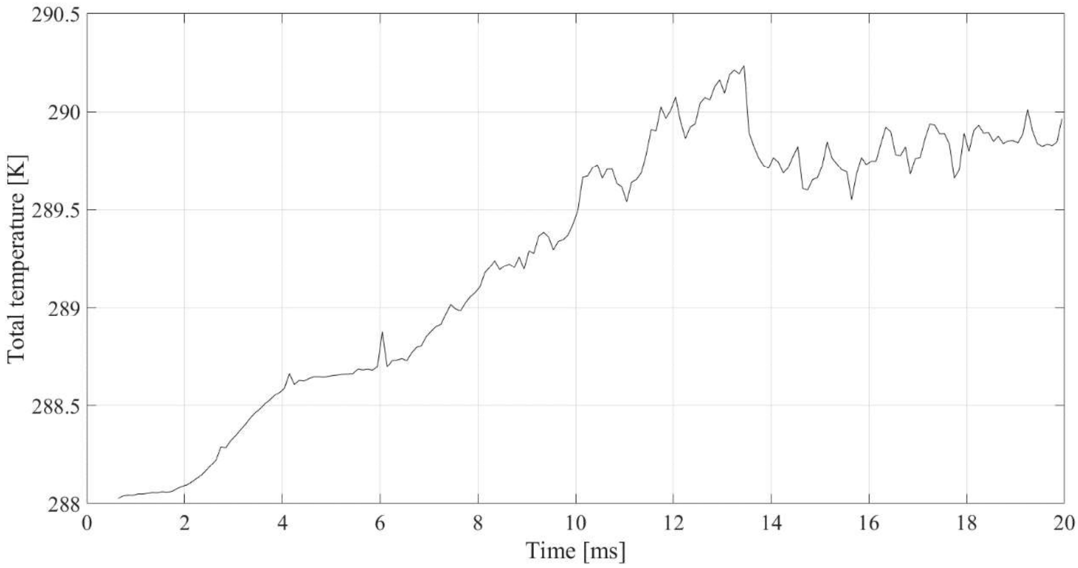


Figure 7. Total temperature of the oil vs. time

In figure 8 is shown field of the total temperature in the hydraulic brake oil at 20 ms.The energy created by viscous disipation is the greatest where the velocity gradient are high [3]. The highest velocity gradient is in the gap between throttling ring and throttling bar, but at that points is also cavitation occurs and the energy for latent heat is taken from surrounding fluid and the temperature rise remains relativly low [4].

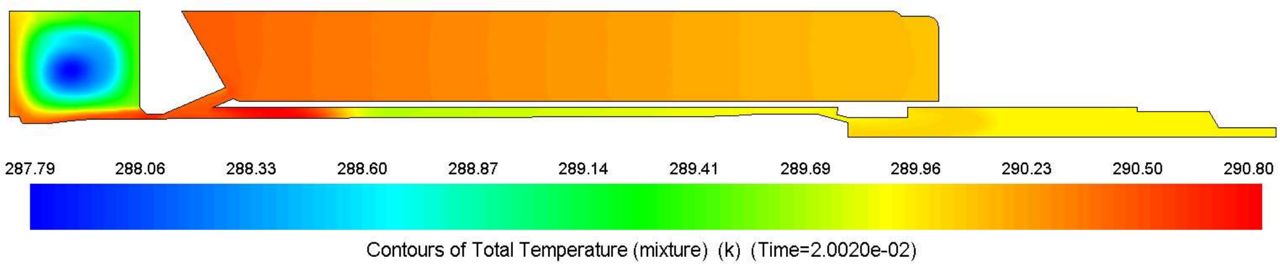


Figure 8. Total temperature field in the hydralic oil at 20 ms

In figure 9 is shown temperature field in the hydraulic brake and surrounding air after 10 seconds. When counterrecoil is finished, until next firing cycle there is heat exchange between metal parts of the hydraulic brake and surrounding air via radiation and natural convection. It was calculated using analytical expression that hydraulic brake temperature rise is approximately 2,9K. From figure 9 can be seen that drop of the hydraulic brake temperature between two firing cycles can be neglected.

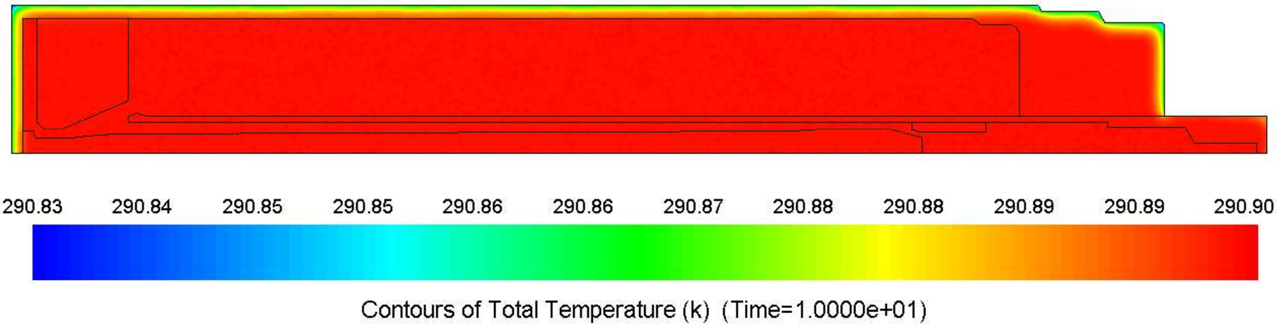
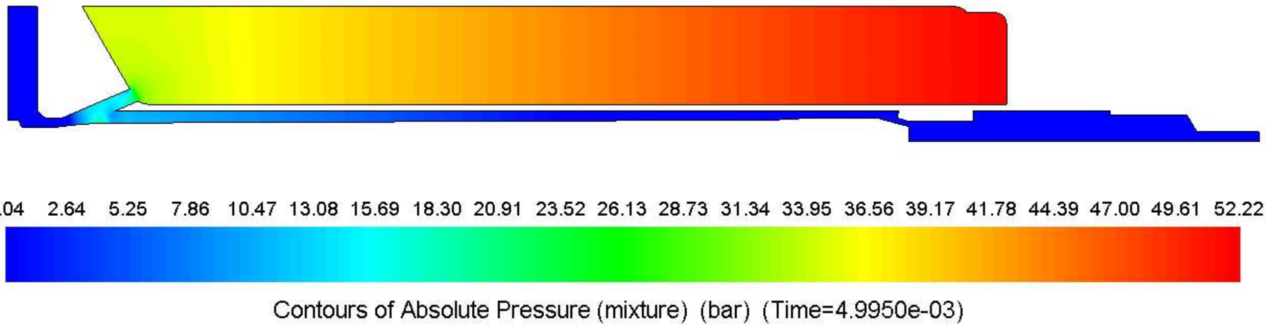


Figure 9. Temperature field in the hydraulic brake and surrounding air after 10 s

In the figure 10a), it is shown the pressure distribution inside the hydraulic brake (volume 1, 2 and 3) in 5ms. Volume 1 shows the pressure difference going from 34 bar at the piston surface to about 52 bar at the bottom of the cylinder. It can be observed that the pressure gradient is opposite to the direction of the moving cylinder. The pressure gradient in volume 1 and channel 1 causes fluid to flow from volume 1 to volumes 2 and 3. In volumes 2 and 3, cavitation can be observed due to the increase of free volume of fluid resulting from the extraction of the piston from the cylinder. In channel 1, it is visible fluid acceleration caused by a change of flow area, where velocity values are going from 37 m/s to 71 m/s, which is shown on the figure 10 b). A change in velocity in channels 3 was also observed, with a value of about 19 m/s. At the observed time of 5 ms, the velocity of the recoil parts is 4.6 m/s.



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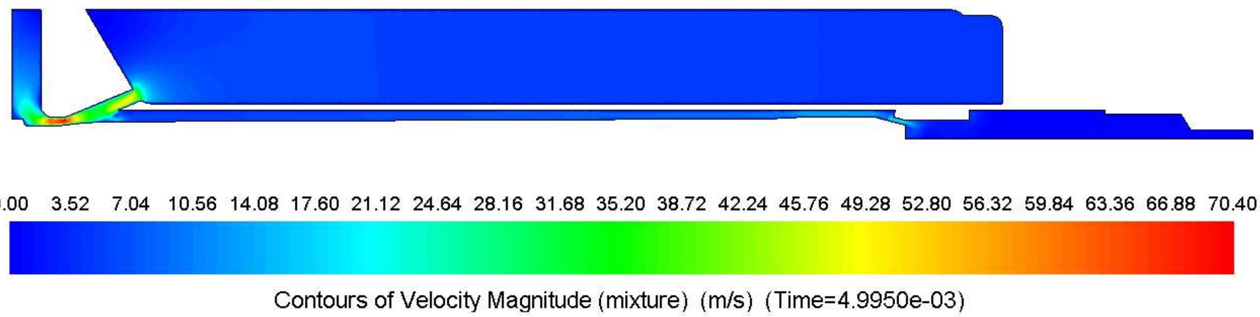
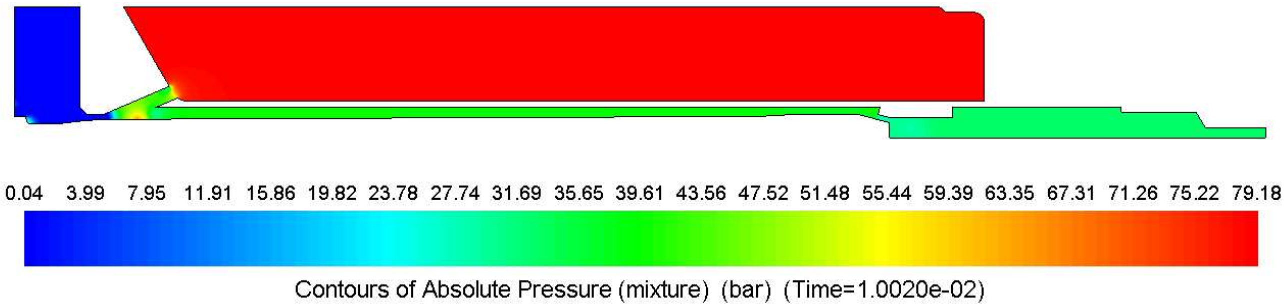


b)

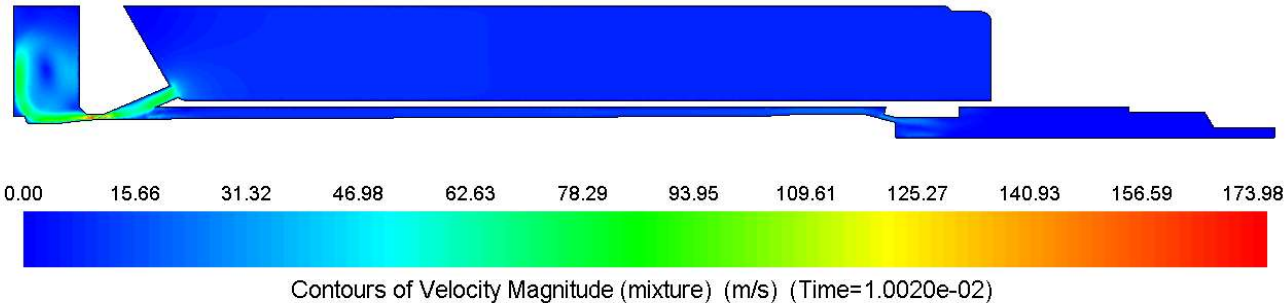
Figure 10: a) pressure distribution in the hydraulic brake at 5 ms;

b) velocity magnitude in the hydraulic brake at 5 ms

At the 10 ms, figure 11a) shows the pressure distribution inside the hydraulic brake. It can be observed that the pressure in volume 1 is approximately constant. The velocity of fluid flow in channels 1 is going from 50 m/s to 174 m/s. In volume 3, there is a significant increase in pressure compared to the previous 5ms time of 34 bar. At the observed moment of 10 ms, the velocity of the recoil parts is 7.35 m/s.



a)

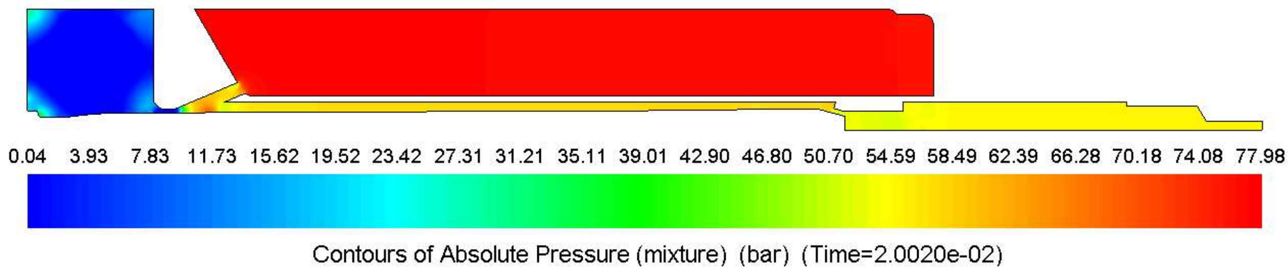


b)

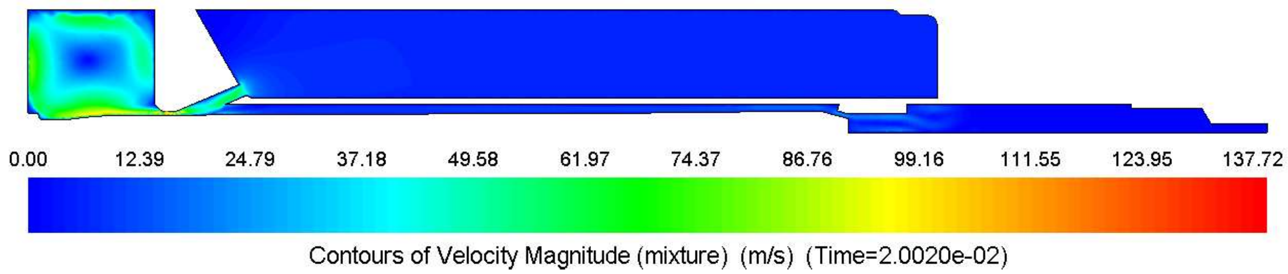
Figure 11: a) pressure distribution in the hydraulic brake at 10 ms;

b) velocity magnitude in the hydraulic brake at 10 ms

In the figure 12a), it is shown the pressure distribution inside the hydraulic brake at 20ms. Due to the decrease in the flow area between volumes 1 and 2, there is an increase in the amount of fluid that moves from volume 1 to volume 3 and the pressure increase in volume 3 (57 bar). The velocity of fluid flow in channel 1 is going from 42 m/s up to 103 m/s. At the observed moment of 20 ms, the velocity of the recoil parts is 5.7 m/s.



a)



b)

Figure 12: a) pressure distribution in the hydraulic brake at 20 ms;

b) velocity magnitude in the hydraulic brake at 20 ms

In figure 13 are shown recoil velocity curves at different temperatures (T=243K, T=288K, T=323K). It can be noticed that the maxim velocity for each temperature is occured in a narrow interval from 8 to 10 ms. Difference of maximum achieved velocities in for considered cases is about 3%. After 10 ms in all considered cases velocities slightly decreases up to 20 ms, where difference between velocities on extreme temperatures is about 8%.

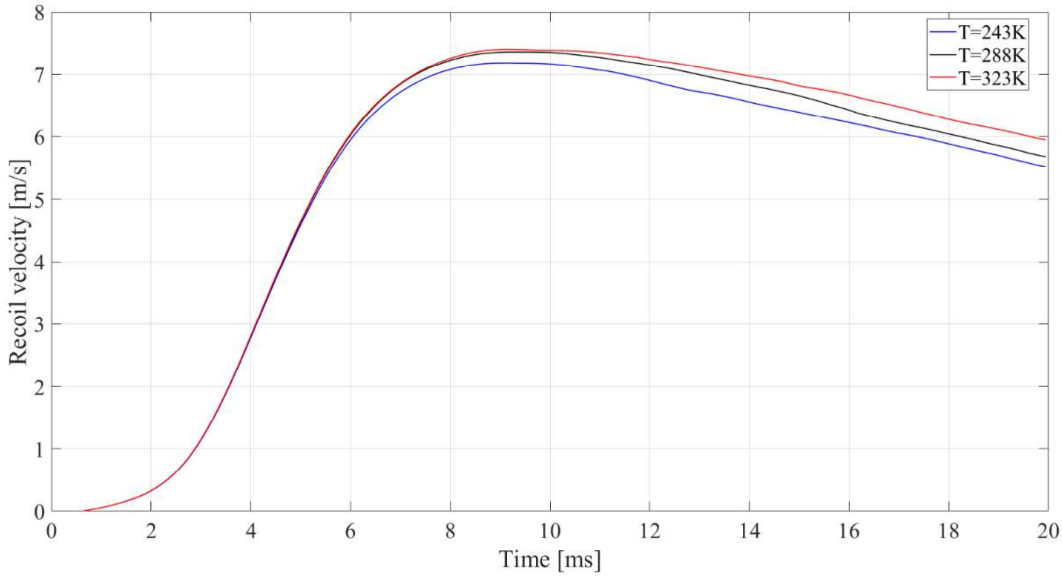


Figure 13. Recoil velocity change in time for different temperatures

Pressures at the normal and extreme temperatures are obtained by numerical method and it is shown in the figure 14. As it can be seen from the equation (2), the pressure depends on fluid density and the square of recoil velocity. The recoil velocities are about the same for normal and extreme temperatures in early phase of recoil (10 ms), which can be seen in figure 13. It means that the pressure in the hydraulic brake at that period will be greater at lower temperature, which can be seen in figure 14. When the velocity of the recoil parts starts to decrease, it will cause the faster pressure decrease at the lower temperature. From figure 14, it can be seen that the maximum pressure at each temperature is about the same value, but it occurs at different time.

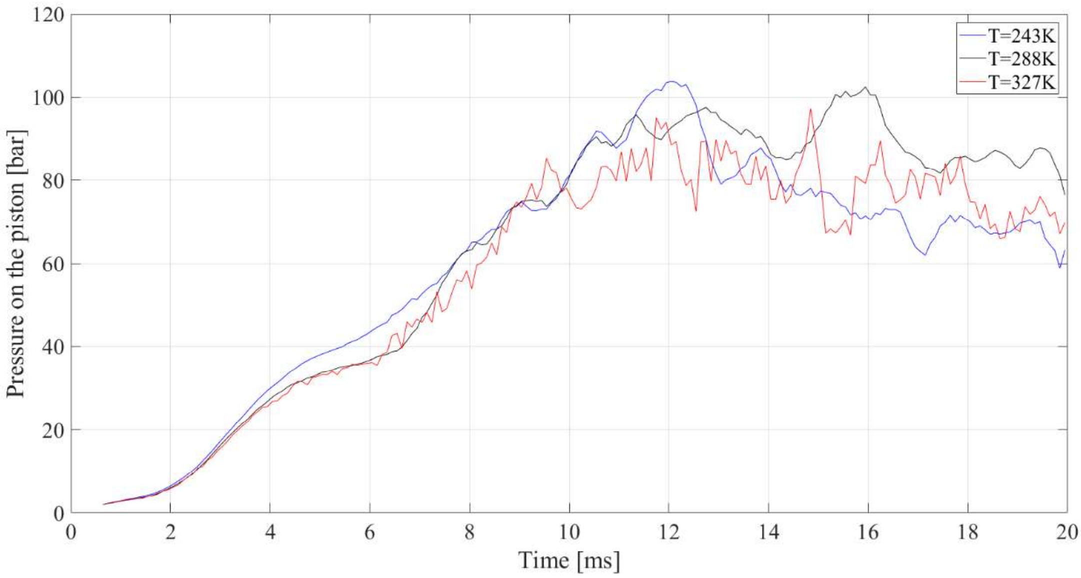


Figure 14. Pressure on the piston vs.time for different temperatures

5. Conclusions

In order to avoid expensive experiments, numerical simulation was applied to get important parameters on different working temperatures that has affects the whole weapon system. Numerical simulation results was verified with experimental results on one temperature. Recoil length was measured with the inductive bar and based on the measured results, recoil velocity was calculated. The recoil velocity obtained experimentally was compared with numerical results and there was a good agreement of the results. Based on that agreement, the hypothesis can be made that the pressure and temperature fields inside the hydraulic brake correspond to realistic conditions. During recoil and counter-recoil cycle, metal parts and hydraulic oil temperature rise approximately 2.9K which is caused by viscous dissipation, work of friction forces, velocity field of the fluid flow inside the hydraulic brake, etc. A temperature drop of the hydraulic brake due to radiation and natural convection with surrounding air can be neglected because of a short period between two firing cycles (10s). The initial temperature of the hydraulic oil has no significant influence on maximum pressure in the hydraulic brake, it only has affects the time when the maximum pressure occurs.

Acknowledgements

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