THERMAL ANALYSIS OF DISK BRAKE IN ORDER TO STUDY THE BRAKE FLUID VAPORIZATION PHENOMENON OF A FORMULA SAE CAR

by

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Abstract

THERMAL ANALYSIS OF DISK BRAKE IN ORDER TO STUDY

THE BRAKE FLUID VAPORIZATION PHENOMENON

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The brake system is inarguable one of the most critical aspect of a vehicle safety. It has always been the major concern for design engineers to develop a system that gives a steady performance with respect to time. In order to achieve that feat, one of the most common problems that arises in maintaining a brake is the problem of brake fluid vaporization.

The race cars of the Formula SAE team at the University of Texas of Arlington face this challenge on a regular basis because of repetitive braking on curved tracks. In order to ensure safety of the vehicles, a study has been proposed in this report that deals with the problem of repetitive braking under extreme (hard) braking conditions and the temperature dependence of the brake fluid on it. The study was concentrated on finding the heat partition towards the brake disk and brake pads when brakes are applied. The theoretical results of the simulation conducted in MATLAB were later verified by the experimental ones performed on a Formula SAE vehicle of the University of Texas at Arlington.

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Chapter 1

Introduction

Braking systems have come a long way since their inception with the advent of transportation system. Over the years, the requirements for safer and reliable vehicles has propelled the necessity of superior engineering designs for the most critical aspect of a vehicle safety, that is, the braking of a vehicle.

The history of brakes goes back a long way with evidences of the Roman empire using a very crude form of braking system in the form of wooden block some 2000 years ago[1] . The wooden block was made in contact with the wheel to slow the vehicles down. The traces of this technology kept on continued to be used till as late as early locomotive period [1]. Loosely speaking, the basis of today's form of braking still is a very advanced improvisation of this technology (wood block to cast iron brake shoes). Come 20th century and there was a rapid development in the automotive industry which emphasized more on the safety. Early automobiles had band brakes which were later followed by drum brakes, both being operated with mechanical linkages. The development of hydraulic lines in the early 1920's -1940's was a major boost to the braking industry. The new concept was disk braking and it became more and more common in the passenger vehicles by 1960's. Formula cars, which were using drum brakes decided to switch to disk brake and a new dawn was started by British Connaught Team IN 1955 which eventually went ahead to win the Syrakus Grand Prix in Italy, though it was not given a world championship status [2]. It was years later in 1958 that the Ferrari team removed the disk brakes from one of their sports car and used it in Italian Grand prix [2]. Most of the Stringent rules governing the weight and safety of formula one vehicles were the major driving forces for a more reliable, durable and comfortable design. More and more number of designers are focusing on the computational optimization of various components of a braking system based on their aerodynamic and material property. Various forms of sensors are being installed to obtain on board data which are crucial for studying and eventually designing for a safer car.

Current study and Objective

The Mechanical and Aerospace Engineering Department of the University of Texas at Arlington has one of the most prestigious Formula SAE program of the world. The rigorous demands of competition and quality as per FSAE rules demands higher performance brakes with stringent safety and cost considerations. Failure of a brake system is more often than not is as a result of brake fluid vaporization under intense braking as per the designated track of the vehicle. Most of the modern day vehicle equipped with disk brakes are using DOT3 and DOT4 brake fluids, depending on their boiling temperature [3], and the formula cars of University of Texas Arlington are no exception to it. Like other categories to brake fluid, they are also susceptible to moisture content and racing conditions. Ideally, the boiling of a pure brake fluid lies between 200-300 degree Celsius [3] . But due to hygroscopic effects, the brake fluid boiling temperature has gone down to 140 degree Celsius and 155 degree Celsius for DOT3 and DOT4 fluid over the years [3]. Keeping this factor and the extreme braking conditions during race, requisite adjustments are needed to be made for the formula cars.

By virtue of frictional contact during braking, heat is generated at the contact surfaces and dissipated to the components of the brakes. Considering the thermal conductivity and surface area of the disk rotor and brake pads, it can be safely assumed that most of the heat generated is dissipated to the disk rotor and a small part goes to the pads. Repetitive braking application of the formula cars and complex aerodynamics causes a sudden rise in temperature of the brake systems. Moreover, for better stability and locking, more torque is applied on the front axle wheels and both the front tires of the formula cars have disk brakes on it. The rear tires has an onboard braking for weight considerations.

Other than the problem of brake fluid vaporization with high brake temperature, there exists many risks such as fading of brake pads and bending of components such as rotor at elevated temperature resulting in uneven brake application. The wearing of frictional lining on the brake pads is directly proportional to the applied pressure and wearing of the lining increases exponentially with elevated temperature [4].

This research is focused is on studying the overall heat distribution of the braking system of a formula racing car. Study of interest will be in observing the overall heat distribution between the disk rotor and the brake pad and consecutively the brake fluid and compare my results with the actual experimental results. To achieve the same, MATLAB will be used to generate a simulation using the governing heat equations and verify the heat generation and temperature of the various components. Validity of the MATLAB model and its stability under noise and other aerodynamics factor will be verified later during the investigation.

Chapter 2

Description of model components

2.1 Physical brake system and its components

The brakes of UTA formula race cars, like every other brakes, serves two basic functions of a brake system i.e. to slow down a vehicle and to stop a vehicle to complete rest. As described earlier, disk brakes are used for their ease in application, reliability, better time response and their ability to withstand a higher temperature tolerance than their counterpart drum brakes [4]. There are two types of disk brakes available in market. Those are floating type disk brake and fixed type disk brake. The description for both of them are given below:

(i) Floating type: In the floating type disk brake type, there is a hydraulic line coming from the master cylinder and entering only one side of the caliper. When brakes are applied fluid travels through the line reaching one side of the caliper and applied pressure against the brake pad, which in return displaces the caliper against its own pressure and thus “floats” on the caliper body and presses the brake pads on the other side too, thus applying brakes.

(ii) Fixed type: In fixed type disk brake type, hydraulic line is present in both sides of the caliper. On application if brakes, the fluid exerts equal pressure on both the sides of the caliper equally, which in turn makes the piston push the brake pads at the same time, thus eliminating any delay, which is a major factor in formula racing cars. Moreover, the smoother experience of braking a fixed type caliper gives them a heads up ahead of their floating type counterpart.

2.2 Thermal dynamics of a disk brake and its components

As mentioned earlier, frictional heating is the major source of heat generation in any brake system. The kinetic energy of the vehicle is converted into frictional heat when a moving vehicle is brought to rest by disk- pad contact. The heat generated in the brakes is equal to the force times the velocity of the vehicle. The force is generated by the inertial force of the deceleration, mass times acceleration. We will denote the deceleration by the term *g's* [5]. The total force applied during deceleration can be expressed as [5]

$$F = \frac{M}{3}\,\dot{v} = \frac{W}{3} g's \qquad \text{Eq. 2.1}$$

Ordinarily in passenger vehicles we use four disk brakes for each tire. But the formula cars of University of Texas at Arlington has three disk rotor, two on the front axle and one on board for the rear axle for weight consideration. In order to accommodate that modification, the weight of the vehicle is divided by 3 so that heat generation for each of the brake can be considered. The brakes are well biased and they take almost equal braking effort [5].

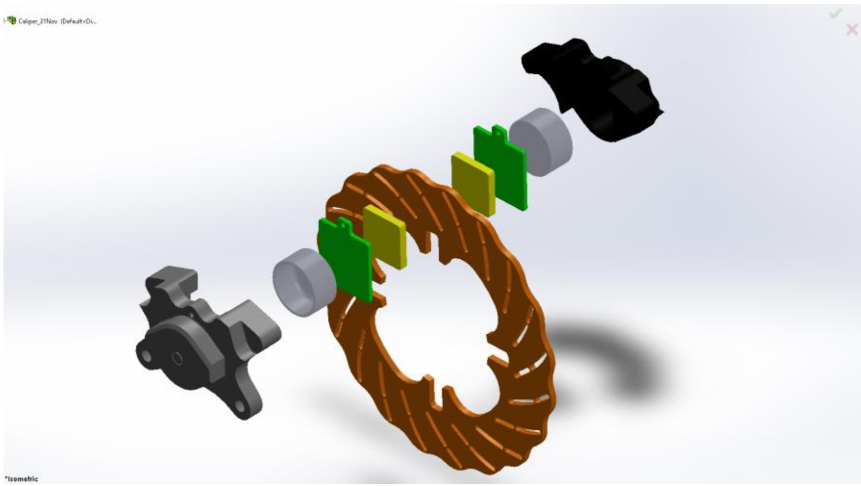

Figure 2-1 Solidworks model of the disk brake arrangement used in the system

The heat generation for each tire can be calculated by the equation

$$Q_{brake} = Fv \qquad \text{Eq. 2.2}$$

Considering that the driver of formula racing team plies a constant deceleration it can be safely presumed that the heat generation will be constant too for a given tire and the velocity decreases linearly over a period of time. So a set of differential equation can be formed and easily calculate the dynamic heat load by simulation in MATLAB. The following are the two equations for heat generation required to be solved [5],

$$\dot{v} = g * g's \qquad \text{Eq. 2.3}$$

And

$$Q_{brake} = \frac{W}{3}\, v\, g's \qquad \text{Eq. 2.4}$$

2.2 Disk pad interaction and heat distribution

Once the heat is generated by virtue of frictional contact between the brake pad and the brake rotor there is a rise in temperature of both the contacting surfaces. This rise in temperature is a result of some part of the heat travelling to the brake pads and rest to the disk. Let's denote the heat partition factor by an expression σ. As per Talati and Jalalifar et al. the heat partition factor for an imperfect contact is given by the equation [6]:

$$\sigma = \frac{\xi_d\, S_d}{\xi_d\ S_d + \xi p\ Sp} \qquad \text{Eq. 2.5}$$

Where ξ_d and ξ_p are thermal effusivities of the disk and pad respectively and S_d and S_p are the frictional contact surface of the disk and pad respectively. Thermal effusivity is given by the equation:

$$\xi = \sqrt{k\rho c} \qquad \text{Eq. 2.6}$$

Where k is the thermal conductivity of the material, ρ is the density of the material and c is the specific heat of the material.

Using the above methodology, Talati and Jalalifar et al. found out the heat partition factor to be around 93.4 % towards the brake disk rotor and the rest of the heat going towards the brake pads. Neys et al. inserted the material properties of the components in the above equation and found the value of heat partitioning actor to be around 98.8%. The validity of this study on a formula race car will be verified later during the study.

2.3 Rotor disk thermal analysis

The rotor disk or simply the rotor of a disk brake is the component is the rotating part of a brake system which is required to be stopped or slowed during a deceleration. The brake pads come in direct contact with the rotor to do the job. Traditionally, cast iron has been the major component for the construction of a rotor disk over the years because of its cheap cost and high thermal stability. But they increase fuel consumption due to high specific gravity [15] and weight of car too. The University of Texas at Arlington racing team procures its rotor disk from Wilwood which is usually a single disk slotted disk. Although double disk slotted rotor has a better cooling property than a single disk one but it increases the weight of the car. The primary construction material of the rotor as obtained from their official website is steel.

Figure 2-2 Solidworks model of the disk used in simulation

The heat input into the rotor disk is the frictional heat from the brake pads. While considering the thermal dynamics of the rotor, heat output from the rotor is in the form of conduction through the mounting tabs of the hub, free convection through air contact (assumed for the sake of simplicity, its validity will be verified later during calculation) and radiation losses.

Considering M as the mass of the rotor disk, C_p as the specific heat of steel and $T_r$as the rotor temperature, balancing the heat equation gives

$$M C_p \dot{T_r} = Q_{brake} - \ Q_{conduction} - Q_{convection} - Q_{radiation} \qquad \text{Eq. 2.7}$$

The pattern of airflow around the tires will be detrimental is finding the convection losses around the disk. Since it is a very complex pattern and changes with velocity of the race car throughout the track we will assume a constant flow and a free flowing stream of air around the disk to make the calculation simple later during the investigation stage. Considering the convection coefficient to be h and A as the surface area of the contact surface of the disk, the convection losses are

$$Q_{convection} = h\, A\, (\, T_r - \ T_\infty) \qquad \text{Eq. 2.8}$$

The conduction loss from the rotor disk is pretty straightforward as it is touching only the tabs of the hub for conduction and is a function of all the four cross sectional area of the mounting tabs. The thermal conductivity of the rotor is k the thickness of the rotor disk is t, w is the width of the mounting hub and L is the length for the same

$$Q_{conduction} = \frac{4\, k\, w\, t}{L}\ (T_r - T_\infty) \qquad \text{Eq. 2.9}$$

There will be considerable radiation losses at the expected level of temperature. The radiation is a function of view factor f_v and emissivity factorf_e, and the Stephen Boltzmann constant σ. The radiation equation for the disk will be

$$Q_{radiation} = f_v\, f_e \sigma\, \mathrm{A}\, (\, T_r^4 - \ T_\infty^4) \qquad \text{Eq. 2.10}$$

Combining all these equation in the heat equation the differential equation for finding the temperature of the rotor becomes

$$MC_p\dot{T}_r + h\,A\,T_r + \frac{4\,k\,w\,t}{L}\,T_r + f_v\,f_e\sigma\,\mathrm{A}\,T_r^4 = Q_{brake} + \frac{4\,k\,w\,t}{L}\,T_\infty + f_v\,f_e\sigma\,\mathrm{A}\,T_\infty^4 + h\,A\,T_\infty$$

Eq. 2.11

This equation is nonlinear and we will use MATLAB to solve this equation later during our study

2.4 Brake pad thermal analysis

The heating of the brake pads will occur in a similar manner like the rotor disk. The brake pads are made of low thermal conductivity, insulating material so as to absorb most of the heat produced during braking and only a small portion of the heat is conducted through it to the piston. Traditionally brake pads are made of organic material like asbestos which are cheaper and gives decent performance for daily use vehicles. But due to their quick wearing nature their use is on a downward scale [7]. Use of semi metallic is becoming more popular now a days because of their lesser wear and being inexpensive. The reason for it being so is its readily availability and better performance than organic materials. Copper brass and steel are some of the most common metals used for embedding in resin. They being less prone to wear make more noise than organic brake pads and also reduces the life of the rotor in some cases [7] .The next class of material used for the construction of brake pads is ceramic. They are by far the best performing brake pads and have very less wear. They are used in high performing vehicles because of their durability and lightweight [7].

The formula cars at the University of Texas at Arlington procures brake pads from Wilwood Incorporation. Though the exact composition of the brake pads is a trade secret but we can assume them to be semi metallic riveted to a steel backing plate.

On application of brakes, major portion of the heat is assumed to travel towards the disk rotor. But whatever amount of heat that reaches the pad is sufficient enough to raise the temperature to an extent sufficient enough to vaporize the brake fluid. So the conduction losses will be through its contact with the backing plate which is made of steel. As for the convection losses, it will experience the same complex air flow phenomenon as experienced by the rotor. For the sake of simplicity we are again going to assume a constant convection coefficient for the losses too. The view factor for the radiation losses will be 1 as its whole surface area will be right in front of the disk rotor. The heat generation will be by virtue of its thermal capacitance like the disk rotor.

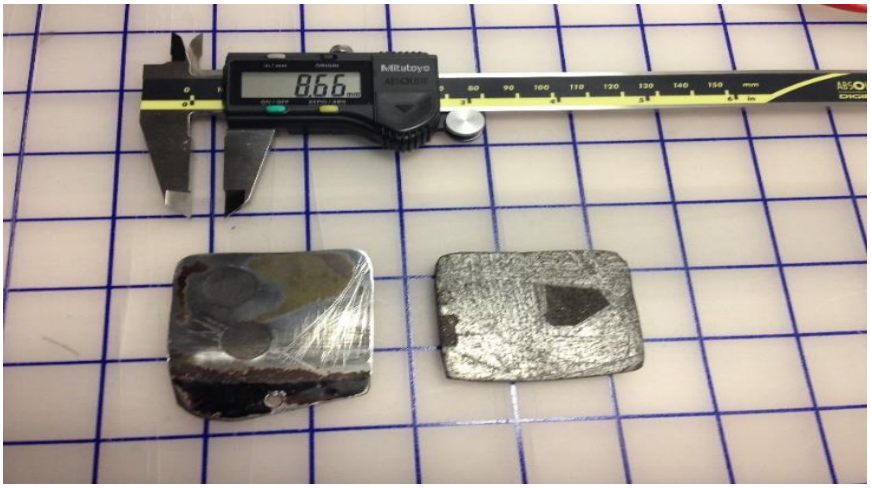

Figure 2-3 Pad material and backing taken apart separately

Let's assume M_{pad} as the mass, C_{pad} as the specific heat and T_{pad} as the temperature of the pad material. The corresponding heat generation equation for the same will be

$$M_{pad}C_{pad}\dot{T}_{pad} = Q_{pad} - Q_{padconvection} - Q_{padcoduction} - Q_{padradiation} \qquad \text{Eq.2.12}$$

Convection losses will be

$$Q_{padconvection} = h\, A_{pad}\,(T_{pad} - T_{\infty}) \qquad \text{Eq. 2.13}$$

Where A_{pad} is the surface area of the brake pad facing the disk rotor side.

Conduction losses will be a function of the cross sectional area of the backing plate and the thermal conductivity k_{pad} for the brake pad material. A_{pad} is the surface area of the backing plate and t_{pad} is the thickness of the backing plate. The equation can be written as

$$Q_{padcoduction} = \frac{k_{pad}A_{pad}}{t_{pad}} \left(T_{pad} - T_{\infty}\right) \qquad \text{Eq. 2.14}$$

Radiation loss will be function of view factor and emissivity. There was no definitive value for the emissivity factor of the pad and hence ended up with assuming one whose validity will be tested later during the investigation

$$Q_{padradiation} = f_{vpad}\, f_{epad}\, \sigma_{pad}\, A_{pad}\, \left(T_{pad}^4 - T_{\infty}^4\right) \qquad \text{Eq.2.15}$$

Where, f_{vpad} is the view factor for the pad and f_{epad} is the emissivity of the pad

The overall differential equation for the brake pad material will be

$$M_{pad}C_{pad}\dot{T}_{pad} + h\, A_{pad}\, T_{pad} + \frac{k_{pad}A_{pad}}{t_{pad}}\, T_{pad} + f_{vpad}\, f_{epad}\, \sigma_{pad}\, A_{pad}\, T_{pad} = Q_{pad} +$$

$$\mathrm{h}\, A_{pad}\, T_{\infty} + \frac{k_{pad}A_{pad}}{t_{pad}}\, T_{\infty} + f_{vpad}\, f_{epad}\, \sigma_{pad}\, A_{pad}\, T_{\infty} \qquad \text{Eq. 2.16}$$

2.5 Thermal analysis of the backing plate

It is interesting to note that though the backing plate is physically is riveted or glued to the brake pad material but still we are required to carry out an analysis for the backing plate. The reason for this exception is the different construction material for the backing plate. Almost universally, backing plate are made of steel. So we need to investigate the thermal behavior of backing plate. The backside of the backing plate is attached to the pistons of the caliper and can be considered for the conduction losses. Since a very small area of the backing plate will be exposed to the surrounding circulating air as the vehicle moves, the same constant value of h can be considered for the exposed portion of the steel of backing plate. The radiation losses will be negligible because of the complex geometry associated with the backing plate and very small surface area.

The thermal dynamics can be considered as follows. The mass of the backing plate is assumed as M_{plate} and the specific heat of steel is C_{pplate}. Q_{plate} is the heat absorbed by the steel due to the braking of the vehicle

$$M_{plate}\ C_{pplate}\ \dot{T}_{plate} = Q_{plate} - Q_{platecoduction} - Q_{plateconvection} \qquad \text{Eq. 2.17}$$

The convection is determined by the same presumed convection coefficient *h* and the exposed surface area.

$$Q_{plateconvection} = h\, A_{plateexposed}\, (T_{plate} - T_{\infty}) \qquad \text{Eq.2.18}$$

Conduction losses will be function of the surface area of the piston touching the back side of the backing plate during the braking application through its thickness. Considering k_{plate} as the thermal conductivity of steel A_{plate} as the surface area of the piston and t_{plate} as its thickness the conduction losses takes the form of

$$Q_{platecoduction} = \frac{k_{plate}\, A_{plate}}{t_{plate}}\, (T_{plate} - T_{\infty}) \qquad \text{Eq. 2.19}$$

The overall differential equation that is needed to be solved to find the temperature of backing plate will be

$$M_{plate}\ C_{pplate}\ \dot{T}_{plate} + hA_{plateexposed}\, T_{plate} + \frac{k_{plate}\, A_{plate}}{t_{plate}}\ T_{plate} = Q_{plate} + h\, A_{plateexposed}\, T_{\infty} + \frac{k_{plate}\, A_{plate}}{t_{plate}}\ T_{\infty} \qquad \text{Eq. 2.20}$$

2.6 Thermal analysis of Piston

The piston is an integral part of the caliper. The racing team of The University of Texas at Arlington procures their calipers from Wilwood Incorporation. The piston are required to be light weighed and are generally made of Aluminum. They have dust boot in order to isolate the brake fluid from impurities that pushes the piston outwards. The function of the oil seal, in addition to prevention of leakage of oil is to pull the piston back and thus release the brakes. The brake fluid under pressure forces the piston to slide out of the caliper. The action deforms the seal towards the direction of the movement of the piston. When the brakes are released, the seal will try to revert back to its original position and shape and pull back the piston with it due to its positioning inside the groove. This action will release the brake pads from the rotor disk eventually [8]. The efficiency of the brakes will depend on the surface area of contact between the pistons and the brake pads. This is achieved by either increasing the number of pistons or by using piston with bigger diameter. They help in increasing the clamping force exerted by the pistons on the brake pads [9]. The formula car under consideration for the University of Texas at Arlington used a caliper with 2 pistons of bigger diameter instead of increasing the piston count on it.

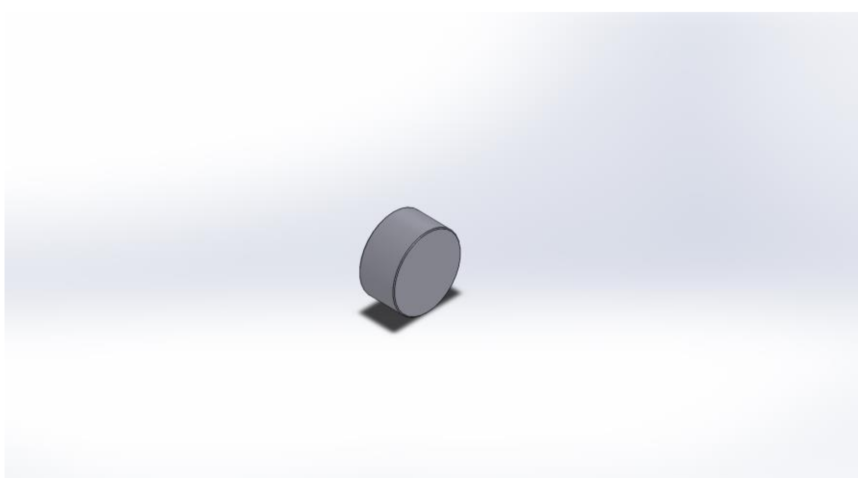

Figure 2-4 Solidworks model of the piston used in simulation

Speaking about the thermal analysis of the piston, it will be again be an interesting observation to make as a lot of aspects will be requiring smart assumptions in absence of standard experimental or textbook data. As earlier, the thermal capacitance will be the mass times the specific heat of the piston. Let be mass of the piston be denoted as M_{piston} and the specific heat of Aluminum is $C_{ppiston}$. The heat generated during braking process will be

$$M_{piston}C_{ppiston}\dot{T}_{piston} = Q_{piston} - Q_{pistonconduction} - Q_{pistonconvection}$$

Eq. 2.21

The conduction losses needs to be simplified because of the uneven geometry of the caliper surrounding it. The surface area of the contact surface is difficult to calculate manually with hand calculation. For the sake of simplicity, the solidworks model of the caliper is drawn to know the exact surface area. And as for the thickness of the caliper, the side where the maximum surface area of the caliper comes in contact with is considered. The validity of these assumptions will be checked during the calculation section. Let k_{piston} be the thermal conductivity of the Aluminum, $t_{pistonlength}$ be the thickness of caliper in contact and $A_{pistontop}$ is the inner surface of the caliper exposed to the piston. Keeping the above criteria under considerations, the conduction losses are calculated as follows

$$Q_{pistonconduction} = \frac{k_{piston}\, A_{pistontop}}{t_{pistonlength}}\,(T_{piston} - T_{\infty})$$ Eq 2.22

The convection losses again possess a challenge with the convection coefficient as no standard value of $h_{fluid}$is present for this particular geometry and liquid solid interacting surface. Again, hopes for finding so are pinned down on suitable presumptions for a stable value of $h_{fluid}$and verifying with the experimental value of its closeness to the experimental temperature of the caliper. The convection losses thus can be stated as

$$Q_{pistonconvection}=h_{fluid}\, A_{pistonexposed}\,(T_{piston} - T_{\infty})$$ Eq. 2.23

There will be no radiation losses as the piston is not exposed to air where it can radiate heat. The overall differential equation for the piston will be

$$M_{piston} C_{ppiston} \dot{T}_{piston} + h_{fluid}\, A_{pistonexposed}\, T_{piston} + \frac{k_{piston}\, A_{caliperexposed}}{t_{pistonlength}}\, T_{piston} = Q_{piston} + h_{fluid}\, A_{pistonexposed}\, T_{\infty} + \frac{k_{piston}\, A_{caliperexposed}}{t_{pistonlength}}\, T_{\infty} \qquad \text{Eq. 2.24}$$

2.7 Thermal analysis of brake fluid

All the above calculations and solving of differential equations was needed to reach to a point where we will be in position to calculate the brake fluid temperature. The brake fluids are classified on the basis of their boiling point [3]. The composition of brake fluids has changed over the years. Previously castor oil based and alcohols (butanol or ethanol) were used for making brake fluids prior to DOT 2 category [10]. Come DOT 3, DOT 4 and DOT 5.1, glycol based brake fluids came into existence. The majority of these fluids were compounds of alkyl ester, aliphatic amine, dithylene glycol, diethylene glycol monoethyl ether, diethyl glycol monomethyl ether, dimethyl dipropylene glycol, polyethylene glycol monobutyl ether, polyethylene glycol monomethyl ether, polyethylene oxide, triethylene glycol monobutyl ether, triethylene glycol monoethyl ether and triethylene glycol monomethyl ether [10]. The most advanced variation of brake fluids are the one that came in market after DOT 5.1 and were silicone based. Various forms of these liquids containing di-2-ethylhexyl sebacate, dimethyl polysiloxane, tributyl phosphate are manufactured [10]. Though DOT 5 is the latest variant of the brake fluid available in the market but still DOT 4 liquid are used on a wide scale.

The classification of brake fluids in the form on the basis of temperature is as following [10]

Table 2-1 various grades of DOT approved brake fluids

	Dry boiling point	Wet boiling point
DOT 2	190°C	140°C
DOT 3	205°C	140°C
DOT 4	230°C	155°C
DOT 5	260°C	180°C
DOT 5.1	260°C	180°C

Wet boiling point is defined for liquid having more than 3.7 % of water by volume. Silicone based brake fluids are far less susceptible to hygroscopic effects than ethyl based liquids [12]. The formula cars at University of Texas at Arlington uses DOT 5 liquids in them.

The mass of the brake fluids acts as the heat reservoir during the braking application where they gain heat from the pistons in whose contact they come. The convection losses from the fluid is when it comes in contact with the piston and the caliper body. Technically speaking, the aluminum grades of the piston and caliper body are different but for the sake of simplicity of our calculation we will consider them as of the same grades so that we can assume the same convection coefficient value of h_{fluid} for it. There will not be any radiation losses again as no part of the fluid is exposed to the environment. Let's denote the mass of brake fluid present inside the caliper body asM_{fluid}, C_{pfluid}as the specific heat of brake fluid and T_{fluid} as the temperature of the fluid to be found out. Considering the aforesaid factors, the heat generation equation will be

$$M_{fluid}\, C_{pfluid}\, \dot{T}_{fluid} = Q_{fluid} - Q_{fluidconvection} - Q_{finlosses} \qquad \text{Eq. 2.25}$$

Another interesting feature about this equation that we can observe are the fin losses. Brake fluids are present not only in the extended hollow cylindrical volume behind the piston but also in the transfer lines from one caliper body to the other side. They also tend to get heated eventually during repeated application of the brakes as we are intending to perform during our simulation and experimentation later. So fin losses should be added in the calculation to compensate for losses.

The convection losses from the fluid surface will be a function convective coefficient and surface area

$$Q_{fluidconvection} = h_{fluid}\, A_{fluidexposed}\, (T_{fluid} - T_\infty) \qquad \text{Eq. 2.26}$$

The fin losses for the system will be considering P_{fin} as the perimeter of fin, A_{fin} as the area of the fin opening,

$$Q_{finlosses} = \sqrt{h_{fluid}\, P_{fin} A_{fin}}\, (T_{fluid} - T_\infty) \qquad \text{Eq. 2.27}$$

The final differential equation for the brake fluid will sum up as follows:

$$M_{fluid}\, C_{pfluid}\, \dot{T}_{fluid} + h_{fluid}\, A_{fluidexposed}\, T_{fluid} + \sqrt{h_{fluid}\, P_{fin} A_{fin}}\, T_{fluid} = Q_{fluid} + \sqrt{h_{fluid}\, P_{fin} A_{fin}}\, T_\infty + h_{fluid}\, A_{fluidexposed}\, T_\infty \qquad \text{Eq. 2.28}$$

2.8 Thermal analysis of caliper

The brake calipers is the housing where all the other brake components are packed. As it has been discussed before, two broad categories of brakes has been classified in the basis of movement of calipers. The floating type caliper works when brake fluid one side of the caliper creates enough pressure to move the caliper on the other side for it operation while the fixed type caliper supplies uniform amount of brake fluid to either side of the caliper for even and smoother application of brakes. The caliper body is usually made of lightweight aluminum. The race team at the University of Texas at Arlington acquires it calipers from Wilwood Incorporation which are made of aluminum.

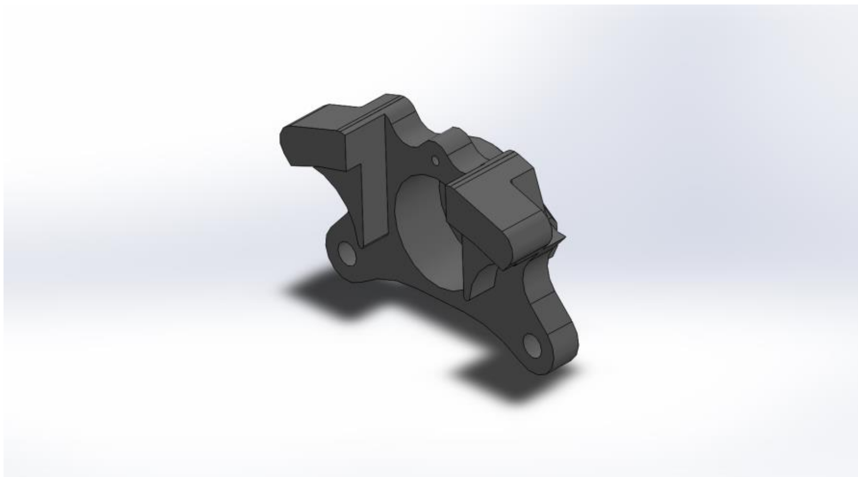

Figure 2-5 Solidworks model of the caliper used in simulation

Considering the elements of governing equations for caliper, the equation will sum up as

$$M_{caliper}\, C_{pcaliper}\, \dot{T}_{caliper} = Q_{caliper} - Q_{caliperconduction} - Q_{caliperconvection} \quad \text{Eq. 2.29}$$

where, $M_{caliper}$ is the mass, $C_{pcaliper}$ is the specific heat, $Q_{caliper}$ is the heat generated by the caliper due to braking, $Q_{caliperconduction}$ are the conduction losses, $Q_{caliperconvection}$ are the convection losses for the caliper accordingly.

The conduction losses will be through the piston to which it comes in contact with. Considering the surface of the caliper which comes in contact with the piston to be $A_{calipercontact}$, $k_{caliper}$ as the thermal conductivity of the aluminum with which the caliper is made of and $t_{caliperback}$, the conduction loss equation is

$$Q_{caliperconduction} = \frac{k_{caliper}\ A_{calipercontact}}{t_{caliperback}}\ (T_{caliper} - T_{\infty}) \qquad \text{Eq. 2.30}$$

The convection losses will be due to the surface area contact of the caliper with air. The *h* is taken as the same for the whole system and $A_{caliper}$ is the area of the caliper exposed to the surrounding air.

$$Q_{caliperconvection} = h\ A_{caliper}(T_{caliper} - T_{\infty}) \qquad \text{Eq. 2.31}$$

Together the governing equation looks like the following

$$M_{caliper}\ C_{pcaliper}\ \dot{T}_{caliper} + h\ A_{caliper}T_{caliper} + \frac{k_{caliper}\ A_{calipercontact}}{t_{caliperback}}\ T_{caliper} =$$

$$Q_{caliper} + h\ A_{caliper}T_{\infty} + \frac{k_{caliper}\ A_{calipercontact}}{t_{caliperback}}\ T_{\infty} \qquad \text{Eq. 2.32}$$

Chapter 3

Theoretical modelling of the problem and solution

3.1 Numerical solution for disk temperature

The following data were being fed into the final governing equation of disk Eq. 2.3.

Ambient temperature=25°C

Mass of vehicle (along with driver) = 290 Kg

Surface area of disk, A, = 0.05 square meter

g's (deceleration) = 1

g's (acceleration) = 0.5

Maximum velocity = 26.83 meters per second (60 miles per hour)

Minimum velocity = 13.41 meters per second (30 miles per hour)

Thermal conductivity of steel [5] k, = 51.88 W/(mK)

Specific heat of steel [5],C_p, = 504 Joules/kg

Thickness of disk,t, = 0.0053 meter

Width of the mounting hub slot,w, = 0.00635 meter

Length of the mounting slot,L, = 0.042 meter

Mass of disk,M, = 0.9906 Kilograms

Convection coefficient [5], h, = 150 W/ ($m^{2\circ}$K) (assumed)

View factor [5], f_v = 1.0

Emissivity [5], f_e = 0.9 (assumed)

Stefan-Boltzmann constant = 5.67 x 10^{-8} N/m^2/C^{-4}

The MATLAB simulation has been done in two parts. The first one is considered that the vehicle was running continuously and there was no time lapse during the racing duration of Formula car. This type of approximation is only ideal and not practical because there will be turns in the tracks and the vehicle needs to adjust accordingly. So, the next simulation is run considering the vehicle to slow down on a curve in a race track after deceleration. The vehicle then moves in a constant speed for around 3 seconds as it covers the curved portion of the track and again accelerates. Brakes are applied when the vehicle reaches to 60 miles per hour at approximately and is turned on until it reaches 30 miles per hour at 1 g's. Once the car attains a velocity of 30 miles per hour, the vehicle is again accelerated to 60 miles per hour at 0.5 g's and is slowed down on the curves where is travels in 30 miles per hour as it covers the whole curve. As the curve ends up, the vehicle is again accelerated to 60 miles per hour and is made to get ready for the next braking application. This process is being simulated numerically in MATLAB in order to get it verified by the experimental procedure.

Another part of the simulation is to check the heat partition factor. Talati and Jalifar et al [6] showed that the heat partition factor to the disk rotor is around 93.4%. This data is being used as a base value and is checked for accuracy during the experiment. Different values of heat partitioning factor at 93%, 95%, 97% and 100% is being directed towards the rotor and the temperature variation is observed with the actual data found out during experimentation.

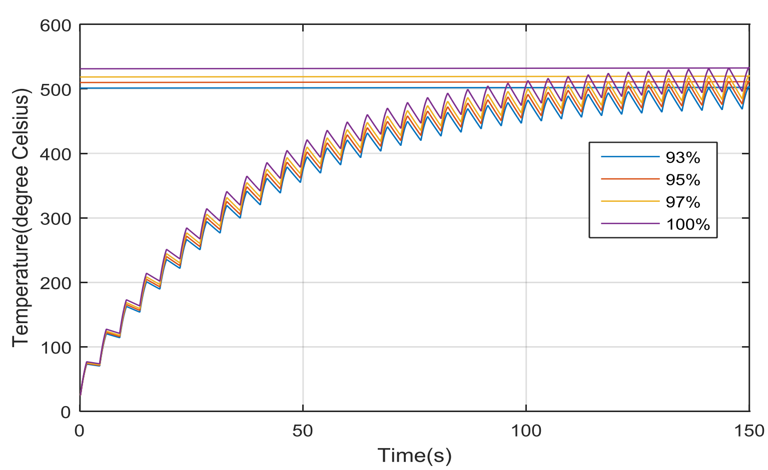

Figure 3-1 Temperature of rotor without time lapse

Table 3-1 Temperature of rotor at different heat distribution factor after 10 braking cycles

Heat distribution	93%	95%	97%	100%
Temperature (°C)	359	366	373	384

Table 3-2 Temperature of rotor after 20 braking cycle without any time lapse for various heat distribution factor

Heat distribution	93%	95%	97%	100%
Temperature (°C)	468	476	485	498

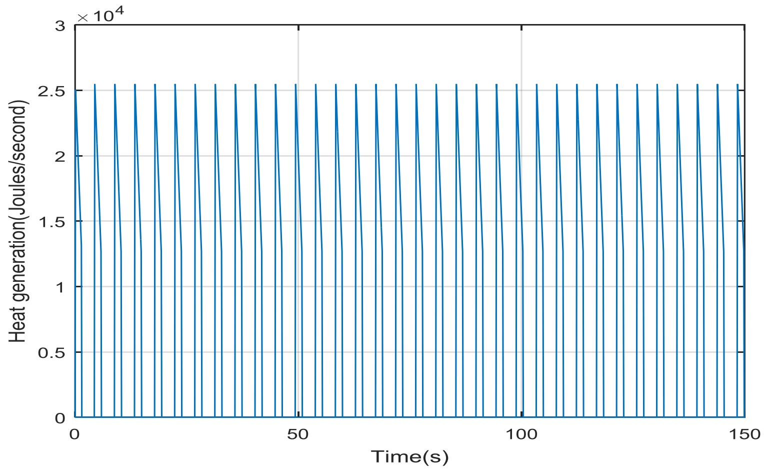

Figure 3-2 Heat generation in rotor without any time lapse

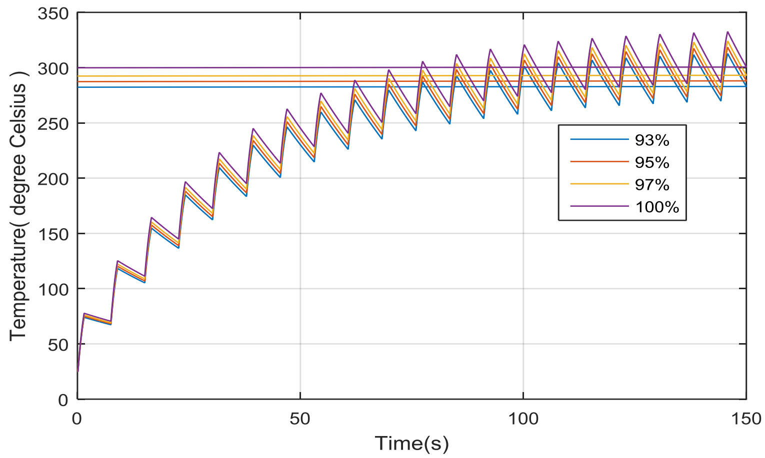

Figure 3-3 Temperature of rotor at a time interval of 3 seconds

Table 3-3 Temperature of rotor after 10 braking with time lapse of 3 seconds

Heat distribution	93%	95%	97%	100%
Temperature (°C)	278	283	288	296

Table 3-4 Temperature of rotor after 20 braking with time lapse of 3 seconds

Heat distribution	93%	95%	97%	100%
Temperature (°C)	331	337	344	353

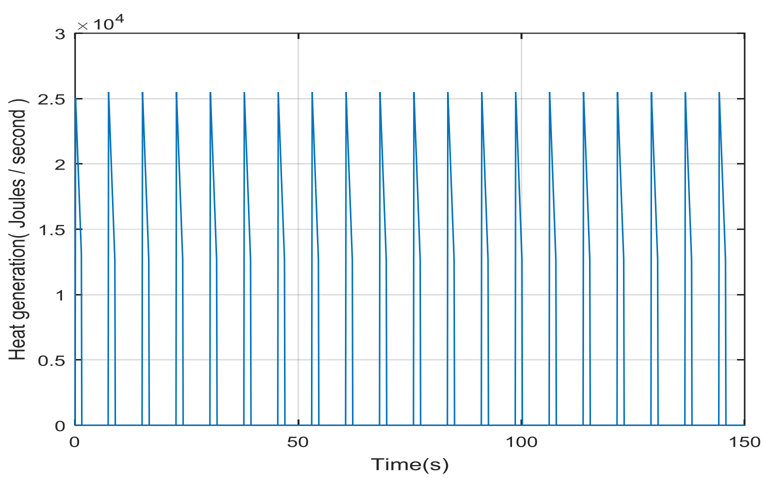

Figure 3-4 Heat generation in rotor during braking

The temperature of rotor is in very good accordance with the experimental value of rotor temperature. So, this mathematical model is a stable method of finding out the rotor temperature is a stable way for analysis temperature behavior of a rotor.

3.2 Numerical solution for brake pad material

Mass of pad material, M_{pad}= .042 kg

Width of pad material = 0.038 meter

Length of pad material = 0.038 meter

Area of pad material,A_{pad}, = .00145 square meter

Thickness of pad material,t_{pad}, = 0.00877 meter

Thermal conductivity of brake pad material [13],k_{pad}, = 5 W/m/˚K

Specific heat of brake pad material, [13] C_{ppad}, = 1000 J/kg/˚K

Convection coefficient, h, = 150 W/m^2/K (assumed)

View factor, f_v = 1.0 (assumed)

Emissivity, f_e = 0.2 (assumed)

Stefan-Boltzmann constant [12] = 5.67 x 10^{-8} N/m^2/˚K^{-4}

The assumed view factor for the pad material is lesser than that for the steel rotor because emissivity is a factor of surface polish along with temperature. Due to a poorer surface finish and lesser temperature rise than the steel rotor, it's being assumed as 0.2. Very limited information is available on the composition of brake pads since it being a trade secret. The validity of the above assumptions are required to be verified. The mass of the brake pad is being calculated by separating it from the backing plate and weighting it on a simple weight scale.

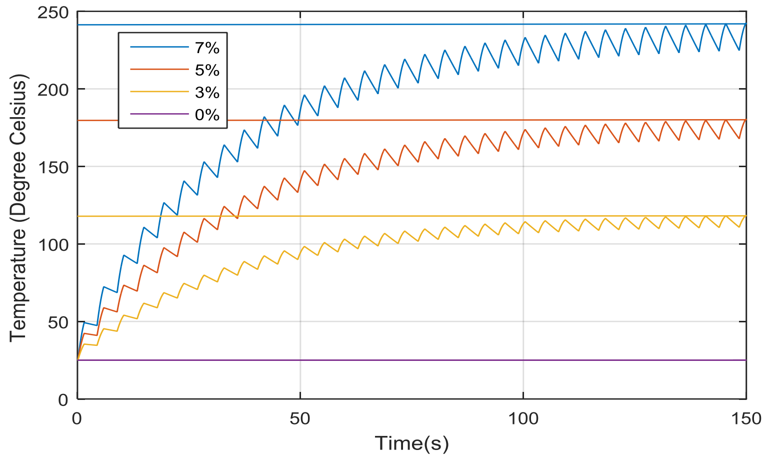

Figure 3-5 Temperature of brake pad material without time lapse

Table 3-5 Temperature of brake pad material with time lapse of 3 seconds after 20 braking

Heat distribution	7%	5%	3%	0%
Temperature (°C)	174	131	88	25

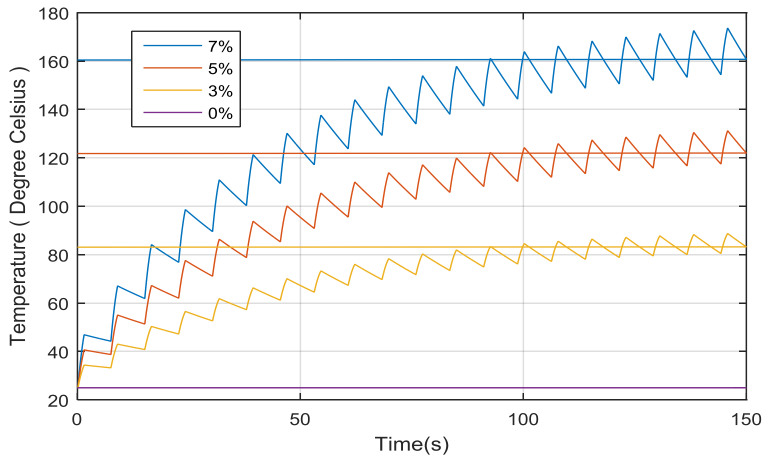

Figure 3-6 Temperature of brake pad material with time lapse

3.3 Numerical solution for backing plate

Mass of backing plate,M_{pad}, = .061kg

Width of backing plate = 0.05 meter

Length of backing plate = 0.05 meter

Area of backing plate,A_{plate}, = .00269 square meter

Area of backing plate exposed, $A_{plateexposed}$ = .00124 square meter

Thickness of backing plate,t_{plate}, = 0.00322 meter

Thermal conductivity of backing plate, [12] k_{plate} , = 54 W/m/˚K

Specific heat of backing plate, [12] C_{pplste} = 465 J/kg/˚K

Convection coefficient, h, = 150 W/(m^2˚K) (assumed)

Theoretically speaking, the backing plate is an integrated part of the brake pad. But since the construction material of both of them are different, they should be considered as different units for analysis. It's usually made of plain carbon steel and it either riveted or glued to the brake pad material. They work as the support material for transferring the brake pedal motion from the piston to the brake pads and the very thin because of packing issues. One more thing that should be mentioned at this point is that while doing the thermal analysis for the backing plate, the radiation losses are ignored because most of the surface area of the backing plate is being covered by the pad material and whatever remaining of the exposed steel should not be much for the heat losses by the radiation to be considered.

Since heat generation is due to direct contact of rotor and brake pads, so its temperature raise can be predicated up to a good accuracy level by these differential equations made of governing equation without considering a coupled capacitance heat storage by the brake pads.

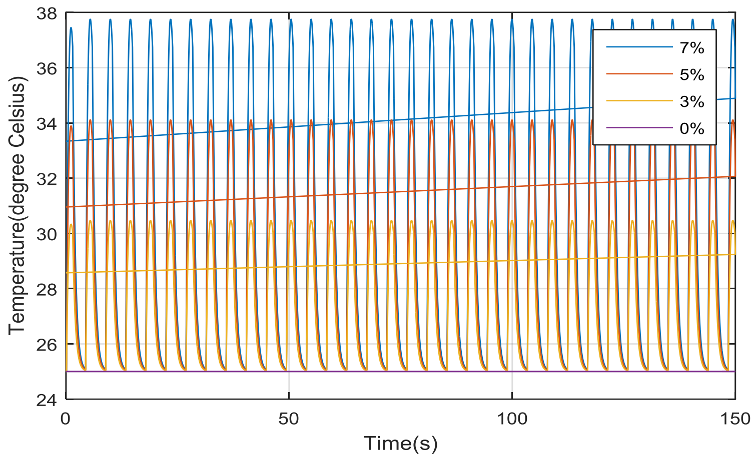

Figure 3-7 Temperature of backing plate without any time lapse

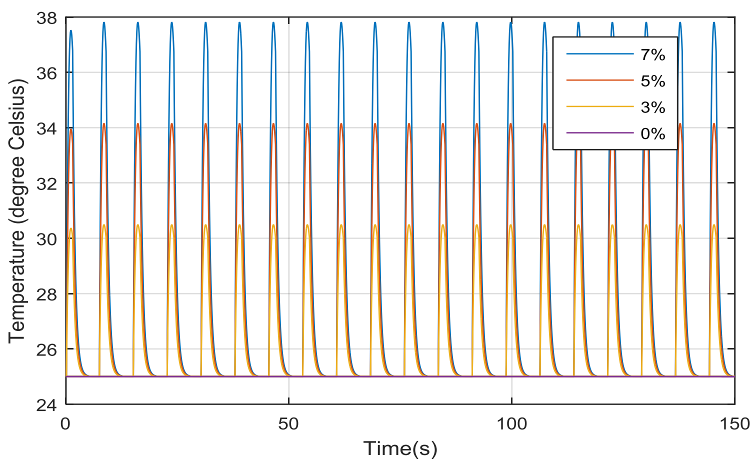

Figure 3-8 Temperature of backing plate after time lapse of 3 seconds

Table 3-6 Temperature of backing plate with after 20 braking cycle with time lapse of 3 seconds

Heat distribution	7%	5%	3%	0%
Temperature (°C)	37	33	30	25

As we can see from the above figure of time lapse, there is not enough temperature raise of the backing plate if we consider it as an individual body having its own heat generation source due to braking application, this can be inferred from the simulation that this modelling method is not suitable for calculating the backing plate temperature and variations from the other bodies should be taken into factor too in order to get an accurate answer.

3.4 Thermal analysis of piston

Mass of piston, M_{piston}= .0295 kg

Area of piston exposed, $A_{pistonexposed}$ = .00497 square meter

Area of piston top, $A_{psitontop}$ = .00148 square meter

Thickness of piston,t_{piston}, = 0.02286 meter

Thermal conductivity of piston, [14] k_{piston} , = 236 W/m/˚K

Specific heat of piston, [14] $C_{ppiston}$ = 896 J/kg/˚K

Convection coefficient,h_{fluid}, = 15 W / (m^{2}˚K) (assumed)

The pistons are usually inverted in order to accommodate maximum brake fluid for safe operation. All the unmentioned values for the above equation have been derived from Solidworks model of the piston. The heat variation due to the presence of dust boot and oil seal is very negligible and can be neglected.

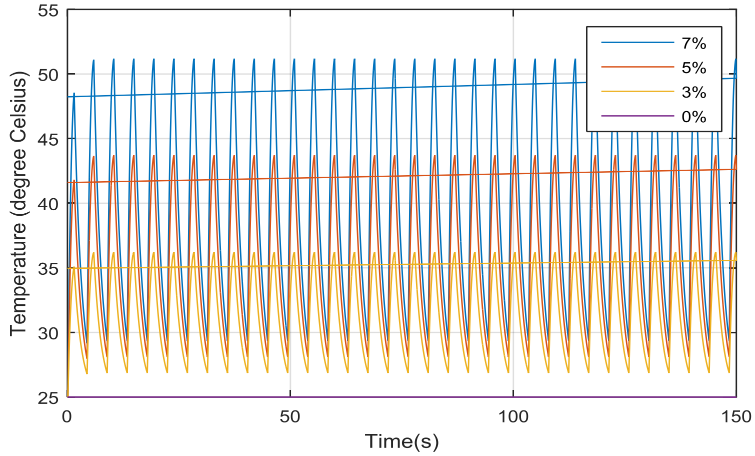

Figure 3-9 Temperature of piston without any time lapse

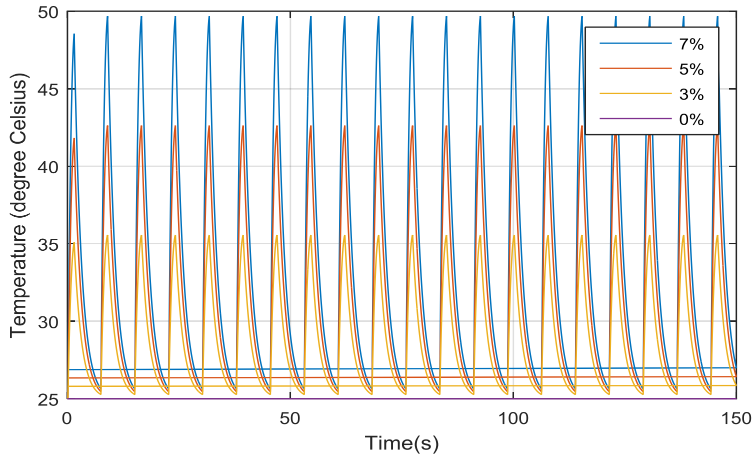

Figure 3-10 Temperature of piston with time lapse

Table 3-7 Temperature of piston after time lapse of 20 seconds at different heat distribution factor

Heat distribution factor	7%	5%	3%	0%
Temperature (°C)	49	42	35	25

Similar temperature profile for the piston as like that of the backing plate can be observed. The incapability of this differential equation to predict a suitable temperature is again due to its inability to incorporate the temperature raise due to other heating elements in the system. So this mode of predicting temperature for the piston too should be rejected.

3.5 Thermal analysis of brake fluid

Mass of brake fluid, M_{fluid} = 0.0030 kg

Area of brake fluid,A_{fluid}, = .00435 square meter

Area of brake fluid top,$A_{fluidtop}$, = .00148 square meter

Area of fluid exposed, $A_{fluidexposed}$ = .00349 square meter

Thickness of fluid,t_{fluid}, = 0.023 meter

Thermal conductivity of fluid, [14] k_{fluid} , = 236 W/m/˚K

Specific heat of brake fluid, [14] C_{pfluid} = 2360 J/kg/˚K

Convection coefficient,h_{fluid}, = 1.5 W / (m^{2}˚K) (assumed)

Perimeter of fin,P_f = .0314 meter

Area of fin, A_f = .0000785 meter

The Formula SAE team of the University of Texas of Arlington uses DOT 5 approved brake fluid which is less resistant to corrosion and have hydrophobic nature. Their major component is diorgano polysiloxane which contains at least 70% by weight at 20°C [11]. Though the exact material properties are not known, so the properties of ethylene glycol are used which have similar, if not exact properties as of DOT 5 fluid and are used as a major component of DOT 4 brake fluid, for the sake of simplicity.

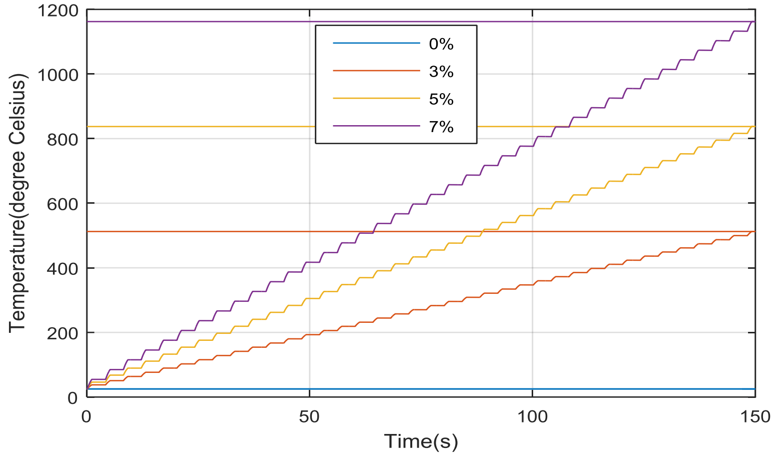

Figure 3-11 Temperature of brake fluid without any time lapse

Table 3-8 Temperature of brake fluid without any time lapse at different heat distribution factor

Heat distribution	7%	5%	3%	0%
Temperature (°C)	613	444	277	25

The brake fluid does not reach steady state temperature within 20 braking application.

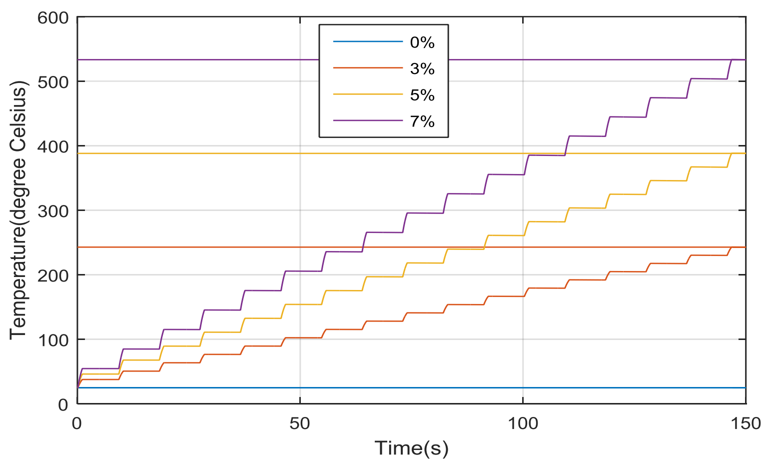

Figure 3-12 Temperature profile with time lapse after 3 seconds

Table 3-9 Temperature of brake fluid with time lapse of 3 seconds at different heat distribution factor

Heat distribution	7%	5%	3%	0%
Temperature (°C)	592	430	268	25

The brake fluid temperature as calculated by the governing equation is far higher than range for vaporization. It can easily melt the piston if the actual case was so. But in reality, due to the heat retention by the fluid and also its dependence to heat up the caliper does not actually raise the temperature to this extent. So a proper modelling should be done by capacitance modelling to account for these phenomenon.

3.6 Thermal analysis of caliper

Mass of caliper, $M_{caliper}$ = 0.308 kg

Area of caliper, $A_{caliper}$ = .01733 square meter

Area of piston exposed, $A_{calipercontact}$ = .000393 square meter

Thickness of caliper back, $t_{caliperback}$, = 0.00572 meter

Thermal conductivity of piston, [14] $k_{caliper}$, = 236 W/m/˚K

Specific heat of caliper, [14] $C_{pcaliper}$ = 896 J/kg/˚K

Convection coefficient, h, = 150 W/ (m^{2}˚K)(assumed)

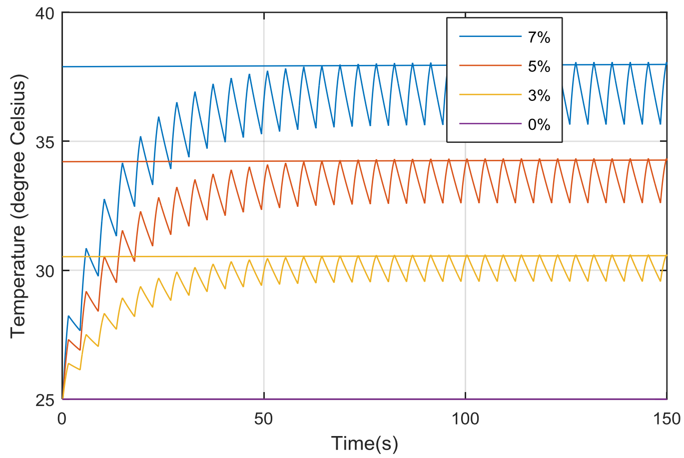

Figure 3-13 Temperature profile of caliper without any time lapse

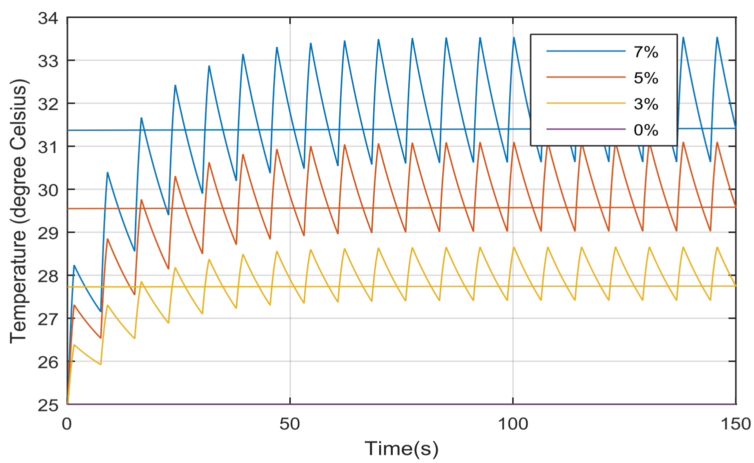

Figure 3-14 Temperature profile after braking with a time interval of 3 seconds

Table 3-10 Temperature of caliper after 20 braking with time lapse of 3 seconds

Heat distribution factor	7%	5%	3%	0%
Temperature (°C)	33	31	28	25

Again, the temperature for the caliper is far lower than the expected temperature and this method of individual modelling to find the temperature of caliper can be rejected too. A new model will be developed in the later sections and will be tested for stability.

Chapter 4

Experimental setup, results and discussions

4.1 The experimental setup

In order to validate the numerical calculation, an experiment should be done with the same condition or at least as close as possible to the actual race track simulation. The car was pushed to its limit braking in order to study the brake fluid property. It was achieved in the following way.

The plan was to accelerate the Formula car from rest (o miles per hour) to 26.82 meters per second (60 miles per hour) at an acceleration of 0.5 *g's*. The requisite *g's* was achieved by opening the throttle on the basis of experience of the driver. Once the vehicle reaches 26.82 meters per second it was subjected to hard braking at 1.0 g's in order to drag down the speed to 13.41 meters per second (30 miles per hour) as hard braking is pretty common in actual race conditions along steep curves. The deceleration g's here too, of course, is based on the intuition of the driver for opening the throttle position. The car is then made to run in a constant speed at 13.41 meters per second along a curve so that it can turn around on the track. The track was designed in such a way that it takes approximately 3 seconds for the car to cover the curve, in the numerical simulation too, this 3 second time lapse has been considered after every single phase of acceleration and deceleration. The points A and B in Figure 4-1 gives an approximate position of the braking points by the driver.

The car that was chosen for the experiment was a 2003 University of Texas at Arlington manufactured Formula SAE vehicle. The following are the racing conditions on that particular day of experiment,

Ambient temperature = 26 degree Celsius

Mass of the vehicle = 220 Kilograms

Mass of the driver = 72 kilograms

Tire pressure of front right = 0.827 bar (12 pounds per square inch)

Tire pressure of front left =0.827 bar (12 pounds per square inch)

Tire pressure of rear left = .758 bar (11pounds per square inch)

Tire pressure of rear right = 0.827 bar (12 pounds per square inch)

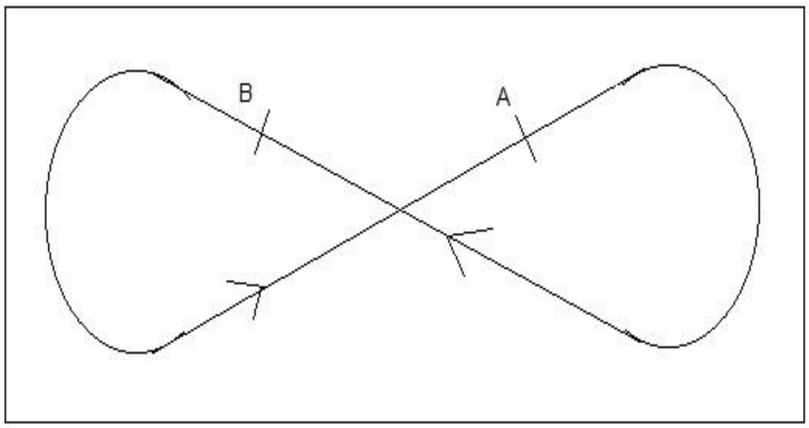

Figure 4-1 Schematic of the track used for experimentation

The next task was to record the temperature. It was decided to make the car run for 10 laps after which it will make a quick stop for the measurement of the rotor and caliper temperature. The idea of attaching a thermocouple to the system was rejected because of their slow response. Another alternative was to use a thermal camera for a more dynamic observation of temperature profile and mount it on the caliper housing. But unfortunately, the thermal cameras available for the experiment could only record up to a maximum of 300 degree Celsius. And the rotor usually reaches a red hot temperature during hard braking (which was basically the requisite of the experiment). Moreover, thermal cameras are very delicate to withstand the vibration of a race car on its full flow.

So it was decided to make the car stop after every 10 laps and make the pit stop for as less time as possible in order to minimize the cooling effect. Laser guns were used to record the temperature which eliminate all the above shortcoming like slow responsiveness and high temperature profile measurement range. It was made sure that the beam was touching the rotor body and the back side of the caliper, since it being the most accessible part for quick measurement.

4.2 Results and discussions

Table 4-1 Experimental value of rotor and caliper after 20 braking cycle with time lapse

	Rotor (° C)	Caliper(° C)
After 20 braking	343	130

The experimentation was stopped after braking for 20 laps as the car lost its brakes and a lot of brake fluid evaporated. But there was sufficient data to validate the theoretical results. Successive calculations were based on the results obtained during these 20 laps.

4.3 Discussions on temperature profile for rotor

The experimental value of rotor temperature was very encouraging when compared with the one derived from the governing equations by simulations. Based on the heat partitioning factor of the rotor and brake pads during the braking application. The graph for temperature profile on the basis of governing equation was as follows

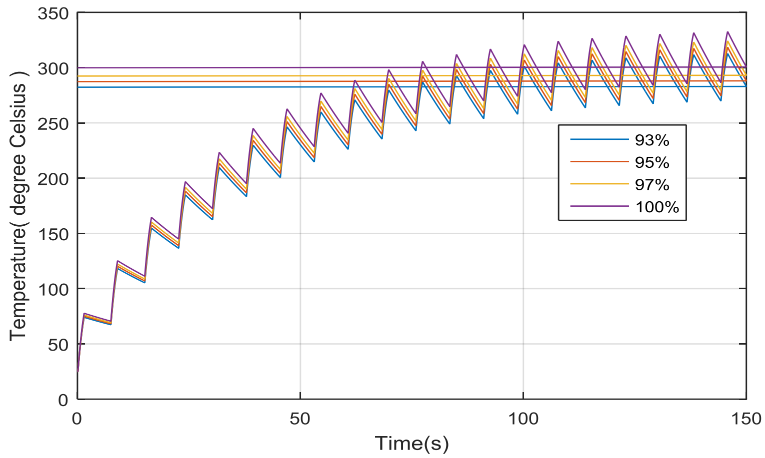

Figure 4-2 Temperature profile for the brake rotor with time lapse of 3 seconds

It was observed that the brake rotor profile responded well to the differential equation formed with actual brake rotor data. The only two unknown data needed for solving the differential equation was convection coefficient of the circulating air around the rotor and the rotor material. From the website of Wilwood Incorporation, the base material for the rotor was determined as steel. But there was no way to determine the convection coefficient for the complex pattern of circulating air around the rotor. The value of the convection coefficient was determined to be 150 Watt/ ($m^{2\circ}K$).

It was further observed that out of all the heat partitioning factor, the most stable temperature range was that for the 97% of the heat going to the rotor body and 3% to the brake pads. Neys [7] further confirmed in this value to be around 98.3% through his experiments. So this value was selected as the base value of heat partitioning factor and the consecutive equations were carried out accordingly.

With 97% of the heat generated travelling towards the disk, variation in temperature after 20 braking cycle of the simulated results from the experimental result is 0.365%.

4.4 Discussions on the temperature profile for caliper and brake fluid

The numerical value of the caliper temperature was way off when it was calculated simply on the basis of the governing equation. This types of problem can be transformed into an electrical circuit analogy for simplicity of solving conjugate equations [12]. Let us consider the heat flow, Q, through a flat plate whose thermal conductivity is k, surface area A, length through which the heat passes L and δt as the temperature difference between the two ends of the plate.

The conduction equation will be [12]

$$Q = \frac{k\,A}{L}\,\delta t \qquad \text{Eq. 4.1}$$

If we consider the heat flow as the flow variable and temperature difference as the effort variable, then we can write the above equation in the form of an impedance relation [12]

$$Q = \frac{\delta t}{R} \qquad \text{Eq. 4.2}$$

In this case, we take the thermal resistance R due to conduction to be [12]

$$R = \frac{L}{kA} \qquad \text{Eq. 4.3}$$

The thermal resistance due to convection is

$$R = \frac{1}{h\,A} \qquad \text{Eq. 4.4}$$

So it is required for the system we have for the brake fluid to be treated as a capacitance and convert the heat equation to its corresponding electrical circuit so that we can take account for the heat storage by the brake fluid. Drawing the circuit for the same

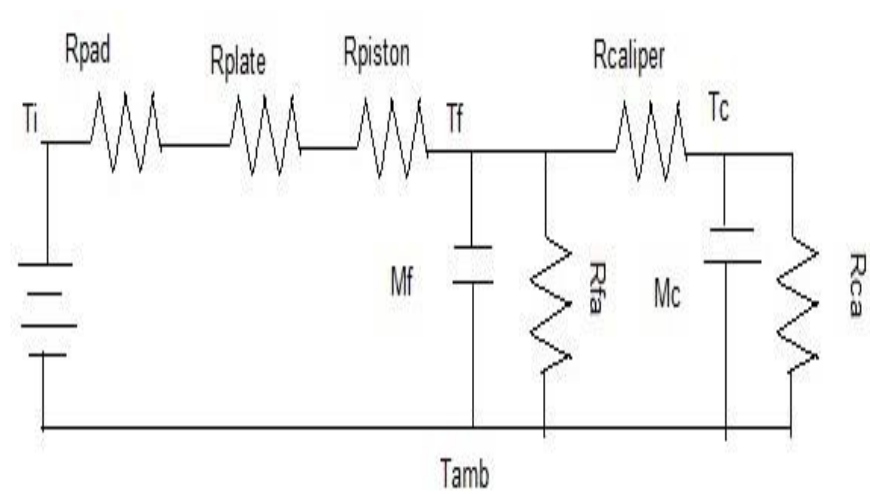

Figure 4-4 Thermal circuit for the brake temperature analysis modeling

Now considering the above circuit, there are two thermal capacitance mass in the form of fluid and caliper and we need to find the temperature of the fluid at the end of the experiment. The validity of the fluid temperature can only be justified if the temperature of the caliper calculated using the above procedure matches or at least is in the range (considering a lot of assumed values) of the experimental one. The above circuit has two unknown and we have to formulate two equations to calculate the two unknowns. Considering the input temperature and the heat input as the one generated during the braking and moving through the brake pads, the above circuit can be further briefed up by summing the resistances in series as one,

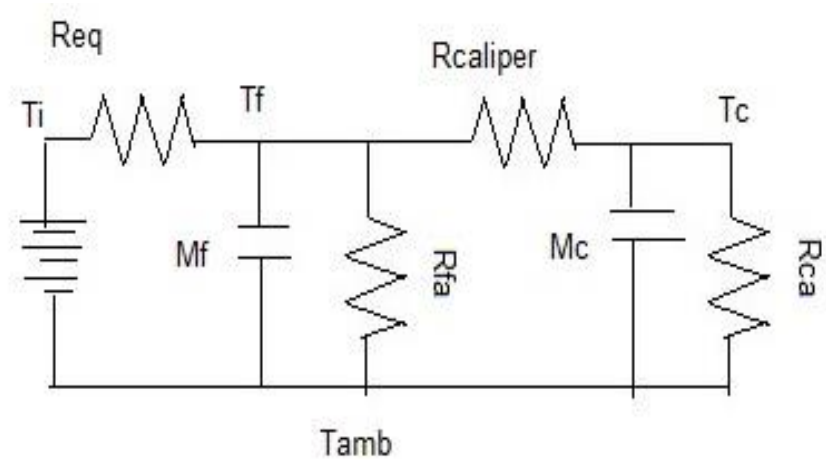

Figure 4-5 Schematic of the circuit with resistances clubbed in as series

By using the first law of thermodynamics, the following are the two equations needed to be solved

Heat input= Heat output + Heat stored

For the first loop,

$$Q_{input} + R_{eq} = R_{fluidlosses} + R_{caliperraise} + M_{fluid}\, C_{pfluid}\, \dot{T}_{fluid} \qquad \text{Eq. 4.5}$$

For second loop,

$$0 = R_{caliperraise} + R_{caliperlosses} + M_{caliper}\, C_{pcaliper}\, \dot{T}_{caliper} \qquad \text{Eq.4.6}$$

Expanding the above two equations

$$Q_{in} + \frac{T_{input} - T_{fluid}}{R_{eq}} = \frac{T_{fluid} - T_{amb}}{R_{fluidamb}} + \frac{T_{fluid} - T_{caliper}}{R_{fluidcaliper}} + M_{fluid}\, C_{pfluid}\, \dot{T}_{fluid} \qquad \text{Eq. 4.7}$$

$$0 = \frac{T_{caliper} - T_{fluid}}{R_{fluidcaliper}} + \frac{T_{caliper} - T_{amb}}{R_{caliperamb}} + M_{caliper}\, C_{pcaliper}\, \dot{T}_{caliper} \qquad \text{Eq. 4.8}$$

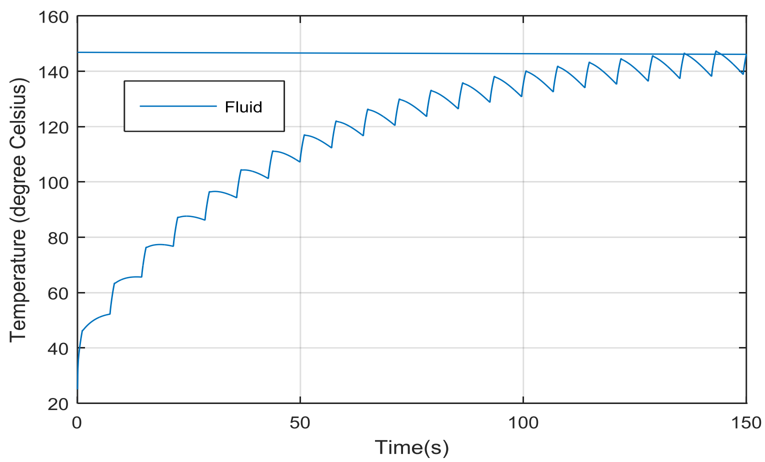

Figure 4-6 Temperature of the brake fluid with time lapse of 3 seconds

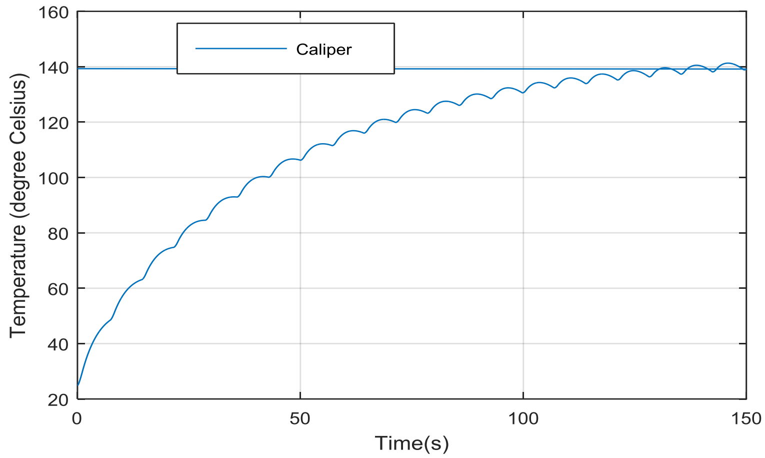

Figure 4-7 Temperature profile for caliper with time lapse of 3 seconds

Table 4-2 Temperature of brake components with lumped capacitance modelling after 20 cycle

	Brake fluid	Caliper
Temperature after 20 braking cycle (°C)	146	138
Variation from experimental value		6 %

It can be seen that the overall temperature readings for both the caliper and brake fluid are lesser than the experimental value. It will be a good idea to figure out every plausible aspects one at a time

(i) Value of convection coefficient: Since no standard value of coefficient of convection is available, a constant value for h has been applied throughout the system. This can be a possible cause of error as the complexity in geometry will give a dynamic range of coefficient of convection for rotor, pad material and the caliper surface too. By changing the value of h from 50 W/(m^2 ˚K) to 20 W/(m^2 ˚K) for brake pad material and surrounding air,

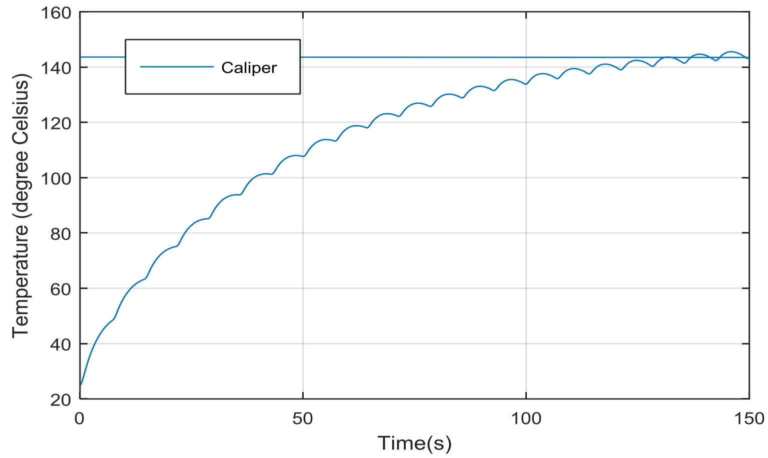

Figure 4-8 Temperature profile of caliper with h =20 W/ (m^2 ˚K)

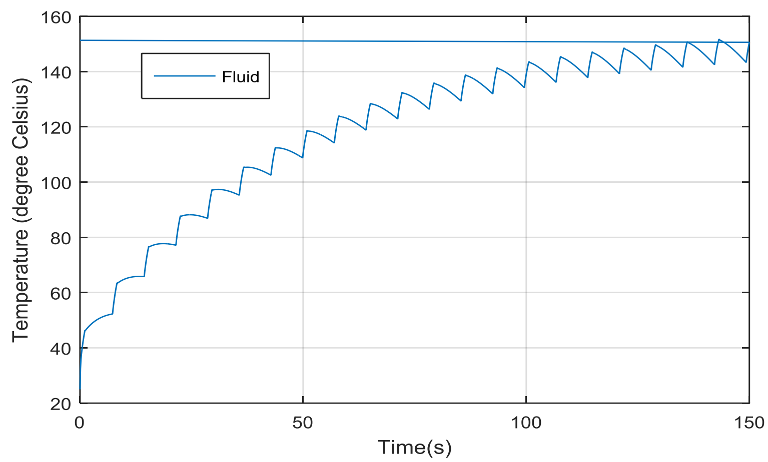

Figure 4-9 Temperature profile of brake fluid with h = 20 W/ (m^2˚K)

Table 4-3 Temperature of brake components with reduced convection coefficient for pad

	Brake fluid	Caliper
Temperature with decrease h=20W/m^2/K after 20 cycles (°C)	150	142
Increase in temperature	3 %	3 %

The temperature by changing the convection coefficient is now in the range of the calculated temperature. So it can be a possible explanation.

(ii) Material property of brake fluid: The property of the brake fluid that has been used for the calculations are that of ethylene glycol whose properties are similar to that of brake fluid, if not close. But the exact composition is a trade secret and the specific heat, density and thermal conductivity can only be found out by experimentations. So assuming values will give a mass approximation of the brake fluid present inside the caliper but not the exact one, which is a vital data for calculating brake fluid temperature. On decreasing the mass of the fluid, the temperature changes as follows

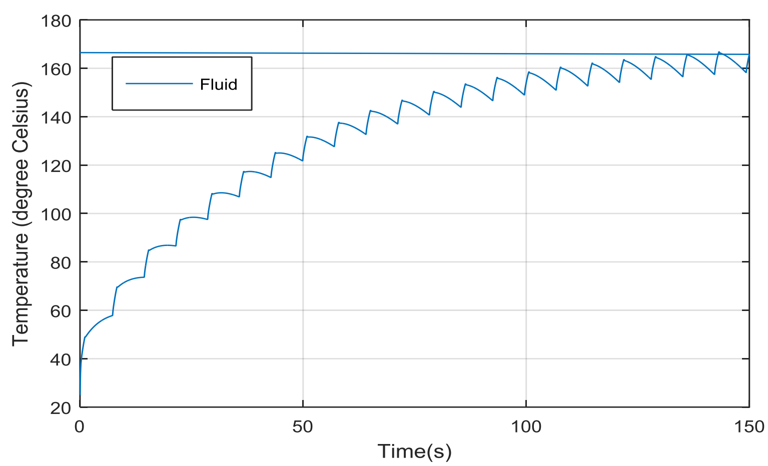

Figure 4-10 Temperature of brake fluid with decreased mass of the brake fluid

Table 4-4 Temperature of brake fluid after decreasing fluid mass after 20 cycles

Temperature of brake fluid after 20 cycles with decrease mass of 0.025 kg (°C)	165
Increase in fluid temperature	13 %

We can see an increase in temperature with decrease in mass of brake fluid from 0.030 Kilograms to 0.025 Kg.

(iii) Unknown material property of caliper: The brake calipers are made of aluminum but again its exact composition is not known. Looking by the temperature profile, it can be assumed the assumption of it being a pure aluminum body is minimum and there can be additives like copper, silica, magnesium etcetera, its thermal conductivity and specific heat is modified accordingly. By altering the standard specific heat value from 896 Joules per kilogram to 750 Joules per kilogram, the temperature profile is as follows

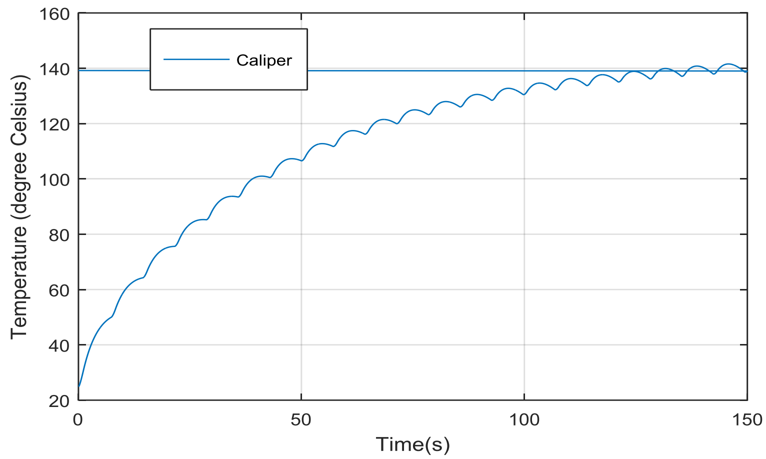

Figure 4-11 Temperature profile after decreasing the specific heat of aluminum

Table 4-5 Temperature of caliper after decreasing specific heat of the caliper after 20 cycle

Temperature after 20 cycles for caliper with decrease in specific heat of 750 J/kg (°C)	138
Increase in caliper temperature	0.6%

The contribution of the caliper specific heat is in increasing the temperature is very less.

(iv) Unknown material property of the brake pad material: The specific heat of the brake pad material has been approximated as 1000 Joules/kg by Belhocine and Bouchetara [15] .Its actual value is again a trade secret and can only be confirmed by experimentation. Still for the sake of understanding, by reducing the specific heat from 1000 Joules/Kilogram to 800 Joules per kilogram, the change in temperature profile will be

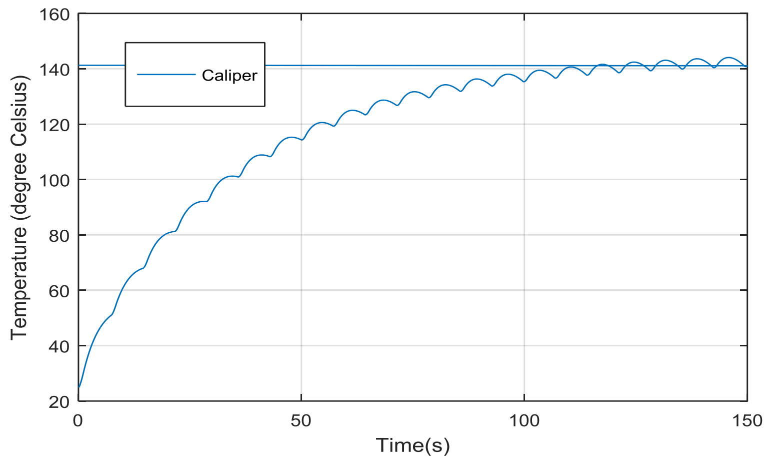

Figure 4-14 Temperature of fluid after 20 braking cycles with reduced specific heat for brake pad at 800 J/Kg

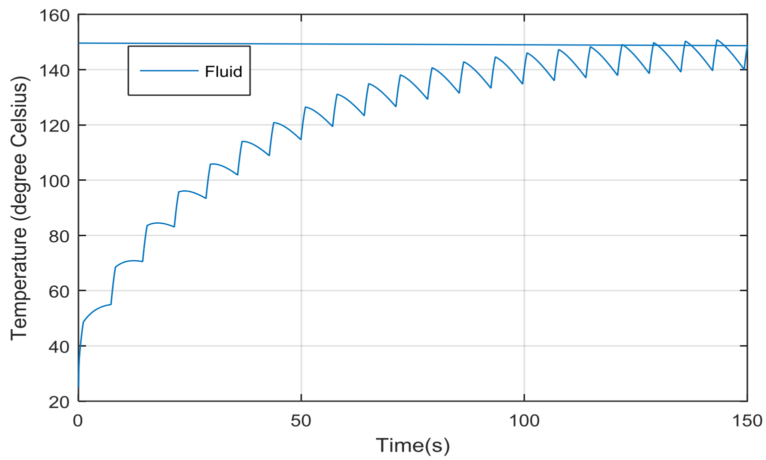

Figure 4-15 Temperature of brake fluid with the specific heat of brake pad as 800 J/kg (°C)

Table 4-6 Temperature of brake components with decrease in specific heat content of brake pads after 20 braking cycle

	Brake fluid	Caliper
Temperature after 20 braking cycles with pad specific heat as 800 J/kg	150	141
Increase in temperature	2.7%	2.2%

Chapter 5

Future scope of work

A lot many conclusions can be derived from this model and experiments can be carried out accordingly to refine this result value in order to develop a better brake system and work on braking pattern accordingly.

5.1 Experimental determination of convection coefficient

There are many kinds of convection coefficient involved in a brake system. First one is between the rotor and the circulating air behind it. The race car will move at different speed during different point of time. The convection coefficient will vary accordingly. Experiments for finding out better methods for calculation of heat generated at rotor is required to be found out.

The next type of convection coefficient is the one between the piston body and the associated fluid. Though the base material for piston is aluminum, most of the times it is an alloy for giving it a more durable property. Experiments can be performed in order to find that value.

One major form of convection coefficient is between the pads and the surrounding air. Since there is a very small gap between the pad and rotor, air flow will be restricted between it. Moreover, in case of a double disked vane rotor, the convection coefficient will be entirely different because of increased air flow.

Same can be said about the value of convection coefficient between the caliper and the surrounding air. In case of a double disk rotor with internal vane, there will be cooling of the caliper from bottom too and convection coefficient will be higher than any presumed value.

5.2 Experimental determination of material property of brake pads

Brake pad material vary from company to company and are trade secret. But their basic physical properties like knowing of specific heat, thermal conductivity and density will give a much more accurate result in calculation of all the brake components. When it comes to thermal properties, radiation losses will be significant in case of brake pads and rotor. Dragomir, Pancu, Bangau, Beles and Goergescu [16] experimentally proved that at elevated temperature of around 230 degree Celsius for cast iron, the value for thermal emissivity is around 0.405. Keeping that factor in mind, since emissivity is also a factor surface polish alongside temperature, the value for the emissivity has been chosen randomly as 0.9 for rotor. But there has been no basis for choosing its value as 0.9 for the brake pad material too and has been a very random value again. Experiments can be done for knowing the surface emissivity of brake pad material for more accurate radiation losses can be carried out.

5.3 Experimental determination of brake fluid properties

Though the material data sheet of various grades of DOT approved brake fluids are easily available, but still those required to plug in into the model formulated in this study are subjected to experiments. Specific heat and thermal conductivity are the two single most important data required to be found out experimentally which make a tremendous impact on the vaporization of brake fluid components.

5.4 Experimental determination of caliper and piston material properties

The caliper and the piston have been considered are made of same material, that is, Aluminum. For the sake of simulation, the Aluminum grade has been considered as pure. But in reality, the possibility of an alloy for the construction of caliper body is highly desirable. In order to reduce weight and decreasing specific heat, many form of alloys embedded with silicon or magnesium may be used which are required to be studied further. Same with the case of piston material too as they play a vital role in transferring heat from the brake pads to the brake fluid. So material properties for the Aluminum grade of piston can be determined experimentally in order to refine the results for the brake fluid temperature by the above model.

5.4 Determination of fin losses

Inside a disk brake caliper, there are hydraulic lines which carry fluid from one caliper body to the other. These lines can be treated as long extended fins coming out of the piston groove of the caliper body and can be tested for fin losses in order to get a more accurate brake fluid temperature. The above approximation of multi lumped capacitance modelling did not consider the fin losses as the mass of the whole brake fluid was considered as one. A mathematical modelling can be devised in order to account for the fin losses.

5.5 Determination of temperature for other brake components

Lumped capacitance modelling approach can be used to find the temperature of the other brake components like backing plate and piston with fair accuracy. As mentioned before the effectiveness of this modelling depend heavily upon the material properties of the components involved.

Chapter 6

Conclusion

The model developed to determine the brake fluid vaporization temperature showed some really interesting patterns. The heat generation and temperature rise of the rotor was well within accordance with experimental results. The governing equations used for the calculation of rotor temperature individually worked because the thermal mass capacitance was not required to be coupled to be with any other body. This provided the rotor the flexibility to reach a temperature irrespective of the temperature of the other components in the brake system.

The experimental results for the temperature of the caliper was not in the range of the one determined through simulation when its temperature was calculated individually. It could be concluded from this anomaly that even the brake fluid temperature will not reach vaporization temperature (which was further testified by the theoretical simulations) if this approach of individual temperature calculation was adopted for components other than rotor disk.

The caliper temperature continued to raise even after the brake fluid temperature to which it is contact with reaches a stable temperature. The reason for such a phenomenon was the high specific heat of the brake fluid. The fluid acted as thermal capacitor storage for the Aluminum caliper even after the fluid temperature itself stopped increasing. In order to accommodate this factor, the improvised model that included a coupled circuit containing two thermal capacitor storage, one each for the brake fluid and the caliper body gave a stable temperature profile for both the caliper and brake fluid and could be tallied to the experimental results.

With the availability of more accurate data for the physical and thermal property of the components of the brake system, the above model can be used reliably to predict the brake fluid temperature using a much simpler approach as discussed in this report. The same modelling approach can be used to find the temperature response of backing plate and piston too. 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.

Appendix A

Theoretical calculations

The circuit is required to be checked for Biot number and to be decided if it can be treated as a single or multi lumped capacitance modelling. Biot number is given by

$$N_b = \frac{R_{cond}}{R_{conv}} = \frac{hL_c}{k} \qquad \text{Eq. A.1}$$

Where, L_c is the characteristic length of solid material

Considering the same for the caliper, the Biot number comes out as

$$N_b = \frac{150\ x\ .00527}{236}$$

=.003349

Since the Biot number is lesser than 0.1, it can be considered for single lumped capacitance modeling too.

The equivalent circuit of the whole system when designed in single lumped capacitance model will be as follows:

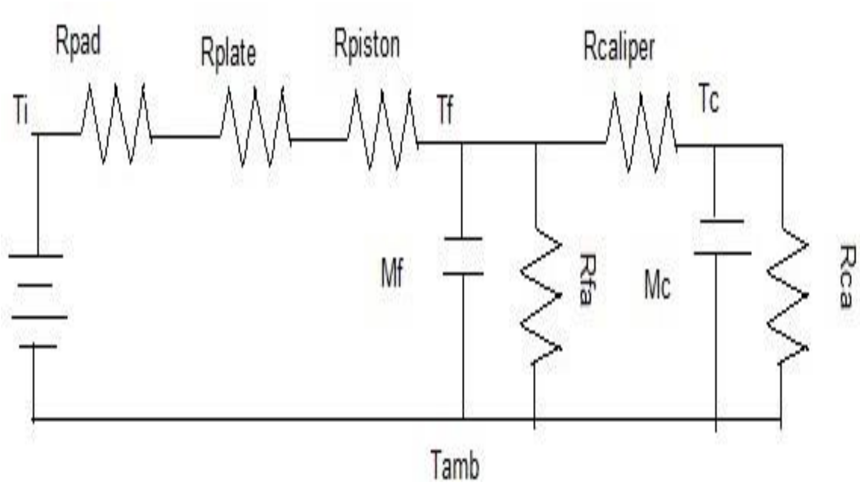

Figure A-1 Thermal circuit with components in series

The resistances in series can be equated by the following formulas

Brake pad: $R_{pad} = \frac{L_{pad}}{A_{pad}\,k_{pad}} = \frac{.00877}{5\ x\ .00145}$ = 1.209 °K/W Eq. A.3

Backing plate: $R_{plate} = \frac{L_{plate}}{A_{plate}\,k_{plate}} = \frac{.00322}{.00269\ x\ 54}$ = 0.022 °K/W Eq. A.4

Piston: $R_{piston} = \frac{L_{piston}}{K_{piston}A_{piston}} = \frac{.02286}{236\ x\ .0014}$ = 0.065 °K/W Eq. A.5

R_{fa}: = $\frac{1}{A_{fluidexposed}h_f} = \frac{1}{1.5\ x\ .00349}$ = 191.1°K/W Eq. A.6

$R_{caliper}$ = $\frac{L_{caliper}}{k_{caliper}\,A_{caliper}} = \frac{.00502}{236\ x\ .002}$= .0106°K/W Eq. A.7

R_{ca} = $\frac{1}{h\ x\ A_{caliper}} = \frac{1}{150\ x\ 0.0173}$ = 0.38 °K/W Eq. A.8

The equivalent circuit resistance will be

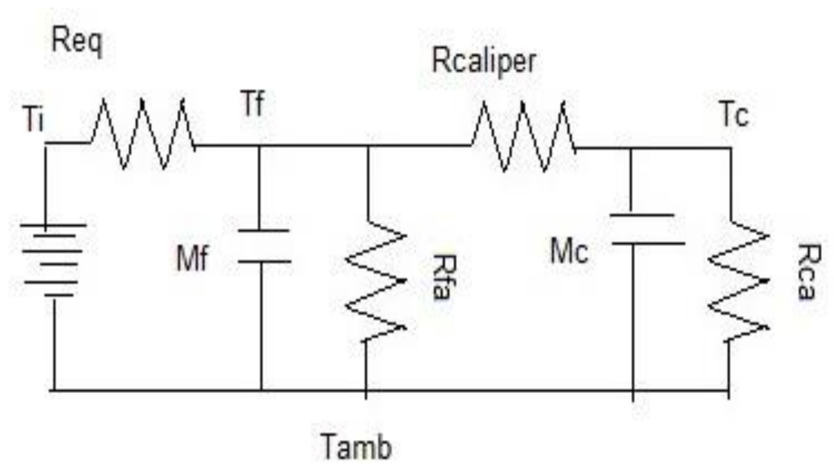

Figure A-2 Thermal circuit with components lumped together

$R_{eq} = R_{fa} + R_{caliper} + R_{ca}$ Eq. A.9

= 1.209 + 0.022 + 0.065

= 1.29 °K/W

Appendix B

Properties of Metals

Properties at 20°C

Metals	ρ, kg/m^3	c_p, KJ/Kg °C	*k,* W/m°C
Aluminum:			
Pure	2707	0.986	204
Al-cu (94-96% Al,			
3-5% Cu, trace Mg)	2787	0.883	164
Al-Si (86.5% Al, 1% Cu)	2659	0.867	137
Al-Si (78-80% Al, 20-22% Si)	0.854	161(97%Al)	7.172
Al-Mg-Si (1% Mg,1% Si)	2707	0.892	177
Iron:			
Pure 7897	0.452	73	2.034
Wrought iron 0.5% C	7849	0.46	59
Steel			
Carbon steel			
C=0.5%	7833	0.465	54
1 %	7801	0.473	43
1.5 %	7753	0.486	36
Nickel Steel			

Properties at 20°C

Metals	ρ, kg/m^3	c_p, KJ/Kg °C	*k*, W/m°C
Ni =0%	7897	0.452	73
20%	7933	0.46	19
40%	8169	0.46	10
80%	8618	0.46	35
Invar 36% Ni	8137	0.46	10.7
Chrome steel			
Cr=0%	7897	0.452	73
1%	7865	0.46	61
5%	7833	0.46	40
20%	7869	0.46	22
Cr-Ni (Chrome-Nickel)			
15%Cr, 10%Ni	7865	0.46	19
18%Cr,8%Ni	7817	0.46	16.3
20%Cr,15% Ni	7833	0.46	15.1
25%Cr,20%Ni	7865	0.46	12.8

Appendix C

MATLAB codes

Code for finding rotor temperature

```
pi=3.14;

Tamb=25;

Wcar=2800;

g=9.8;

gees=-1;

Vmax=26.82;

Vmin=13.41;

Dout=.2413;

Din=.1524;

Thick=.0053;

Wslot= .00635;

Lslot= .042;

Wmount= .0254;

Lmount=.02159;

rho=7850;

k=51.88;

Cp=510;

tdisk=Thick;

Adisk=  pi/2*(Dout^2-Din^2)+ pi*tdisk*(Dout+DinAslot=2*15*tdisk*(Wslot+Lslot);

A=Adisk+Aslot

Ap=(pi*.02225^2*2) + (2*pi*.02225*.02794)

Vpiston= pi*.02225*.02225*.02794;
```

```
rhowater=1000;

Mf= rhowater*Vpiston;

Vdisk= (((pi/4*(Dout^2-Din^2))-15*Wslot*Lslot)*tdisk);

V=Vdisk

M= rho*V

h=150;

Afin=pi*.005*.005;

Pfin=3.14*.01;

hA=h*A;

kAL= 4*k*Wmount*tdisk/Lmount; hAkAL=hA+kAL;

MCp=M*Cp;

MfCpf=Mf*Cpf;

Fv=1; Fe=.9;

SB= 5.67e-8; FvFeSBA= Fv*Fe*SB*A;


n=7;

Tstep=0.1;

Tfinal=300;

X= zeros (n,Tfinal/Tstep);

X(1,1)=26.82;

 X(3,1)=0;

X(4,1)=25;

X(5,1)=25;

X(6,1)=25;

X(7,1)=25;

Gdec=-1.0*9.8;
```

```
Gacc=.5*9.8;

tset=0;



for t=0:0.1:300;

    i=round((t*10)+1);

    X(2,i)=t;

     if X(1,i)>=26.82

         X(1,i+1)=X(1,i)+Gdec*Tstep;

         X(3,i+1)=X(3,i) - (Wcar/3)*X(1,i)*gees;

         tset=0;

     elseif 26.82>X(1,i) && X(1,i)>13.41 && X(1,i)<X(1,i-1)

         X(1,i+1)=X(1,i)+Gdec*Tstep;

         X(3,i+1)=X(3,i) - (Wcar/3)*(X(1,i+1)-X(1,i))*gees;

     elseif 26.82>X(1,i) && X(1,i)>13.41 && X(1,i-1)<X(1,i)

         X(1,i+1)=X(1,i)+Gacc*Tstep;

         X(3,i+1)=0;

     elseif X(1,i)<=13.41 && tset<=30

         tset = tset+1;

         X(1,i+1)=X(1,i);

         X(3,i+1)=0;

     elseif X(1,i)<=13.41 && tset>30

         X(1,i+1)=X(1,i)+Gacc*Tstep;

         X(3,i+1)=0;

     end
```

```
        X(4,i+1)=X(4,i)+((X(3,i)*.93)-hAkAL*(X(4,i)-Tamb) + FvFeSBA*((Tamb+273)^4-
(X(4,i)+273)^4))*.1/MCp;
        X(5,i+1)=X(5,i)+((X(3,i)*.95)-hAkAL*(X(5,i)-Tamb) + FvFeSBA*((Tamb+273)^4-
(X(5,i)+273)^4))*.1/MCp;
        X(6,i+1)=X(6,i)+((X(3,i)*.97)-hAkAL*(X(6,i)-Tamb) + FvFeSBA*((Tamb+273)^4-
(X(6,i)+273)^4))*.1/MCp;
        X(7,i+1)=X(7,i)+((X(3,i)*1)-hAkAL*(X(7,i)-Tamb) + FvFeSBA*((Tamb+273)^4-
(X(7,i)+273)^4))*.1/MCp;

    end


    %plot(X(2,:),X(4,:),X(2,:),X(5,:),X(2,:),X(6,:),X(2,:),X(7,:))
    plot (X(2,:),X(4,:))


    grid
    xlabel('Time(s)')
    ylabel('Heat generation for rotor( Joules per second )')
```

Code for finding the pad temperature

```
    pi=3.14;
    Tamb=25
    Wcar=2800;
    g=9.8;
    gees=-1;
```

```
Vmax=26.82;

Vmin=13.41;

kpad=5;

Cp=1000;

tpad=0.0087

Apad=  .00145

A=Apad;

M= .042

h=50;

hA=h*A; kAL= kpad*A/tpad; hAkAL=hA+kAL;

MCp=M*Cp;

Fv=1;

Fe=.2;

SB= 5.67e-8;

FvFeSBA= Fv*Fe*SB*A;


n=7;

Tstep=0.1;

Tfinal=200;

X= zeros (n,Tfinal/Tstep);

X(1,1)=26.82;

X(3,1)=0;

X(4,1)=25;

X(5,1)=25;

X(6,1)=25;

X(7,1)=25;
```

```
Gdec=-1.0*9.8;

Gacc=.5*9.8;

tset=0;



for t=0:0.1:200;

    i=round((t*10)+1);

    X(2,i)=t;

     if X(1,i)>=26.82

         X(1,i+1)=X(1,i)+Gdec*Tstep;

         X(3,i+1)=X(3,i) - (Wcar/3)*X(1,i)*gees;

         tset=0;

     elseif 26.82>X(1,i) && X(1,i)>13.41 && X(1,i)<X(1,i-1)

         X(1,i+1)=X(1,i)+Gdec*Tstep;

         X(3,i+1)=X(3,i) - (Wcar/3)*(X(1,i+1)-X(1,i))*gees;

     elseif 26.82>X(1,i) && X(1,i)>13.41 && X(1,i-1)<X(1,i)

         X(1,i+1)=X(1,i)+Gacc*Tstep;

         X(3,i+1)=0;

     elseif X(1,i)<=13.41 && tset<=30

         tset = tset+1;

         X(1,i+1)=X(1,i);

         X(3,i+1)=0;

     elseif X(1,i)<=13.41 && tset>30

         X(1,i+1)=X(1,i)+Gacc*Tstep;

         X(3,i+1)=0;

     end
```

```
X(4,i+1)=X(4,i)+((X(3,i)*.035)-hAkAL*(X(4,i)-Tamb) + FvFeSBA*((Tamb+273)^4-
(X(4,i)+273)^4))*.1/MCp;
X(5,i+1)=X(5,i)+((X(3,i)*.025)-hAkAL*(X(5,i)-Tamb) + FvFeSBA*((Tamb+273)^4-
(X(5,i)+273)^4))*.1/MCp;
X(6,i+1)=X(6,i)+((X(3,i)*.015)-hAkAL*(X(6,i)-Tamb) + FvFeSBA*((Tamb+273)^4-
(X(6,i)+273)^4))*.1/MCp;
X(7,i+1)=X(7,i)+((X(3,i)*0)-hAkAL*(X(7,i)-Tamb) + FvFeSBA*((Tamb+273)^4-
(X(7,i)+273)^4))*.1/MCp;

end

plot(X(2,:),X(4,:),X(2,:),X(5,:),X(2,:),X(6,:),X(2,:),X(7,:))
%plot (X(2,:),X(3,:))

grid
xlabel('Time(s)')
ylabel('Temperature for brake pad material( Degree Celsius )')
```

Code for finding lumped capacitance modelling for finding the fluid and caliper temperature

```
pi=3.14;
Tamb=25;
Wcar=2800;
g=9.8;
gees=-1.5;
```

```
Vmax=26.82;
Vmin=13.41;
ksteel=12;
kalum=236;
Cpsteel=510;
Cpfluid=2360;
tpad=0.0087;
kpad=5;
Aopwt=0.00315;
Aip=0.00349;
Aop=0.00463;

Vpiston=0.0000109;

Rhow=1000;
Rhoeg=1113;

Mfluid= 0.030;
Mcal=0.308;
Cpcal=896;
Md=0.9906;

h=150;
hf=1.5;

Apad=0.00145;
```

```
Ad=0.05;
hAd=h*Ad;
hAp=50*Apad;

kAL=ksteel*(((0.03*0.0053)/0.02)*8
hAdkAL=hAd+kAL;
kpAL=(kpad*Apad)/tpad;
hApkpAl=hAp+kpAL;



Fv=1;
Fe=.2;
SB= 5.67e-8;
FvFeSBA= Fv*Fe*SB*Apad;

n=10;
Tstep=0.1;
Tfinal=300;
X= zeros (n,Tfinal/Tstep);
X(1,1)=26.82;
X(3,1)=0;
X(4,1)=25;
X(5,1)=25;
X(6,1)=25;
X(7,1)=25;
X(8,1)=25;
```

```
X(9,1)=25;
X(10,1)=25;
Gdec=-1.5*9.8;
Gacc=.5*9.8;
tset=0;


for t=0:0.1:300;
    i=round((t*10)+1);
    X(2,i)=t;
     if X(1,i)>=26.82
         X(1,i+1)=X(1,i)+Gdec*Tstep;
         X(3,i+1)=X(3,i) - (Wcar/3)*X(1,i)*gees;
         tset=0;
     elseif 26.82>X(1,i) && X(1,i)>13.41 && X(1,i)<X(1,i-1)
         X(1,i+1)=X(1,i)+Gdec*Tstep;
         X(3,i+1)=X(3,i) - (Wcar/3)*(X(1,i+1)-X(1,i))*gees;
     elseif 26.82>X(1,i) && X(1,i)>13.41 && X(1,i-1)<X(1,i)
         X(1,i+1)=X(1,i)+Gacc*Tstep;
         X(3,i+1)=0;
     elseif X(1,i)<=13.41 && tset<=30
         tset = tset+1;
         X(1,i+1)=X(1,i);
         X(3,i+1)=0;
     elseif X(1,i)<=13.41 && tset>30
         X(1,i+1)=X(1,i)+Gacc*Tstep;
```

```
X(3,i+1)=0;
        end
        X(4,i+1)=X(4,i)+((X(3,i)*.015)-hApkpAl*(X(4,i)-Tamb)+FvFeSBA*((Tamb+273)^4-
(X(4,i)+273)^4))*0.1/(.042*1000);

        X(9,i+1)=((X(4,i)-X(9,i))/1.29+((X(3,i)*.015)-(X(9,i)-Tamb)/191.1 - (X(9,i)-
X(10,i)/0.010))*0.1/(0.03*1113))+X(9,i);
        X(10,i+1)=(((-1*(X(10,i)-X(9,i))/0.010) + (-1*(X(10,i)-
Tamb)/0.38))*0.1/(0.307*896))+X(10,i);

    end


    %plot(X(2,:),X(8,:),X(2,:),X(5,:),X(2,:),X(6,:),X(2,:),X(7,:))
    plot (X(2,:),X(10,:))




    grid
    xlabel('Time(s)')
    ylabel('Temperature of caliper (degree Celsius)')
```

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Biographical Information

Anurag Dey, was born to Mala Dey and Ashish Kumar Dey on 6th of September 1988 in the small county of Jagiroad, Assam, India. After completing his schooling from Kendriya Vidyalaya, H.P.C.L Jagiroad, he decided to pursue his Bachelor's degree from Assam Engineering College in the field of Mechanical Engineering in 2006. His professional career started with Punj Lloyd Limited, a Fortune 500 company in 2010 where he worked as a maintenance engineer for 2 year. His next leap of faith was for teaching when he returned to the noble profession as an Assistant Professor in The Department of Mechanical Engineering in 2012 and continued there for a year. In search of excellence, he decided to pursue his masters in the field of Mechanical Engineering from University of Texas at Arlington in 2013. His plans are to return back to India after his graduation and continue with his old job in order to develop a better society.