OVERHEATING ANALYSIS OF THE SPECIAL VEHICLES BRAKING SYSTEMS

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Abstract: *This paper shows the basis of thermal calculation of the special vehicle brake. During the braking the vehicle's kinetic energy is converted to heat. Calculation of a special vehicle brake heat can be based on experimental data on actual conditions in the cooling of the brakes braking or can be simulated using finite element method. The thermal regime has a great importance for the smooth operation of the brakes.*

Keywords: *braking system, heat transfer, brake temperature analysis, thermal stress.*

1. INTRODUCTION

Braking is the process to reduce all or part of a vehicle velocity. The braking system of a vehicle is one of the most important structural components thereof, representing the main element in ensuring traffic safety, given the continued growth of the dynamic qualities of the car and traffic intensification.

Depending on the destination of the vehicle and the level of technological development there are met different braking systems solutions. From their analysis one can see that there is a tendency of using complex braking systems for special vehicles.

As is known, during the braking the vehicle's kinetic energy is converted to heat, mainly taken up by the friction elements of the brake mechanisms of the wheels. Usually this heat is released to the environment, the heat transfer can be accelerated by a suitable configuration of the various components of the wheel brake mechanism, leading to the design of brakes ventilated. But if the service brake is applied extensively, heat removal capability to environment is limited, and thus brake components will overheat and this will conduct to some undesirable phenomena.

First, braking mechanisms being in the vicinity of wheel bearings will rise its temperature, and the lubricant will lose its qualities and can determine their seizure.

Second, by increasing the temperature of the wheel brake mechanism components can lead to loss of its physico-mechanical properties. Finally, thirdly and most importantly undesirable phenomenon by increasing temperature conjugate material (steel or cast iron / lining friction) friction coefficient decreases and hence braking effect is reduced dangerously with serious implications for traffic safety.

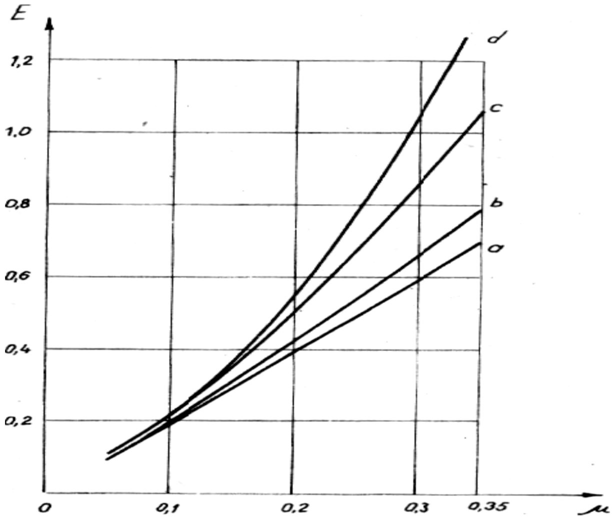

Fig. 1 The efficiency characteristic of drum brake mechanisms
a - simplex, with equal displacement of shoes;
b - simplex, with independent displacement of shoes; c - duplex; d – servo

after Marinescu M, (2006)

2. THERMAL CALCULATION

2.1 Considerations on thermal stresses of the brakes. Calculation of a vehicle brake heat can only be based on experimental data on actual conditions in the cooling of the brakes braking [1]. The amount of heat released in a second is determined by the relation:

$$Q = \frac{F_{ft} v_a}{427} = \frac{\mu p_0 \Sigma A v_a}{427} [kcal/s] \quad (1)$$

where: v_a is the slip speed between the drum and the friction lining ($v_a = (V/3,6) \cdot (r_t / r_r)$; F_{ft} — the braking force acting on the drum ($F_{ft} = \mu p_0 \Sigma A$); ΣA – lining friction area; p_0— average pressure.

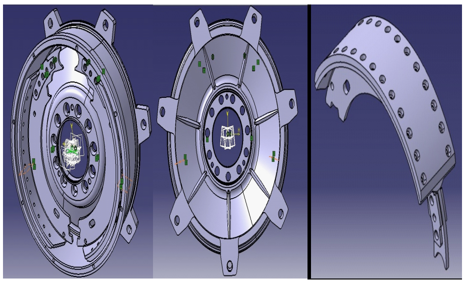

Fig. 2 Braking system with shoe and drum

2.2 Thermal calculation on intensive brakes. In case of short isolate intensive braking, heat exchange with the outside can be neglected, considering that the entire amount of heat that is released contribute to raising the temperature of the brake itself. Due to the very low thermal conductivity of friction, almost all of the heat is taken from drum or disc.

Thermal balance in case of intensive braking from a velocity V to a stop is given by:

$$\frac{1}{2} \cdot \frac{G_a}{g} \cdot \frac{V^2}{3{,}6^2} \cdot \frac{1}{427} = \xi \cdot G_t \cdot c \cdot n_f \cdot \Delta\tau \quad (2)$$

where: ξ is a coefficient representing the fraction of the heat produced and taken to the drum or disc (in the case of drum brakes ξ = 90 ...95% and for disc brakes ξ = 99%); G_t - weight drum (disc); c - heat mass; G_a - vehicle weight; n_f – number of braked wheels; Δτ - the increasing temperature of the drum (disc).

From equation (2) results temperature rise Δτ of drum (disc) in case of an isolated intensive braking from velocity *V* to a stop:

$$\Delta\tau = \frac{G_a V^2}{108500 \xi c n_f G_t} [^o C] \quad (3)$$

If the difference in thermal regime between front and rear brakes of wheels is large temperature increase Δτ determination must be made separately for front and rear brakes, the total energy has to be distributed in the same ratio as brake force distribution on axle.

It is recommended that intensive braking from 30 km/h until the vehicle stops, temperature rise Δτ does not exceed *15*°C.

2.3 Thermal calculation brakes in case of long braking. On long braking it takes account of heat exchange with the environment.

Thermal balance corresponding to a time interval *dt* is given by

$$dQ = dQ_1 + dQ_2 \quad (4)$$

where: dQ is the amount of elementary heat resulting from brake; dQ_1 - the amount of elementary heat transferred to external environment; dQ_2 - the amount of elementary heat consumed in heating the drum (disc).

If elementary heat quantities are replaced dQ, dQ_1 and dQ_2 heat balance becomes

$$q_d A dt = \alpha A_r \tau dt + c G_t d\tau \quad (5)$$

where: q_d is the density of heat flow at the beginning of prolonged braking; A — lining friction surface; α — heat exchange coefficient between the drum and the air; A_r — cooling surface of the drum; *τ* — relative temperature of the drum in relation to the environment; *c* — the heat mass of the material the drum (disc) is made from; G_t — drum weight; dτ — temperature increase.

By integrating relation (5) and setting the initial condition that at *t* = 0 and τ = 0 it results time required for the drum temperature to reach a predetermined value:

$$t = \frac{c G_t}{\alpha A_r} \ln \frac{q_d A}{q_d A - \alpha A_r \tau} [s] \quad (6)$$

The heat flux density q_d is given by

$$q_d = \frac{G_a}{g \sum A} \cdot \frac{V}{3{,}6} \cdot \frac{a_f}{427} \left[\frac{kcal}{cm^2 \cdot s} \right] \qquad (7)$$

where a_f is braking deceleration.

Heat exchange coefficient α varies with the relative speed of the drum (disc) and ambient air and can be calculated with the relation:

$$\alpha = 1{,}25 \cdot 10^{-7} + 1{,}6 \cdot 10^{-10} nr - 3{,}237 \cdot 10^{-12} (nr)^{1{,}32} \, [kcal/cm^2 \cdot s \cdot {}^\circ C] \qquad (8)$$

The cooling surface of the drum A_r consists of the front surface A_f and the surface of the crown A_c. In the calculations front surface have to be equated with the surface of the crown (because both temperature, and α varies with the radius *r),* to give

$$A_r = A_c + \frac{2\pi}{\alpha_s \tau_s} \cdot \int_{r_i}^{r_s} \alpha_{(r)} \tau_{ri} \, r dr \qquad (9)$$

where $\tau_{\max} = \frac{\tau_e}{r_e} \cdot r$

if prolonged braking, maximum temperature of the drum - (disc) can be calculated with approximate relation

$$\tau_{\max} \approx 56{,}5 \frac{\chi q_d}{\rho c} \sqrt{\frac{V}{3{,}6} \cdot \frac{1}{\pi a_f \alpha_i}} \qquad (10)$$

where: ϰ is a coefficient of heat distribution between the friction linings and drum or disc (ϰ = 1 if it is considered that the insulating seals, ϰ = 0,5 if it is considered that the insulating seals); q_d — heat flux density, in kW/cm^2 and is determined by the relation (7) ; ρ — drum or disc material density in kg/m^3; c — heat the drum or disc mass in kJ/kg°C; a_f — vehicle braking deceleration in m/s^2; *V* — the speed of the vehicle, in km/h; $\alpha_t = \lambda/(c\rho)$ — thermal diffusivity m^2/s (λ thermal conductivity W/m °C. It must, under the test conditions prescribed by Regulation no. 13 C.E.E. UN brake temperatures not exceeding 300°C.

2.4 Thermal calculation brakes for repeated braking. Repeated braking, when the number is high, it strikes a balance between the heat and the heat discharged, leading to the saturation temperature of the drum (disc) given by

$$\tau_s = \tau_0 + \frac{\Delta\tau}{1 - e^{-b_0}} \approx \tau_0 + \frac{\tau}{b_0} \qquad (11)$$

where: τ_0 is the ambient temperature; Δτ— temperature rise due to brake; *b* — coefficient characterizing the brake cooling conditions; t_0 - range of braking.

Increased temperature Δτ is calculated with

$$\Delta\tau = \frac{\Delta E}{c m_t} \qquad (12)$$

where ΔE is the energy absorbed from a single braking (during which the vehicle velocity decreases from V_1 to V_2) and m_t is the mass of the drum (disc).

If the brakes are to stopping the vehicle, the temperature increase Δτ is determined by equation (3).

The coefficient *b* depends on the size of the friction surfaces, installation conditions of the drum (disc) on the block, vehicle velocity, and so on. For *V* = 30 km/h the coefficient *b* = 0,001 ... 0,004 [s^{-1}], higher values correspond to better ventilation brakes (open disc brake).

At heat checking, the saturation temperature does not exceed values that can modify the properties of the friction linings and the drum (disc).

The functioning properly of brake depends on the use of appropriate materials for construction.

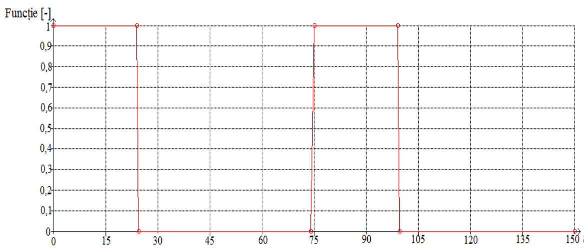

Fig. 3 The law of variation of the heat flow required for a period of 25 seconds brake

As a result of thermal expansion, tensile stresses occur in the drum is determined by the relationship

$$\sigma^- = \frac{E \alpha_l \tau}{1 - \delta_p} \qquad (13)$$

where: E is Young module, in daN/cm^2; α_l— the coefficient of linear expansion, the m/m • °C; τ - drum temperature in °C; δ_p— Poisson's constant (δ_p = 0,26).

3. COMPUTER-BASED BRAKE TEMPERATURE ANALYSIS

The system (the drum or disc thickness) is divided into a number of discrete nodal points, as illustrated in fig. 4, for a one-dimensional temperature analysis [2]. In fig. 4 the temperature is analyzed only as a function of distance x and time t. Application of the first law of thermo dynamics, or energy balance, to each individual node results in a set of algebra equations whose solution will yield individual nodal temperatures for each finite time interval.

It is therefore necessary to calculate the temperature distribution at some future time from a given distribution at an earlier time, the earliest time being associated with the known initial temperature distribution existing at the onset of braking. The relationship expressing heat conduction between two nodes is known as Fourier's Conduction Law and may be expressed in the form of an exact integral:

$$q_{ij} = \int_{\Delta y} -k\left(\frac{\partial T}{\partial x}\right) b dy \approx -k\left(\frac{dT}{Dv}\right)_{average} b\Delta y \tag{14}$$

where b = width of plate, m;

q_{ij} heat flow between nodal points i and j, Nm/ h

x = horizontal distance between two adjacent nodal points, m;

y = vertical distance between two adjacent nodal points, m;

$\frac{\partial T}{\partial x}$ = temperature gradient, K/m;

The distances Ax, Ay, and b designate control volume size, and k the thermal conductivity of the material. Eq. (14) may be rewritten in the form of the temperature of the two nodal points

$$q_{ij} = -\frac{k\left(T_j - T_i\right)b\Delta y}{\Delta x}, Nm/h \tag{15}$$

where T_i = temperature of node i, K; T_j = temperature of node j, K;

For two-dimensional temperature problems, where T = f(x, y, t), for example, and a square grid size with Ax = Ay, the basic heat conduction between two nodal points becomes

$$q_{ij} = k\left(T_j - T_i\right)b, Nm/h \tag{16}$$

For one-dimensional systems such as a solid disc brake, the basic heat conduction equation with Ay equal to unity becomes:

$$q_{ij} = \frac{k_R\left(T_i - T_j\right)b}{Ax}, Nm/h \tag{17}$$

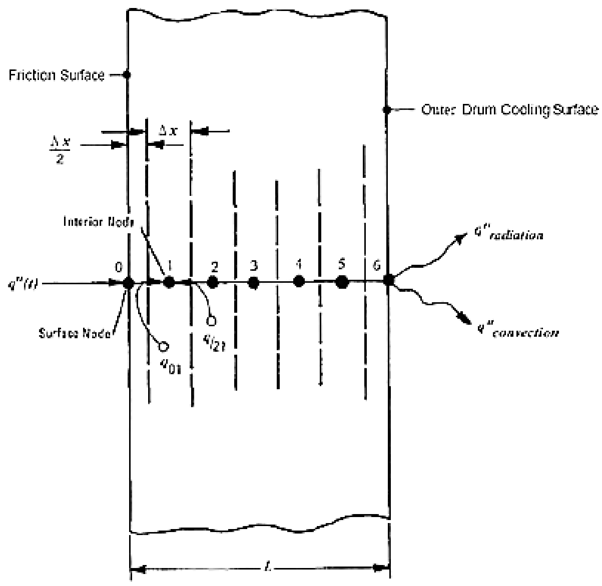

Fig. 4 - Thermal model for finite difference computation (drum brake shown)

after Limpert R, (2011)

CONCLUSIONS

Because of the significant role that the braking installation has to ensure the safety of movement, it is essential that it has a close to 100% reliability.

To meet this requirement a series of constructive measures were taken, in order to permit vehicle braking effectiveness sufficient in the event of the appearance of damage in a section of the braking device or warn in time the driver about an eminent reduction in efficacy.

Brakes are checked on the mechanical and thermal loads. These tests aim to establish their sustainability in terms of wear and the variation coefficient of friction between the friction surfaces along with increasing temperature.

Analytical calculation verification of thermal stresses in the design phase cannot be calculated precisely because it does not meet all the necessary data. It is therefore recommended to be made on the basis of existing data from similar products for predimensioning. Modeling using simulation programs will lead eventually to the establishment of the actual thermal load brake mechanism components.

The thermal regime has a great importance for the smooth operation of the brakes. Reduce heat brake system, in addition to the measures listed, it is also obtained by:

- Correlation between the outer diameter of the drum (the disk) and the diameter of the crown of the wheel;
- Increasing the cooling surface by using drums with circumferential outer ribs (tests indicate a reduction in temperature with 45-65% ribbed drum to drum simple);
- Creation of radial channels in highly loaded disc brake thermal (at rotating disc it creates its interior ventilation, helping to clear heat in the environment);
- By forced cooling of the brakes.

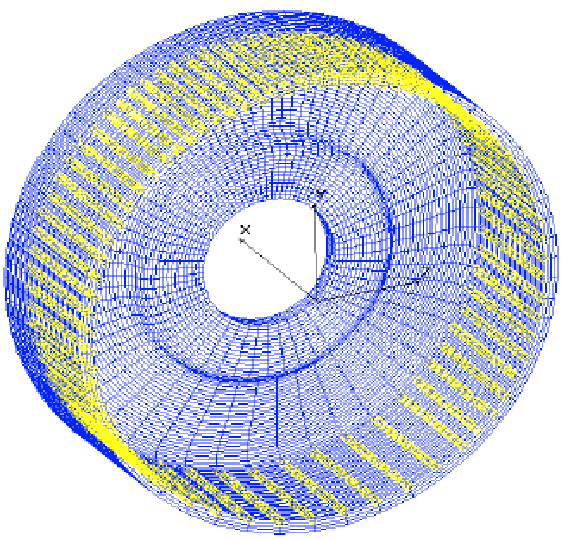

Fig. 5 Scheme of thermal field distribution (internal temperature of 150^0 C) due to intense friction between the shoes and brake drums for armored personnel carrier

after Marinescu M, (2006)

Modeling using simulation programs will lead eventually to the establishment of the actual thermal load brake mechanism components.

For example, in fig. 5, is shown thermal field distribution scheme (for indoor temperature by the heat generated on intense braking 150^o C) for brake drum of armored personnel carrier [4].

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BIBLIOGRAPHY

1. Frățilă, Gh., Mărculescu, Gh., *The braking systems of motor vehicles*. Bucharest: Technical Publishing House (1986).
2. Limpert, R., *Brake design and safety, third ed.,* SAE International, Warrendale, 2011.
3. Marinescu, M., *Modern Solutions in Vehicle Manufacturing*., Bucharest: Military Technical Academy Publishing House (2002).
4. Marinescu, M., *Theory, calculation and construction of armored personnel carrier*. Bucharest: Military Technical Academy Publishing House (2006).
5. Puncioiu, A., Truță, M., *Determine the parameters of the braking capacity of the special vehicles braking systems.*, Review of the Air Force Academy, No 3 (2014) 83-88.