CAR BRAKE SYSTEM ANALYTICAL ANALYSIS

Wojciech Kowalski, Zbigniew Skorupka, Rafał Kajka, Jan Amborski,
Institute of Aviation
02-256 Warsaw, Poland
E-mail: zbigniew.skorupka@ilot.edu.pl
wojciech.kowalski@ilot.edu.pl
rafal.kajka@ilot.edu.pl
jan.amborski@ilot.edu.pl

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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 lover 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

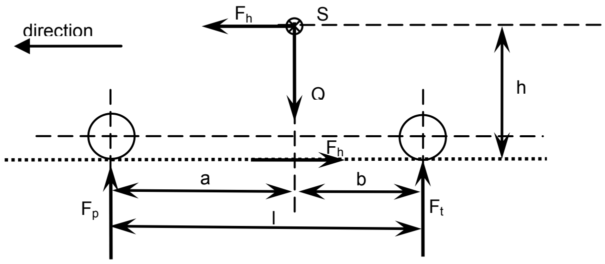

Figure 1. Car Load Distribution

Table 1. Used designations

Q	Vehicle weight	[N]
F_p	Front axle static reaction (for both wheels)	[N]
F_t	Rear axle static reaction(for both wheels)	[N]
F_h	Braking force (four wheels)	[N]
l	Wheel base	[mm]
h	Center of gravity to ground distance	[mm]
a	Front axle to center of gravity distance	[mm]
b	Rear axle to center of gravity distance	[mm]
m_s	Vehicle mass	[kg]
m_1	Vehicle mass for front axle	[kg]
m_2	Vehicle mass for rear axle	[kg]
g	g force = 9,81	[m/s^2]

Table 2. Preliminary Data

m	62931,2	[N]
m_1	27615,2	[N]
m_2	35316,0	[N]
h	1169	[mm]
a	1783	[mm]
l	3191	[mm]
Q	62931,2	[N]
a	1783	[mm]
h	1169	[mm]
b	1408	[mm]
F_p	27767,8	[N]
F_t	35163,3	[N]
r	0,426	[m]
F_p/F_t	0,790	

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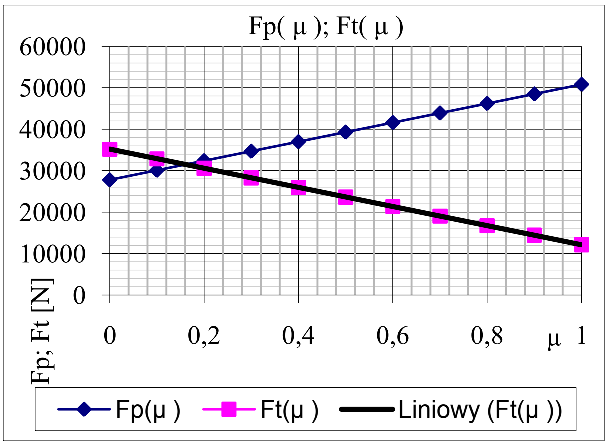

Figure 2. Dynamic Balancing of the Car

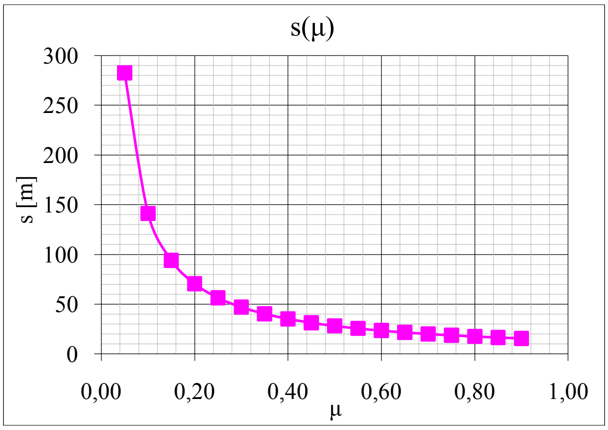

Figure 3. Braking Distance Versus Tyre-Ground Friction Coefficient

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:

$$F_t = \frac{16720}{2} = 8360\ [N]$$

$$F_p = \frac{46212}{2} = 23106\ [N]$$

$$F_{Ht} = \mu * F_t = 6688\ [N]$$

$$F_{Hp} = \mu * F_p = 18485\ [N]$$

$$r = 0{,}426[m]$$

$$M_{ht} = F_{Ht} * r = 2850\ [Nm]$$

$$M_{hp} = F_{Hp} * r = 8300\ [Nm]$$

$$m = \frac{M_{ht}}{M_{hp}} = 2{,}91$$

Let's assume, that every braking pressure will give us constant braking moment ratio m=const

Kinetic energy of the vehicle with mass m_s = 6400 [kg] and speed V=60 $\left[\frac{km}{h}\right]$=16,67 $\left[\frac{m}{s}\right]$ is equal to

$$E = \frac{m_s * V^2}{2} = 888888{,}9[J]$$

Second parameter is a braking distance.
Vehicle braking test shown that braking distance from velocity V =60 $\left[\frac{km}{h}\right]$ is equal to s_h=30[m]
For vehicle stop in desired distance (assuming constant decelerated motion) we need braking force equal to:

$$F_h = \frac{E}{s} = 29629\ [N]$$

or total braking moment equal to:

$$M = F_h * r = 12622\ [Nm]$$

assuming that

$$m = \frac{M_{Ht}}{M_{Hp}} = 2{,}91$$

result is

$$M_{Ht} = \frac{12622}{2*(1+2{,}91)} = 1614[Nm]$$

$$M_{Hp} = 2{,}91 * 1614 = 4697[Nm]$$

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 ÷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:

$$E_s = \frac{mV^2}{2} = 888888{,}9\ [Nm]$$

All wheels braking force:

$$F_{ch} = \frac{E_s}{S_h} = 29629{,}63[N]$$

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

$$M_{hpśr} = 342{,}01 * p_{hp} + 28{,}329[Nm]$$

$$y = 342{,}01 * x + 28{,}329$$

$$F_{hpśr}=\frac{342{,}01*p_{hp}+28{,}329}{R_{kp}}[N]$$

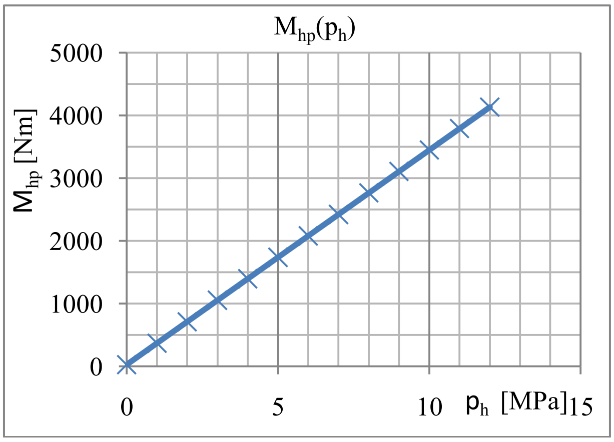

Figure 4. Front Axle Braking Moment Versus Braking Pressure – Linear Interpolation

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 $M_h(p_h)$ relation caused by drum brake operating principle.

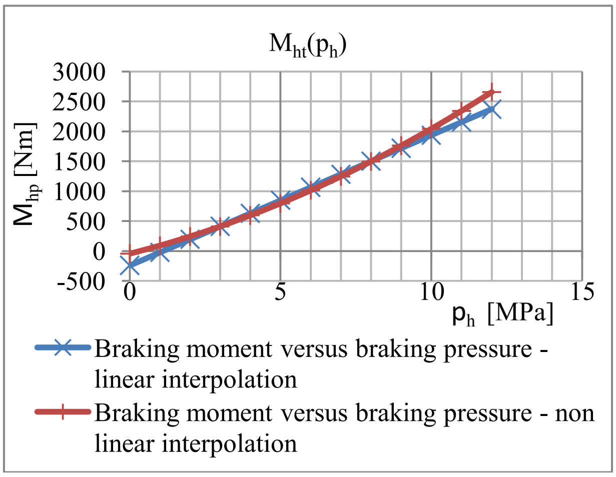

Figure 5. Rear Axle Braking Moment Versus Braking Pressure – Interpolations

Linear interpolation:

$$M_{htśr}=217{,}59*p_{ht}-239{,}2[Nm]$$

$$y=217{,}59*x-239{,}2$$

$$F_{htśr}=\frac{217{,}59*p_{ht}-239{,}2}{R_{kt}}[N]$$

Non -linear interpolation:

$$M_{htśr2}=8{,}0975*(p_{ht})^2+127{,}92*p_{ht}-44{,}657[Nm]$$

$$F_{htśr2}=\frac{8{,}0975*(p_{ht})^2+127{,}92*p_{ht}-44{,}657}{R_{kt}}[N]$$

Overall braking force is equal to:

For linear interpolation:

$$\frac{2*\left(342{,}01*p_{hp}+28{,}329\right)}{R_{kp}}+\frac{2*\left(217{,}59*p_{ht}-239{,}2\right)}{R_{kt}}-F_{hc}=R=0$$

$$R=0{,}731872718$$

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

For nonlinear interpolation:

$$\frac{2*\left(342{,}01*p_{hp}+28{,}329\right)}{R_{kp}}+\frac{2*\left(8{,}0975*(p_{ht})^2+127{,}92*p_{ht}-44{,}657\right)}{R_{kt}}-F_{hc}=R=0$$

$$R=1{,}978344$$

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

verification

$$F_{hc}=29629{,}62963$$

$$F_{hpśr}*2+F_{htśr}*2=29630{,}36153$$

Summary

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

2.4. Braking With Constant Tyre-ground Friction Coefficient (μ = 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).

$m_s=6400$ [kg] car mass

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

$s_h=30[m]$ braking distance

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

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

$E_s=\frac{mV^2}{2}=888888{,}9$ [Nm] car energy in the beginning of braking

$F_{ch}=\frac{E_s}{s_h}=29629{,}63[N]$ all wheels braking force

$p_h=11{,}5[Mpa]$ average braking pressure from chapter 2.3

Braking forces and moments for one wheel in the actual brakes' configuration.

$$M_{hpśr05}=3961[Nm] \quad F_{hpśr05}=9299[N]$$

$$E_{hp05}= 278970[Nm]$$

$$M_{htśr05}=2497[Nm] \quad F_{htśr05}=5862[N]$$

$$E_{ht05}= 175860[Nm]$$

$E_{hc05}=$ 909820[Nm] - overall braking energy (4 wheels)

Difference:

$$\frac{(E_{hc05}-E_s)}{E_s} * 100 = 2{,}4\ [\%]$$

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:

$$F_{p05}=39295[N]$$

$$F_{t05}=23636[N]$$

For friction coefficient $\mu = 0{,}8$ vertical axis loads will be equal to:

$$F_{p08}=46211[N]$$

$$F_{t08}=16720[N]$$

Braking forces and moments for one wheel ($\mu = 0{,}8$) are equal to:

$$F_{hpśr08}=\frac{F_{p08}*\mu}{2}=18485[N]$$

$$F_{htśr08}=\frac{F_{t08}*\mu}{2}=6688[N]$$

$$M_{hpśr08}=F_{hpśr08}*R_{kp}=7874[Nm]$$

$$M_{htśr08}=F_{htśr08}*R_{kt}=2849[Nm]$$

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:

$$n_p=\frac{A_{pnew}}{A_{pold}}=\frac{F_{hpav08}}{F_{hpav05}}=\frac{M_{hpav08}}{M_{hpav05}}=1{,}99$$

$$n_t=\frac{A_{tnew}}{A_{told}}=\frac{F_{htav08}}{F_{htav05}}=\frac{M_{htav08}}{M_{htav05}}=1{,}14$$

Braking distances for tyre-ground friction coefficient $\mu = 0{,}8$

Overall braking force

$$F_{hc}=F_{hpśr08}*2+F_{htśr08}*2=50345[N]$$

Braking distance for braking with start speed equal to $V_s=60\ \left[\frac{km}{h}\right]=16{,}67\left[\frac{m}{s}\right]$:

$$s_h=17{,}66[m]$$

Braking distance for braking with start speed equal to $V_s=100\ \left[\frac{km}{h}\right]=27{,}78\left[\frac{m}{s}\right]$:

$$s_h=49{,}04[m]$$

Summary

For effective use of tyre-ground friction coefficient equal to $\mu = 0{,}8$ can be achieved by using p_h=11,5[Mpa] braking pressure for car m_s = 6400 [kg] of mass, area of the braking pistons has to be multiplied by:

- Front brake

  $n_p=2 \qquad A_{pnew}=n_p*A_{pold}$
- Rear brake

  $n_t=1{,}15 \qquad A_{tnew}=n_t*A_{told}$

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 Układy hamulcowe i kierownicze" - A. Reński 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_{kp}=557940[J] and for rear axle brakes should be equal to E_{kt}=351720[J] when tyre-ground friction coefficient is equal to $\mu \approx 0{,}5$.

During dynamic test of the rear drum brake $M_{ht}\approx$1750Nm 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:

$A_p = 3040\ mm^2$, $A_t = 314\ mm^2$

When tyre-ground friction coefficient is equal to $\mu \approx 0{,}5$ braking pistons area will not change and will be equal to:

$A_{p0,5} = 3040\ mm^2$, $A_{t0,5} = 314\ mm^2$

Therefore when tyre-ground friction coefficient is equal to $\mu \approx 0{,}8$ braking pistons area will change and will be equal to:

$A_{p0,8} = 6080\ mm^2$, $A_{t0,8} = 361{,}1\ mm^2$

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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AUTHOR BIOGRAPHIES

WOJCIECH KOWALSKI, born in Warsaw, Poland and went to the Warsaw Academy of Technology, received his M.Sc. in Engineering in 1970, Faculty of Power and Aeronautical Engineering in Applied Mechanics and 2000 PhD from Warsaw University of Technology, Faculty of Automotive and Construction Machinery Engineering. Now he is working at Warsaw Institute of Aviation. His current professional area are: load and stress analysis, construction dynamics, test parameters determination, results analysis and interpretation of tests and calculations

E-mail: `wojciech.kowalski@ilot.edu.pl`

ZBIGNIEW KONRAD SKORUPKA, born in 1977 in Warsaw, Poland. Finished Warsaw Academy of Technology, where he studied Shaping Machines Design and Steering. He received his M.Sc. in 2004. Currently he is working at Landing Gear Department of Warsaw Institute of Aviation. His current professional area are: use of smart materials in landing gear, general design and testing of landing gear, brakes and test stands.

E-mail: `zbigniew.skorupka@ilot.edu.pl`

RAFAŁ KAJKA born in 1977, received his M.Sc. in Engineering (2000) from Warsaw University of Technology, (Poland). Directly after graduation he started his doctoral studies and in 2005 he received PhD from Warsaw University of Technology, Faculty of Automotive and Construction Machinery Engineering. Since 2001 he is working at Institute of Aviation, Warsaw initially as a strength engineer and from 2007 as a Manager of Landing Gear Department, Institute of Aviation, Warsaw, Poland.

E-mail: `rafal.kajka@ilot.edu.pl`

JAN AMBORSKI born in 1972 received his M.Sc. in Engineering in 1999 from Warsaw University of Technology, Faculty of Power and Aeronautical Engineering and 2006 PhD from Warsaw University of Technology, Faculty of Automotive and Construction Machinery Engineering. Now he is working at Warsaw Institute of Aviation.

E-mail: `jan.amborski@ilot.edu.pl`