Publishing House
AKAPIT

COMPUTER METHODS IN MATERIALS SCIENCE
Informatyka w Technologii Materiałów

Vol. 15, 2015, No. 1

NUMERICAL RESEARCH ON BRAKE CALLIPER PISTON'S WEIGHT REDUCTION

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Abstract

Under understanding the necessity of the weight reduction without decrease in strength properties of a construction part the authors have tried to declare the methodology of the parts' construction on the example of the brake calliper piston. Taking commonly used high strength aluminium wrought alloy of the 6xxx-serie the behaviour of the piston under plain compression load was investigated separately for elastic and plastic deformation. After that it was compared with the results for carbon steel C10. Under definition of the maximal allowed parameter's value for elastic and minimal allowed parameter's value for plastic deformations, represented by factor of safety (FoS) and yield stress respectively, the sensitivity maps of the proposed piston constructions were calculated. Opposite to the total topology optimization the outer geometry of the original piston was unchanged. The results of the study have shown that the changing of the inner geometry can be done in a certain range without any significant strength decrease of the construction part but not arbitrary.

Key words: aluminium wrought alloys 6061, carbon steel C10, brake calliper piston, construction optimisation, numerical simulation, FEM, QFORM 3D, T-Flex Analysis

1. INTRODUCTION

Braking systems is permanent developing system, which is responsible for the vehicle stop after human reaction and pushing the brake pedal or button. For different vehicle types it represented by different constructions. But only two main types of the breaks are widely exist, namely drum and disc brakes. Due to low weight, better heat dissipation and more durability under severe usage the disc brakes superior to drum brakes, that generally the state of art and described in many engineering books on the problem, e.g. by Heissing and Ersoy (2011), Gilles (2005). To protect the pressure fluid against overheating the calliper pistons have to be done regarding the material and geometry. Used in disc brake calliper pistons driven by fluid push the friction pads to clamp the rotor or brake disc during the movement of the brake pedal (figure 1). The generated heat will be dissipated inside the rotor and partly will be transmitted through the piston into the fluid. It causes the increase of the temperature inside the fluid volume and building the gas bubbles that prevent the movement of the brake pedal.

Today's market is overflowed with the pistons of different constructions, verified experimentally. There is no unique technique for construction of the optimal pistons due to distinguished driving mode of each car driver. And commonly there are only three main construction material groups with just a few representative materials per group, which are widely used through the numerous geometries of the pistons. These groups are grey cast iron, steel and aluminium. For flexible production more information about material properties regarding the application case is required. For example, the correlation between the geometry and density or weight of the piston is needed to calculate the generated and transmitted heat through the piston.

ISSN 1641-8581

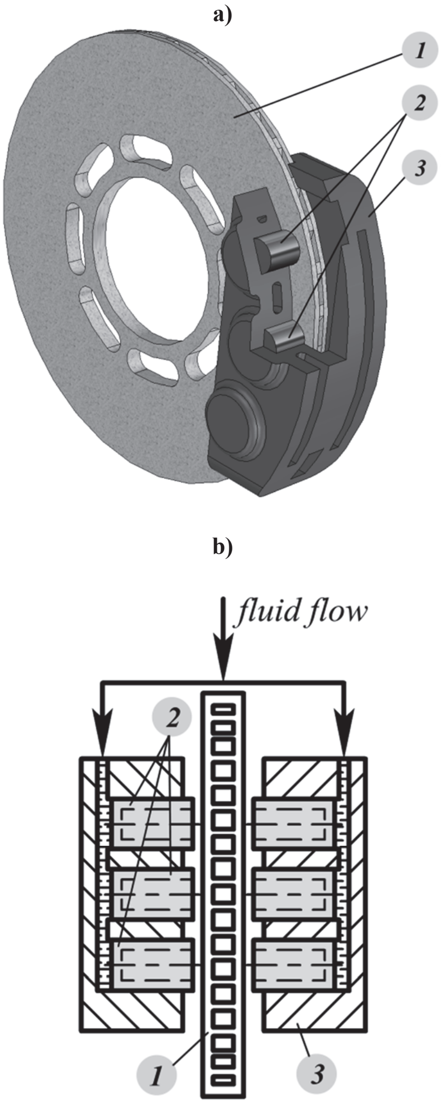

Fig. 1. *Fixed calliper: 1 – brake disc, 2 – brake pistons, 3 – calliper (hydraulic connections, bleed screws and friction pads are not presented). 3D view (a), cross-section (b).*

1.1. Fundamental ways of weight reduction of technical constructions

There are several methods in the optimization practice for making new construction parts. Among them the methods like topology optimization and lightweight materials' application. Topology optimisation means that non-relevant material volume will be selected and neglected during the production stage. It means that the pistons with thin walls, smaller radiuses and complex surfaces can be created. Lighter materials can be also used but without decrease in strength of the end product. The most popular materials stay carbon steels and aluminium wrought alloys of the 5xxx, 6xxx and 7xxx-series thanks their good till excellent strength properties and high corrosion resistance against aggressive brake fluids.

The application of new material needs to be technically feasible that can be achieved by development of new technological processes, for instance processes based on the cold bulk metal forming. Philippov et al. (2011, 2013), Gnevashev et al. (2003), Philippov and Molodov (2012) have shown the practicable realisation of the numerical simulation of the developed cold bulk metal forming operations and good agreement of the simulation results with the experiments for many part produced by cold bulk forming operations.

1.2. Construction for brake calliper pistons under study

In the ideal case the new optimized piston should have the reduced weight and the same or higher strength values. The authors tried to couple in one theoretical function two main influence parameters (arguments), i.e. geometry and material. Each argument was considered commonly and designated as a criterion in the following equation:

$$F = F(G, M), \tag{1}$$

where G – geometry criterion (G-criterion), M – material criterion (M-criterion).

Taking into account this fact and based on the two mentioned above methods the geometry and consequently the weight of the original mass produced pistons (figure 2, case A-1 and B-1) was changed. For the cases A-2 and B-2 the thick wall of the initial pistons was replaced by two thin walls. For the cases A-3 and B-3 the volume of the inner free space was increased. In both investigated modifications the replaced material volume was approximately the same (table 1). From the physical point of view two thin walls represented one connected shell and the induced during the brake operation stresses are closed inside it (compare A-1 vs. A-2). Stress distribution in one thin wall is comparable to one thick wall but with the correction on the ultimate material properties (compare A-1 and A-3). In principle the case A-2 can be converted into case A-3 then there is no gap between two thin walls of the piston in case A-2.

The number of investigated materials was restricted to two (one per main material group), namely aluminium alloy (EN AW-6061) and steel (carbon steel C10) with the chemical composition and mechanical properties presented in tables 2 and 3, respectively.

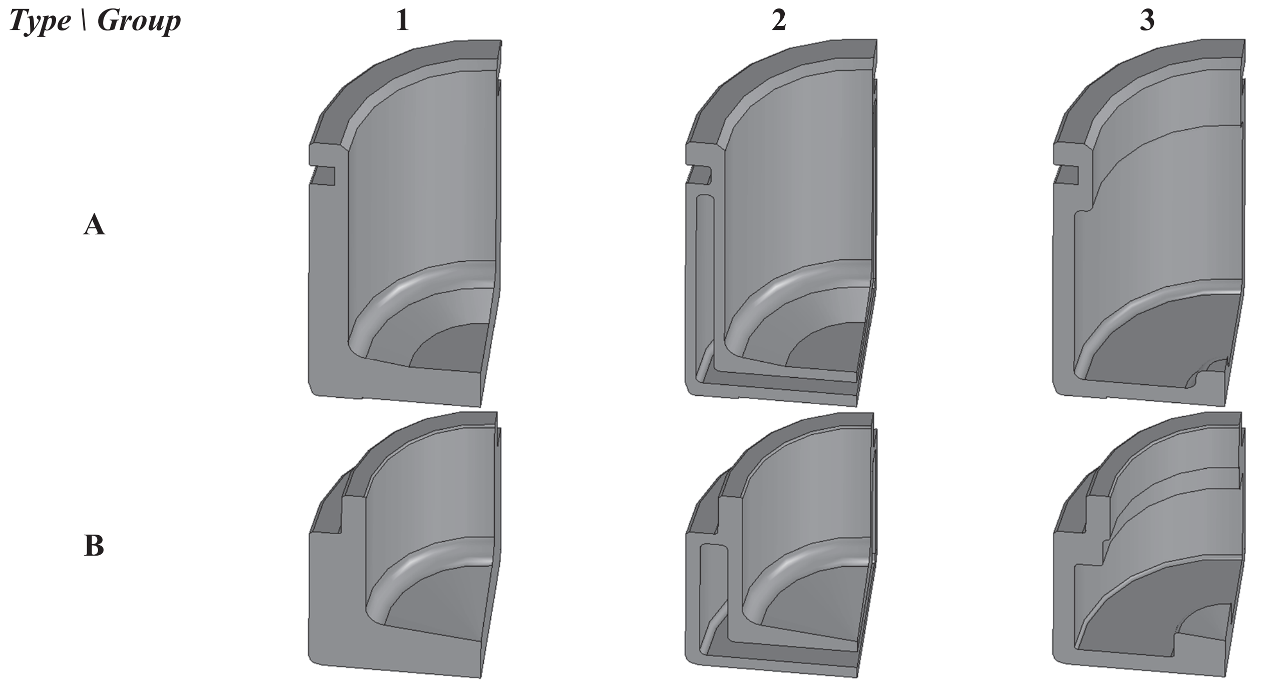

Fig. 2. *Classification of the brake calliper pistons incl. weight-reduced examples (A-1 and B-1 represent initial geometries of the real pistons).*

Table 1. *Comparison of the weight characteristics of the pistons.*

Variant	Volume, mm^3	Weight, kg	
		C10	AW-6061
A-1	42.050,8	0,328	0,118
A-2	26.800,2	0,209	0,075
A-3	27.411,2	0,214	0,077
B-1	35.003,9	0,273	0,098
B-2	21.412,3	0,167	0,060
B-3	21.321,1	0,166	0,060

Table 2. *Chemical composition of the materials (GOST 4784-97, DIN EN 573-3).*

carbon steel C10								
C	Si	Mn	Ni	S	P	Cr	Cu	As
0,07-0,14	0,17-0,37	0,35-0,65	max. 0,3	max. 0,04	max. 0,035	max. 0,15	max. 0,3	max. 0,08
aluminium wrought alloy EN AW-6061								
Fe	Si	Mn	Cr	Ti	Cu	Mg	Zn	Al
max. 0,7	0,4-0,8	max. 0,15	0,04-0,35	max. 0,15	0,15-0,4	0,8-1,2	max. 0,25	rest

1.3. Loading schemes

Two loading schemes were applied separately to the top surface: one to perform the static simulation in T-Flex Analysis (the problem is solved only in scope of Hooke's law by the system of linear algebraic equations) and one to perform the simulation of the plastic deformation in QForm 3D. To check the construction reliability of each case, presented in figure 2 the maximal (in elastic area) and the minimal (in plastic area) values of deformation stresses or forces have to be determined.

Firstly the material of the piston has to withstand the middle force value of 10.000 N under uniaxial compression stresses, which are normally exists on the working surface rotor-friction pad with piston (figure 3) and are necessary boundary condition for elastic problem formulation.

The stress-strain state of the piston's material during the brake pedal activation is quit the same as for the classical compression test except the temperature fields, which are induces by severe friction between the rotor represented by brake disk and pads, and small friction on the contact surfaces, which can be neglected. Secondly, if the material of the investigated piston obtains plastic deformation or stays close to that transient point, i.e. near to the yield stress point, the stresses increase inside the material rapidly. As a result the factor of safety (FoS) against equivalent stresses tends to zero, follows from the equation (2) and (3). Additionally the authors specified a construction coefficient through equation (4), which should obviously correctly describe the dependency between critical deformation force and FoS.

$$K_{fs} = \frac{[\sigma]}{\sigma_{eq.}} \tag{2}$$

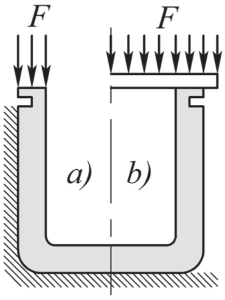

Fig. 3. *Loading schemes with restrains: loading in T-Flex Analysis (a) and loading in QForm 3D (b).*

Table 3. *Mechanical properties of the materials (databases of the QForm and T-Flex).*

Parameter	EN AW-6061		C10	
	T-Flex	*QForm 3D*	*T-Flex*	*QForm 3D*
Young module, GPa	69	-	210	-
Poisson ratio	0,33	-	0,28	-
Thermal conductivity, W/(m K)	170	250	43	28
Linear thermal expansion coefficient, 10^{-5} 1/K	2,4	-	1,3	-
Density, kg/m^3	2.700	2.800	7.800	7.550
Flow stress, MPa	55,15	table function	220,6	table function
Ultimate stress, MPa	124,1	-	399,8	-
Heat capacity, J/(kg K)	1.000	1.230	440	649

$$\sigma_{eq.} = \frac{1}{\sqrt{2}}\sqrt{(\sigma_x - \sigma_y)^2 + (\sigma_y - \sigma_z)^2 + (\sigma_z - \sigma_x)^2 + 6(\tau_{xy}^2 + \tau_{yx}^2 + \tau_{xz}^2)} \tag{3}$$

$$K_c = K_{fs} \cdot F_d \tag{4}$$

where K_{fs} – factor of safety against equivalent stresses; $\sigma_{eq.}$ – equivalent stress value, which is calculated from the stress tensor components; $[\sigma]$ – allowed stress value, which was equal to the flow stress of the material during the calculation, K_c – construction coefficient, F_d – deformation force.

If the same factor is equal to minimal (recommended) allowed value of 1 – 1.5 and is more higher the material works properly in elastic area and the construction is robust. For ultimate deformation, i.e. the situation, which can occur during the staff clamping of the friction pads, each piston was loaded according to the material's flow law and the maximal obtained force, which resolves stresses to perform plastic deformation, was determined on the first calculation steps.

2. NUMERICAL SIMULATION

Firstly, the elastic problem was numerically solved in finite element (FE) code T-Flex Analysis. It deals with the isotropic material with a specified in program's database mechanical properties. Under isotropic material the material with an invariant with respect to orientation of the body in the space, i.e. identical properties in all directions, has to be understood. According to the loading schemes showed in the figure 3a full restraints were assigned to the bottom and side surfaces of the piston.

Following the stress-strain diagram the simulation runs were continued by the solving of the plastic problem, which was done with the help of the FE commercial code QForm v.7. It is based on flow formulation, with independent variables, represented by velocity vector and mean stress. In rigid-visco-plastic model the material is considered as incompressible, isotropic continua, whereby elastic deformations are neglected. The restrains were applied only to the top and bottom surfaces and represent the piston contacts with both tools (figure 3b).

2.1. Process setup

The simulation runs were done for normal environmental conditions, presented in the table 4. For investigation the material behaviour near yield stress point the hydraulic press was chosen. To eliminate the influence of the friction during the simulation the friction factor was set to zero.

Table 4. *Process parameters (for QForm 3D and T-Flex Analysis).*

Parameter	Value
Tools' temperature, °C	20
Environmental temperature, °C	20
Workpiece temperature, °C	20
Friction factor	0
Press nominal capacity, MN	1
Nominal ram velocity, m/s	0,001

Table 5. *Information about the number of finite elements/nodes.*

Case	T-Flex Analysis	QForm 3D	Case	T-Flex Analysis	QForm 3D
A-1	90.457/139.372	5.872/11.746	B-1	61.906/95.774	5.426/10.854
A-2	143.532/224.750	11.922/23.846	B-2	73.950/120.335	9.316/18.634
A-3	97.460/152.737	7.954/15.910	B-3	190.549/288.182	12.462/24.926

2.2. Mesh preparation

All models were discretised by finite-elements generated automatically, whereby meshes based on the 4-node tetrahedral elements in QForm and on the 10-node tetrahedral elements with curved sides in T-Flex Analysis were prepared. Moreover finite-elements in T-Flex were generated automatically with global size propagation factor, which refers to the speed control of the mesh variation from reduced-size mesh cells to large cells of the general size. Its default value of 1 was accepted. This number means that the size of the element will double with each next calculation step until its size reaches the largest allowed mesh size. The generated mesh is not homogeneous and has clearly determined stacking places. During simulation this problem was corrected automatically just in QForm by a self-consistency algorithm of the program, that allows to increase mesh density locally for small materials volumes. Thanks to the internal algorithm of the program the number of the finite elements generated in T-Flex Analysis has not increased the calculation time.

3. RESULTS

As it was expected, the obtained simulation results have shown the dependency between the level of deformation force and geometry-material criteria. From the figure 4 it could be noticed that the pistons made from aluminium alloy did not reach the FoS of 1 – 1.5 and due to the construction recommendations such pistons could not be applied for real loaded brake systems. But also two representatives made from steel (A-2 and A-3) do not correspond this criterion.

Cases with the designations B-1 and B-3 are the most robust and trusted. They show that the piston with one thin wall (B-3) is much better than the piston with hollow area created by two thin walls (B-2), whereby the summarized wall thickness stays the same. Now analysing the maximal deformation forces, that can follow to the plastic deformation of the piston it is obviously that cases A-2 and B-2 cannot be applied due to high possibility of its plastic deformation during clamping stage and the A-3 case with low FoS can be applied conditionally because the construction is more rigid (figure 5). All investigated pistons' constructions made from aluminium alloy could not be recommended for application due to too low total elastic ability.

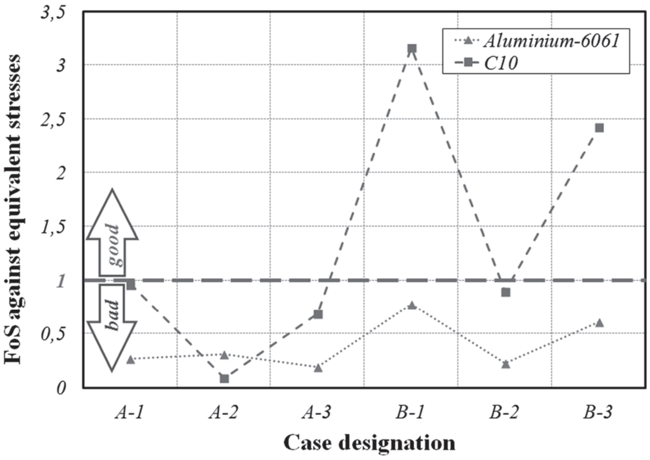

Fig. 4. *Minimal allowed value of the FoS (dashed horizontal line) against equivalent stresses show the fracture possibility of the piston (calculated in T-Flex Analysis).*

There were two sensitivity maps calculated (figure 6), that had shown two separate areas for correspondent material, whereby the first map (a) include only single function based on the elastic solution, since the second map (b) deals with both elastic and plastic solutions. It was found out that the same point of the diagram (e.g. B-2 for C10) changes its position and can move within the geometrical figure (circle or ellipse) of the certain radius.

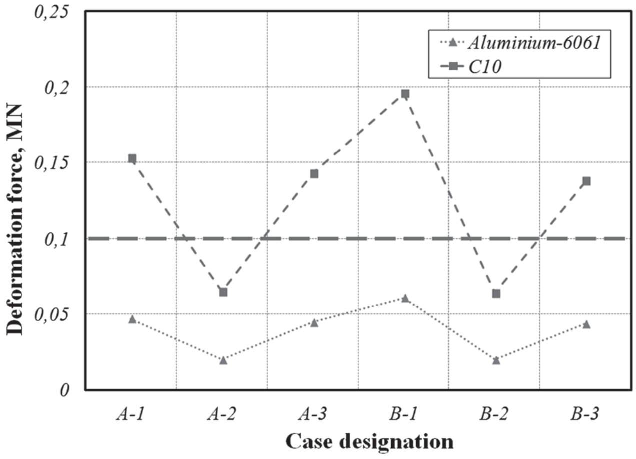

Fig. 5. Maximal allowed values of the deformation force (F_d) before the plastic deformation take place (dashed horizontal line); reference value of 0,1 MN corresponds the loading force value of 10.000 N applied during the elastic problem simulation (calculated in QForm v.7).

4. SUMMARY

In the carried out investigation the authors tried to complete the fundamental task on the coupling of the deformation force with geometry and material. Numerical simulation was performed in two programs to solve different tasks from one hand – on searching of the maximal allowed elastic stresses represented by the FoS against equivalent stresses and from the other hand – the minimal resolved stresses follows to the undesirable plastic deformation of the piston. Both results had shown quit similar results and strongly cut the aluminium alloy as not desirable at least for the investigated geometry. This tendency can be changed if the wall thicknesses of the aluminium piston will be increased. Moreover the small gaps between the correspondent points from the figures 4 and 5 for the A-2 and B-2 cases confirm such assumption although the investigated cases are outside of the allowed ranges. The proposed construction coefficient K_c is included into the M-criterion in equation (1) and can be the interface parameter between the elastic and plastic problem solution and also be used as a criterion for real constructions of the pistons for disc brakes in sense of the used materials.

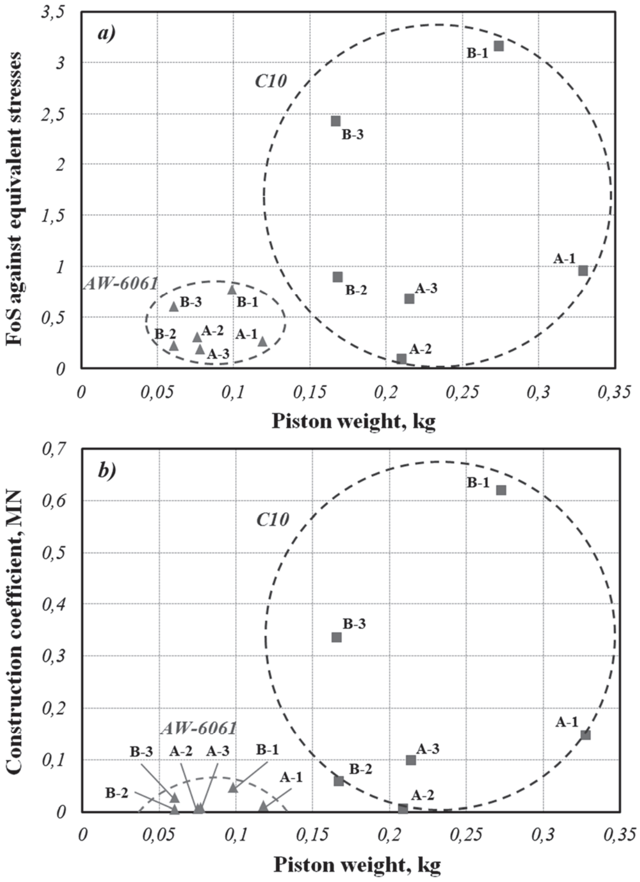

Fig. 6. Sensitivity maps of the piston's construction: one criterion (a), two criteria (b).

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WYKORZYSTANIE MODELOWANIA NUMERYCZNEGO DO BADAŃ NAD REDUKCJĄ WAGI TŁOKA W ZACISKU HAMULCA

Streszczenie

Rozumiejąc konieczność redukowania masy elementów części samochodu przy jednoczesnym zachowaniu ich własności wytrzymałościowych, autorzy pracy podjęli próbę opracowania metodologii konstrukcji tego typu elementów na przykładzie tłoku w zacisku hamulca. Do produkcji tłoka powszechnie stosowane są stopy aluminium z serii 6xxx, charakteryzujące się wysoką wytrzymałością. Stąd dla tego materiału przeprowadzono badania w płaskim stanie odkształcenia w zakresie odkształceń sprężystych oraz plastycznych. Następnie wyniki doświadczeń porównano z wynikami otrzymanymi dla stali węglowej C10. Na podstawie współczynnika bezpieczeństwa (FoS), definiowanego z wykorzystaniem maksymalnej dopuszczalnej wartości dla odkształcenia sprężystego i minimalnej dozwolonej wartości dla odkształcenia plastycznego, oraz granicy plastyczności, obliczono mapy wrażliwości dla zaproponowanych wariantów konstrukcji tłoków. W przeciwieństwie do zastosowania globalnej optymalizacji, zewnętrzny kształt oryginalnego tłoka pozostał niezmieniony. Wyniki przeprowadzonych badań pokazały, że zmiana wewnętrznego kształtu tłoka jest możliwa w pewnym zakresie bez znaczącego obniżenia własności wytrzymałościowych konstrukcji.

Received: October 5, 2014
Received in a revised form: November 7, 2014
Accepted: November 12, 2014