SZENT ISTVÁN UNIVERSITY

KINETICS OF DISC BRAKE CALIPER

Thesis of PhD work

Horváth Ádám Tamás

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Leader: Dr. Farkas István, DSc
Faculty of Mechanical Engineering
Szent István University
Gödöllő, Hungary

Supervisor: Dr. Kalácska Gábor, DSc
Institute of Mechanical Engineering Technology
Faculty of Mechanical Engineering
Szent István University
Gödöllő, Hungary

Co-supervisor: Dr. Oldal István, Phd
Institute of Mechanics and Machinery
Faculty of Mechanical Engineering
Szent István University
Gödöllő, Hungary

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Affirmation of head of school Affirmation of supervisors

CONTENTS

1. INTRODUCTION AND OBJECTIVES

Statistics show that vehicle companies manufactured more and more vehicles in the last few years. From 2001 to 2007 the number of cars increased, in 2007 there were 73 million cars manufactured, this number decreased and in 2009 the number of manufactured cars was 62 million. After the economic crisis the quantity of manufacturing increased again and by the end of 2014 there were 90 million cars produced. There are no cars without a brake system, which brake system can be of two types: disc brake and drum brake.

The applications of the disk brake motivate researches, because disc brakes are used in automobiles, freight vehicles and agricultural machines. The aim of these researches is to realize the optimal function of the given construction, in order to increase the lifetime or the performance. Most brake-researches examine the thermal and tribological behaviour, where the behaviour of the different friction materials was checked at high temperature.

In the examination various methods were used to check properties of the parts of the brake system. Sometimes a real system, sometimes a model was used to examine the properties of the parts. Nowadays we often use computer software in research because these programs are suitable to model the environment and can compare lots of different constructions.

Objectives

The aim of my research is to find different properties that might help us in future to optimize the working parameters and to increase the performance of the brake system.

1. The first step to examine the caliper opening is to find construction points that determine the allowable opening of the caliper; in this case the caliper stiffness is adequate. In this research caliper has more parts where some bolts contact different parts and the effect of preload of bolts was examined how it would change the caliper's opening.

2. The next step was to investigate the geometry of the piston to find values suitable to make the optimal design. Optimal wall thickness and optimal position of the top face was investigated.

3. As the last step, I examined the consistent pressure distribution on the friction surface of the brake pad to determine optimal diameter ratio in four pistons caliper. The aim of these cases is to increase the efficiency of brake system and increase the lifetime of brake pad.

2. MATERIAL AND METHOD

In this chapter the material and finite element models are shown. Three finite element models were used in this research (allowable deformation of caliper, optimal design of piston, optimal diameter ratio in four pistons caliper).

2.1. Properties of different parts of disc brake

Materials of different parts (caliper, disc, brake pad, and piston) were defined about the working parameter. The unsprung mass effects on the manoeuvrability, so there were very often low density materials used in the brake system. In this research the caliper material is aluminium alloy (7075T6) and bolts are M10 10.9 steel. The caliper' and bolts material properties are shown in Table 1.

Table 1. Properties of parts of caliper

Mechanical properties	Aluminium, 7075T6	M10 10.9 steel bolt
Yield strength	503 MPa	940 MPa
Modulus of Elasticity	71,7 GPa	220 GPa
Poisson ratio	0,33	0,3

In disc brake the other important part is/parts are piston/pistons that press the brake pad to the brake disc. Literature shows the three most popular kinds of material of pistons, so these materials (aluminium alloy, steel, titanium alloy) have been investigated. Table 2 shows properties of this tree pistons material.

Table 2. Properties of pistons

Physical and mechanical properties	Aluminium alloy (AlZn4.5Mg1)	Steel (S235JRH)	Titanium alloy (Ti6Al4V)
Density	2770 kg/m^3	7850 kg/m^3	4620 kg/m^3
Yield strength	280 MPa	251 MPa	930 MPa
Modulus of Elasticity	71 GPa	210 GPa	96 GPa
Poisson ratio	0,33	0,3	0,36

The friction elements (disk, brake pad) are the two most important parts of disc brake. Brake pad has two different parts: friction material and steel plate. The properties of material of brake pad and disc are shown in Table 3.

Table 3. Properties of brake pad and disc

Mechanical properties	Friction material	Steel plate of brake pad	Disc
Modulus of Elasticity	1GPa	210GPa	110GPa
Poisson ratio	0,25	0,3	0,28

2.2. The formation and validation of the model of the caliper

Caliper was designed and manufactured to examine the opening. Opening of caliper was checked in two ways. In one case manufactured caliper was used, in the other case finite element model was used to check the caliper's opening (Fig. 1.).

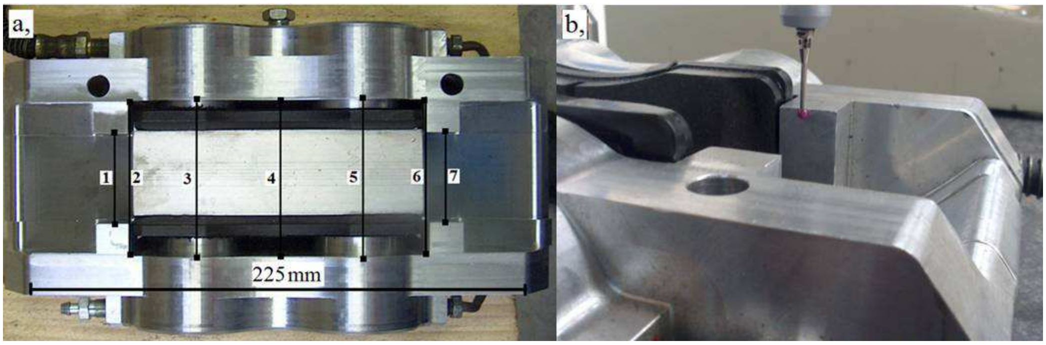

Fig. 1. Defining the opening of caliper: a) pistons of measuring locations; b) measuring on coordinate measuring machine

The opening at different locations on the manufactured caliper was defined from the statistical assessment of the three measurements (Fig. 2a). Fig. 2b shows the calculated opening results.

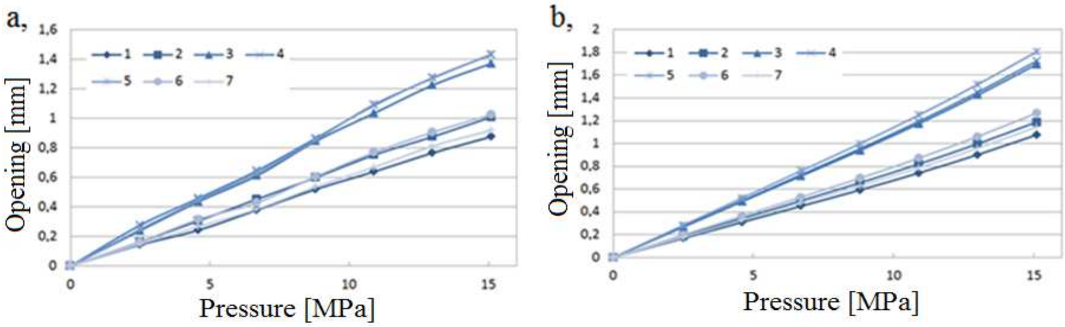

Fig. 2. Different cross-sections opening a) opening of manufacture caliper, b) Opening of FE model

The simulation results and measurement results were compared to check the difference between real-caliper and model, because the input parameters (material properties) of simulation come from technical hand book. The result of the opening rate was different in two cases (real-caliper, model) (Fig. 3a). The input parameter was changed (modulus of elasticity of aluminium alloy and bolts) and the opening rate became similar in both two cases (Fig. 3b).

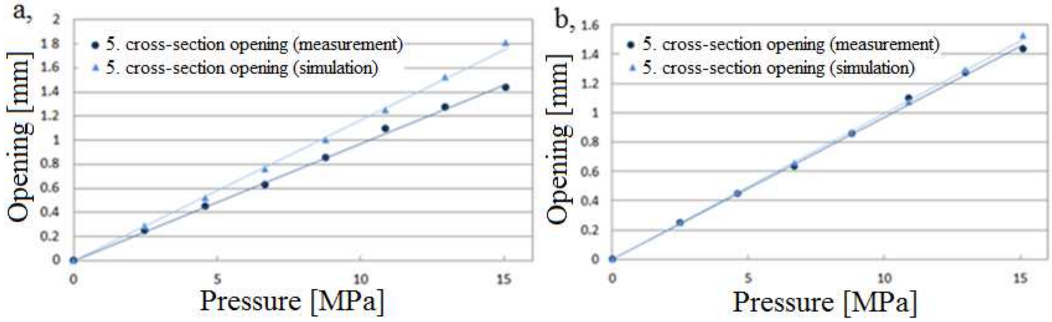

Fig. 3. Opening in 5th cross-section (measured, simulation)

All cross-section opening rates were defined in real and simulation caliper and functions gradient was compared. The difference between functions gradient in the first set up came to 13 - 20%. After changing of the input parameter the received opening result was similar and the difference between functions gradient was between 0% and 4%.

After the validation of the model, the total deformation of the caliper was defined and the position of the cylinder of pistons was examined.

2.3. Model of pistons of caliper

The investigation of piston geometry was carried out in finite element software. In this investigation there was a 2D model used where hydraulic pressure works in different surface depending on the sealing ring position (sealing ring in caliper (SIC), sealing ring in piston (SIP)). Fig. 4 shows the surface where hydraulic pressure works.

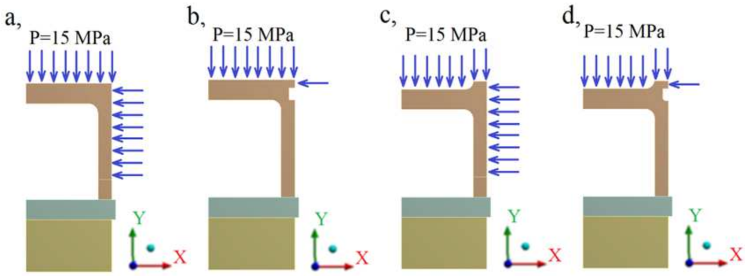

Fig. 4. Piston's surface where hydraulic pressure works in different cases

First, deformation of the piston's wall was checked. In this investigation a simple model was used where the wall thickness had been changed. Wall thickness was between 0.5 mm to solid piston in this examination.

Second, optimal top face position was examined where top face position had been changed to find out, which case gives the smallest deformation of the wall. In this case the wall thickness is constant (3.5 mm), the top face thickness is 5 mm. Fig. 5 shows how the top face position is changing from top to bottom.

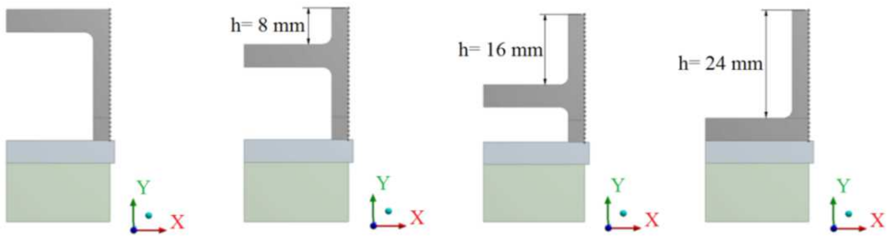

Fig. 5. Position (h) of top face

2.4. Model to define optimal diameters ratio

In order to find optimal diameter of pistons in four pistons caliper there was a simple model used. The model used to define the optimal diameter ratio is shown in fig. 6.

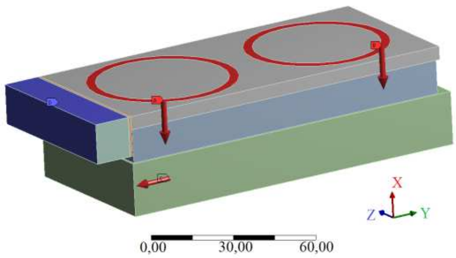

Fig. 6.Constrain and loads on the simple model

In this investigation the first piston's (P1) (according to a direction of rotation) diameter was observed between 32 mm and 44 mm, the second piston's (P2) (according to a direction of rotation) diameter was changed between 32 mm and 64 mm.

The pressure distribution was examined on friction surface. Pressure distribution was defined along 7 lines (13 points per line) and different constructions were compared to find optimal diameter of pistons (fig. 7). In this case friction coefficient between the brake pad and the caliper was changed to investigate how the diameter of pistons changes in different cases.

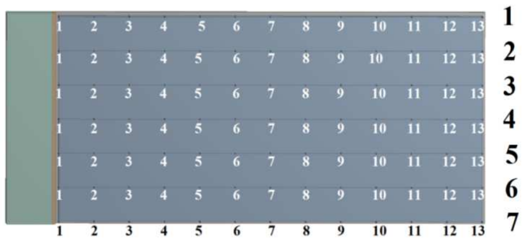

Fig. 7. Check pressure distribution on friction surface along 7 lines (13 points per line)

3. RESULTS

In this chapter results on the deformation of caliper was presented, optimal wall thickness of piston, optimal top face position of piston and optimal diameter ratio in four pistons caliper.

3.1. Defining the allowable deformation of caliper

The investigation of deformation of caliper under different pressures was used to check the degree of deformation. Fig 8. shows the caliper opening when pressure was changed between 0 and 15.1 MPa.

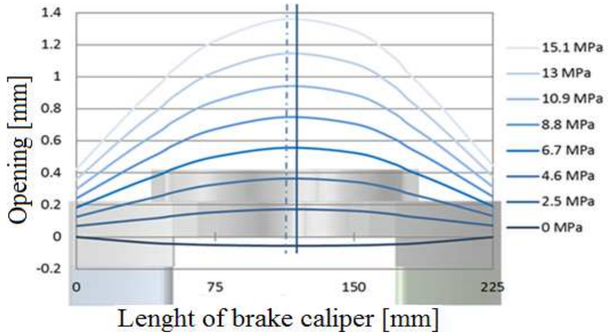

Fig. 8. Total cross-section deformation of caliper at different pressures

The positive and negative opening has an effect on the brake system's working and on the lifetime of the brake pad. The deformation of the caliper changes the cylinder bore of pistons in caliper and in critical case it changes the pistons optimal position. Cylinder bore position was check in *x* and *z* direction.

Pistons position and working parameters show what caliper opening is allowable or not. The finite element simulation shows how the cylinder of pistons changes in the caliper. Fig. 9a shows the cylinder bore degree in *x* direction and fig. 9b shows cylinder degree in *z* direction when different pressure was used.

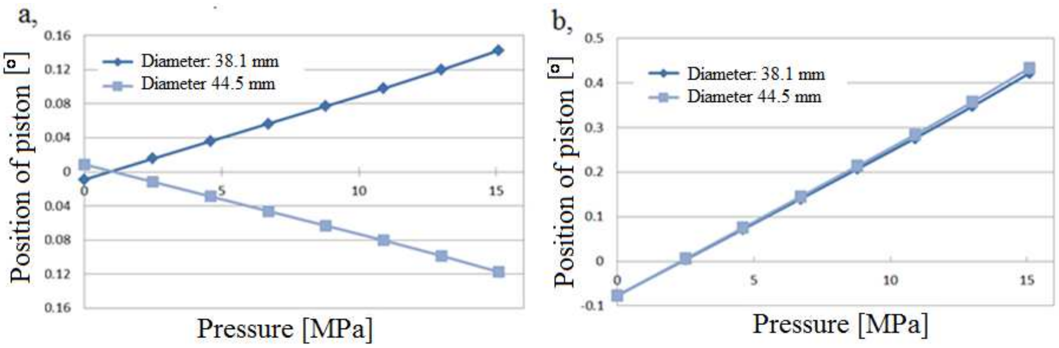

Fig. 9. Cylinder bore of piston bending angle in a) X direction, b) and in Z direction

Fig. 10 shows the allowable deformation. These diagrams show that bolts preload and hydraulic pressure effect the deformation. The results show that bolts preload makes negative deformation when hydraulic pressure does not work. When hydraulic pressure works, the caliper opens. The results show that the allowable opening (both pistons) is under 4.56 MPa.

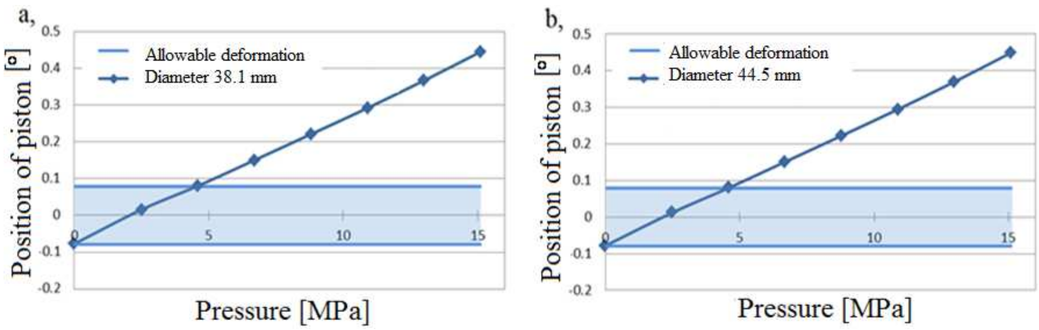

Fig. 10. Cylinder bore of pistons bending angle, a) diameter 38.1 mm, b) diameter 44.5 mm

Bolts preload effects caliper deformation, so caliper deformation was checked when bolts preload was changed. Preload of bolts is between 20 kN and 52 kN in agreement with the standard M10 10.9 bolts preload. The preload depends on the bolt's design and friction is coefficient. Fig. 11a shows the smaller piston's (diameter 38.1) angle when different preloads of bolt were used. Fig. 11b shows the bigger piston's (diameter 38.1) angle when different preloads of bolt were used.

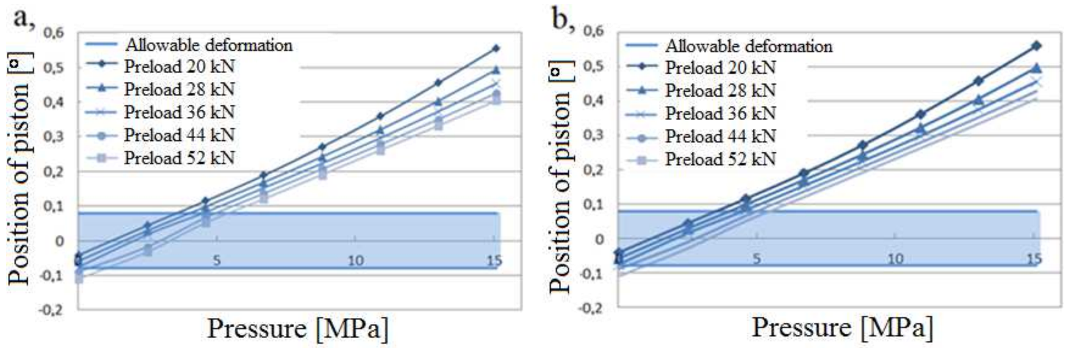

Fig. 11. Cylinder bore of pistons bending angle when different preloads of bolts were used, a) diameter 38.1 mm, b) diameter 44.5 mm

Results show that bigger preload of bolts eventually uses bigger hydraulic pressure. The bolt preload has a limit, because if cylinder bore of pistons bending angle is bigger than allowable, the pistons position is not optimal, that means the efficiency of brake system is low (for example small hydraulic pressure was used). The other board of limit is stress, the stress in bolt is not bigger than the yield strength.

3.2. Defining the optimal geometry of piston

3.2.1. Defining the optimal wall thickness of piston

After the investigation of the cylinder position, stress and wall deformation of piston was examined when hydraulic pressure affects piston. The sealing ring position effect to the stress and deformation so results were compared, what happened when sealing ring is in caliper or in piston.

In different cases the hydraulic pressure (15 MPa) gives various results. The stress becomes different if the sealing ring position and wall thickness is changed. Fig. 12 shows the cases where the stress is bigger then the yield stress in different kinds of material and different constructions.

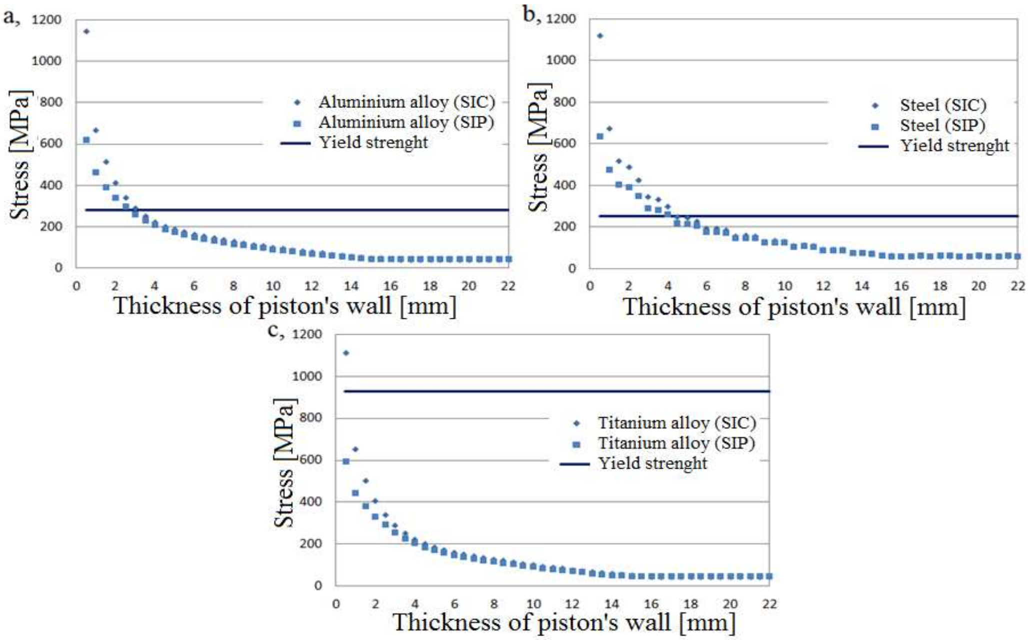

Fig. 12. Maximum stress in aluminium alloy (a), steel (b), titanium alloy (c) piston in the function of wall thickness

Stress is one of the most important parameter, the other one is the wall deformation of piston. The deformation has an effect on the working condition because in a critical case (when deformation is too big) the piston gets stuck into caliper and the braking force is not increasing if hydraulic pressure increases. The allowable deformation depends on the tolerance between piston and cylinder in caliper. The investigation of wall deformation is important to design a safe brake system.

The result of deformation demonstrates what kind of wall thickness gives suitable stiffness of wall (small deformation) and what kind of wall thickness is that top face makes a change on the deformation of wall. The results show how the deformation of the wall changes by changing the wall thickness. The deformation results (aluminium alloy, steel, titanium alloy) show the limit of wall thickness where the wall has stiffness and in what case the top face changes into deformation results.

Fig. 13 demonstrates how the deformation of wall changes if thickness of wall was changed when sealing ring is in caliper. Different parameters were made (ΔD/D and D/v) to compare different constructions. ΔD is deformed diameter, D is original diameter and v is wall thickness. Fig. 13a shows results of aluminium alloy, fig. 13b demonstrates deformation of steel and fig. 13c shows the results of titanium alloy.

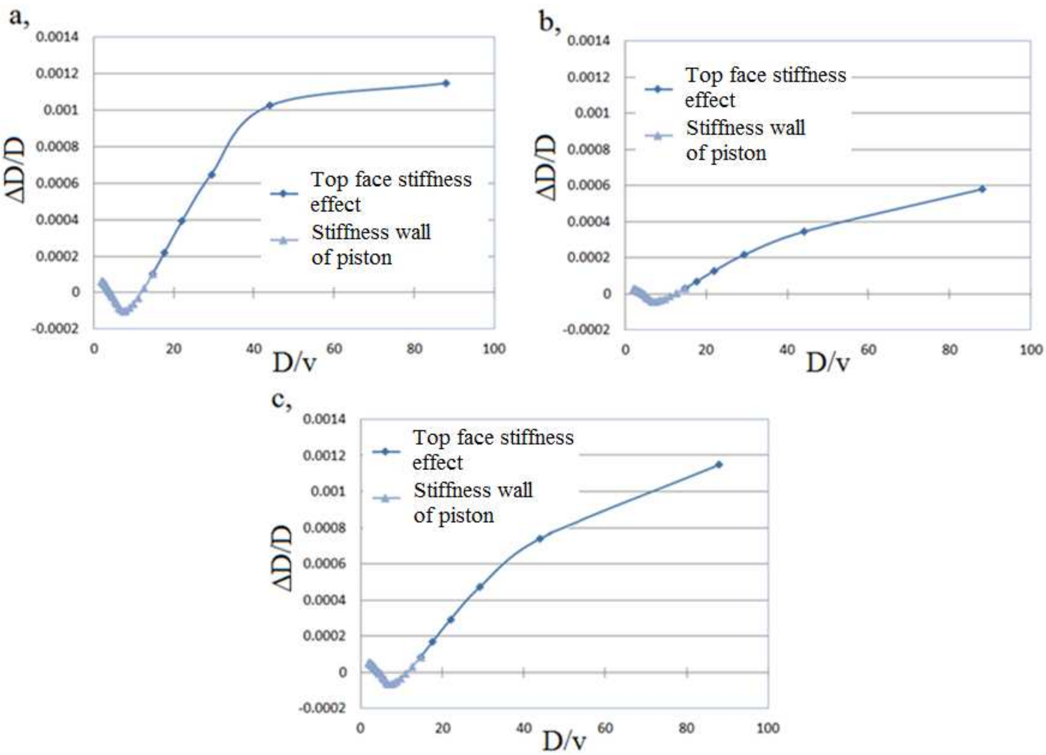

Fig. 13. Stiffness of wall if sealing ring is in caliper:
a) aluminium alloy, b) steel, c) titanium alloy

Fig. 14 shows the deformation results when sealing ring is in piston: a) aluminium alloy, b) steel, c) titanium alloy

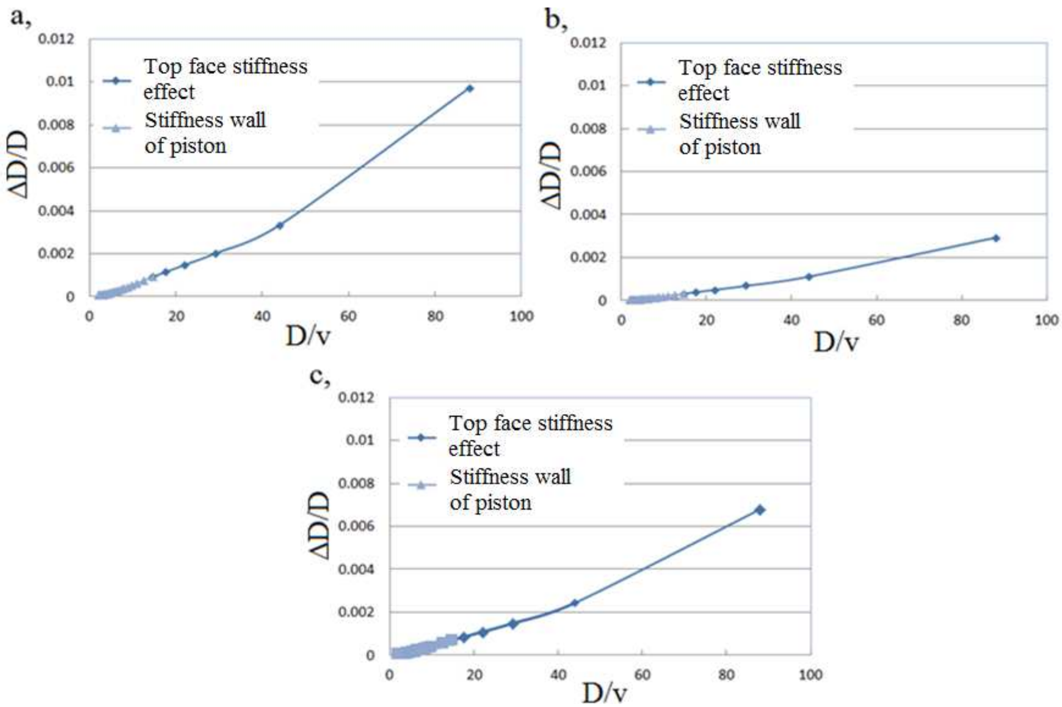

Fig. 14. Stiffness of wall if sealing ring is in caliper:
a) aluminium alloy, b) steel, c) titanium alloy

In different cases curves have two different parts. Various ways of behaviour cause different effects and two functions were approximated. The point where the two functions connect is the transition point between stiffness of wall and where the top face modifies the deformation results because wall stiffness is not suitable. The results show that this transition point is similar in all cases. The transition wall thickness is 3 mm that means that over 3 mm the wall has suitable stiffness, less than 3 mm wall is weak and top face increases the stiffness of wall.

Fig. 15 demonstrates the necessary wall thickness that defines stress and deformation. The yield strength and allowable deformation define the necessary deformation. Usually the stress determines the necessary wall thickness. Steel piston case has the biggest difference between the stress and deformation wall thickness results. In case of titanium alloy pistons we can see that deformation determined the necessary wall thickness.

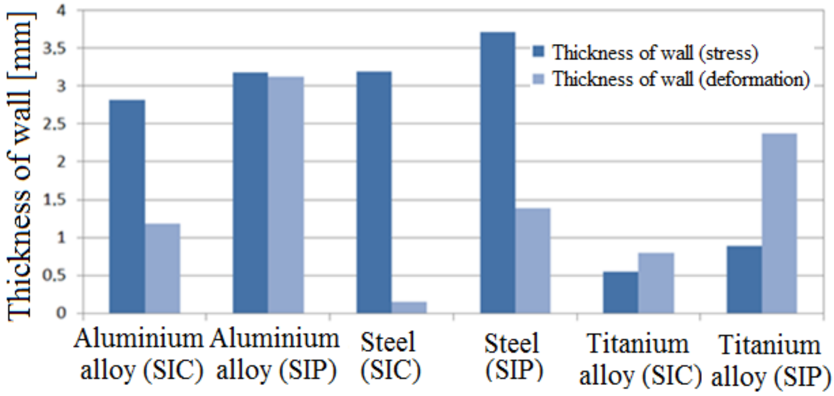

Fig. 15. Necessary wall thickness what define stress and deformation results

3.2.2. Defining the optimal top face position of pistons

When top face optimal position was defined aluminium alloy material was used to pistons. The optimal position of top face of piston is when wall deformation is the least one. In this investigation the wall thickness is 3.5 mm the thickness of top face is 5 mm. In this case two constructions (SIC, SIP) were investigated and the optimal position is different in the two cases.

The deformation results in different cases (SIC, SIP) determined the optimal position of top face, so in critical cross section (where deformation is the largest) *x* direction deformation was defined. The optimal position of top face is found where the *x* direction deformation of cross-section is the least one.

The result of the change of the piston's geometry seems to show that deformation is different in all cases and the measure of deformation depends on the sealing ring's position. This study demonstrates where the optimal position of the circular top face is. When the sealing ring is in the caliper the optimal position of top face is 20% of piston's length (Fig. 16a). In the other case, when sealing ring is in the piston the optimal position of top face is 30% of piston's length (Fig. 16b).

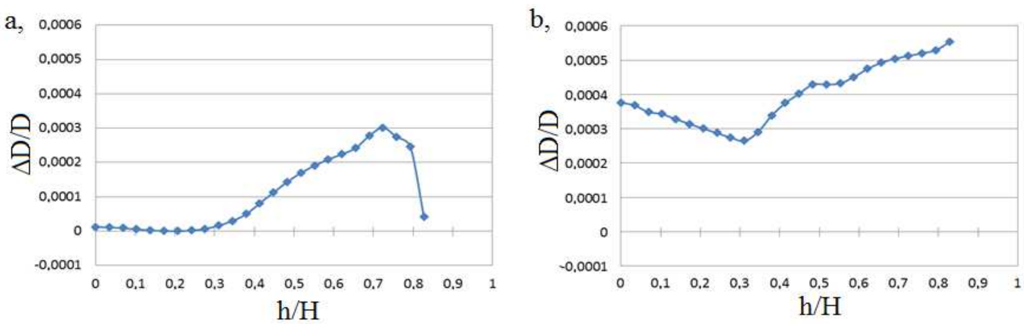

Fig. 16. Optimal construction when (a) sealing ring in caliper (SIC), (b) sealing ring in piston (SIP)

The results show that the optimal position is determined by the sealing ring's position. The deformation of wall defines the optimal top face position where deformation is the smallest.

3.3. Defining the optimal diameter ratio of pistons

In case of the brake pad the aim is that pressure distribution is consistent on the friction surface to increase performance of brake pad and extend the lifetime of the brake pad (equable wear). The size of diameter of pistons has an effect on endurance, so it is important to find the best construction. The best construction gives that wear of friction parts of brake becomes low and consistent. As a result of consistent pressure distribution performance and lifetime will increase.

In the following investigation four pistons caliper were examined to find the optimal diameter of pistons that would give homogenous pressure distribution on friction surface. That means two pistons on one side generate similar force that presses the brake pad to the brake disc. Different diameters of pistons and different friction coefficient were used to find the optimal diameter ratio. Pressure distribution was examined along several lines to compare different constructions (various diameters of pistons) to find optimal diameter ratio.

In this investigation optimal diameter-ratio was checked: how the optimal diameter ratio changes if friction coefficient was changed between brake pad and caliper. Fig. 17a shows how the pressure distribution (13 points) is modified on friction surface centre line of break pad when first piston according to the direction of rotation (P1) is 40 mm and second pistons according to the direction of rotation is between 40 mm and 64 mm. The friction coefficient between the caliper and the brake pad is 0.15. Fig. 17b shows the result (11 points) what was used to compare different construction. The first point and last point of the result are disparate because edge effect distorts results, so when different results were compared the first and last point results were omitted.

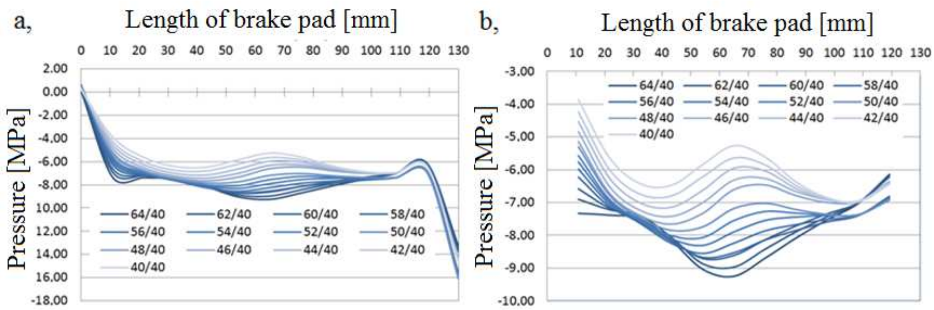

Fig. 17. Pressure distribution of brake pad: a) centre line pressure along the entire length (13 points); b, centre line pressure without outsides point (11 points)

The results show that different diameters of pistons give various pressure distributions on the centre line of friction surface of the brake pad. Figure of merit was made to compare the different construction to find the best diameter ratio where pressure distribution is consistent. Equation 1. helps to find the best construction that shows what diameters of ratio give consistent pressure on centre line of brake pad’s friction surface. This formula helps to define the relative standard deviation

$$V = \frac{s}{\bar{x}}, \quad (1)$$

where V is the relative standard deviation, s is scatter of pressure and $\bar{x}$ is the average of pressure.

Relative standard deviation was used to define figure of merit where the higher figure of merit shows which construction gives optimal wear of brake pad (Formula 2.):

$$Q = 1 - \frac{s}{\bar{x}}, \quad (2)$$

where Q is the figure of merit, s is scatter of pressure and $\bar{x}$ is the average of pressure.

Figure of merit was determined by all the designs (different diameter of pistons, different friction coefficients between brake pad and caliper). Figure 18. shows the figure of merit when different friction coefficients were used between brake pad and caliper ($\mu = 0.1$; $\mu = 0.15$ and $\mu = 0.2$). In allthree cases the results were similar and it is independent from friction coefficient. Results show that all cases have an optimal point where figure of merit is higher, that means pressure distribution is more consistent.

In this study friction coefficient between brake pad and caliper was investigated to check how the diameter of pistons changes if friction coefficient is changed. Figure of merit helps to find the optimal diameter ratio of pistons in different cases (friction coefficient: $\mu = 0.1$; $\mu = 0.15$ and $\mu = 0.2$) where pressure distribution is more consistent. Fig. 18 demonstrations figure of merit in different construction. All cases find the optimal point to compare how friction coefficient changes the optimal diameter ratio of pistons.

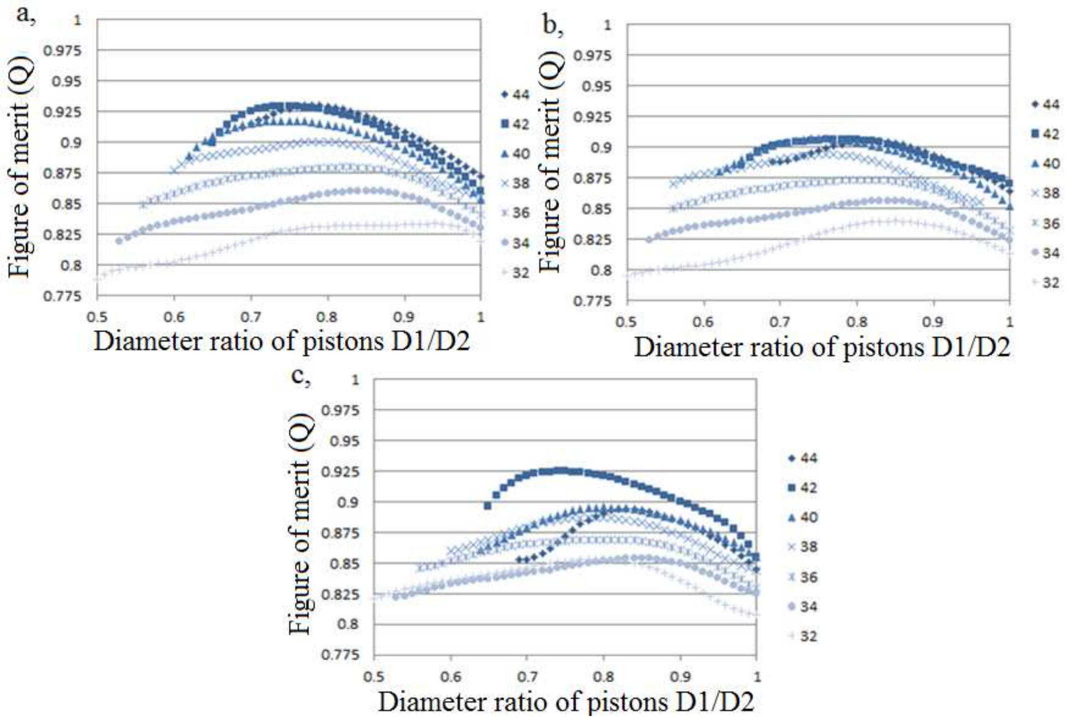

Fig. 18. Figure of merit in different cases, where diameter of pistons is different and coefficient of friction is different between brake pad and caliper: a) μ = 0.1; b) μ = 0.15; c) μ = 0.2

The friction coefficient does not change the optimal diameter ratio of pistons. Fig. 19 shows how the optimal diameter ratio (higher figure of merit) changes when different coefficient of friction (μ = 0.1; μ = 0.15; μ = 0.2) were used between brake pad and caliper. The average of diameter ratio of pistons is 0.805 where scatter was examined and the results (optimum point) are in scatter.

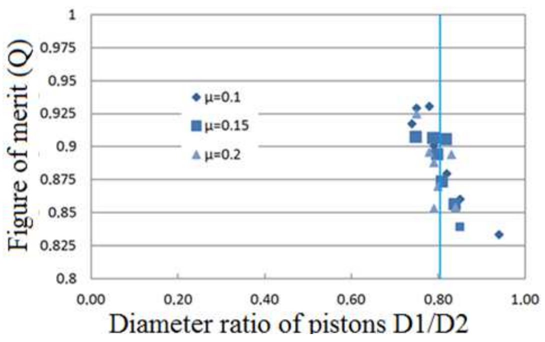

Fig. 19. Highest figure of merit in different case (Different diameter, different friction coefficient) which define the optimal diameter ratio

4. NEW SCIENTIFIC RESULTS

In this study the brake system of high performance cars was investigated where various parameters were defined. These new scientific results help engineers to design or/and optimize the brake system. Finite element models used in this research were identified and validated by measurements. My new scientific result is:

1. *Defining the allowable deformation of caliper*

   I defined the load limit of fix caliper where the opening of caliper changes the pistons position (not optimal position), that means pistons do not press the brake pad's total surface and the performance of brake system decreases. The effect of pre-load of bolts in caliper was determined, if the pre-load of bolts is increased, load limit of caliper will increase. The deadline of bolt's pre-load is if the caliper is unloaded, the piston does not change optimal position and total face presses the brake pad to the brake disc.

2. *Defining the optimal wall thickness of piston*

   I demonstrated limit of wall thickness of piston (not solid piston) where piston's wall is rigid and top face of piston has no effect on deformation of piston's wall. Limit of wall thickness depends on the sealing ring's position (sealing ring is in caliper, sealing ring is in piston).

3. *Defining optimal top face position*

   In case of 0.16 wall thickness-radius ratio certificates that top face optimal position depends on the sealing ring's position (sealing ring is in the caliper, sealing ring is in piston). I proved when sealing ring is in the caliper the optimal position of top face is 20 % of piston's length. In the other case, when sealing ring is in the piston the optimal position of top face is 30 % of piston's length.

4. *Method to define optimal diameters ratio of pistons*

   The pressure distribution made by the two pistons was independent from friction coefficient (friction of coefficient range is 0.1 to 0.2). I defined the optimal diameters ratio of pistons, this optimal diameter ratio is independent from friction coefficient. In this case the pistons pressure centres are on the centre line of brake pad and distance of the pistons is 25% and 75 % of the length of brake pad. Figure of merit was used to define the optimal diameter ratio:

   $$Q = 1 - \frac{s}{\bar{x}},$$

   where Q is the figure of merit, s is scatter of pressure and $\bar{x}$ is the average of pressure.

5. *Defining optimal diameters ratio of pistons*

   I certified with my experiments that in the case of diameter of piston being between 32 mm and 64 mm, optimal diameter ratio is 0,805. In this case the pistons pressure centres are on the centre line of brake pad and distance of the pistons is 25% and 75 % of the length of the brake pad.

5. CONCLUSION AND SUGGESTION

In this study the results give various parameters, which can help to make an optimal brake system. These parameters are good to optimize the working parameter or to increase the performance of brake system. The disc brake is a complex system. In my work small parts were examined. Finite element models used in this investigation are suitable for the deduction of general inferences.

In this investigation three parts were checked: opening of caliper, deformation of piston and pressure distribution on friction surface of brake pad. The opening results of caliper show that opening has an extent that determined the permissible opening. In mounting caliper this permissible opening changes if preload of bolts was change.

The other part of my research dealt with geometry of pistons of caliper. In this case the target is to make optimal design of piston with suitable working parameter when higher hydraulic pressure works into brake system. The investigation results prove that positions of sealing ring change the optimal geometry of piston (where wall's deformation is small). That means sealing ring position determined the optimal wall thickness and the optimal position of top face of piston.

The third part of my research optimized the pressure distribution on friction surface of brake pad. The goal is to define optimal diameter ratio of pistons making consistent pressure distribution on friction surface that increases the performance of brake system and increases the lifetime of the brake pad.

Models used in the research are upgradeable and more parameters use to simulated the real environment. The heat load effect was not checked in my work, so the heat load may modify the results in a little measure, but my models are good to check how results change, if the temperature is getting higher.

In my work tight environment of brake system was examined (caliper, piston, brake pad), but in this model deformation of disc and other movable parts (suspension, wheel, stabilizer bar, etc) deformation was not in my model. In the future this effect will have to be checked to make exact model that gives more real results.

The optimal diameter ratio was defined but if more points and real brake pad geometry would be examined the result would give more accurate value about the pressure distribution of brake pad. In my investigation new brake pad was examined but the wear of friction material changes the pressure distribution so it will be necessary to check how the wear changes the pressure distribution and how it changes the optimal diameter ratio of pistons.

6. SUMMARY

Breaking system is important in automobiles (cars, trucks, agricultural machines) as it regulates the speed of a vehicle. In high performance vehicles, disc brakes are used because their working property is better than that of drum brakes. The disc brake is a complex system with several parts within (caliper, disc, brake disc, pistons, and brake fluid). If these parts function and co-operate with each other in an efficient way the performance of the brake system will be higher.

The target of this study is to examine the different parts of the brake system in order to define the behaviour of parts and to define some parameters that will be used in its design. These parameters will be used to optimize geometry and material of the parts to find the best solution to increase performance and lifetime. In this study, four pistons caliper – consisting of several parts – were examined. Caliper's deformation was examined when hydraulic pressure is applied on the system. Deformation of piston's cylinders was also investigated for its influence on the performance of the system. If deformation of piston cylinder is too large, the piston's position changes and it does not push the brake pad with full surface. The effect of preloading condition of bolts inside the caliper was analyzed with varying hydraulic pressure. This study showed that the higher preload of bolts elevated the stress level in the caliper (10% increase of preload of bolts elevated the stress level in the caliper with 5.2%).

Furthermore, the deformation in the piston wall due to the applied pressure was scrutinized. In the course of application of the brake, piston comes in contact with the brake pad and hence to the brake disc. The walls of piston are deformed and in critical cases, the piston adheres inside the caliper because of its high deformation and performance of brake system diminishes. The wall thickness of the piston was studied. It could be inferred that stiffening effect was observed at the wall thickness less than 3 mm. Position of top face depends on the sealing ring's position. If the sealing ring is in caliper, the optimal position of the top face is 5.6 mm to the top, when the sealing ring is in the piston the optimal position of top face is 8.7 mm to the top. When the sealing ring is in caliper, the hydraulic pressure affects the wall's deformation; in the other case this effect can't be experienced. This effect influences the optimal wall thickness and the top face position.

The third part of this research is focused on the optimal wear of the brake pad, where optimal diameter ratio of pistons was defined. In case of four piston caliper, optimal diameter ratio provides a constant pressure distribution across the friction surface. Different diameter ratios were examined when friction coefficient differs between caliper and brake pad (μ=0.1; μ=0.15, μ=0.2). The diameter ratio does not change when friction coefficient varies and hence friction coefficient does not affect the optimal diameter ratio of pistons. The optimal diameter ratio is 0.805 (D1/D2, where D1 is the first piston according to the direction of rotation, D2 is the second pistons according to the direction of rotation).

7. MOST IMPORTANT PUBLICATIONS RELATED TO THE THESIS

Referred articles in foreign languages:

1. **Horváth, Á.**; Csik, Z.; Sukumaran, J.; Neis, P.; Andó, M. (2012), Development of brake caliper for rally-car, Sustainable Construction and Design, vol. 3, issue 3, pp. 191-198, ISSN 2032-7471
2. **Horváth Á.**, Oldal I., Kalácska G., Andó M., (2014), Thermal analysis of caliper's pistons in terms of brake fluid warming in finite element software, Mechanical Engineering Letters, pp. 136-142, ISSN 2060-3789
3. **Horváth Á.**, Oldal I., Kalácska G., Andó M., (2015) Multi parameter optimalization of brake piston, Sustainable Construction and Design vol. 6, issues 2, pp. 1-8, ISSN 2032-7471
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1. **Horváth Á.**, Kalácska G., Oldal I., (2015) Traktorokban alkalmazott tárcsásfékek dugattyúinak konstrukciós vizsgálata, Mezőgazdasági Technika, 56. évf. 7. sz., 2-4.o.
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