Modelling Of Automobile Brake Pad Wear

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Abstract- In the optimization of wear resistance of automotive brake pads, the braking temperature as a result of contact surface of the disc and the pads of a friction brake during its operation has significant impact on brake performance. An interaction between a brake disc and brake pads of automobile brake is characterized by a number of dry contact phenomena; these phenomena are influenced by brake operation conditions (applied pressure, speed and brake interface temperature) and material characteristics of friction couple at a given time. The temperature measurement techniques, which are always available under laboratory test conditions at different radial distance enable obtaining relatively accurate values of temperature at the friction surface using thermo couples. However, measuring the sliding surface temperature at different radial distance during the entire lifetime of the brake pads is very necessary due to the demanding operating conditions of the brakes. Therefore, an appropriate mathematical model was developed in order to enable estimate of the sliding surface temperature between the brake disc and brake pads throughout the entire duration of brake application. This is achieve by infinite element method using the results of the of temperature measurement at different radial and axial distance within the brake pad and its processing, by means of an originally developed mathematical model which aided the analysis of results and validation of the mathematical model. The finite element analysis is simplified by utilizing the inherent symmetry of disc brake and applying symmetric boundary conditions. The finite element analysis results presented herein illustrate that the brake pads temperature varies at different radial distance. While making comparison of temperature at different radial distance of the pad and disc ,the maximum and minimum values and the difference between them were: pad highest surface temperature was 880^{0}C and the lowest surface temperature value was 700°C, the difference of temperature was 130°C. Considering the wearing rate in radial and axial distance when the rate of wear rises most in radial direction, the temperature occurring on the surface and the temperature difference between two surfaces also rises. At a time of 1.5 to 2.5 sec maximum heat was generated and the temperature of pad and disc rises. Above 2.5 however, convection heat lost set in, this is as a result of nature trying to obey Newton's law of cooling which reduced the temperature in the interval 2.5 to 4 sec. Disc surface temperature in the radial interval 75-85 mm, where the disc is exposed to air flow is relatively low. But in the interval 85-105 mm, where the disc is in contact with friction linings, the temperature increased because in the case of uniform pressure distribution, heat generation grows in the radial distance. It was finally observed that if the thickness of the pad is reduced due to excessive wear, influence of heat into the pad and caliper assembly increases and the risk of brake fluid vaporization will increase leading to brake temperature rise. Therefore, it was recommended that the material with low thermal conductivity for the pad and caliper components be used and to build a test rig for real braking system to study the heat dissipation experimentally so that the analysis result obtained from finite element method can be validated.

Indexed Terms- Modelling, Automobile, Brake Pad, braking temperature, Wear

I. INTRODUCTION

The overall aim of this study is allow for prediction of pad performance based on intrinsic external influences such as temperature and heat dissipations and assess its thermal performance in the early design stage. The study would seek to fulfill the following objectives; To determine the contact temperature distribution on the working surface of a brake with the aid of thermocouples. Develop mathematical models for describing the thermal behavior of a disc brake wear in a vehicle. Develop a time dependent heat equation from energy balance for the brake rotor and the pads.

The automobile industry is one of the most economically vibrant industries in national development. Development of durable, effective, healthy and reduced cost brake pads is not only going to satisfy environmental and safety requirements but will contribute to national development. In this study, analytical and finite element analysis approach will be conducted in order to identify the temperature distributions and behaviors of disc brake rotor in transient response of automobile brake system will be developed and some experiment will be carried out on the brake pad materials to understand heat mechanism with related variables and the performance. To characterize the temperature fields of the solid rotor with appropriate thermal boundary conditions to ascertain the heat conduction in a disk brake system in respect to time.

An automotive brake functions by converting the vehicle's kinetic energy into heat energy. During braking, the heat energy is first born by the two contact surfaces of the brake, namely the brake disc and the brake pad (or drum and shoe in the case of the drum brake), and is the transferred to the contacting components brake such as calipers of the brake, as well as the surroundings. According to a research study by (Frost, N.& Sullivan F.2002), there is likelihood that after 2010, the global automotive industry will start using brake-by-wire systems instead of hydraulic braking systems. The various technologies such as electro mechanical braking system and the electronic wedges brake are soon going to replace the older braking systems. With the help of these brake-by-wire systems, automobile drivers will be having more control on their vehicles particularly in case of sheer emergency. The braking system used in automobiles is mainly used for helping the driver control the deceleration of the vehicle. It is one of the crucial systems, which is especially designed for decreasing the speed of the fast moving vehicle. Shahril,K., Monarita N., and Aliff , S., 2012).

According to (Mr. Ajit B. Jain, Dr. S.G. Taji and Prof. H.S. Bawiskar, 2008). For Better transportation there is need for automobile manufacturers to provide good safety systems. The brake system has become important active safety systems. The disc brake is a device for deaccelerating or stopping the rotation of a wheel. This work presents the analysis of the contact pressure at the disc interfaces using a detailed 3-dimensional model of a brake in FEM. Finite element (FE) model of the brake-disc is created using solid modeling Software i.e. Pro-E and Analyzed in ANSYS which is based on the finite element method (FEM). It also investigates contact pressure distributions at varying load. Due to rise in temperature the change in angular velocity and the contact pressure is also studied. Wear reduces the life of friction material. Due High wear, the frictional material needs to be replaced Continuously. Problems such as Continuous wear of brake pads and thermal cracks in brake discs are due to high temperatures. Continuously controlling the temperature and thermal and mechanical stresses are critical to proper functioning of the braking system. Cooling of Brake System is further an important aspect to consider for brake disc durability and performance. Different materials for brake pads is tested as compared with the existing one. Finally comparison between analytical results and result obtained from Analyses is carried out and all the values obtained from the analysis are less than their Permissible values. Hence on the basis of thermal and contact stress analysis best suitable material is suggested.

Nagy et al. (1994) pioneered the transient analysis of disc brakes using the finite element method. The nonlinearity of the friction at the rotor/pads contact was considered. They found that the stability of the system was mainly influenced by the friction coupling the rotor and the pads. Hu and Nagy (1997) developed the method of Nagy et al. (1994) and tackled large finite element models of disc brakes. Dynamic contact was considered by incorporating a penalty function to model the contact surface in the virtual work statement. Another merit of this work was the use of pressure-dependent friction coefficient obtained from dynamometer tests. To speed up the explicit numerical integration and yet maintain accuracy, they used a one-point-integration solid element with stiffness hourglass control that they developed with Belytschko. The numerical time-domain information was converted to power spectral density data through FFT. Several predicted frequencies of high sound level compared well with squeal frequencies found from experiments. Chargin et al. (1997) also performed a transient analysis for a simple finite element model of a brake. They used implicit integration with tangent matrices of the steady-state solution. This was the point to start the complex eigenvalue analysis. Again, the contact constraint was imposed by the Lagrange multiplier. A stiff equation solver was necessary to solve the resultant differential equations. Hu et al. (1999) investigated the effects of friction characteristics and structural modifications (such as pad chamfer or slot). An optimal design was found using Taguchi method. Mahajan et al. (1999) used the same method of Nagy et al. (1994) and the procedure of Hu et al. (1999) and compared the transient analysis, complex eigenvalue analysis and normal mode analysis and found that they were useful at different stages of design. Following Nagy and Hu's approach, Chern et al. (2002) performed a nonlinear transient analysis of a disc brake including rotor, pads, two pistons, sliding pins, and anchor bracket using LS/DYNA with constant dynamic friction coefficient's the squeal frequency and the corresponding operational deflection shape of the rotor, respectively.

A moving vehicle possesses an amount of kinetic energy depending on the weight and speed of the vehicle. This energy must be dissipated in order for the vehicle to slow down or stop. Brakes serve to reduce the velocity of a vehicle in this scenario. The focus of this project is on the mechanisms of braking, primarily the physical interaction between the friction components in a disc brake system. Braking torque and contact forces of different brake pad materials under a constant brake pressures are studied in order to analyze this interaction. The goal of this project is to allow for the prediction of pad performance based on intrinsic material properties. While such an approach does not completely study the external influences or sophisticated details of wear and heat dissipation, it will still provide a firm basis and potential starting point for the study of brake pad materials (Jared Feist, 2014).

Voller, et al.(2003) perform a Analysis of automotive disc brake cooling characteristics. The aim of this investigation was to study automotive disc brake cooling characteristics experimentally using a specially developed spin rig and numerically using finite element (FE) and computational fluid dynamics (CFD) methods. All three modes of heat transfer (conduction, convection and radiation) have been analyzed along with the design features of the brake assembly and their interfaces. The influence of brake cooling parameters on the disc temperature has been investigated by FE modelling of a long drag brake application. The thermal power dissipated during the drag brake application has been analysed to reveal the contribution of each mode of heat transfer.

Zaid, et al. (2009) presented a paper on an investigation of disc brake rotor by Finite element analysis. In this paper, the author has conducted a study on ventilated disc brake rotor of normal passenger vehicle with full load of capacity . The study is more likely concern of heat and temperature distribution on disc brake rotor. In this study, finite element analysis approached has been conducted in order to identify the temperature distributions and behaviors of disc brake rotor in transient response. ABAQUS/CAE has been used as finite elements software to perform the thermal analysis on transient response. Thus, this study provide better understanding on the thermal characteristic of disc brake rotor and assist the automotive industry in developing optimum and effective disc brake rotor.

In a study, where the heat conduction problems of the disk brake components (pad and rotor) are modeled mathematically and was solved numerically using Finite Difference Method. In the discretization of time dependant equations the implicit method is taken into account. In the derivation of the heat equations, parameters such as the duration of braking, vehicle velocity, geometries and the dimensions of the brake components, materials of the disk brake rotor and the pad and contact pressure distribution have been taken into account. Results show that there is a heat partition at the contact surface of two sliding components, because of thermal resistance due to the accumulation of wear particles between contact surfaces. This phenomenon prevents absorption of more heat by the discs and causes brake lining to be hot. As a result, heat soaking to the brake fluid increases and may cause brake fluid to evaporate (Mazidi, S.Jalalifar, S. And Chakhoo, J. 2010).

Recently, disk brakes have been widely used in light vehicles (Faramarz, T.,Salman J. 2009). Proper performance of a vehicle brake system is one of its advantages. Long repetitive braking leads to temperature rise of various brake components of the vehicle that reduces the performance of the brake system. Long repetitive braking, such as one which occurs during a mountain descent, will result in a brake fluid temperature rise and may cause brake fluid vaporization. This may be a concern particularly for passenger cars equipped with aluminum calipers and with a limited air flow to the wheel brake systems. Braking performance of a vehicle can be significantly affected by the temperature rise in the brake components. High temperature during braking may cause brake fade, premature wear, brake fluid vaporization, bearing failure, thermal cracks, and thermally excited vibration. Therefore, it is important to predict the temperature rise of a given brake system and assess its thermal performance in the early design stage. Recently, brake fluid vaporization has been suspected as a possible cause of some collisions and a proper inspection procedure has been recommended.

Currently, brake pads in automobiles are made of composite materials composed of more than ten different ingredients. Sometimes, up to 20 or 25 different constituents are used. These ingredients are categorized into four broad classes: binders, structural materials, fillers and frictional additives/modifiers (Mohanty, and Chugh, 2007). The binders bind together rest of the ingredients, structural materials provide the structural reinforcement to the composite matrix, fillers make up the volume of the brake lining, while keeping the cost down, and friction modifiers stabilize the coefficient of friction (Eriksson and Jacobson, 2000). Investigations into the effect of different ingredients on brake performance are available in literatures (Hee K W, Filip, 2005, Cho et al., 2005, Cho et al., 2006, Gudmand- Hoyer and Bach, 1999, Bijwe et al., 2005, Jang et al., 2000, Kim and Jang, 200, Jang et al., 2001, Jang et al., 2003).

A mathematical model to describe the thermal behavior of a brake system (Naji et al,2002) was presented which consists of the shoe and the drum. The model was solved analytically using Green's function method for any type of the stopping braking action. The thermal behavior is investigated for three specified braking actions which were the impulse, the unit step and trigonometric stopping action.

An analytical model presented by (Gao and Lin, 2002) for the determination of the contact temperature distribution on the working surface of a brake. To consider the effects of the moving heat source (the pad) with relative sliding speed variation, a transient finite element technique is used to characterize the temperature fields of the solid rotor with appropriate thermal boundary conditions. Numerical results shows that the operating characteristics of the brake exert an essentially influence on the surface temperature distribution and the maximal contact temperature.

In a study by (Talati and Jalalifar, 2008), two major models are used for calculation of frictional heat generation: namely macroscopic and microscopic model. In the macroscopic model, the law of conservation of energy or first law of thermodynamics is taken into account. And for the microscopic model, parameters such as the duration of braking, velocity of the vehicle, dimensions and geometry of the braking system, material of the disk brake rotor and the pad are taken into account. For calculation of prescribed heat flux boundary condition in their model, two kinds of pressure distribution are considered: uniform wear and uniform pressure.

II. MATERIALS

Brake disc are traditionally made from grey cast iron due to its strength and durability, stable mechanical and frictional properties, wear resistance, heat absorption, thermal conductivity, vibration damping capacity and low cost. While brake pads are typically a manufacturer proprietary composite of different friction materials based on the desired performance. Brake pads material blends are categorized as metallic, semi-metallic, organic, or carbon/ceramic depending upon the material composition. The Finite

Element Analysis (FEA) applied elastic material properties of Society of Automotive Engineer (SAE) SAE J430 G10H19 grey cast iron for the disc. The brake pad material compositions are varied throughout this project to investigate the variation in braking performance. The material properties used in the FEA for each component are summarized in Table 1.

Table 1: The material properties used in the FEA

Friction component	Mass Density (Kg/m^3)	Elastic Modulus (Gpa)	Poission Ratio
Disc (Steel)	7150	140	0.24
Pad (Ceramic Al 204)	3800	325	0.22

A mathematical model is use for the determination of the contact temperature distribution on the working surface of the brake. First to consider the effects of the moving heat source (the pad) with relative sliding speed variation, a transient finite element technique is used to characterize the temperature fields of the solid rotor with appropriate thermal boundary conditions.

For calculation of heat generation due to friction, rate of dissipated heat via friction should be taken into account. This is all to do with the calculation of friction force and rate of work done by friction force.

III. METHODOLOGY

As previously stated, temperature measurement in the friction surfaces of a brake is a difficult task. This is due to numerous influencing factors specific to rubbing surfaces such as those in friction brakes, especially since it is necessary to provide temperature measurement with an appropriate accuracy and minimum delay. When comparing all the available techniques of temperature measurement, the method using thermocouples shows significant advantages over others; they are very effective for measuring the temperature in the contact of the friction pair. In this case, a so-called “hot end” or hot junction is located very close to the friction surface. However, it must be taken into consideration that the thermocouple should not at any time be exposed to direct rubbing over the friction surface in order to eliminate the potential impact on the quality of the measuring signal of the thermocouple sliding itself over the metal surface as much as possible. This kind of problems may be avoided if the thermocouple is positioned within the pad, very close to the sliding surface, e.g. 0.5 mm deep from it (SAE J843). In the present study, one temperature sensor was located in such a position, and it will be used to measure the temperature on the friction surface (T1). However, given the requirement that the temperature at the friction surface be measured throughout the lifespan of brake pads, this position is not satisfactory due to wearing phenomena.

Therefore, another thermocouple (T2) will be positioned 12.5 mm deep from the contact surface, within the pad. The position of this thermocouple is close to the backing plate, and it is defined by the thickness of the brake lining material. This thermocouple is not exposed to the effects of wearing, which ensures its use throughout the operating period. It is important to note that both thermocouples were placed at the friction radius of the pad. The measurement was carried out at the Kaduna automobile brake system laboratory, with a car disc brake tested at a single-ended full-scale inertia dynamometer. Temperature measurements were carried out repeated with full-stop brake applications over a total time of 600 seconds, with an initial brake speed corresponding to linear vehicle speed of 60 km/h, and with the control line pressure of 60 bar, while the initial brake temperature at the beginning of the measurement cycle was 100 ºC (±3 ºC). The brake was subject to cooling by means of the fan operating throughout the measurement period. In each brake application, a brake disc was first accelerated until it reached the predetermined initial brake speed, and consequently braked to a full stop. After completing a single brake application, the brake disc remained at a standstill.

IV. FINITE ELEMENT MODEL

The finite element method is a tool to get the numerical solution of wide range of engineering problem. The method is capable to handle any complex shape or geometry, for any material under

different boundary and loading conditions. The analysis requirement of engineering problems is overcome with help of finite element method. The difference between the analytical solution and finite difference method is that the solution is obtained only at discrete points. Therefore, dividing the region into small regions and assigning each region a reference index will be provide very accurate results. Solving the circular boundary problem solved using the finite difference approach defined in polar co-ordinates (r,θ) is more convenient than using cartesian co-ordinates as it avoids the use of awkward differentiation formulae near the curved boundary. Defining the mesh in the r-θ plane by the points of intersection of the circles r=r1 + i*h, where i=0, 1, 2, 3 ..., r1 is the inner radius of the disc and h=(r2-r1)/M, M is the number of
circles defined for the mesh and the straight lines θ= j * l where j=0, 1, 2, 3.... and l=(θ2-θ1)/N , where N is the number of divisions. These lines and circles are called grid lines and their intersections are referred to as mesh points of the grid or nodal points. The distance between two nodal points is the grid sizes.

The developed finite element analysis model contains a total of 278 elements and 597 degrees of freedom, while the time step used during the numerical computation was 0.01sec. The initial temperature used during the simulation was set as 20ºC. Table 3.1 and 3.2 shows how the temperature increases further from the centre of the disk rotor to the point of the maximum temperature within the contact area between the disk and the pad, and then it decreases.

There are two methods to solve finite element problems, implicit and explicit. The implicit method requires solving for a static or quasi-steady state processes, while the explicit method uses an explicit direct-integration procedure to solve dynamic response problems. The finite element code used to analyzed the disc and pad brake system herein utilizes ABCQUS/Explicit, Abaqus 6.12 –EF3 to solve a dynamic simulation of brake pad to disc contact.

The assumption detailed to this point significantly simplify the numerical analysis. However, a disc brake has fairly symmetric configuration about Z-axis as shown in Figure 2. The final finite element model (FEM) assembly developed for this study is thus 3D axisymmetric representation of disc brake. The FEM geometry has half of the disc thickness, and engages the disc on one surface with one pad, utilizing a symmetric boundary condition about the Z-axis.

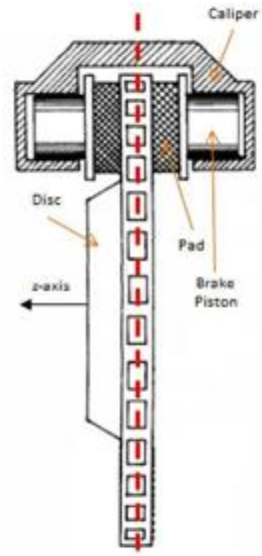

Figure 2 – End View of Disc Brake Highlighting the Symmetry

V. FINITE ELEMENT FORMULATION FOR HEAT CONDUCTION

The unsteady heat conduction equation of each body for an axisymmetric problem described in the cylindrical coordinate system is given as follows;

$$\varrho c \frac{\partial T}{\partial t} = \frac{1}{r}\ \partial\partial\mathrm{r}\left(rkr\frac{\partial T}{\partial t}\right) + \frac{\partial}{\partial z}\left(kz\frac{\partial T}{\partial z}\right) \qquad (1)$$

With the boundary conditions and initial condition

T= T* on T_0, (2)
q_n=h(T-T∞) on T_1(3)
q_n=q*n onT_2(4)
T=T_0attime=0(5)

Where,
Q, c. Kr and kz are the density, specific heat and thermal conductivities in r and z direction of the material respectively.

T* is the prescribed temperature
h is the heat transfer coefficient

q*n is the heat flux at each contact interface caused by friction

$T\infty$ is the ambient temperature
T_0 is the initial temperature
t_0 , t_1 and t_2 are the boundaries on which temperature, convection and heat flux are imposed.
A finite element formulation of unsteady heat equation can be written in the following matrix form as;
$C_T\dot{T} + KH_T T = R,$ (6)

Where C_T is the capacity matrix
KH_T is the conductivity matrix
T and R are the nodal temperature and heat source vector

The above equation is solved with the direct integration method based on the assumption that the temperature Tt at time t and the temperature $Tt + \Delta t$ at time $t + \Delta T$ have the following relation:

$Tt + \Delta t = Tt +[(1 – \beta)\ \dot{T}t + \beta\dot{T}t + \Delta t]\ \Delta t.$ (7)
Equation above can be used to reduce the ordinary differential equation of equ. (7) to the following implicit algebraic equation:

$(C_T + b_1\ KH_T)\ Tt+ \Delta t = (C_T – b_2\ KH_T)\ Tt+ b_2Rt + b_1Rt + \Delta t ,$ (8)

Where the variable $b_1$and $b_2$are given by
$b_1= \beta\ \Delta t,$ $b_2 = (1- \beta)\ \Delta t$ (9)

For different values of β, a well-known numerical integration scheme can be obtained.
β ranges from $0.5 \leq \beta \leq 1.0$

Bearing all this in mind, the model for the brake contact surface temperature estimation was developed in the following general form based on polynomial regression:
$T_E(t) = T_2 (t).k + dt_2/dt .t.k_v + d^2T_2 / dt^2.t^2/2 .ka.$ (10)

Where T_E is the estimated brake interface temperature, T_2 the temperature measured within the pad, t the time, k the temperature ratio, and kv, and ka are coefficients representing speed and acceleration of temperature increase intensity, i.e. representing the coefficients of heat transfer through the friction material. These coefficients also depend on the composition of the brake pad friction material, wear status, brake geometry and operating conditions, as well as the position of thermocouple T_2 in the depth of the friction material. The coefficient of the speed of temperature increase intensity kv is determined by fitting the estimated temperature T_E to the measured temperature T_1. This process is performed in the sequence of monotonous temperature change in order to avoid the influence of coefficient ka from Eq. (2). Because the estimated temperature T_E change does not fully correlate to that of the temperature T_1, neither by value nor by shape, the value of the coefficient kv was calculated with a simple linear regression method based on the least squares criterion using measurement data of T_1 and T_2 temperatures. Now, the coefficient ka remains the only unknown parameter in Eq. (2). The least squares criterion will give best value for parameter ka for the process model using the polynomial regression method and between the measured temperatures T_1 and T_2. After this procedure, the estimated temperature T_E could be calculated. The thus obtained estimated (i.e. calculated) brake contact surface temperature T_E is presented and it can be seen that estimated temperature T_E follows both the shape and the character of the behaviour of measured temperature T_1 very well, which seems to deviate from the actual value by not more than ± 3 ºC. This deviation can be assumed to come from the noise of the measurement signal, being the result of conditions by which measurement was carried out, and a high sampling rate selected, even though it is a relatively slow process.

VI. PART GEOMETRY AND MESH

The brake FEM uses a simplified version of a typical domestic passenger vehicle's disc brake system. Figure 3 shows a simplified 3D solid disc, which allows the model to use coarser meshes than would be normally required to model the details of a typical brake disc, given its complicated geosmetrical features such as cooling ducts and bolt holes. Figure 3 also shows simplified 3D pad geometry and is modeled such that it can only contact part of the friction surface of the disc.

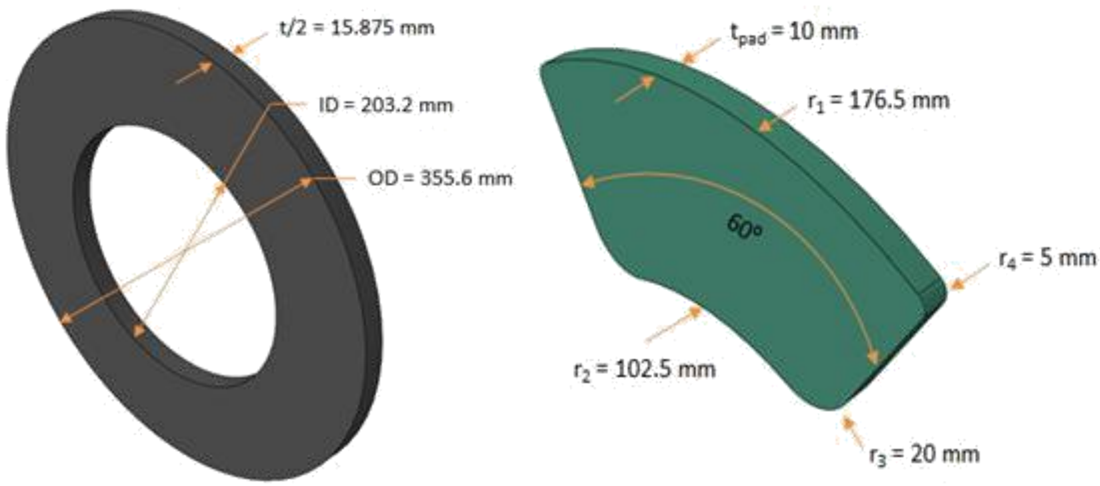

Figure 3 – 3D FEM Geometry Disc (left) and Pad (right)

The disc has an outside diameter of 355.6 mm, an inside diameter of 203.2 mm, and a thickness of 31.75 mm. The pad surface area is 10,700 mm^2. Each part is meshed independently of the FEM assembly. Figure 4 illustrates the disc and pad mesh, and using the ABAQUS mesh quality verification tool all 1576 elements (1408 disc and 168 pad) were queried resulting in zero analysis errors. The simplicity of the uniform mesh improves solver performance considering the complexity that contact adds to the FEA.

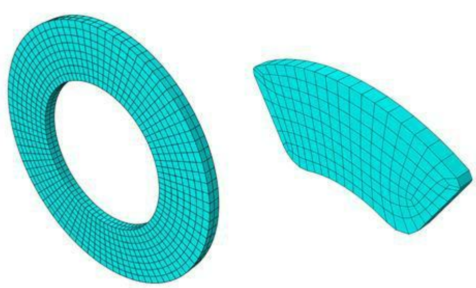

Figure 4 – FEM Mesh Disc (left) and Pad (right)

Both disc and pad utilize eight-node linear brick (C3D8) fully integrated elements in order to give more resolution at the surface than reduced integration (C3D8R); and artificial incompressible bending or locking is not a concern in this analysis. Both disc and pad contact surfaces are modeled as flat, so a quadratic or curved surface definition is unnecessary

VII. UNIFORM PRESSURE DISTRIBUTION:

When the pad is new and short enough the pressure is distributed uniformly in the contact area that is:

$$p = p_{max} \tag{11}$$

- Uniform Wear

After several braking action when the pad is rundown this type of pressure distribution is taken into account because work done by friction force grows when the radial distance increases and eventually the farther radial position is more encountered to wear and the assumption of uniform pressure distribution is no longer dominant.

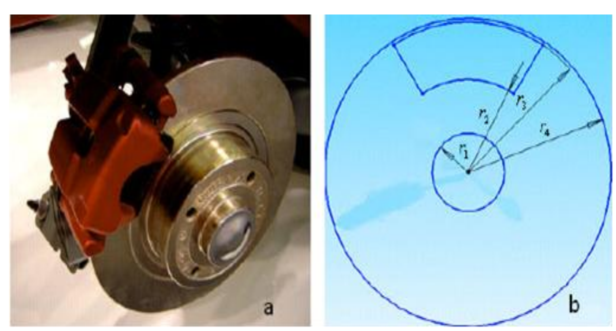

Fig. 1:(a) Disk brake system and caliper assembly and (b) schematic shape of the disk and the pad in sliding contact

In this situation pressure distribution in the pad is proportional to 1/r and maximum pressure occurs at r = r_2.

$$\delta = kpr = const \Rightarrow p = p_{max}\frac{r_2}{r} \tag{12}$$

Where, δ is wear, p_{max} is the maximum pressure distributed in the pad and p is the pressure at radial position r.

VIII. HEAT GENERATION DUE TO FRICTION

In the contact area of brake components; the pads and the disk; heat is generated due to friction. For calculation of heat generation at the interface of these two sliding bodies' two methods is suggested:

- At the basis of law of conservation of energy the kinetic energy of the vehicle during motion is equal to the dissipated heat after vehicle stop
- By knowing the friction coefficient, pressure distribution at the contact area, geometric characteristics of the pad and the disk, relative sliding velocity and duration of braking action one can calculate the heat generated due to friction

IX. HEAT EQUATION

Contact surface element of the disk and the pad is shown in Fig. 2a and b, respectively. The rate of heat generated due to friction between these surfaces is calculated as follows:

$$d\dot{E} = dP = VdF_f = r\omega\mu p\phi_0 rdr \tag{12a}$$

$$d\dot{E} = d\dot{E}_p + d\dot{E}_d \tag{12b}$$

$$d\dot{E}_p = (1-\sigma)dP = (1-\sigma)\mu p\omega\phi_0 r^2 dr \tag{12c}$$

$$d\dot{E}_d = \sigma dP = \sigma\mu p\phi_0\omega r^2 dr \tag{12d}$$

where, $d\dot{E}$ is the rate of heat generated due to friction between two sliding components, V is the relative sliding velocity and dF_f is the friction force. The terms $d\dot{E}_p$ and $d\dot{E}_d$ are the amount of absorbed heat by the pad and the disk, respectively. The parameter σ is defined as:

$$\sigma = \frac{\xi_d S_d}{\xi_d S_d + \xi_p S_p} \tag{13}$$

where, ξ_p and ξ_d are the thermal effusivities of the pad and the disk and S_p and S_d are frictional contact surfaces of the pad and the disk, respectively. Thermal effusivity is defined as:

$$\xi = \sqrt{k\rho c} \tag{14}$$

Heat flux: To obtain the heat flux at the surfaces of two components of the brake system, we divide rate of thermal energy by the surface contact area of each component.

- Heat flux in the pad

$$q_1(r,t) = \frac{d\dot{E}_p}{dS_p} = (1-\sigma)\mu p(t) r\omega(t) \tag{15}$$

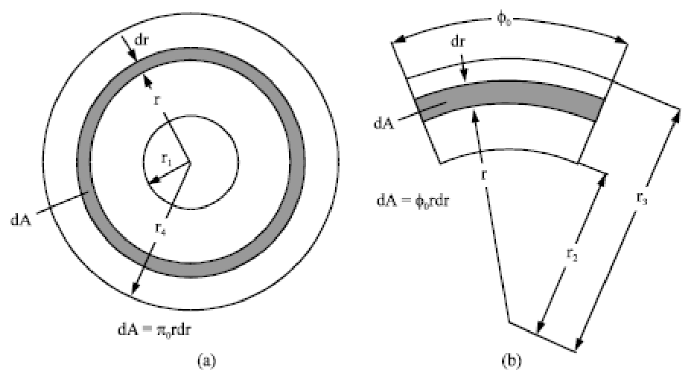

Fig. 2:Contact surface element of two components (a) the disk and (b) the pad

- Heat flux in the disk

$$q_2(r,t) = \frac{d\dot{E}_d}{dS_d} = \frac{\phi_0}{2\pi}\lambda\mu p(t) r\omega(t) \tag{16}$$

Heat flux for uniform pressure is a function of time and space variable r; the angular velocity decreases with time during braking action and the work done by friction force grows as radial space variable increases. Heat flux obtained for the uniform wear is just a function of time and it is independent of the space variable; the work done by friction force is the same at radial direction.

X. HEAT EQUATION FOR THE PAD

Figure 3shows the two dimensional thermal problem of the pad for two assumption of pressure distribution; uniform wear and uniform pressure in Fig. 3a and b, respectively. Heat equation and the appropriate boundary conditions for the pad may be written in the following form:

$$\frac{\partial^2 T}{\partial r^2}+\frac{1}{r}\frac{\partial T}{\partial r}+\frac{\partial^2 T}{\partial z^2}=\frac{1}{\alpha}\frac{\partial T}{\partial t} \ ; \ r_2<r<r_3 \ , \ 0<z<d_1 \ , \ t>0 \quad (16a)$$

$$-\frac{\partial T}{\partial r}+HT=HT_{\infty} \ ; \ r=r_2 \ ; \ 0\le z\le d_1 \ , t\ge 0 \quad (16b)$$

$$\frac{\partial T}{\partial r}+HT=HT_{\infty} \ ; \ r=r_3 \ ; \ 0\le z\le d_1 \ , t\ge 0 \quad (16c)$$

$$\frac{\partial T}{\partial z}=q_1(r,t) \ ; \ z=0 \ ; \ r_2\le r\le r_3 \ , t\ge 0 \quad (16d)$$

$$\frac{\partial T}{\partial z}=0 \ ; \ z=d_1 \ ; \ r_2\le r\le r_3 \ , t\ge 0 \quad (16e)$$

$$T(r,z,0)=T_0 \ ; \ r_2\le r\le r_3 \ , \ 0\le z\le d_1 \quad (16f)$$

- Discretization of heat conduction problem of the pad

Figure 4 shows meshing of the finite difference method of the pad. For discretization of the equations the variables r, z and t are defined as:

$$r=i\Delta r; \ i=0,1,2,\dots,48 \quad (16g)$$

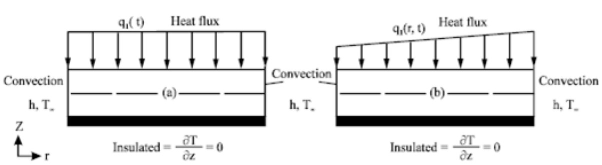

Fig. 3:Boundary conditions for the pad (a) uniform wear and (b) uniform pressure

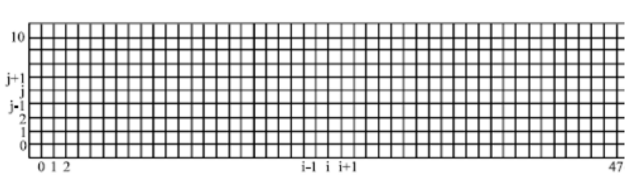

Fig. 4:Finite difference discretization of the pad

$$z=j\Delta z; \ j=0,1,2,\dots,10 \quad (16h)$$

$$t=n\Delta t; \ n=0,1,2,\dots \quad (16i)$$

Energy absorbed by the brake

Energy absorbed by a brake depends on the type of motion either pure translation or pure rotation or combination of both.

When it is pure translation;

$$E_1 = \frac{1}{2} M \{(V_1)^2 - (V_2)^2 \} \quad (17a)$$

At a stop,

$$E_1 = \frac{1}{2} M \{(V_1)^2 \quad (17b)$$

When pure rotation;

$$E_2 = \frac{1}{2} I \{ (\omega_1)^2 - (\omega_2)^2\} \quad (17c)$$

At a stop,

$$E_2 = \frac{1}{2} I \{ (\omega_1)^2 \quad (17d)$$

XI. RESULTS AND DISCUSSION

From Table 3.1, the pad highest surface temperature value is 780°C and the lowest surface temperature value is 500°C, the difference of temperature is 280°C. The reason for this is that the thermal conductivity parameter is low and specific heat value is not high. Taking the wearing rate into account, the surface temperature of the axial position from Table 3.2 the pad highest surface temperature is 8800C tand the lowest surface temperature value is 700°C, the difference of temperature is 130°C. Considering the wearing rate in radial and axial distance when the rate of wear rises most in radial direction, the temperature occurring on the surface and the temperature difference between two surfaces also rises. In other radial distance as well, with the increase of wear rate, the temperature value occurring on the pad surface and the temperature difference between two surfaces rises as well.

Table 3.1 Temperature of Pad and Disc at different time (uniform wear)

Max Disc Temp	Max Pad Temp	Time (sec)
100	500	0.5
180	730	1.0
190	770	1.5
200	780	2.0
200	760	2.5
205	730	3.0
205	700	3.5
205	700	4.0

Table 3.2 Temperature of Pad and Disc at different time (uniform pressure)

Max Disc Temp	Max Pad Temp	Time (sec)
180	700	0.5
200	840	1.0
250	870	1.5
300	880	2.0
300	870	2.5
310	840	3.0
300	800	3.5
280	700	4.0

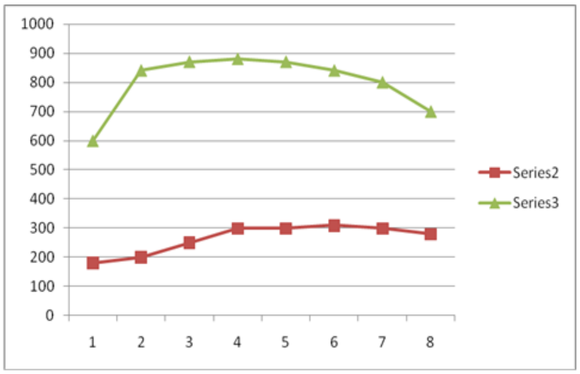

Fig 4.1 Temperature of Pad and Disc at different time (uniform pressure)

Fig 4.1, a higher temperature of 870°C is achieved. That is the case for uniform pressure is more pronounced than the case of uniform wear. At the start of brake friction was low. So heat generated was also low. At a time of 1.5 to 2.5 sec maximum heat was generated and the temperature of pad and disc rises. Above 2.5 however, convection heat lost will set in, which will reduced the temperature in the interval 2.5 to 4 sec. In addition, it is the kinetic energy of the moving car or vehicle that is converted to heat energy. When the brake is applied the kinetic energy gradually reduces which also lead to fall in temperature in the region 2.5 to 4 sec

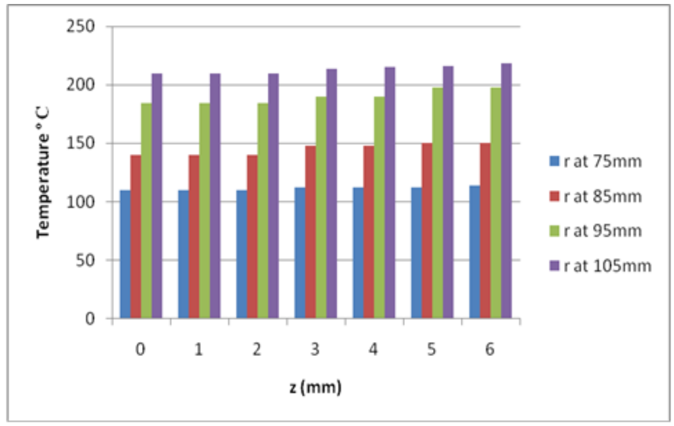

Fig 4.2 Disc Temp in axial direction at different radial position

As it is clear from Fig 4.2, disk surface temperature in the radial interval 75-85 mm, where the disc is exposed to air flow is relatively low. But in the interval 85-105 mm, where the disc is in contact with friction linings, the temperature will increase because in the case of uniform pressure distribution, heat generation grows in the radial distance. However, because the disk is exposed to air, convection heat lost will set in, which will reduced the temperature in the interval below 85mm

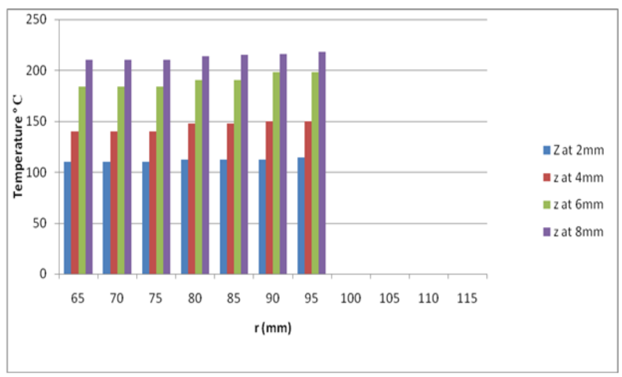

Fig 4.3 Disc Temp in radial distance at different axial position

As it is clear from Fig 4.3, disk surface temperature in the axial interval 2-4 mm, where the disc is exposed to air flow is relatively low. But in the interval 6 - 8 mm, where the disc is in contact with friction linings, the temperature will increase because in the case of uniform pressure distribution, heat generation grows in the axial distance.

XII. CONCLUSIONS AND RECOMMENDATION

Results for contact surface temperature of the pad and the disc show that there is a heat partition at the contact surface of two sliding components. This is because of thermal resistance due to the accumulation of wear particles between contact surfaces of the pad and the disc and lack of necessary provisions to convection heat loss through the disk to the surrounding.

Contact thermal resistance at the surfaces of the pad and the disk prevent absorption of more heat by the discs and causes brake lining to be hot. As a result, heat soaking to the brake fluid increases and may cause brake fluid to evaporate. So, the brake fluid regarding to minimum wet and dry boiling point with appropriate DOT ratings should be used.

As you know, caliper assembly is located at the end of the pad. If the pad becomes too hot, it may increase the temperature of the brake fluid and causes evaporation. Therefore, it is recommended that the material with low thermal conductivity for the pad and caliper components is used.

If the thickness of the pad is reduced due to excessive wear, influence of heat into the pad and caliper assembly increases and the risk of brake fluid vaporization will increase.

In this study, the temperature distributions and stress situations of the pad materials during 2s occurring as a result of permanently braking were examined using finite element method. In the study, in order to consider the impact of pad's wear, the pad thicknesses were examined as well.

In this paper, the governing heat equations for the disk and the pad are extracted in the form of transient heat equations with heat generation that is dependant to time and space. In the derivation of the heat equations, parameters such as the duration of braking, vehicle velocity, geometries and the dimensions of the brake components, materials of the disk brake rotor and the pad and contact pressure distribution have been taken into account. The problem is solved analytically using Green's function approach. It is concluded that the heat generated due to friction between the disk and the pad should be ideally dissipated to the environment to avoid decreasing the friction coefficient between the disk and the pad and to avoid the temperature rise of various brake components and brake fluid vaporization due to excessive heating.

It is recommended building a test rig for real braking system to study the heat dissipation experimentally so that the analysis result obtained from finite element software package can be validated. This small step towards the development of a more thermally stable brake system will give design engineers a hands on quick on the go option during the initial stages of prototype designing for the dimension and material selection which could be further modified during the later stages of product development.

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