CAR BRAKE SYSTEM ANALYTICAL ANALYSIS

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Car brakes, braking system, brake analysis.

ABSTRACT
This paper contains numerical analysis of brake system for heavy (mass m=6400[kg]) transport car. Analysis was performed in order to correct existing non optimal brakes in mentioned car. Analysis was based on results of the brake system dynamic tests made in Landing Gear Laboratory Institute of Aviation in Warsaw, Poland. Authors describe analytical process which led to generate results for new parameters of more efficient braking system for heavy transport car.

1. INTRODUCTION
In 2009 Institute of Aviation Landing Gear Department was asked for redesigning brake system for heavy transport car. Car brake system was not efficient enough and too costly in maintenance because of extensive wear of braking shoes (rear brake) and braking pads (front brake).

Analyzed car brake system used two types of brakes: rear drum brakes and front disc brakes. Such configuration of brakes is common but not very efficient due to lower efficiency of drum brakes in general. Another problem was that brakes weren’t made especially for this car but were taken from another car which was similar in parameters but had much lower nominal mass.

Landing Gear Department was supplied with some brake parameters (made by car testing facility) but full characteristics of brakes were unknown. Landing Gear Department has its own laboratory equipped with test stand capable to perform tests of brake systems.

Based on results from car test facility new set of tests were made in Landing Gear Laboratory in order to achieve full characteristics of existing brakes. Obtained results were used in calculation for new brake system. Calculations assumption was made that disc brakes will be used on both axles.

Below chapters shows analysis and calculations which led to generate parameters for improved brake system for heavy transport car.

2. ANALYSIS AND CALCULATIONS

2.1. Preliminary Data

Table 1. Used designations

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Unit</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>Q</td>
<td>N</td>
<td>Vehicle weight</td>
</tr>
<tr>
<td>F_p</td>
<td>N</td>
<td>Front axle static reaction (for both wheels)</td>
</tr>
<tr>
<td>F_t</td>
<td>N</td>
<td>Rear axle static reaction (for both wheels)</td>
</tr>
<tr>
<td>F_h</td>
<td>N</td>
<td>Braking force (four wheels)</td>
</tr>
<tr>
<td>l</td>
<td>mm</td>
<td>Wheel base</td>
</tr>
<tr>
<td>h</td>
<td>mm</td>
<td>Center of gravity to ground distance</td>
</tr>
<tr>
<td>a</td>
<td>mm</td>
<td>Front axle to center of gravity distance</td>
</tr>
<tr>
<td>b</td>
<td>mm</td>
<td>Rear axle to center of gravity distance</td>
</tr>
<tr>
<td>m_s</td>
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<tr>
<td>m_1</td>
<td>kg</td>
<td>Vehicle mass for front axle</td>
</tr>
<tr>
<td>m_2</td>
<td>kg</td>
<td>Vehicle mass for rear axle</td>
</tr>
<tr>
<td>g</td>
<td>m/s^2</td>
<td>g force = 9,81</td>
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Table 2. Preliminary Data

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<tr>
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<td>m</td>
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</tr>
<tr>
<td>F_p/F_t</td>
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<td>0,790</td>
</tr>
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2.2. Brake Selection Analysis

Let’s assume that brake is optimized for tyre-ground friction coefficient equal to μ=0.8 (see fig. 2) then:

\[ F_t = \frac{16720}{2} = 8360 \text{ [N]} \]
\[ F_p = \frac{46212}{2} = 23106 \text{ [N]} \]
\[ F_{lh} = \mu F_t = 6688 \text{ [N]} \]
\[ F_{hp} = \mu F_p = 18485 \text{ [N]} \]
\[ r = 0.426 \text{[m]} \]
\[ M_{lh} = F_{lh} r = 2850 \text{ [Nm]} \]
\[ M_{hp} = F_{hp} r = 8300 \text{ [Nm]} \]
\[ m = \frac{M_{lh}}{M_{hp}} = 2.91 \]

Let’s assume, that every braking pressure will give us constant braking moment ratio \( m = \text{const} \)

Kinetic energy of the vehicle with mass \( m_v = 6400 \text{ [kg]} \) and speed \( V=60 \text{ [km/h]} = 16.67 \text{ [m/s]} \) is equal to:

\[ E = \frac{m_v \cdot V^2}{2} = 888888.9 \text{[J]} \]

Second parameter is a braking distance. Vehicle braking test shown that braking distance from velocity \( V = 60 \text{ [km/h]} \) is equal to \( s_h = 30 \text{[m]} \)

For vehicle stop in desired distance (assuming constant decelerated motion) we need braking force equal to:

\[ F_h = \frac{E}{s_h} = 29629 \text{ [N]} \]

or total braking moment equal to:

\[ M = F_h r = 12622 \text{ [Nm]} \]

assuming that \( m = \frac{M_{lh}}{M_{hp}} = 2.91 \)

result is:

\[ M_{lh} = \frac{12622}{2*(1+2.91)} = 1614 \text{[Nm]} \]
\[ M_{hp} = 2.91*1614 = 4697 \text{[Nm]} \]

2.3. Braking Pressures Estimation

After front brake tests, it turned out, (despite first suggestions) that pressure in braking system can be around 3[MPa] and pressure needed to achieve braking moment at level of 3000[Nm] is around 9 \( \pm \) 10[MPa].

According to results of the tests for front and rear brakes distribution of the rear and front braking moments can be checked in assumption that pressures in both front and rear brakes are equal. Initial parameters are taken from previous chapter

Vehicle energy at the start of braking process:

\[ E_v = \frac{m V^2}{2} = 888888.9 \text{[Nm]} \]

All wheels braking force:

\[ F_{ch} = \frac{E_v}{s_h} = 29629.63 \text{[N]} \]

According to dynamic tests linear interpolation of front axle braking moment (disc brake) versus braking pressure was made:

\[ M_{hp} = 342.01 \cdot p_{hp} + 28,329 \text{[Nm]} \]
\[ y = 342.01 \cdot x + 28,329 \]
For rear axle drum brake, two braking moments versus braking pressure two interpolations were made. One is linear as in the case of front brake while the second is non linear. Such analysis was made because of non linear drum brake $M_s(p_h)$ relation caused by drum brake operating principle.

**Linear interpolation:**

$$F_{hpir}=\frac{342.01*p_h+28,329}{R_{kp}} [N]$$

**Non-linear interpolation:**

$$F_{hpir2}=\frac{8,0975*(p_h)^2+127,92*p_h-44,657}{R_{kt}} [N]$$

Overall braking force is equal to:

For linear interpolation:

$$2*\left(\frac{342.01*p_h+28,329}{R_{kp}}\right)+2*\left(\frac{127,92*p_h+44,657}{R_{kt}}\right)-F_{hpir}=R=0$$

$$p_h=11,655[\text{MPa}]$$

For non-linear interpolation:

$$2*\left(\frac{8,0975*(p_h)^2+127,92*p_h-44,657}{R_{kt}}\right)-F_{hpir}=R=0$$

$$p_h=11,275[\text{MPa}]$$

**Summary**

Car (mass $m_c = 6400 [\text{kg}]$) braking from speed $V_s = 60 \left[\frac{\text{km}}{\text{h}}\right]$, will stop within $s_b=30[\text{m}]$ when braking pressure is equal to $p_h=11,655[\text{MPa}]$ (linear interpolation) or $p_h=11,275[\text{MPa}]$ (non linear interpolation) what gives us average braking pressure $p_h \equiv 11,5[\text{MPa}]$

**2.4. Braking With Constant Tyre-ground Friction Coefficient ($\mu = 0.8$)**

Aviation regulations (ex. FAR, JAR) recommend taking $\mu = 0.8$ tyre-ground friction coefficient for permissible side loads.

Where permissible loads are the ones which can be present during standard operation.

Let’s take an assumption that analyzed car uses friction tyre-ground coefficient $\mu = 0.8$

Rest of the parameters are the same as in previous chapters (repeated below for better overview).

$m_c = 6400 [\text{kg}]$ car mass

$V_s = 60 \left[\frac{\text{km}}{\text{h}}\right] = 16,66667 \left[\frac{\text{m}}{\text{s}}\right]$ car speed

$s_b=30[\text{m}]$ braking distance

$R_{kp}=0,426[\text{m}]$ front wheel radius

$R_{kt}=0,426[\text{m}]$ rear wheel radius

$E_c = \frac{m_c*V_s^2}{2} = 888888.9 \left[\text{Nm}\right]$ car energy in the beginning of braking

$F_{h_b} = 29629.63 \left[\text{N}\right]$ all wheels braking force

$p_h=11,5[\text{MPa}]$ average braking pressure from chapter 2.3
Braking forces and moments for one wheel in the actual brakes’ configuration.

\[
\begin{align*}
M_{hp05} &= 3961[Nm] \quad F_{hp05} = 9299[N] \\
E_{hp05} &= 278970[Nm] \\
M_{bh05} &= 2497[Nm] \quad F_{bh05} = 5862[N] \\
E_{bh05} &= 175860[Nm] \\
F_{hc05} &= 909820[Nm] - overall braking energy (4 wheels)
\end{align*}
\]

Difference:
\[
\frac{f_{hc05} - f_{hc}} {f_{hc}} * 100 = 2.4 [%]
\]

Derives from approximations during analysis and calculations. Such a difference is fully acceptable from engineering point of view.

Vertical loads on one axle for friction coefficient \( \mu = 0.5 \) (chapter 2.1. Preliminary Data.) are equal to:

\[
F_{p05} = 39295[N] \\
F_{p05} = 23636[N]
\]

For friction coefficient \( \mu = 0.8 \) vertical axis loads will be equal to:

\[
F_{p08} = 46211[N] \\
F_{p08} = 16720[N]
\]

Braking forces and moments for one wheel (\( \mu = 0.8 \)) are equal to:

\[
\begin{align*}
F_{hp08} &= 8485[N] \\
F_{bh08} &= 6688[N] \\
M_{hp08} &= 7874[Nm] \\
M_{bh08} &= 2849[Nm]
\end{align*}
\]

In redesigned brakes pressure will be the same as in previous version, main change will be the area of the brake pistons.

Pistons area coefficient will be:

\[
\frac{A_{new}} {A_{old}} = \frac{F_{hp08}} {F_{hp05}} = \frac{M_{hp08}} {M_{hp05}} = 1.99
\]

\[
\frac{A_{new}} {A_{old}} = \frac{F_{bh08}} {F_{bh05}} = \frac{M_{bh08}} {M_{bh05}} = 1.14
\]

Braking distances for tyre-ground friction coefficient \( \mu = 0.8 \)

\[
\begin{align*}
F_{hc} &= F_{hp08} * 2 + F_{bh08} * 2 = 50345[N] \\
V_s = 60 \frac{km} {h} &= 16,67 \frac{m} {s} \quad s_n = 17,66[m] \\
V_s = 100 \frac{km} {h} &= 27,78 \frac{m} {s} \quad s_h = 49,04[m]
\end{align*}
\]

Summary

For effective use of tyre-ground friction coefficient equal to \( \mu = 0.8 \) can be achieved by using \( p_0 = 11.5[Mpa] \) braking pressure for car \( m_p = 6400[kg] \) of mass, area of the braking pistons has to be multiplied by:

- Front brake 
  \[
  n_p * A_{new} = n_p * A_{old}
  \]

Rear brake 
  \[
  n_t = 1.15 \quad A_{new} = n_t * A_{old}
  \]

Comments

Friction coefficient \( \mu = 0.8 \) is required by aviation regulations. It is taken for calculations of airplane landing gears and during laboratory tests of landing gears it is proven that friction coefficient is no less than \( \mu = 0.8 \).

In some cases friction coefficient \( \mu > 1 \) is achieved during landing gear laboratory tests performed in Institute of Aviation Landing Gear Department. This is also proven by literature, for example by „Budowa samochodów Układ hamulcowe i kierownice” - A. Ręski Oficyna Wydawnicza Politechniki Warszawskiej 2004 [1].

3. SUMMARY

Based on dynamic analysis of the car - dissipated energy by the front axle brakes should be equal to \( E_p = 557940[J] \) and for rear axle brakes should be equal to \( E_p = 351720[J] \) when tyre-ground friction coefficient is equal to \( \mu = 0.5 \).

During dynamic test of the rear drum brake \( M_0 = 1750[Nm] \) was obtained with braking pressure \( pht = 9[Mpa] \). For tyre-ground friction coefficient \( \mu \approx 0.5 \) (or braking distance equal to 30m), braking moment taken from numerical analysis is equal to \( M_{hta} = 2567[Nm] \) while the same moment taken from test results non linear analysis should be equal to \( M_{hta} = 2497[Nm] \) for braking pressure equal to \( pht = 11.5[Mpa] \).

Due to 2 % less moment value obtained during dynamic tests compared to calculated during numerical analysis it can be assumed that original brake system was calculated for tyre-ground friction coefficient \( \mu \approx 0.5 \) (or braking distance equal to 30m). In this case front disc brake should generate braking moment equal to \( M_{hp} = 3961[Nm] \) what can be achieved by existing brake system with the braking pressure \( p_{hp} = 11.5[Mpa] \).

With constant braking pressure equal to \( p_{hp} = 11.5[Mpa] \) current area of the braking pistons (for one wheel) is equal to:

\[
A_p = 3040 \text{ mm}^2 \quad A_r = 314 \text{ mm}^2
\]

When tyre-ground friction coefficient is equal to \( \mu = 0.5 \) braking pistons area will not change and will be equal to:

\[
A_{0.5} = 3040 \text{ mm}^2 \quad A_{0.5} = 314 \text{ mm}^2
\]

Therefore when tyre-ground friction coefficient is equal to \( \mu = 0.8 \) braking pistons area will change and will be equal to:

\[
A_{0.8} = 6080 \text{ mm}^2 \quad A_{0.8} = 361,1 \text{ mm}^2
\]

New braking system will be optimized in order to make braking process more effective what can result with reducing braking distance from about 30 [m] to about 18 [m] (according to analytical data) what gives 41% improvement.
4. REFERENCES
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