

Design, Analysis and Behavioural Assessment of Hydraulic Braking System

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ABSTRACT: Optimum design of any system leads to an improvement in efficiency and its reliability. Thus an effort is made to design an optimum hydraulic braking system thus improving the efficiency, ease of assembly and reliability of the system. Many parameters play an important role for achieving the same. There is also the case of compatibility of OEM components with customized components. Many simulations including the computational fluid dynamics has been carried out for this. To validate the obtained results an experimental test rig is manufactured and the results are validated using flexi force sensors, thermocouples and thermal imaging camera. These results are further used to enhance the performance of the braking system and of the overall vehicle.

KEYWORDS: Efficiency, Hydraulic, Braking, Rig, Sensors, Thermocouples, Performance

I.INTRODUCTION

Safety in any vehicle is of prime importance and is given utmost priority in its design and development. Braking system is one of the most critical system in the vehicle. If brakes malfunction, the result can be disastrous. Braking system works on the energy converting system that converts the kinetic energy of moving vehicle into heat by stopping all the four wheels. When pressure is applied to lever, it pressurises the fluid inside master cylinder and transmits the pressure to all the four wheels by use of hydraulic lines. Piston in the caliper pushes the brake pads and friction between brake pads and disc reduces the speed of the vehicle. Hydraulic brakes work on principle of Pascal's law and the advantage of it is that it is easy to achieve force multiplication or force division. The importance of hydraulic system is that it provides reliable braking action on all wheels, increased braking force, flexibility of brake linings and increased mechanical advantage.

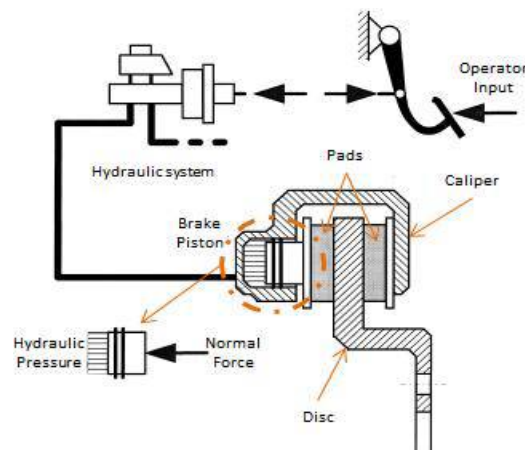


Fig 1 Schematic of a disc brake system

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The brake disc, also called the rotor, is the rotating component within the brake system connected to the wheel hub. The disc provides the friction surface for the pads, which generates braking torque. Brakes disc are commonly vented or cross drilled to facilitate the dissipation of heat.

II. LITERATURE SURVEY

This literature review aims to cover an overview on the published research by Liles, G.D (2007) [1] into brake system which includes research on frictional evolution of brake pads materials and temperature analysis. The purpose of this study is to determine methodology by identifying potential area and working on it.

A dynamic instability is induced in the braking system due to the friction between the brake pads and the disc. Consider a vehicle moving at a certain speed. Sometimes on application of brake there is a high frequency sound or noise generated. This noise in mechanical terms is known as a squeal. The factors responsible for the generation of squeal phenomenon can be coupling of two adjacent vibrations modes. If the frequency modes of the braking system and the vehicle match, then resonance occurs and the system becomes unstable. These unstable modes can be determined by extracting them from their eigenvalues. If the value corresponding to unstable mode is positive, then it can be identified. The instability generated can be eliminated by-

- Changing the geometry of the braking system
- Changing the material of the Brake pads

To determine the squeal Liles, G.D (2007) carried FEA (Finite Element Analysis). In this method static analysis of the disc is done. Then a contact is established between the pad and the disc by applying pressure to the piston of the caliper. A contact pressure distribution is obtained between disc and the pad when it is not rotating. Then a steady state rotational motion is obtained by giving some rotational velocity to the disc as a predefined field variable. This gives the modeling of steady state frictional lining between the bodies that are moving with different velocities. At last a steady state braking condition is obtained. Initial stress in the model and steady state braking condition is obtained by this step. The real eigenvalues are then extracted and they provide a subspace to determine the complex eigenvalues. To determine this, complex modes analysis is used.

The real part of the complex eigenvalue becomes positive for certain parameters. The corresponding mode for that parameter becomes the unstable mode and can lead to the generation of squeal. This unstable mode is also known as unstable squeal. This has been tested experimentally. The parameters like caliper pressure and friction coefficient on the unstable modes were studied. Fig 2 shows the determined value of the complex eigenvalue when the coefficient of friction is 0.4 and caliper pressure is 1.5 MPa. This graph shows the eigenvalues at different frequencies. Advantages of determining squeal was it will improve the performance of braking system, less wearing of brake pads, quiet functioning of brakes.

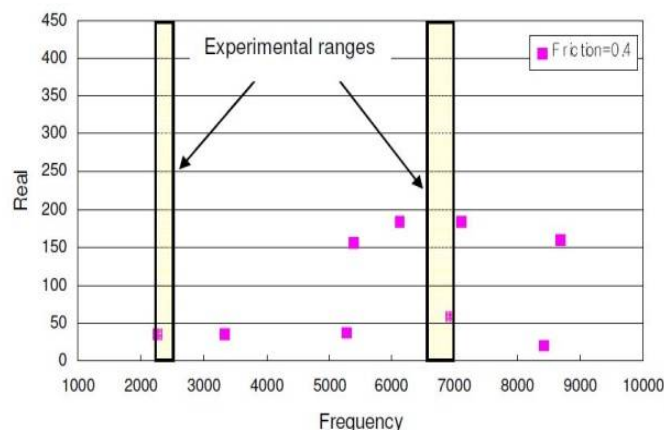


Fig 2 Predicted Squeal modes at 2.2 kHz and 6.9 kHz lie in the bands observed experimentally

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The disc or rotor used in disc brake is usually made up of gray cast iron due to its excellent heat dissipation properties. Earlier asbestos was used for the manufacturing of brake pads due to its high strength, resistance to heat and fire-proof properties. But it was banned from the use as it causes lung cancer and other health problems. So, instead of asbestos, non-asbestos materials were used such as Kevlar, glass fibre, graphite etc. Friction materials such as binder resin and reinforcing fibres have a substantial influence in determining the frictional characteristics. On application of brakes the disc gets heated up and it transfers the small amount of friction material onto the disc or pad. Application of brakes can easily raise the temperature at the interface beyond the glass transition temperature of the binder resin which can change the braking force. This happens because of degradation of binder resin and other constituents. Usually two or three different fibres from metallic, glass, ceramic and other fibres are used to improve physical strength and frictional properties.

K.W. Liew, Umar Nirmal (2013) [2] evaluated and investigated the tribological properties of two brake pad materials, NABP (non-asbestos brake pad material) and ABP (asbestos brake pad material) and compared them with CMBP (commercial brake pad material). A pin-on-disc Tribo-test machine was used to carry out Tribo-test and the frictional materials were characterized by measuring their hardness and ultimate compression stress. All the tests were conducted under gray cast iron brake disc.

The composition and dimensions (9.5 mm x 9.5mm x 20 mm) of the ABP and NABP is shown in the fig 3(a) and fig 3(b) respectively [2]. Brinell hardness tester with a load of 2 kN was used to measure the surface hardness of the brake pad materials to ensure uniform mixing and proper curing of brake pads during manufacturing. Compression test was also performed on all the three materials (ABP, NABP and CMBP) to obtain the ultimate compressive stress.

Raw materials	Sample code (Vol.%)	
	ABP	NABP
Asbestos fiber	10	-
Metal fiber-steel fiber	15	15
Friction modifiers-brass, cashew dust	16	16
Solid lubricant-graphite (C)	8	8
Abrasive-zircon ($ZrSiO_4$)	3	3
Binder (matrix)		
Phenolic resin	20	20
Styrene butadiene rubber (SBR)	-	10
Fillers, reinforcements	28	28



(a)

(b)

Fig 3(a) Composition of ABP and NABP (b) Specimen dimensions

Brinell hardness tester with a load of 2 kN was used to measure the surface hardness of the brake pad materials to ensure uniform mixing and proper curing of brake pads during manufacturing. Compression test was also performed on all the three materials (ABP, NABP and CMBP) to obtain the ultimate compressive stress.

Aleksandar Grkić, Davorin Mikluc, Slavko Muždeka, Živan Arsenić and Čedomir Duboka(2015) have discussed the possibilities of estimating both the character in changes and the behavior of the disc and the brake pad with change in temperature [3].The model developed here is based on the continuous temperature readings achieved from the braking process. For achieving continuous temperature data two thermocouples are placed in the brake pad of the braking system. Thermocouples were used because they are very effective for measuring the temperature in the contact of the friction pair. However, the following precautions were taken care of. The thermocouple was not exposed to any time rubbing directly. This problem was avoided by placing the thermocouple inside the brake pad within the thickness of 0.5mm, let this be Thermocouple T1. However, the problem with the thermocouple was, after certain operations the brake pad would wear off giving exposure to the Thermocouple therefore another Thermocouple T2 was placed within 12.5mm deep from the contact surface. The following Fig 4 shows the placement of both the thermocouples inside the

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brake pad.

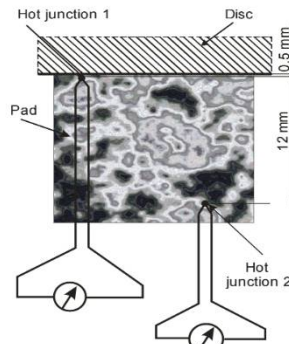


Fig 4 Position of thermocouples in brake pad [3]

The results of both the thermocouples are placed below. The solid line shows temperature measurement result at sliding surface whereas the dashed dot line shows the temperature measurement result at 12.5 mm deep from the friction pad. The model for estimation of temperature depends on the contact surface of the disc and the pad is based on the temperature ratio (k), which is determined experimentally as a rate between temperature T_1 and temperature T_2 at hot junction 1 and 2 respectively. The value of k factor varies between $K_{min}=1.14$ and $K_{max}=1.32$. It may be seen from figures that k factors certainly depend and varies on whether the brake was in the warming or cooling phase.

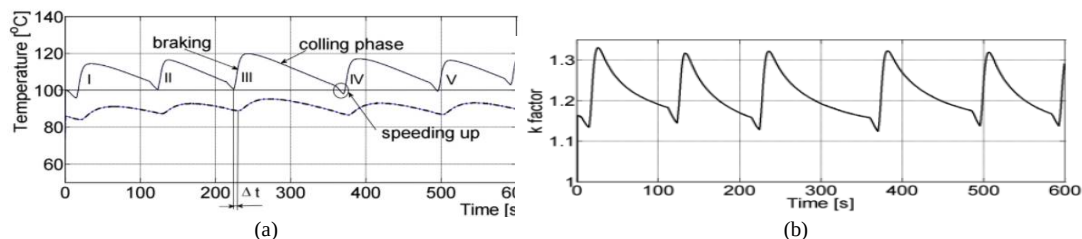


Fig 5(a) Temperature measured in the disc pad for 5 consecutive brake applications (b) K factor between temperatures T_1 and T_2

Temperature measurement were carried out during 5 consecutive full stop brake application over a total time of 600 seconds, with an initial brake speed corresponding to 60 km/h with the control line pressure of 60 bar.

Sanket Kothwade, Aditya Patankar, Rohit Kulkarni and Sameer Ingale have determined the heat transfer coefficient of brake rotor disc using computational fluid dynamics simulation [4]. Basic goal was to design, analyze and investigate the temperature distribution along rotor disc with help of numerical tools. Further this analysis is used to identify the critical temperature of the rotor and heat flux distribution for rotor with various grooves pattern and also evaluated performance under severe braking conditions which were helpful for design. For designing, braking force taken into consideration was more than required to generate because it directly affect good control and safety feature. Analytically heat transfer is calculated by assumed heat transfer coefficient, as forced convection had taken place on contact surface during rotation and on cylindrical external and internal surface.

For CFD simulation a quarter disc model was created as the disc is symmetrical about both axis and also solver time was reduced. Methodology used for CFD analysis was in three steps

- Preprocessing - Geometric modeling, Grid Generation
- Solver - Geometric modeling, initial Condition and Boundary Condition, Selection of flow model
- Post - Processing Contour Plots

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Disc with 4 grooves	182W/m ² k
Disc with 8 grooves	187W/m ² K
Disc with 16 grooves	187W/m ² K
Disc with 24 grooves	234W/m ² K

Table 1 Results showing variation of heat transfer coefficient

From results, the provision of grooves improved the heat transfer coefficient at the proximity of the grooves. Heat dissipation rate increased with no of grooves. Analytical solution and Numerical solution differed slightly; it is due to provision of grooves which is not taken into consideration in analytical calculation.

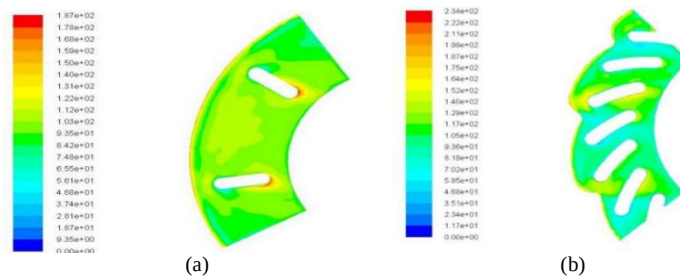


Fig 6(a) Contour plot for disc with 8 grooves (b) Contour plot for disc with 24 grooves [4]

III. METHODOLOGY

The basic aim is to provide sufficient torque to overcome the mass transfer of the vehicle. Brake biasing is done in the ratio of 50:50 so that equal pressure is generated in the front as well as the rear caliper, pedal ratio, caliper bore size plays very crucial role in braking performance. Pressurized fluid creates frictional force which squeezes the rotor attached to rotating wheel. Comfortable and ergonomic vehicle controls eliminate driver fatigue and ensures safety during panic braking situations.

A. Function of brakes subsystem:

1. Reduction of velocity of the vehicle
2. Prevent unwanted acceleration
3. Stopping of vehicle
4. Aid manoeuvrability

Specifications	Value
Pedal Ratio	6:1
Master Cylinder (bore)	Wilwood (19.05mm)
Caliper	Custom single piston floating (front) Custom dual piston floating(rear)
Brake bias	50:50
Pedal Force	250 N
Braking Torque	205.9 Nm(Front), 185.6 Nm (Rear)
Average working pressure	35 bar
Maximum system working pressure	70 bar
Minimum hose burst pressure	810 bar
Minimum hose bend radius	25 mm
Tire ground friction coefficient	0.7 (gravel)
Disc pad friction coefficient	0.45
Braking Distance	8.6m

Table 2 Parameters of the braking system

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IV. CALCULATIONS

Static Load Calculations:

Front weight:

$$W_f = \%W_f \times W_t / 100$$

$$W_f = 105 \text{ kg}$$

Rear Weight:

$$W_r = \%W_r \times W_t / 100$$

$$W_r = 65 \times 300 / 100$$

$$W_r = 195 \text{ kg}$$

Dynamic Load Calculations:

$$F_d = H \times W_t \times A_x / L$$

$$F_d = 533.4 \times 300 \times 9.81 / 1574.8$$

$$F_d = 996.822 \sim 997 \text{ N}$$

Where, F_d = Dynamic force; H = height of CG; A_x = deceleration; L = wheel base

Total Normal Load on Axles:

$$L_f = (W_f \times g) + F_d$$

$$L_f = 105 \times 9.81 + 996.822$$

$$L_f = 2026.872 \text{ N}$$

$$L_r = (W_r \times g) - F_d$$

$$L_r = 195 \times 9.81 - 996.822$$

$$L_r = 916.12742 \text{ N}$$

Where, L_f = Load on front axle; L_r = Load on rear axle

Braking Force at Tyres:

$$F_{bf} = \mu_{road} \times L_f / 2$$

$$F_{bf} = (0.7 \times 2026.87258) / 2$$

$$F_{bf} = 709.4054 \text{ N}$$

$$F_{br} = \mu_{road} \times L_r / 2$$

$$F_{br} = (0.7 \times 916.12742) / 2$$

$$F_{br} = 641.289 \text{ N}$$

Where, F_{bf} = Braking force on front tyre; F_{br} = Braking force on rear tyre

Braking Torque:

$$T_{tf} = F_{bf} \times R_{teff}$$

$$T_{tf} = 709.4054 \times 0.29$$

$$T_{tf} = 205.727566 \text{ Nm}$$

$$T_{tr} = F_{br} \times R_{teff}$$

$$T_{tr} = 641.289 \times 0.29$$

$$T_{tr} = 185.97381 \text{ Nm}$$

Where, T_{tf} = front torque due to weight; T_{tr} = rear torque due to weight; R_{teff} = tyre effective radius

$$T_{bf} = T_{tf} + T_{if}$$

$$T_{bf} = 205.727566 + 0$$

$$T_{bf} = 205.727566 \text{ Nm}$$

$$T_{br} = T_{tr} + T_{ir}$$

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$$T_{br} = 185.97381 + 0$$

$$T_{br} = 185.97381 \text{ Nm}$$

Where, T_{bf} = total braking torque front; T_{br} = total braking torque rear; T_{if} = Torque due to moment of inertia front; T_{ir} = Torque due to moment of inertia rear.

Pedal Force Calculation:

$$F_p = P_{eff} \times \text{Pedal Ratio}$$

$$F_p = 250 \times 6$$

$$F_p = 1500 \text{ N}$$

Caliper Side Calculations:

$$\phi_{cal} = 32.05 \text{ mm}, A_{cal} = 0.0008 \text{ mm}^2$$

$$\phi_{mc} = 19.05 \text{ mm}, A_{mc} = 0.00029 \text{ mm}^2$$

$$F_p = 1500 \text{ N}$$

$$P_{mc} = (F_p) / (2 \times A_{mc}) = (1500) / (2 \times 0.0029)$$

$$P_{mc} = 2.5862 \text{ MPa}$$

$$P_{mc} = P_{cal} = 2.5862 \text{ MPa}$$

$$F_{cal} = P_{cal} \times A_{piston} = 2.5862 \times 0.0008$$

$$F_{cal} = 2068.96 \text{ N}$$

$$F_{tb} = F_{cal} \times N = 2068.96 \times 4$$

$$F_{tb} = 8275.862 \text{ N}$$

$$F_{rotor} = F_{tb} \times \mu_{rotor} = 8275.862 \times 0.4$$

$$F_{rotor} = 3310.3448 \text{ N}$$

$$T_b = F_{rotor} \times R_{eff} = 3310.3448 \times 0.0875$$

$$T_b = 289.655 \text{ Nm}$$

Where, F_p = force generated at pistons; PR = Pedal Ratio; P_{mc} = pressure in mc; P_{cal} = pressure in caliper; F_{tb} = total force for braking; N = no of pistons; F_{rotor} = force exerted on rotor; T_b = braking torque.

V. SELECTION CRITERIA FOR MASTER CYLINDER

Master Cylinders were selected over Tandem Master Cylinder to provide varying brake biasing as required. Master cylinders which generates minimum pressure of 2.5 Mpa were to be selected. Sufficient displacement has to be provided for obtaining complete travel of pedal. Considering the above parameters OEM Master Cylinders (Made: Wilwood) having the bore diameter of 19.05mm is selected. This master cylinder has stroke length of 28.4mm which gives the sufficient displacement required. This master cylinder is compatible to DOT3/DOT4 which are hygroscopic in nature. Various master cylinders were seen from the Wilwood catalogue. 260-10372 was the master cylinder chosen best suited for its operation.

VI. DESIGN PROCEDURE FOR PEDAL

An overhanging pedal has been designed to get maximum leverage and considering the ergonomic perspective of the driver. Leverage in the ratio of 6:1 is provided so that sufficient pressure is generated with minimum pedal effort. Proper positioning of the pedal is done to ensure that ball of foot point of the driver follows the arc made by pedal. Also its position can be varied upto few angles considering different drivers. Weighted point analysis has been carried out to determine the pedal material. Aluminium 6082 T6 is used as the pedal material.

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VII. SELECTION CRITERIA FOR BRAKE HOSE

DOT3/DOT4 is opted as the brake fluid considering its thermal properties and its hygroscopic nature. Hoses having rating of SAE 100 R17 are opted since they are compatible with the brake fluid. They are having the working pressure of 210 bar and the burst pressure of 810 bar. Flexible hoses are used considering the ease in its routing and for safety purposes.

VIII. DESIGN PROCEDURE FOR BRAKE CALIPER

Line geometry of the caliper and the disc was made to verify that maximum effective area is obtained between the brake pads and the rotors. Bore diameter was selected as 32.05mm which generates the sufficient amount of the braking force and the displacement required. Floating type of caliper is designed considering the packaging constraints. Slider pins are the integrated part of the upright acting as the mounting points of the caliper and reducing its overall weight. While designing of the caliper it was ensured that the bleed screw position is on the topmost direction to get its maximum efficiency and to reduce the bleeding issues. Proper grooves with sufficient tolerances has been provided to avoid any leakages. Oil seals and dust seals has been selected in reference to the Parker handbook for oil seals. Seals having material of Neoprene Butadiene Rubber is selected because of its high compatibility with the brake fluid. After doing thorough weighted point analysis Aluminium 7050-T6 was selected as the caliper material. OEM brake pads were selected.

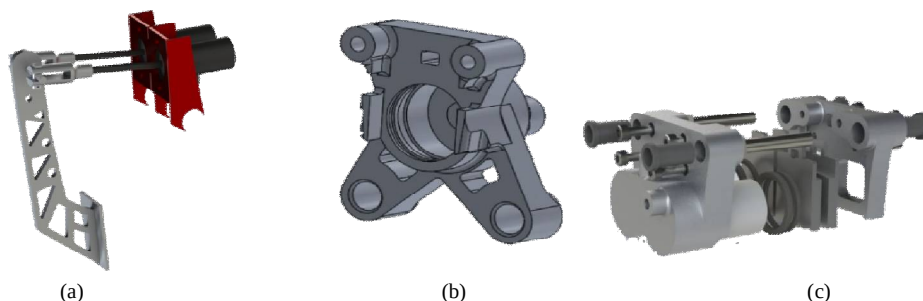


Fig 7(a) Pedal and Master Cylinder Assembly (b) Front caliper (c) Dual piston rear inboard caliper

IX. DESIGN PROCEDURE FOR BRAKE ROTOR

Brake rotor has been designed to get the minimum torque required and to ease the integration of the rotor in the wheel assembly. By varying the rotor diameter, the torque can also be varied. Front discs are having larger diameter than the rear discs since the torque required at the front is higher than the rear because of the dynamic weight transfer. Since heat generated during braking is very high slotted vents has been provided to sufficiently dissipate the heat generated. Optimization of the rotors has been done to reduce its weight. Stainless Steel 420 is selected as the material for the brake disc. Hardening of the discs is done to increase its tensile and ultimate strength.

X. ANALYSIS

Structural analysis plays a crucial role when it comes to the design of any component. A component has to be simulated before it is sent for manufacturing. The component which has to be fabricated is simulated using various simulation software like ANSYS, Hypermesh, Abaqus etc. The basic method is to provide boundary constraints wherever necessary and apply the loads which are going to act in real time. We have performed structural as well as thermal analysis of brake components and have validated the same.

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A. Structural analysis of brake pedal

The design of overhanging pedal of material Al 7050 was considered and the constraints were given on the hinged point. A force of 800N was applied on the footrest and displacement of magnitude 0.8 mm was observed. The stresses induced had a magnitude of 90.37 MPa.

B. Structural and thermal analysis of rotor

The material selected for rotor was SS420 having S_{yt} of 320MPa and S_{ut} of 400MPa having a thickness of 4mm. The material was hardened to achieve S_{ut} of 1400 MPa. The rotor was given constraints at bolting point and a torque of 200Nm was provided. Results obtained were: Maximum Stress induced: 458.8 MPa Factor of Safety: 3.5. Thermal analysis of the rotor was performed to analyse the temperature attained by the rotor for the maximum braking time of 3 seconds for 5 cycles. Temperature attained by the disc was 69°C.

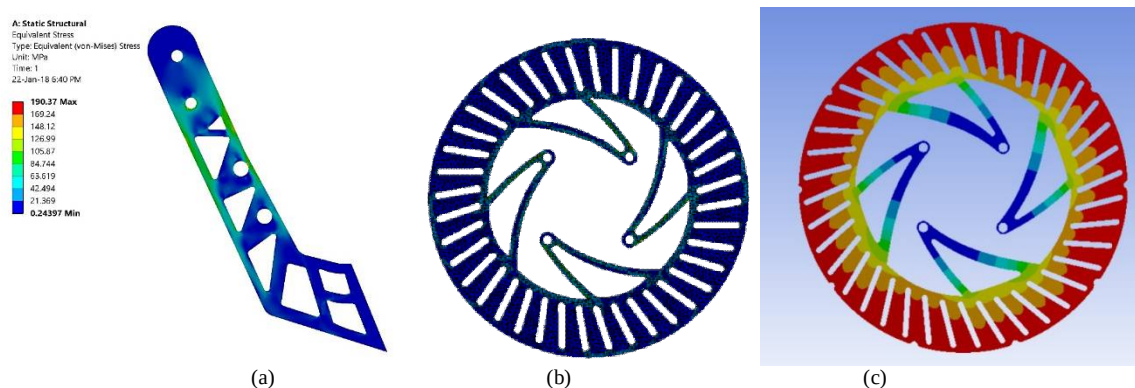


Fig. 8(a) Structural Analysis of Rotor (b) Structural analysis of Brake pedal (c) Temperature analysis of rotor

C. Computational fluid dynamics of wheel assembly

Computational Fluid Dynamics of the wheel assembly was performed to determine the air velocity flowing through the wheel assembly. This velocity of 12.5 m/s was given as an input to determine optimum number of slots in the rotor thereby maximizing the convection heat coefficient and structural stability as well.

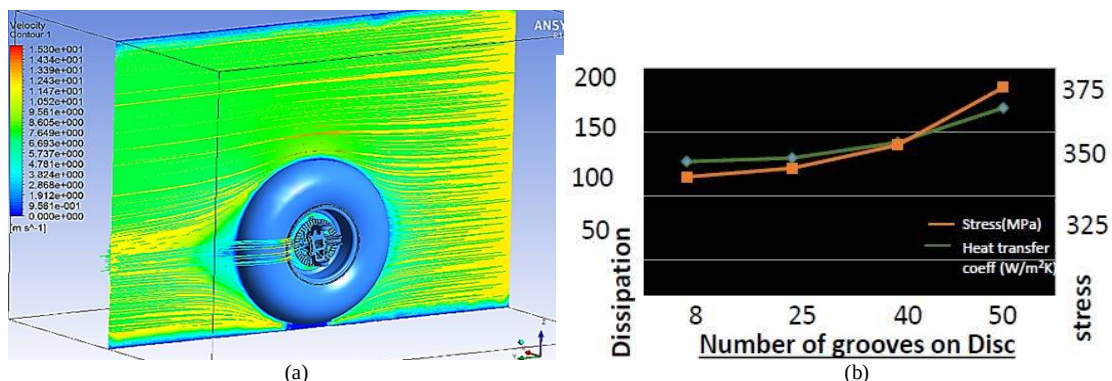


Fig. 9(a) CFD of Front Wheel Assembly to Determine Velocity at the Front Disc (b) Graph of number of slits V/S heat dissipation rate and stress distribution in the rotor.

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As the system was designed for an ATV the weight factor played a major role. In order to achieve that slits were provided on the rotor. The number of slits was decided on the basis of heat dissipation rate and structural rigidity. A graph of number of slits against heat dissipation rate and stress distribution was plotted. It was found that the heat dissipation rate increased linearly with the increasing number of slits but the stress concentration increased exponentially. So the optimum numbers of grooves or slits was determined at the intersection of these two line graphs. Figure shows the graph of number of slits V/S heat dissipation rate and stress distribution in the rotor.

D. Structural analysis and CFD of caliper

The material used for caliper was Al 7050 having S_{yt} of 469 MPa. Constrains were provided at the slider pin holding points and a pressure of 3.5 MPa was applied on the inner surface of piston bore in radially outward direction. The results obtained were: Maximum stresses induced: 289.7 MPa, Factor of Safety: 1.8.

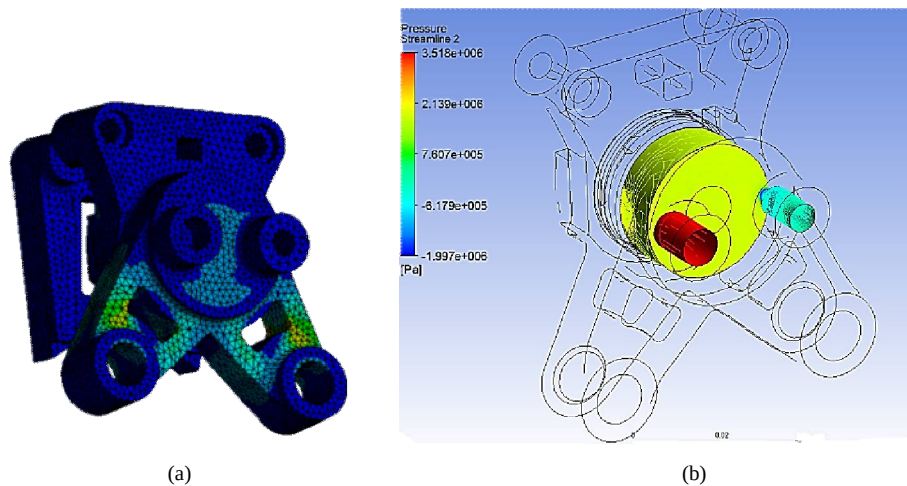


Fig. 10(a) Structural analysis of caliper (b) Computational Fluid Dynamics of Caliper to determine the pressure generated at piston end

Computational Fluid Dynamics was carried to determine the pressure that needed to be applied at piston end. This gave us a brief idea of how much pressure was needed at piston end. Then these results were validated by using flexi-force sensors which were placed between the Rotor and the Brake pads to determine the amount of force generated at Rotor-Pad interface.

E. Structural and thermal analysis of brake pads

Brake pads are made up of non-asbestos material and their analysis was performed. Simulation of a vehicle coming to a halt from 45-0 Km/h was performed. The stress induced in the pad was found to be 43 MPa. Thermal analysis was also carried out using same software and the rotor pad interface temperature was also determined. The maximum temperature attained by the brake pad was 53°C for the braking time of 3 seconds and resting time of 6 seconds for 5 cycles.

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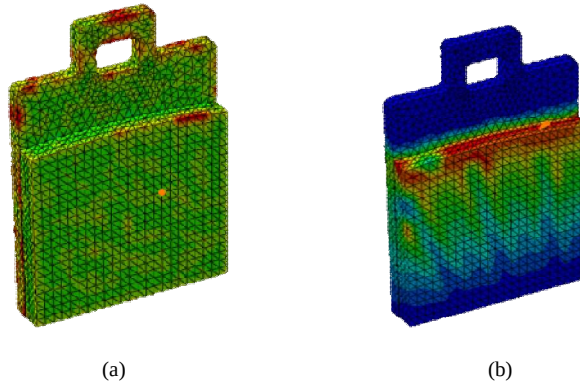


Fig. 11 (a) Structural analysis of brake pad (b) Thermal analysis of brake pad

XI. EXPERIMENTAL SETUP AND RESULT

A. Hydraulic Brake Dynamometer

A brake dynamometer is used to measure the braking force or pressure available while braking. In this case hydraulic brake dynamometer was used to measure the pressure applied at the caliper piston on the brake pads and ultimately on the rotor at different static loading condition. A static test rig was set up to calculate this pressure and validate the theoretical results. The braking system was integrated with transmission subsystem comprising of a 304cc Briggs and Stratton engine coupled with a CVT and two stage reduction gear box. The role of the braking system is to simulate variable load on gear box which is similar to the load acting on the ATV in dynamic conditions through wheel assembly.

B. Test Rig Set-up

The below set was mounted on a CRC frame. The structure holds the Briggs and Stratton engine, CVT, gear box, splined shaft with a fork welded to it. A rotor is coupled with the fork by Allen bolts and the dual piston floating type caliper is mounted on the frame at an optimum distance to ensure proper pad-rotor interface for optimum braking. This braking assembly is actuated by a M22 nut and bolt type arrangement unlike the conventional pedal mounting actuator. Wilwood master cylinder is used with the piston rod coupled with M22 bolt. A nut is placed against the mounting plate which restricts the reciprocating motion of the piston rod. This type of arrangement helps in providing varying loading condition on the gear box and the brake pads for the required amount of time.

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Fig. 12 Test Rig of hydraulic brake dynamometer

This set up was used to measure the pressure generated at the caliper end. To acquire this data flexi force sensors were used. The required parameters and the Data Acquisition (DAQ) circuit was as follows:

Engine specifications:

Specification	Value
Engine Displacement(cc)	305
Bore(mm)	80
Stroke(mm)	62
Compression Ratio	8:1
Peak Torque(Nm)	19.6Nm
Peak Power(HP)	10HP

Table 3 Engine specifications

Force Sensors

- Range: 0-1000N
- Resolution: 1N

Considering the above requirement, the Sensors and Circuits selected for the setup were A201 Flexi force pressure sensors. The sensors were controlled by Arduino Uno Microcontroller and the output was taken on the serial monitor. While acquiring the data the force applied by the caliper is measured by the flexi force sensor. Varying load was provided by the nut and bolt arrangement considering 3 seconds as braking time for 5 cycles. Fig 13(a) refers to the circuit diagram of flexi-force sensor circuit. This sensor was placed between pads and rotor and was connected to Arduino board. A code was generated in Arduino to obtain magnitude of force at pad-rotor interface. IC mounted on the board computes all the data and displays the results on the monitor. A graph of force applied against force generated was also obtained based on the results. Fig 15 shows the mounting position of flexi-force sensor.

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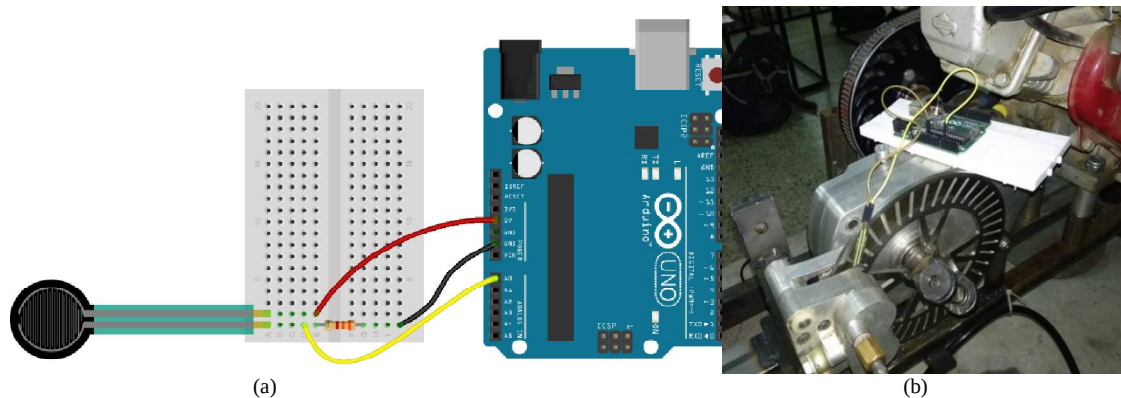


Fig.13(a) Circuit layout for flexi force sensors and Arduino for force measurement (b) Fig 5.4 Flexi-force sensor mounted on the setup

```
/* FSR testing sketch.

Connect one end of FSR to 5V, the other end to Analog 0.
Then connect one end of a 10K resistor from Analog 0 to ground
Connect LED from pin 11 through a resistor to ground

For more information see www.ladyada.net/learn/sensors/fsr.html */

int fsrAnalogPin = 0; // FSR is connected to analog 0
int LEDpin = 11;      // connect Red LED to pin 11 (PWM pin)
int fsrReading;       // the analog reading from the FSR resistor divider
int LEDbrightness;

void setup(void) {
  Serial.begin(9600); // We'll send debugging information via the Serial monitor
  pinMode(LEDpin, OUTPUT);
}

void loop(void) {
  fsrReading = analogRead(fsrAnalogPin);
  Serial.print("Analog reading = ");
  Serial.println(fsrReading);

  // we'll need to change the range from the analog reading (0-1023) down to the range
  // used by analogWrite (0-255) with map!
  LEDbrightness = map(fsrReading, 0, 1023, 0, 255);
  // LED gets brighter the harder you press
  analogWrite(LEDpin, LEDbrightness);

  delay(100);
}
```

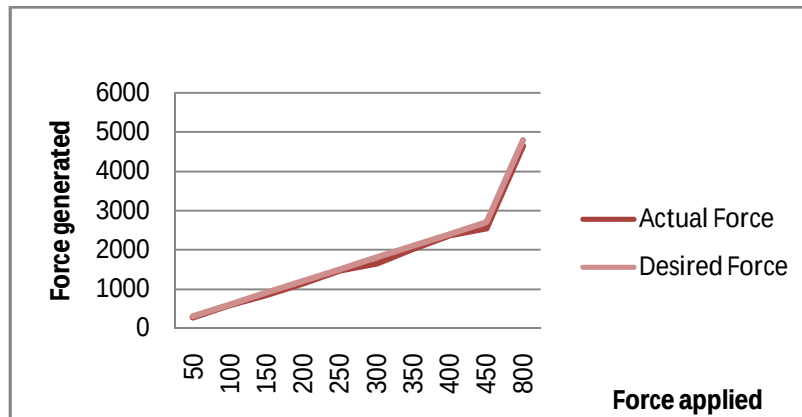
Fig. 14The code generated

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Force Applied	Force Generated
50	276
100	486
150	870
200	1140
250	1475
300	1680
350	2016
400	2360
450	2610
800	4720

Fig 15 Graph of force applied v/s force generated

From the results obtained the deviation of 6.6% is obtained from the desired results.

C. Temperature of the brake disc:

To validate the simulation results, thermal imaging of rotor was performed with the help of TESTO imaging camera. After aggressive and successive braking at 45 Km/h for 10 cycles, the temperature of the rotor was recorded. It was found that the actual value differed by just 4.4%. After completing the bleeding procedure of the hydraulic brake system brake tests were conducted to find out the stopping distance of the vehicle. It was close to that of the distance inferred from calculations. This also helped us to validate the value of coefficient of friction assumed by us and gave us a better insight of other frictional losses.

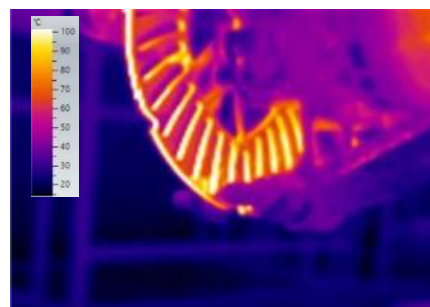


Fig 16 Image of heated disc captured using thermal imaging camera

XII. CONCLUSION

Our aim was to design, simulate, fabricate and validate hydraulic braking system. To achieve that, detailed study, design and analysis of the complete braking system was carried out. Various parameters were studied and were implemented in the design to improve the efficiency of the system. Computational fluid dynamics has also been performed to optimize the design and subsequently structural and the thermal analysis of the rotor was performed giving us the maximum temperature and stresses attained. The components were designed and manufactured not only based on their performance but also to provide the ease in the assembly. All the mounting fasteners of the caliper are

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eliminated, thus reducing the overall weight. This resulted in overall decrease in the weight of ATV. A test rig was set up for the validation of the caliper design which involved building a brake dynamometer as well as CVT dynamometer. Driver ergonomics was also considered while designing the brake pedal leading to a comfortable pedal positioning for the driver. Hence an effective hydraulic braking system was manufactured and assembled on an ATV.

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