SIMULATION TESTS OF PERIPHERAL FRICTION BRAKE USED IN AGRICULTURAL MACHINERY SHAFTS

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Abstract. Shafts are commonly used parts in agricultural machinery. They are responsible for transmitting the torque from the power source to the receiver, which can be, e.g. a mower blade, hydraulic pump or something completely different. In the agricultural industry, where the operator is very often near a rotating part, it is important to have the ability to quickly stop the rotating shaft. This has a direct impact on work safety. Therefore, the health or even life of both the operator and third parties nearby may depend on the proper functioning of the braking system. A characteristic feature of all friction brakes is their heating up during operation. High rotational speeds of shafts combined with the need to quickly stop the rotational element can lead to the generation of very high temperatures in the friction node. Overheating the brake can lead to the so-called fading phenomenon, which reduces the value of the friction coefficient practically to zero. The purpose of the work was to develop a mathematical model of the agricultural machine shaft breaking process and check how the initial speed affects the heating of friction steam. In this work, a mathematical model of a typical peripheral brake was developed, and simulation tests of the shaft braking process from different initial speeds based on it were performed. The CAD model of the brake and the finite element method were used to determine temperature profiles. It was also assumed that the inertia resulting from the non-zero mass of the system is constant for all cases. The stopping time was also the same for each case. The obtained results allow to assess the dependence of heating on the initial rotation speed of the shaft, and the developed model can be extremely helpful at the design stage of agricultural machinery.

Keywords: mechanical engineering, brakes, friction, agricultural machinery.

Introduction

Agricultural machinery of various types and sizes are widely used in today’s agricultural industry [1; 2]. This is due to, among others, growing farm acres [3]. They are usually driven through the PTO shaft by the internal combustion engine of the agricultural tractor [4, 5]. There is a certain rotational speed and a certain torque at the input of the device. In the most cases they are transmitted to the rest of the device by using shafts, which can be connected by various types of gears (traction, gear, friction, etc.) or clutches [6; 7]. Ultimately, the drive goes to the receiver, which can be, e.g. a cutting knife, press or any other device.

Most of moving parts of agricultural machines are made of steel [8]. It is a very good construction material, but it has a high density, which simply makes the components heavy [9-11]. In turn, mass affects inertia [12-14]. The high inertia of the rotating elements means that both their acceleration and stopping require significant amounts of energy [15]. For the drive, as already mentioned, the energy generated by the farm tractor engine is needed. For stopping, the phenomenon of friction is most often used [16-19]. In both cases, energy conversion takes place. During acceleration, the chemical energy of the fuel is converted into the kinetic energy of the rotating element [20-23], while during braking, because of the friction, kinetic energy is converted into thermal energy [24].

For agricultural machinery shafts, the most common brake solution is peripheral friction brakes. They are made of a friction pair, which is a disc as an integral part of the shaft or an element mounted on the shaft, and a belt that clamps on the plug when braking.

As a result of their cooperation, kinetic energy is converted into thermal energy, which is then discharged into the atmosphere and into the immediate surroundings (to nearest elements of the structure). Unfortunately, large amounts of heat can be harmful [25; 26]. A real threat of the phenomenon of the so-called fading appears, as a result of which the value of the coefficient of friction drops practically to zero [27].

In this work, it was decided to check how the initial rotational speed of the shaft affects the course of the friction pair heating process during emergency stopping of the rotary mower shaft.

Materials and methods

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The research object was the main working shaft of a popular rotary mower in Poland equipped with a peripheral brake. The brake disc is a separate element mounted on the shaft using shape connection. The shaft assembly diagram is shown in Fig. 1.

![Scheme of the test object](image)

**Fig. 1. Scheme of the test object:** 1 – disc acting as a flywheel and brake element; 2 – brake band; 3 – brake actuator; 4 – belt drive pulley (shaft source of drive); 5 – mower shaft; 6 – mower disk; 7 – cutting knives

The CAD model (Fig. 2) of the tested element (the shaft brake) was developed (using SolidWorks) and then it was tested by using FEM. Simplification consisted in removing elements that do not have a significant impact on the test results (like shaft inlet connection or bends in the strap for attaching the actuator). Otherwise it could introduce unnecessary high mesh density, and thus - among others, extension of the test time. It was also assumed that the nominal curb weight of the braking element is 46kg.

![Simplified CAD model of the shaft brake](image)

**Fig. 2. Simplified CAD model of the shaft brake**

The brake disc material is cast iron, and the band material is metal wool pressed with synthetic rubber. The determined by the experiment coefficient of friction between these materials is 0.56. The rest of material data used in the study are summarized in Tab. 1.
Of the many available research methods (field tests, bench tests, complete component tests, parts tests and simulation tests based on model – the advantages and disadvantages of each method are described more specific in a separate paper [28]), it was decided to perform simulation tests because of numerous benefits, e.g. low cost of research and short time of obtaining results [29-32]. Unfortunately, this method also has its drawbacks. It is necessary to develop a mathematical model that accurately reflects the actual physical phenomenon, otherwise the results may be loaded with a large error [33-35].

Simulation research was carried out using the finite element method. The COMSOL Multiphysics software was used for this purpose (geometry was imported from SolidWorks). The mesh imposed on the developed model consisted of about 3000 elements, mostly triangular shaped, which gave nearly 17000 degrees of freedom. It is commonly used software in the braking process phenomenon [36-40] and in many other features [41-43].

It has been assumed that in the research the initial speed of the shaft will be: 1800, 2000 and 2200rpm. This speed range is due to different versions of the rotary mower prototype design – the designers experimented with various gear ratios. The simulation will cover only the braking process, not including the brake system turning on time. It was also assumed that there will be no additional sources of forces and torques that would reduce the rotational speed (like contact with the ground). Braking will take place at an ambient temperature of 20 ºC, and the stopping time in each case is 1s. The diameter of cooperation is 100mm, and the width of the band is 20mm. Moreover, the following assumptions were made: unchanging coefficients of friction, stable and homogeneous contact pressure, homogeneous material of the brake pair and full surface contact, constant braking deceleration.

The retardation power of the braking process can be described as a negative derivative of the shaft’s kinetic energy [44]:

$$ P = -mR^2 \omega(t) \alpha $$

(1)

where  
- $m$ – shaft mass, kg;  
- $R$ – radius of the brake disc, m;  
- $\omega$ – angular velocity, RPM  
- $t$ – time, s;  
- $\alpha$ – angular deceleration, rad·s$^{-2}$.

Assuming that the braking deceleration is constant, we may say that:

$$ \omega(t) = \omega_0 + \alpha t $$

(2)

The retardation power is also expressed as the relation [45]:

$$ P = f_f(t) \cdot \int_0^t \omega(t) \cdot \alpha(t) \cdot r_m \cdot dA $$

(3)

where  
- $r_m$ – radius of friction force (in this case $R = r_m$), m.  
- $f_f$ – frictional force per surface unit;  
- $A$ – disc and pad contact surface area, m$^2$.

Comparison of the two equations makes it possible to determine the $f_f$ coefficient:

### Table 1

<table>
<thead>
<tr>
<th>Shaft material</th>
<th>Thermal conductivity</th>
<th>Density</th>
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<tbody>
<tr>
<td></td>
<td>47 W·(m·K)$^{-1}$</td>
<td>7870 kg·m$^{-3}$</td>
</tr>
<tr>
<td>Thermal capacity at constant pressure</td>
<td>498 J·(kg·K)$^{-1}$</td>
<td></td>
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<table>
<thead>
<tr>
<th>Band material</th>
<th>Thermal conductivity</th>
<th>Density</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>30 W·(m·K)$^{-1}$</td>
<td>4985 kg·m$^{-3}$</td>
</tr>
<tr>
<td>Thermal capacity at constant pressure</td>
<td>320 J·(kg·K)$^{-1}$</td>
<td></td>
</tr>
</tbody>
</table>
\[ f_f = - \frac{mR\alpha}{A} \]  \hspace{1cm} (4)

Assuming that the shaft deceleration occurs only through the action of the brake, the heat flux can be expressed using the equation [46]:

\[ q(r, t) = -f_f \cdot v_d(t) \]  \hspace{1cm} (5)

where \( v_d = \omega(t) \cdot r_m \) – brake disc linear speed at radius \( r_m \), m·s\(^{-1}\);

or

\[ q(r, t) = -\frac{mR\alpha}{A} r_m (\omega_0 + \alpha t) \]  \hspace{1cm} (6)

Amonton-Coulomb friction law [47, 48] makes it possible to determine the contact pressure, which in the analysed case is as follows:

\[ p = \frac{P}{\mu \cdot v} \]  \hspace{1cm} (7)

where \( \mu \) – coefficient of friction between the friction pair;

\( P \) – total braking force, N.

Also, the study takes into account the heat exchange occurring between the disc and pad, expressed as the following relation [49]:

\[ \rho \cdot C_p \frac{\partial T}{\partial t} + \nabla \cdot (-k \cdot \nabla T) = Q - \rho \cdot C_p \cdot u \cdot \nabla T \]  \hspace{1cm} (8)

where \( k \) – thermal conductivity, W·(m·K)\(^{-1}\);

\( C_p \) – thermal capacity, J·(kg·K)\(^{-1}\);

\( u \) – heat flux rate, W·m\(^{-2}\);

\( Q \) – heating power per density unit;

\( \rho \) – density, kg·m\(^{-3}\);

\( T \) – temperature, K.

During the action of the brakes, the following amount of heat is released through convection and radiation:

\[ q_d = -h(T - T_r) - \varepsilon \sigma (T^4 - T_r^4) \]  \hspace{1cm} (12)

where \( (T - T_r) \) – temperature difference between the friction material and ambient temperature, K;

\( h \) – convection coefficient, W·(m\(^2\)·K)\(^{-1}\);

\( \varepsilon \) – emissivity of material;

\( \sigma \) – Stefan-Boltzman constant.

The relation between the convection coefficient and the linear speed of the brake disc at the radius \( r \) is described as follows:

\[ h = \frac{0.037k}{l} \left( \frac{\rho \cdot l \cdot v}{\mu_v} \right)^{0.8} \left( \frac{C_p \cdot \mu_v}{k} \right)^{0.33} \]  \hspace{1cm} (13)

where \( \mu_v \) – viscosity, Pa·s;

\( l \) – shaft diameter, m.

**Results and discussion**

The direct result of simulation tests is temperature data of the tested elements. The measuring points were set 0.1 mm below the contact surface in the geometrical centre of the cooperation area. The temperature profiles of the brake disc and pad are shown in Fig. 3 and Fig. 4. Also, dissipated heat was calculated (Fig. 5). In addition, the amount of energy generated in the braking process was estimated for each case (Tab. 3).
The obtained data were compared with the results of similar works by other researchers [50-52]. Although the results were presented in a slightly different form, the values are proportional. This allowed to assess the correctness of the research.

Fig. 3. **Temperature profile of brake disc**: 1 – initial speed 1800 rpm; 2 – initial speed 2000 rpm; 3 – initial speed 2200 rpm;

Fig. 4. **Temperature profile of brake band**: 1 – initial speed 1800 rpm; 2 – initial speed 2000 rpm; 3 – initial speed 2200 rpm;

Fig. 5. **Dissipated heat**: 1 – initial speed 1800 rpm; 2 – initial speed 2000 rpm; 3 – initial speed 2200 rpm;
Table 3

<table>
<thead>
<tr>
<th>Initial speed, rpm</th>
<th>Generated energy, W</th>
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</thead>
<tbody>
<tr>
<td>1800</td>
<td>312.81</td>
</tr>
<tr>
<td>2000</td>
<td>386.87</td>
</tr>
<tr>
<td>2200</td>
<td>468.72</td>
</tr>
</tbody>
</table>

Analyzing the obtained results, it can be seen how little energy is needed to stop the shaft in question, and how low temperature rise is generated on the friction elements of its brake. At measuring points, for all considered cases it increased by less than 1K. The initial speed difference did not significantly affect the results obtained. Between the lowest and highest initial speed, the difference in the brake temperature is only ~0.3 °K. The charts also show characteristic “breaks”. They are a result of different properties and volumes of interacting materials. The heat dissipated graph shows that heat is given off the fastest during the braking process. After its completion, it stabilizes in each case studied.

It should be remembered that in the most cases brake elements are made of a composite material, which resin is the matrix. It is a material that has limited resistance to heat load. Depending on the type of resin, damage to its structure may occur when heated to 400-500 °C [53; 54]. The weakest of them burn already at 260 °C (533 °K) [55]. In the studied cases, the temperature increase is so low that it will not cause a threat of material damage, and even less the phenomenon of fading. Mechanical stress may be more significant, however, it was not the subject of this study.

Conclusions

Braking systems of all kinds are extremely important for the safety of users. The health or even life of many people may depend on their correct functioning. It is also very important to correctly design the brakes, which should not overheat. Too high temperature can lead to the fading phenomenon, with can cause a COF value drop, even to zero. This study proposes a methodology for simulation research of the working elements of agricultural machinery braking systems heating up process. Based on this method several different cases with different initial shaft rotation speeds were analysed. A number of simplifying assumptions were made to avoid unnecessary mesh compaction and to reduce the time needed to perform a single test.

As the result of the tests it was found that:
1. Development of a mathematical model requires knowledge of a number of facts, both geometrical of the examined element, as well as related to the considered phenomenon.
2. The performed simulation tests based on the developed model allowed to measure the temperature of the braking system components at any time and at any point.
3. The highest temperatures were achieved when braking from 2200 rpm. Temperature rise is 0.76 °K on the shaft brake disc surface, and 0.9 °K in the band measuring point.
4. The degree of heating is low for all considered cases, so there is no risk of overheating and thermal damage to the material.

The obtained results show that the examined peripheral brake has been designed with a high safety margin. Therefore, it can be successfully used to brake much heavier shafts or rotating with higher speeds.

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