Analysis of heat conduction in a drum brake system of the wheeled armored personnel carriers

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Abstract. This paper is an integrated study performed over the Braking System of the Wheeled Armored Personnel Carriers. It mainly aims to analyze the heat transfer process which is present in almost any industrial and natural process. The vehicle drum brake systems can generate extremely high temperatures under high but short duration braking loads or under relatively light but continuous braking. For the proper conduct of the special vehicles mission in rough terrain, we are talking about, on one hand, the importance of the possibility of immobilization and retaining position and, on the other hand, during the braking process, the importance movement stability and reversibility or reversibility, to an encounter with an obstacle. Heat transfer processes influence the performance of the braking system. In the braking phase, kinetic energy transforms into thermal energy resulting in intense heating and high temperature states of analyzed vehicle wheels. In the present work a finite element model for the temperature distribution in a brake drum is developed, by employing commercial finite element software, ANSYS. These structural and thermal FEA models will simulate entire braking event. The heat generated during braking causes distortion which modifies thermo-elastic contact pressure distribution drum-shoe interface. In order to capture the effect of heat, a transient thermal analysis is performed in order to predict the temperature distribution transitional brake components. Drum brakes are checked both mechanical and thermal. These tests aim to establish their sustainability in terms of wear and the variation coefficient of friction between the friction surfaces with increasing temperature. Modeling using simulation programs led eventually to the establishment of actual thermal load of the mechanism of brake components. It was drawn the efficiency characteristic by plotting the coefficient of effectiveness relative to the coefficient of friction shoe-drum. Thus induced thermal loads determine thermo mechanical behavior of the structure of wheels. Study the transfer of heat generated during braking is useful because results can improve and validate existing theory or may lead to the development of a mathematical model to simulate the behavior of the brake system for various tactical and operational situations. Conclusions of this paper are relevant because theoretical data analysis results are validated by experimental research.

1. Introduction
Land vehicles were used in ancient times for the transport of persons and goods, satisfying transportation needs on longer distances, using increasingly higher speeds of increasing loads with weights and volume increased.
Figure 1. Vehicle’s braking system diagram: 1, 23 – amplifying cylinder; 2 – braking cylinder; 3 – brake pedal; 4 – braking regulator; 5 – pressure gauge; 6, 13 – pressure limit valve; 7, 14 – electric valves; 8 – air tank; 9 – safety valve; 10 – compressor; 11 – tail lamp; 12 – pressure regulator; 15 – air pipe for engine stopping; 16 – air pipe of the transfer case’s differential locking device; 17 – exhaust valve; 18 – CTIS valve; 19 - warning lamp; 20 – hydraulic warning sensor; 21 – air exhaust pipe; 22, 24 – braking circuits.

Armored personnel carriers are a necessity of the modern battlefield, because they allow movement on undeveloped land at high speeds, proving its usefulness especially in attacks. The high dynamics of warfare involves not only considerable start-up or high speed travel but also the ability to stop on short distances and in safety conditions. We must not forget that it is sometimes necessary to travel on paved roads, in civil traffic conditions, in which case the existence of an effective and reliable braking system is strictly necessary and regulated by the law. Finally, conduct combat actions in the field requires the existence of tight wheel braking mechanisms that do not allow access to indoor contaminants and compromised quality brake pad operation.

The braking system is designed to reduce vehicle velocity faster or even to stop safely, to immobilize the stationary vehicle and to maintain a reasonable speed during downhill. The kinetic energy gained by the vehicle to maintain braking action turns on the one hand in heat (friction), energy which is then dissipated into the environment, and on the other hand is consumed for overcoming rolling and air resistance, which always opposing the movement of the vehicle. The braking system’s diagram is given in figure 1. It is a hydraulic braking system assisted in parallel by a pneumatic servomechanism.

In case of a car braking from a higher velocity \( V_1 \) to a lower velocity \( V_2 \), the braking energy is:

\[
E = \frac{m}{2} \left( V_1^2 - V_2^2 \right) + \frac{L}{2} \left( \omega_1^2 - \omega_2^2 \right) Nm
\]

Where: \( m \) – the vehicle mass, [kg]; \( V_1 \) – the velocity at the beginning of braking, [m/s]; \( V_2 \) – the velocity at the end of braking, [m/s]; \( \omega_1 \) – The angular speed of the parts that rotate at the beginning of breaking [1/s]; \( \omega_2 \) – The angular speed of the parts that rotate at the ending of breaking [1/s].

2. Thermal calculation
Thermal calculation of armored transporter braking system- see figure 2, is based on experimental data relating to real conditions of cooling of the brakes in the braking phase.

Figure 2. The drum brake equipping armored transporter.
The amount of heat released in a second is determined by:

\[ Q = \frac{F_f v_a}{427} = \frac{\mu p_0 \Sigma A v_a}{427} \text{[kcal/s]} \]  

(2)

In which: \( v \) is the speed of sliding of the drum on the friction lining \((v_a = (V/3.6) (r_t/r_r))\); \( F_f \) - the braking force acting on the drum \((F_f = \mu p_0 \Sigma A)\); \( \Sigma A \) - friction surface; \( p_0 \) - average pressure [2].

The heat flux density \( q_d \) is determined by the relation:

\[ q_d = \frac{Q}{\Sigma A} = \frac{\mu p_0 v_a}{427} \]  

(3)

It results that, in the case of braking, the heat flux density that is released is proportional to the specific friction power at the drum sliding the friction lining \((\mu p_0 v_a)\).

From the relation:

\[ p_0 = \frac{G_a}{\Sigma A} \cdot \frac{r_f}{r_t} \cdot \frac{a_f}{\mu g} \]  

(4)

Results:

\[ \mu p_0 = \frac{G_a}{\Sigma A} \cdot \frac{r_f}{r_t} \cdot \frac{a_f}{g} \]  

(5)

Considering the relation between \( v_a \) and the vehicle velocity \( V \), the specific friction power at the drum sliding the friction lining will be:

\[ \mu p_0 v_a = \frac{G_a}{\Sigma A} \cdot \frac{V}{3.6} \cdot \frac{a_f}{g} \]  

(6)

When comparing vehicles of a certain class, in case of intensive braking to the limit of adhesion \((\text{when } a_f = g \phi)\) it can be considered that \( V a_f = V g \phi \) is constant. It means that the specific friction power at the drum sliding the friction lining, and therefore, the heat flux density, equation (3) is proportional to the ratio \( G_a/\Sigma A \).

It is considered that the ratio \( G_a/\Sigma A \) on an average brake shall have the following values:
- cars, \( G_a/\Sigma A = 1.5 \pm 2.0 \text{ daN/cm}^2\);
- trucks, buses and trailers, \( G_a/\Sigma A = (2.0 \pm 4.0) \text{ daN/cm}^2\).

If the ratio \( G_a/\Sigma A > 3 \text{ daN/cm}^2 \) the brakes will be intensive used. This is allowed in vehicles equipped with retarders, which reduce the usage period of the service brake and therefore heating process.

3. **Theoretical aspects of non-stationary heat transfer**

In non-stationary processes, temperature and heat flow in any point can take variable values in time. The general equation of non-stationary state heat transfer is as follows [3]:

\[ \rho c \frac{\partial T}{\partial t} = \frac{\partial}{\partial x} \left( \lambda_x \frac{\partial T}{\partial x} \right) + \frac{\partial}{\partial y} \left( \lambda_y \frac{\partial T}{\partial y} \right) + \frac{\partial}{\partial z} \left( \lambda_z \frac{\partial T}{\partial z} \right) + Q \]  

(7)

*Hamilton's Principle* states that \( J = 0 \) in relation to the temperature \( T \). Thus, the functional sought is:
The condition \( J = 0 \) is equivalent formulation to equation (6). It is necessary to be determined the extreme condition for functional \( J = 0 \).

It considers thermal gradients vector \( \{ \mathbf{T}'_{x,y,z} \} \) and thermal characteristics matrix:

\[
[\lambda] = \begin{bmatrix}
\lambda_x & 0 & 0 \\
0 & \lambda_y & 0 \\
0 & 0 & \lambda_z \\
\end{bmatrix}
\]

(9)

Record the temperature derivative depending on time \( \dot{T} \) and organize \( T \) and \( \dot{T} \) as vectors with one single component. In this case, functional \( J = 0 \) can be written as a matrix:

\[
\begin{align*}
J(T) &= \frac{1}{2} \left[ \int_V \left( \lambda_x \left( \frac{\partial T}{\partial x} \right)^2 + \lambda_y \left( \frac{\partial T}{\partial y} \right)^2 + \lambda_z \left( \frac{\partial T}{\partial z} \right)^2 \right) dV + \int_V \left( \alpha \frac{\partial T}{\partial t} - Q \right) dV + \int_{A_i} \alpha(T - T_2)^2 dA + \int_{A_i} q dA \right] = 0 \\
\end{align*}
\]

(8)

It is admitted that the calculation of the whole structure is divided into a finite number of subdomains called finite element small enough so that each element in the finite temperature function \( T = T(x, y, z) \) can be approximated well using interpolation functions \( \{ N \} = \{ N_1, N_2, N_3, \ldots \} \), and \( i \) is the counter for the number of finite element nodes. For a finite element the tetrahedral relationship \( \{ N \} = \{ N_1, N_2, N_3, \ldots \} \) can be written as \( \{ T \} = \{ N_1 \ldots N_4 \} \{ T_1 \ldots T_4 \}^T \).

Interpolation functions \( N_i \) are functions of \( x, y, z \), so one can build the temperature derivatives \( T \) in relation to \( x, y, z \). It results:

\[
\frac{\partial T}{\partial x} = \sum_i \frac{\partial N_i}{\partial x} T_i \begin{bmatrix} T_1 \\ T_2 \\ T_3 \\ T_4 \end{bmatrix}, \text{ also similar for } \frac{\partial T}{\partial y} \text{ and } \frac{\partial T}{\partial z}
\]

(11)

It results:

\[
\begin{align*}
J(T) &= \frac{1}{2} \left[ \{ T_1 \}^T \int_V [\lambda] [B] dV \{ T_1 \} + \{ T_1 \}^T \rho c \{ N \} \{ N \} dV \{ T_1 \} - \\
&- \{ T_1 \}^T \int_V \{ N \}^T \{ Q \} dV + \frac{1}{2} \int_{A_i} \alpha \left( \{ T_1 \}^T \{ N \} - \{ T_2 \}^T \{ N \} \right) \{ T_1 \} dA + \\
&+ \{ T_1 \}^T \int_{A_i} \{ N \}^T \{ q \} dA \\
\end{align*}
\]

(12)

It requires minimal functional condition in equation (12) with respect to the nodal temperatures \( T_i \). [3]:
\[
\delta J = \{T_i\}^T \left( \int_V [B]^T \lambda [B] dV \{T_i\} + \int_V \rho c \{N\}^T \{N\} dV \{T_i\} - \right.
\]
\[
-\int_V \{N\}^T \{Q\} dV + \int_{A_i} \alpha \{N\}^T \{T_2\} dA \{T_i\} + \int_{A_i} \alpha \{N\}^T \{T_2\} dA - \int_{A_i} \{N\}^T \{q\} dA \bigg) = 0
\]

(13)

where \( \delta T_i \) is a small but finite temperature variation to the thermal equilibrium temperature. It follows that it is necessary that value between brackets in equation (13) to be null. It will be obtained:
\[
\int_V [B]^T \lambda [B] dV \{T_i\} + \int_V \rho c \{N\}^T \{N\} dV \{T_i\} =
\]
\[
= \int_V \{N\}^T \{Q\} dV - \int_{A_i} \alpha \{N\}^T \{T_2\} dA \{T_i\} + \int_{A_i} \alpha \{N\}^T \{T_2\} dA - \int_{A_i} \{N\}^T \{q\} dA
\]

(14)

By grouping the terms it results:
\[
\int_V \rho c \{N\}^T \{N\} dV \{T_i\} + \left( \int_V [B]^T \lambda [B] dV + \int_{A_i} \alpha \{N\}^T \{T_2\} dA \right) \{T_i\} =
\]
\[
= \int_V \{N\}^T \{Q\} dV + \int_{A_i} \alpha \{N\}^T \{T_2\} dA - \int_{A_i} \{N\}^T \{q\} dA
\]

(15)

The following notations are introduced:
\[
\int_V \rho c \{N\}^T \{N\} dV = [C]
\]
\[
\int_V [B]^T \lambda [B] dV + \int_{A_i} \alpha \{N\}^T \{N\} dA = [K]
\]
\[
\int_V \{N\}^T \{Q\} dV + \int_{A_i} \alpha \{N\}^T \{T_2\} dA - \int_{A_i} \{N\}^T \{q\} dA = [R]
\]

(16)

where: \([C]\) is called heat capacity matrix; \([K]\) is called thermal rigidity matrix; \([R]\) is called free vector terms.

With this notation, equation (14) becomes
\[
[C] \{T_i\} + [K] \{T_i\} = \{R\}
\]

(17)

Matrix differential equation (17) is a linear ordinary differential equation system associated with the given element, which means that the heat transfer is linear both in the stationary as well as in dynamic conditions [3, 4]. Following a process of assembly systems it results a linear system of ordinary differential equations associated with the entire structure, whose solution approximates the temperature variation in time and space depending on the external thermal stresses.

4. Modeling of heat transfer in the brake system

The heat generated by the friction depends on the pressure of contact between the friction lining and the drum and also the relative velocity between the parts in contact, the temperature of the component
parts that influence, as well as the velocity, the coefficient of friction between the drum and the friction lining. An estimate of the heat that depends on these parameters is difficult to predict. Energy approach to this problem is much simpler, if some simplifying assumptions are adopted:
- work done by friction brake it is fully converted into heat,
- all wheels contribute equally to the work of friction,
- heat flux is distributed on the contact surfaces constantly, being only variable in time,
- heat taken up by the drum is considered to be evenly distributed over the inner surface thereof, the width of contact with the friction lining, so resulting a heat flux equivalent. In reality this amount of heat is transferred to the drum only on the contact surfaces with friction linings that are changing the position due to the rotation of the drum.

Figure 3. Thermal analysis of drum brake.

5. Conclusions
The modeling of the friction shoe-drum shown in figure 3 allowed simulations of the thermal performance of the two components, in assembly with the metallic element of the shoe. The main conclusions arising from the performance of simulations are:
- the friction lining knows a rapid heating of the surface contact during the period the shoe is applied to the drum. However, once the shoe is withdrawn, the friction lining surface temperature decreases due to the process of convection and transmissibility of the shoe metallic element;
- reducing the temperature of the friction lining surface is enhanced by increased convection because of the high tangential speed of the air driven by the rotation of the drum;
- at the friction lining surface in contact with the rollers is made large increases in temperature during application of the shoe to the drum, followed by intense cooling by heat transfer, so that at the end of a braking cycle the temperature rise does not exceed 40 °C for an intense braking situation;
- through simulation was determined that by acting intermittent braking the friction lining is kept at an accepted level;
- during braking it is an increase of about 16 °C ÷ 18 °C of the drum temperature, field sites being equalized quickly between the range of braking;
- in the shoe was found by simulation the existence of 3 ÷ 8 °C gradients temperature between the areas studied, the lowest temperatures being recorded in the ribs. This is explained by the local mass increase and by higher surface heat convection adjacent space;
- performed simulations allow identifying the mode of transmission of heat flow, enabling further development of measures to optimize its design of the shoe so as to reduce the operating temperature of friction.

The final conclusion is that, by finite element modeling of the mechanical and thermal simulations allow highlighting the areas localization module with thermal accumulation and intensity of heat transfer processes, both inside the drum and shoe friction lining as and by convection.
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