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Interface Pressure Distributions through Structural Modifications

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ABSTRACT

Due to the friction forces acting at the rotor and pads interface, the pressure distribution at the interface is asymmetric in a disc brake system of normal floating-type caliper design. The asymmetry and the high unevenness of the interface pressure distribution cause uneven wear and shorten life of pads. It has been speculated that these undesirable features promote disc brake squeal.

This paper investigates the contact (interface) pressure distributions at the rotor and piston-pad interface in response to several ideas of simulated structural (geometric or material) modifications. These modifications are made on the pads and/or at the interface between the piston and the back plate or at the pad guide. A detailed finite element model is constructed taking into account all significant contact interfaces between disc brake components. Sliding frictional contact is analyzed to obtain the interface pressure distributions. A plausible modification is identified which offers improved interface pressure distributions against wear. This work may also help create a good design of disc brakes for improved noise performance as well.

INTRODUCTION

A disc brake of floating caliper design typically consists of pads, calliper, carrier, rotor (disc), piston, and guide pins. One of the major requirements of the calliper is to press the pads against the rotor and should ideally achieve as uniform interface pressure as possible. A uniform pressure between the pads and rotor leads to uniform pad wear and brake temperature, and more even friction coefficients [1]. Unevenness of the pressure distribution could cause uneven wear and shorter life of pads. It has also speculated that they may promote disc brake squeal.

The interface pressure distributions have been investigated by a number of people. Tirovic and Day [2] studied the influence of component geometry, material properties and contact characteristics on the interface pressure distribution. They used a simple and non-validated, three-dimensional model of the disc brake. Tamari et al. [3] presented a method of predicting disc brake pad contact pressure for certain operating condition by means of experimental and numerical method. They developed a quite detailed model and validated the model by fitting the numerical deformations of the disc brake components with experimental results. Hohmann et al. [4] also presented a method of contact analysis for the drum and disc brakes of simple three-dimensional models using ADINA software package. They showed a sticking and shifting contact area in their results. Like [2], validation of their model was not made. Ripin [5] developed a simple, validated three-dimensional finite element model of the pad, and applied rather simple piston and finger force onto the back plate interface in his analysis. He studied the contact pressure distribution at the disc/pad interface, where gap elements were used to represent contact effect.

It has been observed that significantly more wear appears on the leading side than on the trailing side of a worn pad. This uneven wear is due to higher pressure between the rotor and the pad on the leading side. It is estimated that pressure at the leading side will be as much as one-third greater than the average pressure. There have been several solutions suggested in order to minimize and/or eliminate tapered pad wear. Amongst them are off-center force application to the back plate, locating the piston towards the trailing edge, using "hammerhead back plate design" as patented by ITT-Teves and using opposed pistons [1]. Other solutions include changing friction material compressibility and back plate stiffness as proposed by Tirovic and Day [2] and modifying shim and offsetting piston position by Tamari et al [3]. Unfortunately, the details of proposed modifications in [3] were not given.

As mentioned earlier, a good pad design should produce more uniform pressure distributions and therefore lead to more even pad wear. This paper examines pressure distributions at the rotor and the piston pad interface at different rotor speeds by using a validated and detailed three-dimensional finite element model. Only the piston pad has been examined because it presents more uneven pressure distributions than those of the finger pad. The paper also investigates several pad configurations plus modifications at the interface between the piston and the back plate. From those simulated modifications that produce favorable contact pressure distributions, physical modifications may be made and tested to establish a good design. The disc brake being studied is illustrated in Fig. 1.

In some recent papers on disc brake squeal [6-10], contact analysis forms part (the first phase) of the whole analysis procedure. In this sense, the present study also offers an improvement on the CAE simulation work.

THE FINITE ELEMENT MODEL

The finite element model consists of a rotor, two pads, a calliper, a carrier, a piston and two guide pins, as shown in Fig. 1. The model uses up to about 8000 solid elements and a total of approximately 35,000 DOFs. Prior to contact analysis, normal mode analysis is performed on the model of the rotor. By adjusting the Young's modulus and the density of the rotor, the numerical and experimental frequencies of the free-free rotor become very close and are listed in Table 1. The material data of the pad comes from an industrial source and has been validated. The whole disc brake is tuned to obtain correct values for the spring elements that connect the brake components.

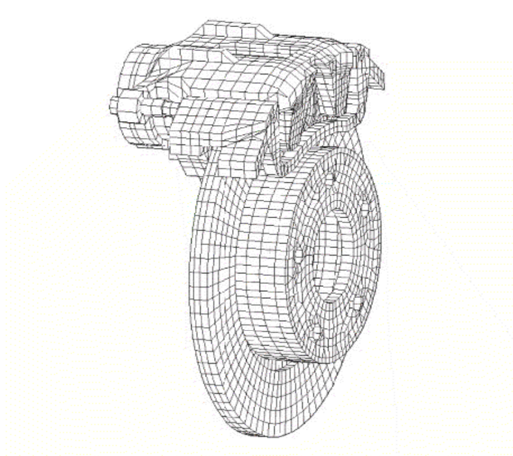


Figure1. Finite element model of the disc brake

Table 1. Modal result of the rotor

Mode	2ND	3ND	4ND	5ND	6ND	7ND
Test (Hz)	937	1809	2942	4371	6064	7961
FEA (Hz)	960	1820	2939	4365	6062	7964
Error (%)	2.4	0.6	-0.1	-0.1	-0.0	-0.0

Having satisfied with the validated model, contact analysis is carried out to obtain the pressure distribution between the rotor and the piston pad. In order to simplify this procedure and reduce computational time of the contact analysis, a rigid surface was used to represent the deformable rotor surface. The piston and the piston pad are included. The model consists of 408 solid elements (738 nodes) for the piston and 1040 solid elements (1343 nodes) for the pad respectively, as shown in Fig. 2. There are a number of spring elements between the piston and the pad back plate with appropriate stiffness values. The outer wall of the piston is rigidly constrained in the radial direction. For the contact interface between the pad and the rotor a friction coefficient of μ=0.6 is prescribed. The structure is loaded in two steps. First, a uniform pressure of 8.0 MPa is applied on the top of the piston. In the second step the rotor is rotated about the central axis at two different angular velocities, i.e., 0.1 rad/s and 6.0 rad/s.

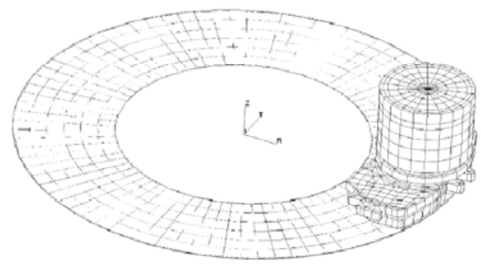


Figure 2. FE model of the pad, piston and rigid surface

CONTACT ANALYSIS

First, contact analysis is carried out for the original piston pad as shown in Fig. 3. Its geometric and material data is given in Table 2.

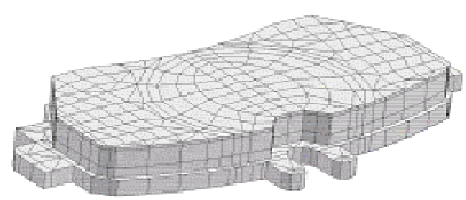
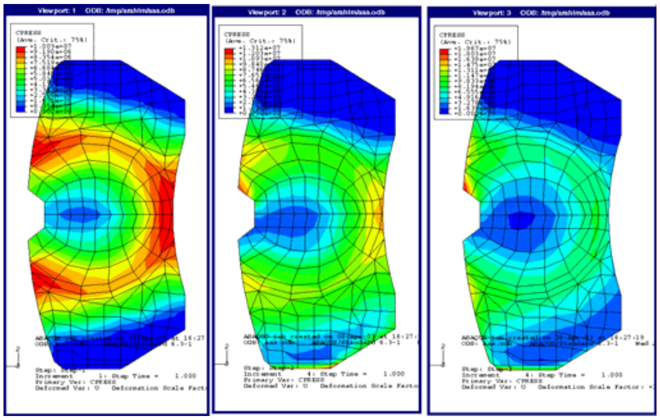


Figure 3. FE mesh of the original pad

Table 2. Geometric and material data of the pad

	Lining	Back plate
Length (m)	0.095	0.118
Width (m)	0.036	0.045
Thickness (m)	0.0095	0.005
Density (kg·m^{-3})	2798	7850
Young's modulus (GPa)	orthotropic	210
Poisson's ratio	0.2	0.3

Then, the interface pressure distributions of a real pad under normal (centralized) piston line pressure at three different rotational speeds Ω are computed as a benchmark. The results are shown in Fig 4. The bottom side in the diagrams is the leading edge.



(a) Ω=0 rad/s (b) Ω=0.1 rad/s (c) Ω=6 rad/s

Figure 4. Interface pressure distributions

Fig 4 shows that when the rotor is at rest, the pressure distribution is symmetric about the geometric centre line of the pad. When the rotor slides, the pressure distributions are no longer symmetric and the highest pressures occur at the leading side of the pad. This is consistent with previous findings [2]. The apparent contact area (geometric area) of the pad is 0.0039 m^2. The predicted contact areas at Ω=0 and at Ω=6rad/s are 0.00327 m^2 and 0.00335 m^2 respectively.

It is interesting to know how the interface pressure distributions vary as a result of structural modifications to the pad and/or the interface between the piston and the back plate. The modifications studied in this paper are summarized in Table 3.

Table 3. Structural or material modifications

Modifications	Descriptions	Changes
A	Softer back plate	E=71 GPa
B	Stiffer back plate	E=331 GPa
C	Lengthened pad	+9%
D	Shortened pad	-9%
E	Thicker back plate	+0.003 m
F	Thinner back plate	-0.002 m
G	C + E	
H	Horned ears	Full constraint
I	Horned ears	Half constraint
J	Braced	See Fig. 5
K	Raised back plate	See Fig. 6
L	Partial connections	See Fig. 9
M	z-constraint	Trailing ear

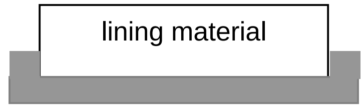


Figure 5. Braced pad modification (steel in gray)

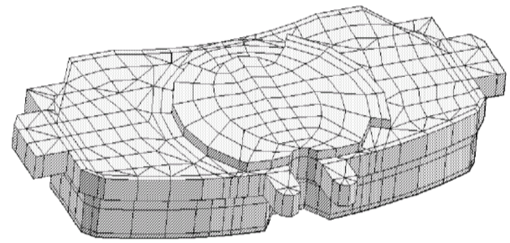


Figure 6. Back plate partly raised by 2mm at the area in contact with piston

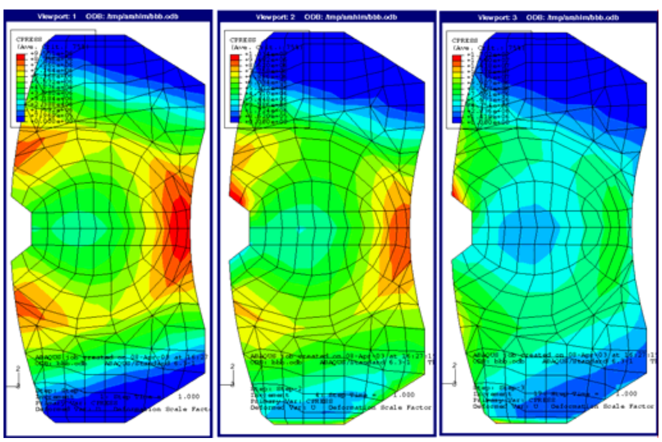
Because of the page limitation, the contact pressure distributions in the form of pictures cannot be displayed for all the cases of modifications. Instead, the contact area and the highest contact pressure, relative to the corresponding results of the original pad, for each modification at Ω=6 rad/s, are shown in Table 4. In it, the area ratio is the ratio of the computed contact area to the apparent contact area and the pressure ratio is the computed highest contact pressure of that particular modification to that of the original pad.

Table 4. Results of contact analysis

Modifications	Area ratio	Pressure ratio
None (original)	86.0%	100%
A	77.4%	114%
B	86.8%	98.5%
C	83.4%	99.4%
D	90.1%	109%
E	89.5%	86.5%
F	80.8%	114%
G	86.5%	85.6%
H	86.0%	104%
I	84.4%	107%
J	85.3%	98.8%
K	88.9%	80.0%
L	87.1%	66.6%
M	92.5%	97.8%

It might be expected that increased contact areas generally mean more uniform pressure distributions. However, the contact area is not the only factor. The highest contact pressure needs to be considered as well when evaluating the merit and drawback of a modification. For example, a stiffer back plate (modification B) leads to a small increase of the contact area (always compared with the original, unmodified case) but a considerable increase of the highest contact pressure. Therefore this is not a desirable modification.

It can be seen that changing the Young's modulus of the back plate does not bring about favorable pressure distributions. Shortening the length of the pad can increase the ratio of the contact area to the apparent contact area but does not reduce the highest contact pressure. It also shortens the pad life because of the reduced apparent area of the lining material. Increasing the thickness of the back plate produces desirable pressure distributions. This result is consistent with the previous finding by Tirovic and Day [2, 5]. However, this modification may lead to higher manufacturing cost [2]. Modification G, which combines modifications C and E, lowers the highest contact pressure but does not alter the contact area. The pressure distributions at three different rotational speeds of the rotor for modification E are illustrated in Fig. 7.



(a) Ω=0 rad/s (b) Ω=0.1 rad/s (c) Ω=6 rad/s

Figure 7. Pressure distributions of thickened back plate

Horned back plates (shown in Fig. 8) do not offer any improvement on the pressure distribution. This result confirms the conclusion made by Ripin [5] that the abutment arrangements do not significantly alter the contact force distribution.

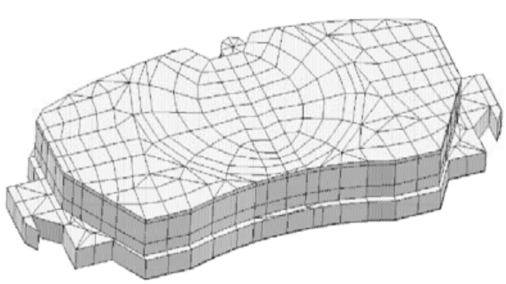


Figure 8. Pad made of back plate with horned ears

An interesting modification is M, in which the DOFs in the *z* direction (the same direction of the piston line pressure) of a few nodes at the trailing edge of the back plate are grounded. This modification results in a considerable increase of the contact area. Even though the highest pressure does not drop much, this modification may be combined with other desirable modification to achieve a favorable pressure distribution. However, it may be difficult to physically implement this modification.

A very good outcome is obtained from modification K, where the area of the back plate that is in contact with the piston head is raised in a flat plateau of special shape (see Fig. 6). This allows the piston line pressure to be transmitted into the pad in desirable paths. The resultant pressure distributions at different rotor speeds are shown in Fig. 9. This modification requires extra machining and incurs extra manufacturing cost.

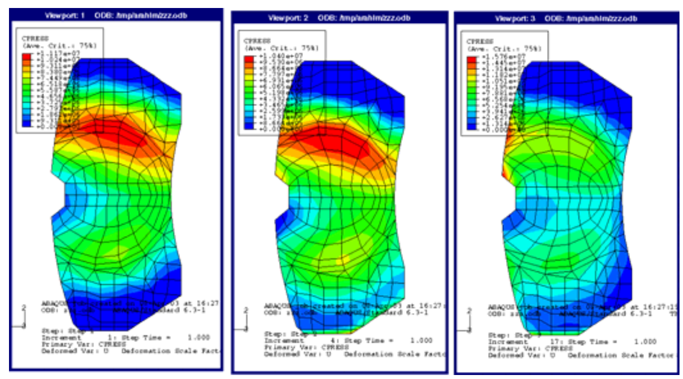


Figure 9. Pressure distributions of raised back plate

The best result seems to come from modification L, where the connection between the piston head ring and the back plate is partial. At this contact interface, there are a number of rigid springs that simulate the contact between the piston head and the pad back plate in the axial direction. If these springs are present over the whole region, where the piston head ring overlaps the back plate, in a dense and symmetric manner, they represent a full contact. If, on the other hand, some of the springs are removed, a partial contact is represented. By choosing different combinations of the locations of the retained springs and/or spring constants, favorable contact pressure distributions at the pad and rotor interface can be achieved. This has the effect of a well-designed piston adapter and affords an opportunity for an improved disc brake design. The contact pressure distributions at three different rotating speeds of the rotor for one particular partial, rigid connection, as shown in Fig. 10, are presented in Fig. 11. Note that the pressure distributions at the three different rotor speeds are nearly the same. This is also an advantage in that a favorable pressure distribution appears at all rotor speeds.

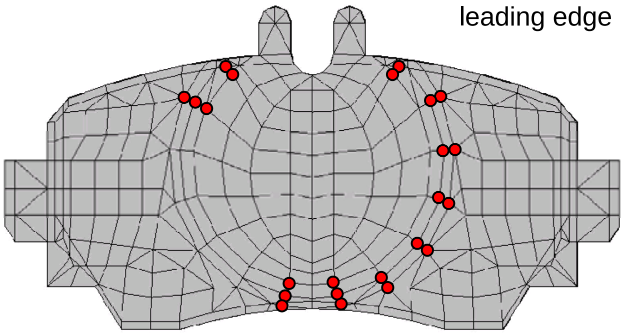
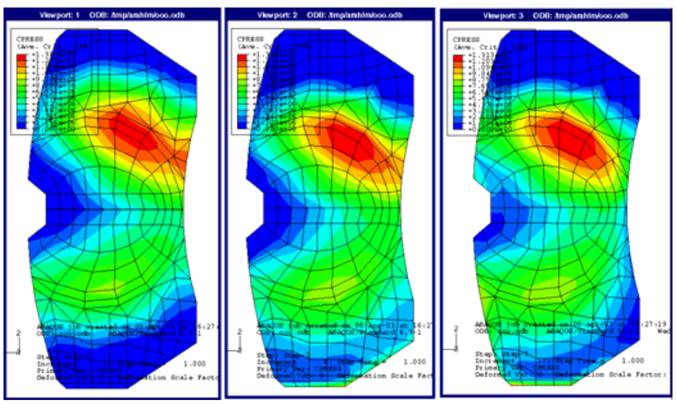


Figure 10. Partial connections in the axial direction at the back plate and piston interface (the red dot represents removal of one axial connection)



(a) Ω=0 rad/s (b) Ω=0.1 rad/s (c) Ω=6 rad/s

Figure 11. Pressure distributions for partial connection

Fig. 11 reveals that by removing the connections between the piston head and the back plate at the edge of trapezium-shaped cutout of the back plate, the high pressure appearing there in other cases vanishes. The remaining high pressure region is located at the leading edge of the pad.

The attention in this paper is focused around the piston pad. Rumold and Swift [11] recently studied the interface pressure distributions due to different positions of the fingers and the pistons of a twin piston disc brake of a medium truck. They found that by shifting the position of the fingers or the pistons to the trailing side rather uniform interface pressure distributions were established, using a multibody code with flexible superelements.

Spurr is the first researcher who studied the influence of the centre of the contact pressure on disc brake squeal [12] and found that squeal was only generated when the contact (a thin strip in his experimental pad) was sufficiently close to the leading edge. Recent experimental work by Fieldhouse showed that when the piston line pressure had a small, suitable offset to the trailing side of the back plate the squeal propensity was reduced [13]. Bergman et al. [14] recently presented experimental results and showed that the shorter pad contact surface reduced squeal occurrence and noise level. They suggested that one of the reasons for the improvement was the change of the interface pressure distribution. Liles [15] simulated a number of simple modifications of system parameters to show their influence on squeal propensity, even though the interface pressure distribution was not included.

Achieving favourable contact pressure distributions at the pad and rotor interface to reduce squeal propensity is not a very new idea. The crux is how to achieve a favourable contact pressure distribution, first in theory and then in practice, in a quantifiable and economic manner. This paper is a step forward toward this goal. The authors intend to conduct next complex eigenvalue analysis based on the results of the interface pressure distributions.

CONCLUSIONS

This paper presents the analysis of the contact pressure distributions at the rotor and piston pad interface of a solid disc brake as a result of structural modifications. A number of modification ideas have been studied using the finite element method. It is found that by making right connections between the piston head ring and the back plate in the axial direction (the direction of the piston line pressure) the contact area can be increased and the contact pressure distribution can be improved (made more even) considerably.

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