Analysis and methods of squeal reduction of disc brake system

I.A. Magomedov, A.I. Mezhieva, M.A. Suleymanova

Chechen State University, Grozny

Annotation: The following work is concentrated on the analysis of dynamic pressure distribution of a brake pad by using ANSYS software. A non-linear contact analysis of brake disc has been performed using Finite Element Method. The finite element model of disc and pads have been developed, and followed by an inclusive simulation of the contact analysis of disc brake system using ANSYS software. A detailed 3D model of a disc brake assembly was built. Contact analysis was performed to work out the pressure distribution, interfacial contact area and normal contact forces under following conditions: with no friction (μ=0) and frictional braking. The effect of changing friction coefficients and the brake pressure (hydraulic) on squeal occurrence is examined. The results illustrated that the contact pressure distribution are far greater near the leading edge of brake system's pads respect to the trailing edge. It was found from results of finite element analysis that the pressure in brake is linearly proportional to the normal contact pressures, moreover the magnitude of the interfacial contact pressures amplified with the greater calliper pressures. It was also known that with a higher coefficient of friction amount of interfacial friction forces will be higher, so leaving with possibility to increase instability in a brake assembly.
Keywords: FE, dynamic pressure, contact pressures, squeal, brake system, non-linear contact analysis

Introduction

Disc brake noise has been a major concern for the automotive industry, even though a lot of research has gone into reducing its occurrence. There are many ways in which disc brake noise can manifest itself, but it can be divided into three general categories. First one is low frequency noise, second is low frequency squeal and finaly high frequency squeal. The low frequency noise occurs between of 100 Hz and 1000 Hz, and some typical types of noise in this region are grunt, groan, grind and moan. This type of noise is known to occur due to friction material excitation at the brake rotor and the brake pad lining contact. This energy is known to transmit to various other components as a vibrational response. Low frequency squeal is known to occur at frequencies higher than 1000 Hz however, lesser than the first circumferential mode of the rotor. This is known to occur due to two reasons, one is the frictional excitation of the system, in which modes of two or more structures couple generating squeal. High frequency squeal has been defined as noise produced by friction induced excitation due to coupled resonances

© Электронный научный журнал «Инженерный вестник Дона», 2007–2018

of two modes. High frequency squeal generally occurs at frequencies higher than 5000 Hz.

Literature review

The topic of interface pressure distributions in brakes system has been investigated by many scientists. Spurr [1] introduced a new concept of brake squeal termed sprag-slip, where unbalanced oscillation in the system can arise even with constant coefficient of friction. Spurr demonstrated this by utilising a shoe of railway brake with an adaptable pin bearing on the wheel. To associate spragging to squeal of disc brake system, an altered brake disc also indicated this to be the case. For this case, the pad of brake system was constrained by studs in the back of the pads. After ground in just to let merely a radial strip to be in interaction with the disc of brake system. Gathered result showed that, squeal in brake system was only produced when the contact strip was sufficiently near to the pad's leading edge. Moreover it was worked out that brake squeal rely upon the magnitude of friction coefficient and it also depends on the point (position) of the areas of contact between the material of frictio and brake disk surface.

Dubensky [2] used pressure-sensitive paper and Tumbrink [3] used a ball-pressure technique. To work out pressure distribution between disc and pad surfaces in static analysis Samie and Sheridan utilised pressure-sensitive film [4]. Fieldhouse performed a test where he used a thin steel wire between pad's backplate and the face of piston to regulate pressure distribution and shift it in the direction of the trailing side [5]. Researchers by investigating the squeal problem noticed that tendency of pressure distribution is toward the leading side.

The centre of pressure theory suggests that the brake squeal propensity of a disc brake system relies on the position of center of pressure with respect to the contact patch between the brake pad and the brake rotor. It has been previously researched that a trailing centre of pressure in a disc brake system successfully

© Электронный научный журнал «Инженерный вестник Дона», 2007–2018

controls the squeal propensity. The amount of control depends on the position of the centre of pressure with respect to the contact patch between the brake pad and the brake rotor, and also on the geometry of the disc brake system itself [6].

Bosch in their research utilized thin film to find out tendency of pressure at the center of pad during static and dynamic analysis. The method used inserted sensitive film in the pad of brake system. Therefore the approach was able to work out dynamic pressure at the centre of the pad. The pads of brake system were equipped to collect the laminate by machining a recess and "plug". The sensitive film (film laminate) was imbedded in the recess and the "plug" as in sandwich. Then pads of brake system were introduced with a level rubbing in it and the assembly of brake used to find out centre of pressure on a test rig. They worked out that with the high pressure acting to the centre of the pad the centre of pressure was shifting (varying) and more stable with less pressure acting. It was also found that at low pressure the pressure shifts to leading side and it is unstable at that point. However pressure barely shifted toward the trailing edge.

Different analysis were performed on brake system utilizing Finite element method to gather results of squeal occurrence and its reduction and build a predicting tool for future investigation. Tirovic and Day [8] calculated the impact of several factors such as: compressibility of friction material, friction coefficient and stiffness of disc on the interface distribution of pressure. Results illustrated coefficient of friction of pad's lining have a major part in the interface pressure distribution, so leading it too squeal occurrence by pressure is being shifted to leading edge.

Abu Bakar [9] proposed a new method of predicting squeal using the finite element method. A finite element model of a disc brake was constructed and validated through contact analysis where static contact pressure distribution and its contact area correlated well with the experimental results. In the FE analysis he considered a real surface topography of which measurements were carried out in

© Электронный научный журнал «Инженерный вестник Дона», 2007–2018

order to obtain a realistic contact interface model. It was shown that the real contact interface model predicted squeal occurrences much better than the perfect contact interface model. The results showed that with the inclusion of wear, squeal events were predicted to appear and disappear as wear progressed even though similar boundary conditions and operating conditions were imposed to apparently the same disc brake model. This phenomenon explains the fugitive nature of squeal behaviour adding complexities to a brake system.

Ibrahim Ahmed [10] investigated the drum brakes and performed the non-linear contact analysis to calculate the normal contact forces and circumferential friction forces. He did contact analysis using surface-to-surface contact to get a closer representation of real contact behaviour. The hydraulic pressure was varied from 20 to 80 Bar and coefficient of friction set as constant 0.42. The result showed that the effect of the piston pressure concentrated on longitudinal centerline of lining for both leading and trialing. It has also been observed that small hydraulic pressure leads to small contact area of lining with rotor. The maximum reaction force of 37 N was captured when 20 bar of hydraulic pressure was applied and 66 N at 40 bar for leading shoe. At 80 bar the maximum reaction force reached to 110 N. The result also showed that 30% of contact between lining and drum was reached when pressure was 20 bar and 90% at 80 bar. However, for trailing shoe the contact area was less than 6% than for leading, but higher forces and therefore higher friction between lining and drum

Methodology

As it was mentioned previously this work implies usage of ABAQUS, Solidworks usage also took place. Disc and pad were built and assembled in Solidworks. Then file was converted to be used in ABAQUS analysis. The finite element model of the rotor and pads were simplified to lessen the computation time. The FE model of brake system composed of a disc, and inboard and outboard pad as illustrated (figure 1). The pad of brake system consists of a back plate and

© Электронный научный журнал «Инженерный вестник Дона», 2007–2018

atached friction lining. Sizes and other data (material properties) for the disk brake system parts are illustrated in Table 1 below. A fixed type calliper will be presumed in the FE model. In addition, another assumption was made that the interactions at the disc/pad interfaces are even and friction lining surfaces of both back plates were smooth.

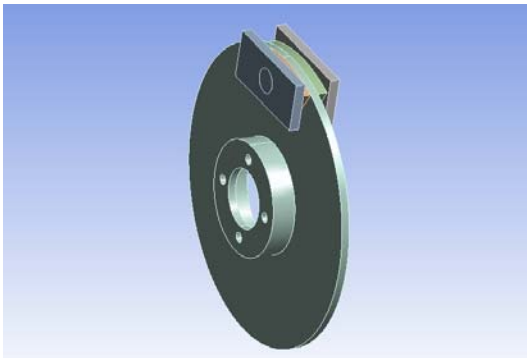


Fig. 1. Assembled cad model view from ANSYS.

Table №1

Assembly with material properties and dimensions

Units	Disc	Lining	Backplate
Density (kg/m^3)	7107.6	2798	7850
Young's modulus (GPa)	115.4	1	210
Thickness (m)	0.08	0.1	0.1
Radius (m)	0.15	-	-
Poisson's ratio	0.211	0.25	0.3

Results

For static analysis presence of coefficient of friction was ignored as for contact pressure distributions it will play no role, in other words it was set to zero. Hydraulic pressure was applied to pistons from both sides with total magnitude of 20 bars. Figure 16 represents non-linear analysis of pressure distribution on the surfaces of both pads. Gathered results of pressure distribution of static analysis

© Электронный научный журнал «Инженерный вестник Дона», 2007–2018

along centerline of pads surfaces can be observed from figure 2. Left and right sides from the center of the pad are leading and trailing sides respectively.

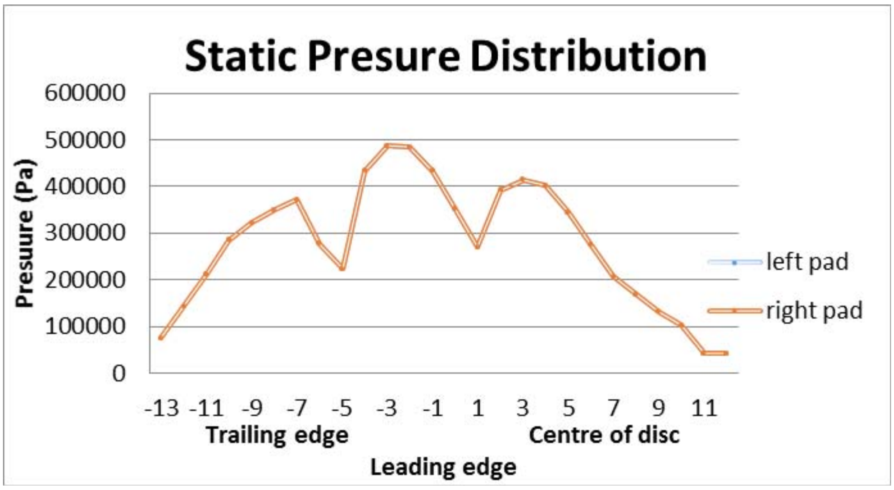


Fig. 2. Static pressure distribution on the pads' surfaces.

For dynamic investigation two main aspects were analyzed: varying coefficient of friction and hydraulic pressure. First dynamic analysis was done with varying coefficient of friction. Three variation took place: 0.1, 0.4, 06 (value of coefficient of friction). Hydraulic pressure was kept as previous with a total value of 20 bar acting on both pads. A slight movement of disc (5 degree) was introduced to perform a dynamic analysis. The following figure 3 represents results from centerline of pad with different values of coefficient of friction. Units of pressure from figure 3 are Pascals. From center to left is trailing side and from center to right is leading side.

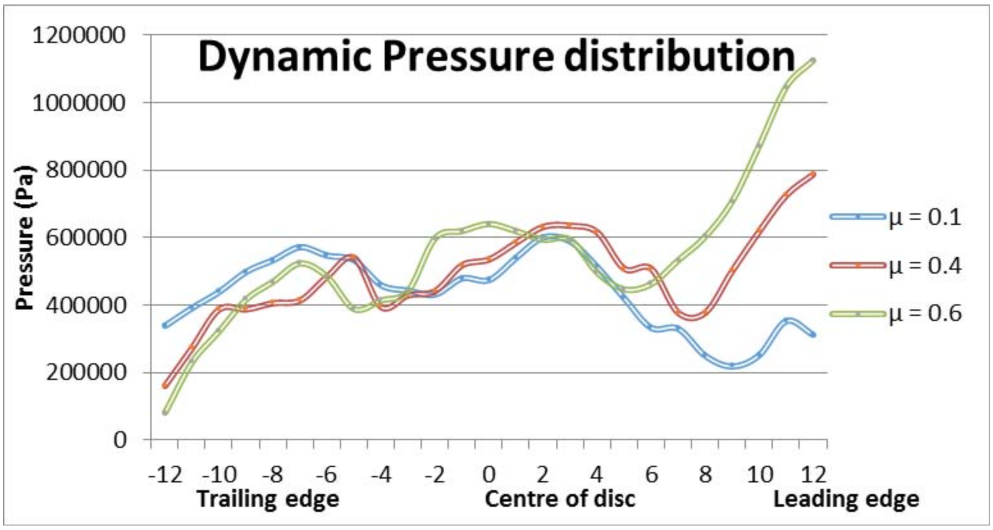
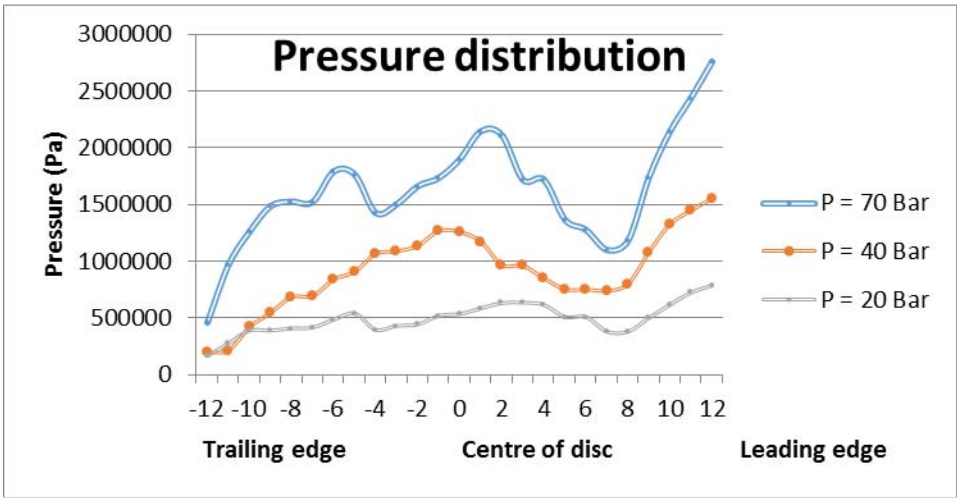


Fig. 3. Dynamic pressure distribution with constant pressure with varying coefficient of friction.

Second dynamic analysis was done to investigate influence of varying hydraulic pressure on pressure distribution. Hydraulic pressure was varied with three different values: 20, 40, 70 bar. However coefficient of friction was constant with a value of 0.4. As previously disc was moved with angle degree of 5. Figure 4 illustrates pressure distribution shift along pad's centerline with three different values of hydraulic pressure.



.

Fig. 25. Pressure distribution.

Discussion

Static analysis illustrated that pressure distribution will be mainly acting to the center of the pads. It also was proven that pressure distribution will be evenly

© Электронный научный журнал «Инженерный вестник Дона», 2007–2018

distributed on both leading and trailing edges. Abu Bakar's static contact pressure distribution analysis illustrated that pressure will be distributed at the center of pads. After static analysis dynamic analysis was performed with varying coefficient of friction. Static results illustrated that the pressure distribution will acts at center of the pad surface. However, in dynamic analysis pressure distribution shifted toward leading edge. Shift to leading edge can be observed from figure 23. Moreover by studying figure 23 one can see, that by increasing coefficient of friction trailing edge will lose contact pressure area. In numbers it can be presented as follows: by increasing coefficient of friction from 0.1 to 0.4 trailing edge will lose its 53 % of contact pressure. Further increase from 0.4 to 0.6 will give a lost of 48%. However as trailing edge loses its contact pressure leading edge will gain contact pressure. Increasing coefficient of friction from 0.1 to 0.4 leading edge will gain 49% of contact pressure, and further increase from 0.4 to 0.6 will gain 44% of contact pressure. Tirovic and Day in their investigation showed that by increasing coefficient of friction trailing edge will lose its contact area while leading edge will gain its contact area. Resulting in as they stated squeal occurrence, which also was shown by Spurr. Also its worth of saying that total contact area will increase with increase of coefficient of friction. Influence of varying hydraulic pressure was investigated. Coefficient of friction was kept constant with value of 0.4, while varying hydraulic pressure. Gathered results showed that increase of hydraulic pressure from 20 bar to 40 bar will lead to gaining 20% of contact area in trailing edge and 49% in leading edge. With further increase trailing adge gained 56% while leading edge gained only 44% of contact area. In last case trailing edge gained 22% more contact area than leading edge, but leading edge have 84% of total contact. It also was observed by Dubensky that the contact pressure distributions shifted towards the leading side. Lee and Ibrahim did the same investigation on varying the hydraulic pressure. The results they got showed that by increasing the hydraulic pressure the normal contact

© Электронный научный журнал «Инженерный вестник Дона», 2007–2018

forces/pressure will increase. Generally, the results follow the trend of pressure distribution when compared with experimental results or FE of other researchers. The comparison showed that the results of this dissertation are likely reliable as they follow the trend of results found within other research.

Discussion

To conclude, finite element method by using software ANSYS can compute contact pressure distribution in the brake system. The results followed the trend of other researchers' work and comparison been done in this work. The following work performed an investigation of brake system by changing the different characteristics to work out the best performing condition. The results illustrate that trend during normal brake will always be shifting toward leading side by reducing contact at trailing side. It been observed by analyzing the results that the tendency to squeal will increase with an increase of coefficient of friction, and remains stable with an increase of hydraulic pressure.

References

1. M. Eriksson, "Friction and contact phenomena of disc brakes related to squeal", Comprehensive summaries of Uppsala dissertations from the faculty of science and technology, Acta University Upsaliensis, Uppsala, 2000, p. 46.

2. Dubensky R.C., Experimental techniques for rotor performance measurements, 1985. p. 32.

3. Tumbrink, H. J., "Measurement of load distribution on disc brake pads and optimization of disc brakes using the ball pressure methods, 1989, p. 4.

4. Samie, F. and Sheridan, D. C., "Contact analysis for a passenger car disc brake", 1990, p. 220.

5. Fieldhouse, J., Ashraf, N., Talbot, C. and Pasquet, T. (2009). Measurement of the dynamic centre of pressure of the disk interface during a braking operation, 51(1, 2), p.73

6. Liles, G., "An analysis of disc brake squeal using finite element methods", 1989. p. 42.

7. Yakovlev M. Y. Yangirova A.V. Inženernyj vestnik Dona (Rus). 2013. №2. URL: ivdon.ru/ru/magazine/archive/n2y2013/1639

8. Tirovic, M. and Day, A. (1991). Disc brake interface pressure distributions. ARCHIVE: Proceedings of the Institution of Mechanical Engineers, Part D, 1989-1996 (vols 203-210), 205(24), pp.137-146

9. Abu Bakar, A., Abdul Hamid, M. and Ouyang, H. (2006). A New Prediction Methodology for Dynamic Contact Pressure Distribution In A Disc Brake, 45(1). p. 196

10. Magomedov I. A. Inženernyj vestnik Dona (Rus). 2018. №2. URL: ivdon.ru/ru/magazine/archive/N2y2018/5009

11. Kinkaid, N.M., O’Reilly, O.M. and Papadopoulos, P., “Automotive disc brake squeal: a review”, Vol. 267, No. 1, pp.105–166, 2003

© Электронный научный журнал «Инженерный вестник Дона», 2007–2018