Development of a Model for Predicting Brake Friction Lining Thickness and Brake Temperature

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ABSTRACT

Road traffic injuries and deaths are a growing public health concern worldwide, majorly in developing countries. Brake failure constitutes to be one of the primary reasons for accidents. The majority of brake failures are caused due to overheating of the brakes, while wear of lining is another big share-holder. Early detection of such causes can prevent these accidents. This study puts forth a model that can be used for onboard monitoring of drum/disc temperature & lining/pad thickness by taking velocity & road inclination in real-time as inputs. Many quantities are interdependent and vary with respect to time/temperature. Therefore, an incremental approach is used. The model is implemented in the Simulink software. Many standard profiles are also fed to compare results for different terrains and driving conditions. The drivers can also be classified based on their driving behavior. The thermal model can give us an early warning about the brake overheating. This model can be used to study the energy distribution while braking. Researchers and designers can also use this model to study & optimize the brake system.

1. INTRODUCTION

The growing number of automobiles across the world has caused a significant increase in road traffic accidents. Brake failure is one of the primary reasons for such accidents. Some of these accidents lead to severe injuries or death and can be avoided if the driver is given an early indication about brake failure. Among many reasons for brake failure, wearing out of the lining and overheating of Drum/Disc are the prominent reasons. Over the period many researchers have studied the brake system by different approaches.

https://doi.org/10.36001/IJPHM.2022.v13i1.3064

An experimental study comparing cooling rates of brake drum & disc has been conducted. The drums or discs were heated to a uniform temperature of 300^{0}-400^{0}C by drag braking, and the rate at which they cooled were measured while the vehicle was driven at a constant speed. Measurements were made at various speeds in the range of 0 to 90 miles/hr (Newcomb & Millner, 1965). The FEA method based on the phase transfer method in the time region is established vehicle braking transient thermal field for single & repetitive braking (Liang, Jian & Xuele, 2005). Another paper presents a thermal analysis of a sub-set drum/shoe brake lining of a braking system in a rear drum brake for a lightweight passenger vehicle using FEM for single & repetitive braking (Chiaroni & Silveira, 2014). A study has been conducted to compute the convective heat transfer coefficient of automotive disc brake rotors using CFD modeling (Belhocine & Wan Omar, 2017). Another paper summarizes challenges faced, discusses the effects that can and cannot be captured, and gives a broader picture of the issues faced when conducting numerical brake cooling simulations using CFD and CAE (Vdovin & Gigan, 2020).

The machine learning approach is used by various researchers for health monitoring and prognostics of various automotive systems. Vehicle Remote Health Monitoring and Prognostic Maintenance System was proposed, which uses ML techniques for fault prediction of four main subsystems of vehicle: fuel system, ignition system, exhaust system, and cooling system (Shafi, Safi, Shahid, Sheikh & Saleem, 2018). Many types of research have been conducted using the ML approach to deal with vibration-based fault diagnosis of automobile hydraulic brake systems (Jayakrishnan, Manghai & Jegadeeshwaran, 2020; Jegadeeshwaran & Sugumaran, 2015). A prediction model was proposed for the brake drum temperature of large trucks on consecutive mountain downgrade routes. It mainly analyses the effect of the four variables, namely, the speed and weight of the truck, and the percentage and length of the grade, on the brake drum temperature (Yan & Jinliang, 2018). In another research, a data acquisition system was designed, constructed, and implemented to measure and store in real time the performance-related parameters of brake systems like distance, velocity, acceleration, brake line pressure, temperature, & effectiveness (Fumi & Sultan, 2009).

In the study titled 'braking system modeling and brake temperature response to repeated cycle', brake characteristic and vehicle dynamic model was generated in MATLAB/Simulink software to estimate friction force and dissipated heat. Furthermore, Arduino-based prototype brake temperature monitoring was developed and tested on the road (Dalimus, 2014). In another paper, the effect of contact force and friction radius is studied with varying conditions of parameters; longitudinal force, caliper force, and torque on piston side as well as non-piston side by using Matlab/Simulink models for drum & disc brakes (Khairnar, Phalle & Mantha, 2016). Another study for estimation of automotive brake drum–shoe interface friction coefficient under varying conditions of longitudinal forces using Simulink was conducted (Khairnar, Phalle, Mantha, 2015).

Existing studies/models discussed above consider either a single braking cycle or repetitive braking cycle of the same kind, uniform terrain, or constant road inclination. Partial braking is not considered. Assumptions made for simplicity, such as considering thermophysical properties constant, make the model deviate from being realistic. Most models cannot be directly implemented onto the vehicle for real-time monitoring & prognosis of brake health. Hence, a simple system that takes input from the speedometer, works in any terrain for any vehicle and monitors the wear and temperature will be an ideal choice without any further modifications in the car.

The proposed model requires only velocity and road inclination profiles as inputs that can be easily acquired. It uses an incremental approach to monitor the wear & temperature, which is explained in the methodology part of the paper along with the mathematical model of the system. MATLAB/Simulink implementation of this model is shown next.

2. METHODOLOGY & MATHEMATICAL MODELING

2.1. Incremental approach

In actual driving, velocity, road inclination & various resistive forces vary w.r.t. time. Physical properties like Specific heat capacity *(C)*, Thermal conductivity *(k)* and Specific wear rate coefficient *(ki)* vary w.r.t. temperature. Convective heat transfer coefficient *(h)* varies with respect to velocity. To take such interdependencies & variations in consideration incremental approach is used as shown in Figure 1.

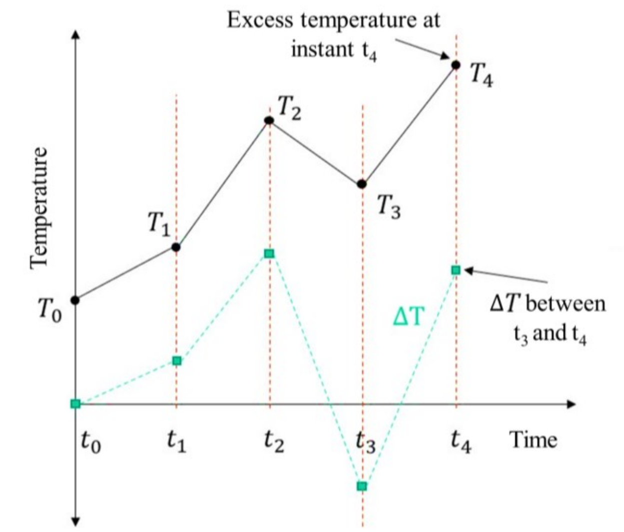


Figure 1. Incremental approach

In this approach, the time is subdivided into several small intervals of length t_{step}& wear/temperature rise is calculated for each time interval. Total wear and total temperature rise were calculated up to that instant considering summation of all such wear displacement and change in temperature, respectively.

2.2. Mathematical Model

2.2.1. Wear Model

Figure 2 shows the flowchart for Wear model for every time step. This is discussed in detail as follows.

By slight modification in Archard's equation,

The equation for wear happening in specified time interval t_{step}, is given by:

$$\Delta h_i = I \times k_i(T) \times \alpha_i \times (P_{avg})_{max} \times \Delta S_d \quad (1)$$

Where, Δh_i is the wear displacement in time interval t_{step}. I corresponds to condition 'is brake applied?' that can be specified by value *'0'* or *'1'* based on whether the brake is applied. $k_i(T)$ is specific wear rate coefficient of a brake pad material which depends on temperature of the drum/disc. α_i is the instantaneous partial braking fraction. $(P_{avg})_{max}$ is the average pressure on lining/pad when brakes are applied at their maximum capacity. ΔS_d is the sliding distance between lining/pad & drum/disc during time interval t_{step}.

The partial braking concept is introduced as for most of the times driver does not apply brakes to its full capacity. This partial braking fraction can be calculated as shown below, without using any sensor.

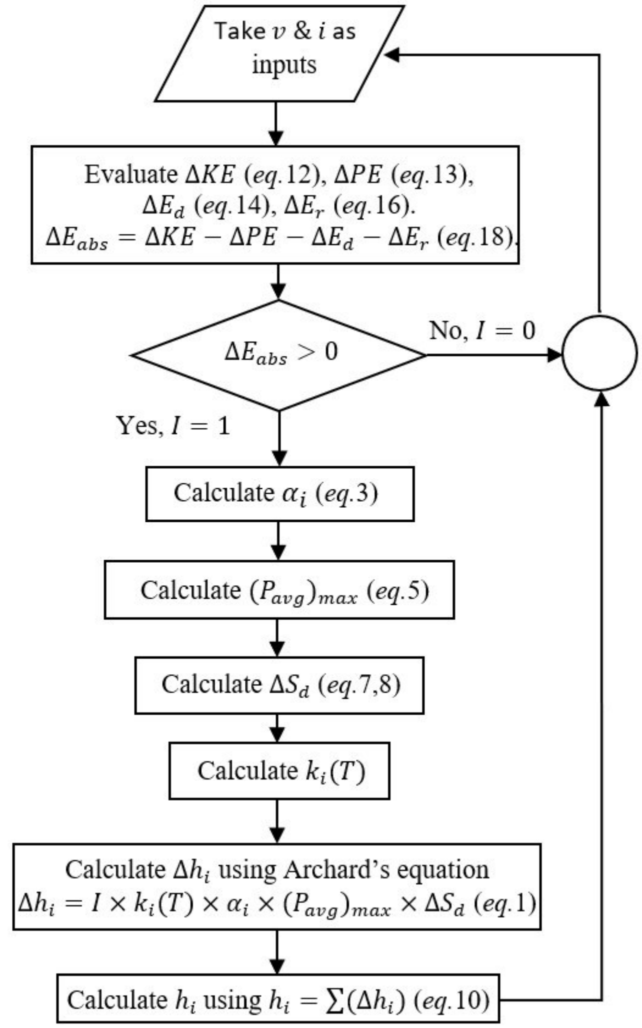


Figure 2. Flowchart for wear model

Instantaneous partial braking fraction (α_i) is defined as:

$$\alpha_i = \frac{P}{(P_{avg})_{max}} \tag{2}$$

But braking torque is directly proportional to pressure applied (P) by the lining/pad on drum/disc (Bhandari, 2010). Braking torque applied is inversely proportional to braking distance. Therefore, we get,

$$\alpha_i = \frac{P}{(P_{avg})_{max}} = \frac{\Delta(S_b)_{min}}{\Delta S_b} \tag{3}$$

$$\Delta(S_b)_{min} = \frac{\Delta(v^2)}{2a_{max}} \tag{4}$$

Where, ΔS_b is actual distance travelled while brakes are applied during time interval t_{step}. $\Delta(S_b)_{min}$ is the distance travelled if the brakes were applied at their full capacity during time interval t_{step}. $\Delta(v^2)$ is the change in square of vehicle velocity (v) during time interval t_{step}. a_{max} is the maximum possible deceleration. S_b is calculated by integrating velocity when acceleration is negative.

In drum brakes, pressure on lining changes from point to point (Bhandari, 2010). Hence average pressure should be considered. P_{max} is the maximum allowable pressure of lining/pad material. The drum brake is shown in Figure 3.

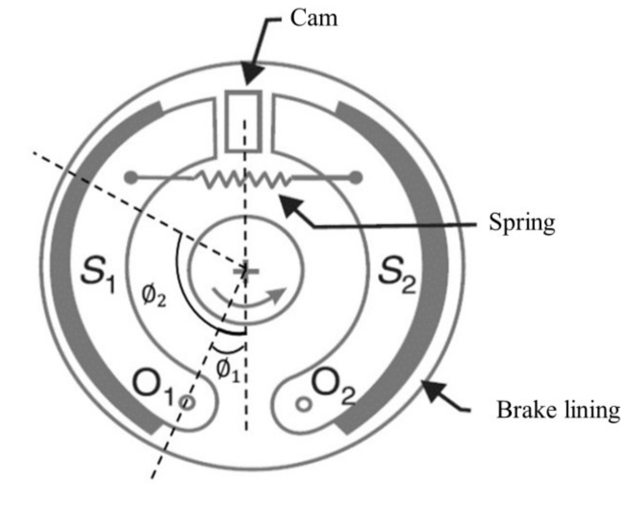


Figure 3. Drum Brake

Average pressure when the brake is applied at its full capacity can be given by:

$$(P_{avg})_{max} = \frac{P_{max}}{\sin \emptyset_{max}} \times \frac{\cos \emptyset_1 - \cos \emptyset_2}{\emptyset_2 - \emptyset_1} \tag{5}$$

In disc brakes, pressure is uniformly applied through hydraulic actuation all over the pad. Therefore, for disc brakes:

$$(P_{avg})_{max} = P_{max} \tag{6}$$

Sliding distance for drum brake can be given by:

$$\Delta S_d = \frac{r_{di}}{r_w} \times \Delta S_b \tag{7}$$

r_{di} is the inner radius of drum & r_w is radius of wheel.

Sliding distance in case of disc brakes is given by:

$$\Delta S_d = \frac{r_e}{r_w} \times \Delta S_b \tag{8}$$

r_e is the effective radius for disc and is given by:

$$r_e = \frac{2({r_o}^3 - {r_i}^3)}{3({r_o}^2 - {r_i}^2)} \tag{9}$$

r_o & r_i are outer radius & inner radius of disc, respectively.

The total wear displacement (h_i) will be the sum of wear displacements at each interval and it can be represented as follows:

$$h_i = \sum(\Delta h_i) \tag{10}$$

$$\textit{Remaining thickness} = \textit{Maximum thickness} - \sum(\Delta h_i) \tag{11}$$

Maximum thickness is the original lining thickness.

2.2.2. Thermal Model

The flowchart for Thermal model is as shown in Figure 4 and the equations shown here are discussed in detail in the following section.

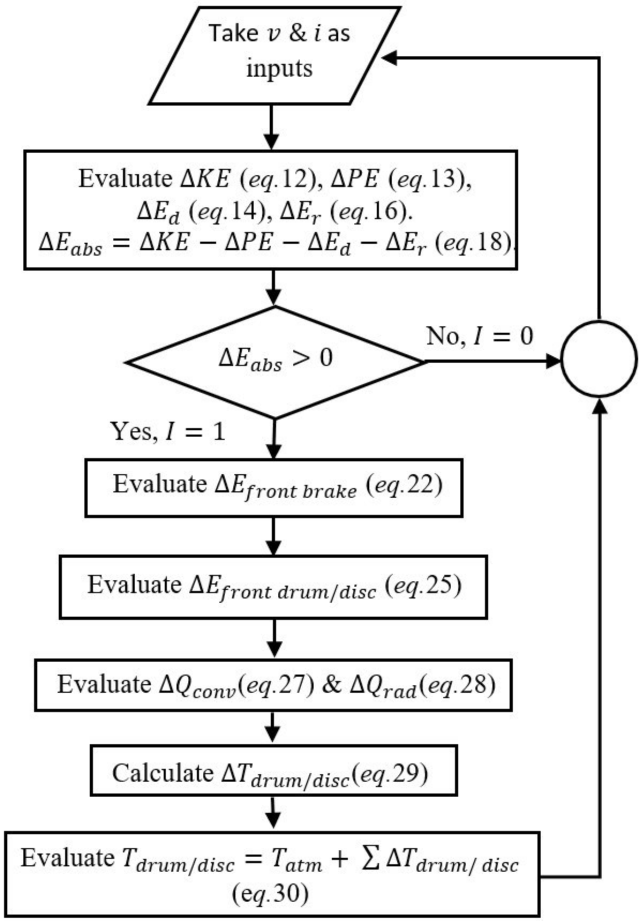


Figure 4. Flowchart for thermal model

Kinetic Energy to be absorbed during time interval t_{step}:

$$\Delta KE = \frac{1}{2} m(K+1) \times \Delta(v^2) \tag{12}$$

m is the total mass of the vehicle including passengers and luggage if any.

K is defined as ratio of rotational kinetic energy to be absorbed to the translational kinetic energy to be absorbed of the vehicle during braking.

Change in Potential Energy in during time interval t_{step}:

$$\Delta PE = m \times g \times v \sin i \times t_{step} \tag{13}$$

g is the acceleration due to gravity.

i is the instantaneous road inclination in radians.

Energy lost in overcoming aerodynamic drag in t_{step}:

$$\Delta E_d = F_d \times \Delta S_b \tag{14}$$

$$F_d = \frac{C_d}{2} \times \rho \times A_p \times v^2 \tag{15}$$

Assumption: Drag Force (F_d) is assumed to be constant over the time interval t_{step} .

C_d is the coefficient of drag.

ρ is the density of air.

A_p is frontal projected area of vehicle.

Energy lost in overcoming rolling resistance in t_{step} : (Yan & Jinliang, 2018)

$$\Delta E_r = F_r \times \Delta S_b \tag{16}$$

$$F_r = 0.001 \times m(C_1 + C_2 v) \tag{17}$$

Assumption: Rolling resistance force (F_r) is assumed to be constant over the time interval t_{step}.

C_1 & C_2 are parameters related to tire types.

Total energy to be absorbed during time interval t_{step} is given by:

$$\Delta E_{abs} = \Delta KE - \Delta PE - \Delta E_d - \Delta E_r \tag{18}$$

If ΔE_{abs} is positive → It should be absorbed by brakes and road-tire interface.

If ΔE_{abs} is zero → KE is already absorbed by the resistive forces and PE without application of brakes.

If ΔE_{abs} is negative →Energy is supplied by the engine to overcome fraction of resistive forces and PE.

This same logic is used to calculate value of I (is brake applied?) in wear model. If ΔE_{abs} is positive, I is 1. Otherwise, I is 0.

Energy absorbed by brake system during time interval t_{step} is given by:

$$\Delta E_{brakes} = x \times \Delta E_{abs} \tag{19}$$

Out of the total energy absorbed during braking, some of the energy is also dissipated at road-tire interface. x is fraction of energy absorbed by brakes (Dalimus, 2014).

Fraction of ΔE_{brakes} absorbed by front and rear brakes (Jazar, 2017):

$$\textit{Fraction of Energy absorbed by front brakes} = \frac{N_f}{N_f + N_r} \tag{20}$$

$$\textit{Fraction of Energy absorbed by rear brakes} = \frac{N_r}{N_f + N_r} \tag{21}$$

Energy absorbed by each front and rear brake during time interval t_{step} is given by:

$$\Delta E_{front\ brake} = 0.5 \times \frac{N_f}{N_f + N_r} \times \Delta E_{brakes} \tag{22}$$

$$\Delta E_{rear\ brake} = 0.5 \times \frac{N_r}{N_f + N_r} \times \Delta E_{brakes} \tag{23}$$

Fraction of heat energy going to drum *(y)* (Chiaroni & Silveira, 2014)

$$y_i = \frac{\xi_d(T) \times A_{drum/disc}}{\xi_d(T) \times A_{drum/disc} + \xi_l(T) \times A_{lining/pad}} \quad (24)$$

$$\Delta E_{front\ drum/disc} = y_i \times \Delta E_{front\ brake} \quad (25)$$

$$\Delta E_{rear\ drum/disc} = y_i \times \Delta E_{rear\ brake} \quad (26)$$

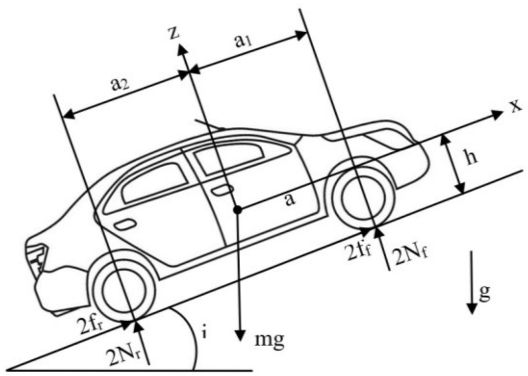


Figure 5. An accelerating car on inclined pavement

Cooling during time interval t_{step}:

Convection:

$$\Delta Q_{conv} = h(v) \times A_{conv} \times (T_{drum/disc} - T_{atm}) \times t_{step} \quad (27)$$

$h(v)$ is the convective heat transfer coefficient at velocity v, A_{conv} is the convective heat transfer area, $T_{drum/disc}$ is temperature of drum/disc at that instant, T_{atm} is atmospheric temperature.

Radiation:

$$\Delta Q_{rad} = \sigma \varepsilon A_{rad} ({T_{drum/disc}}^4 - {T_{atm}}^4) \times t_{step} \quad (28)$$

σ is Stefan-Boltzmann constant, ε is emissivity of drum/disc, A_{rad} is radiative heat transfer area.

Temperature of drum increased during time interval t_{step} (ΔT_{drum} *and* ΔT_{disc}):

$$m_{drum} \times C_{drum}(T) \times \Delta T_{drum} = \Delta E_{drum} - (\Delta Q_{con} + \Delta Q_{rad}) \quad (29)$$

Here, while calculating ΔT_{drum}, use m_{drum} and $C_{drum}(T)$ and while calculating ΔT_{disc} use m_{disc} and $C_{disc}(T)$.

Temperature of drum/disc at any instant is given by:

$$T_{drum/disc} = T_{atm} + \sum \Delta T_{drum/\ disc} \quad (30)$$

2.3. Classification of drivers

Wear is a quantifiable parameter that has its dependency on driving behavior. Wear in case of harsh driving is significantly higher than its good counterpart. Wear rate could therefore be used as an indicator of the driving behavior.

Drivers could also be classified based upon their wear rates. Indication to the driver about their driving can improve the driver's driving behavior, improving the vehicle's performance and fuel efficiency (Bhandari, 2010). This would also be of uttermost importance in the case of rental vehicles where drivers could be charged based upon their driving.

Driving behavior can be indicated with the help of the slope of the linear fitted curve of 'Wear displacement' vs 'Distance travelled by the vehicle'.

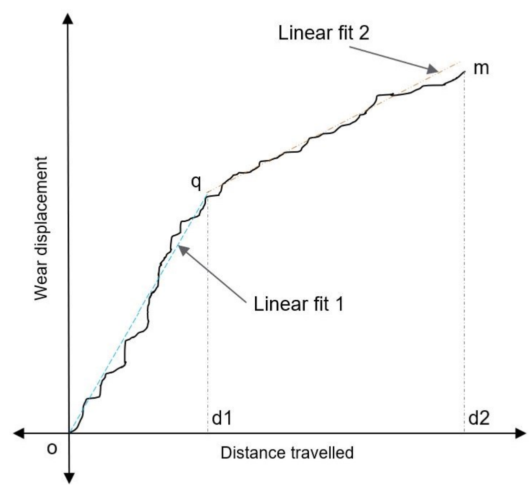


Figure 6. Comparison between two drivers

In Figure 6,

$$Slope\ of\ curve = \frac{wear\ displacement}{Distance\ travelled\ by\ the\ vehicle} \quad (31)$$

The slope of the linear fit to the graph of "Wear displacement" vs "Distance travelled" indicates average wear displacement per unit distance travelled. Therefore, lesser the value of slope, better the driving behavior and vice versa. As seen from figure 6, the slope of linear fit 1 is more than linear fit 2. Hence, the driving behavior between the distance od1 is harsher than that of d1d2.

3. SIMULINK MODEL AND MATLAB PROGRAMMING

Figure 7 shows the starting window of the model. The input parameters (on left), outputs (on top right) and their pictorial & graphical representations can be seen on a single panel. Figure 8 shows the thermal part of the Simulink model. All the equations discussed in mathematical modeling are implemented using various Simulink blocks. This model takes velocity and road inclination as inputs and calculates change in temperature in given instant, then sums it up to that instant, to show excess temperature of drum/disc over atmosphere. After adding atmospheric temperature to this, we get instantaneous drum/disc temperature.

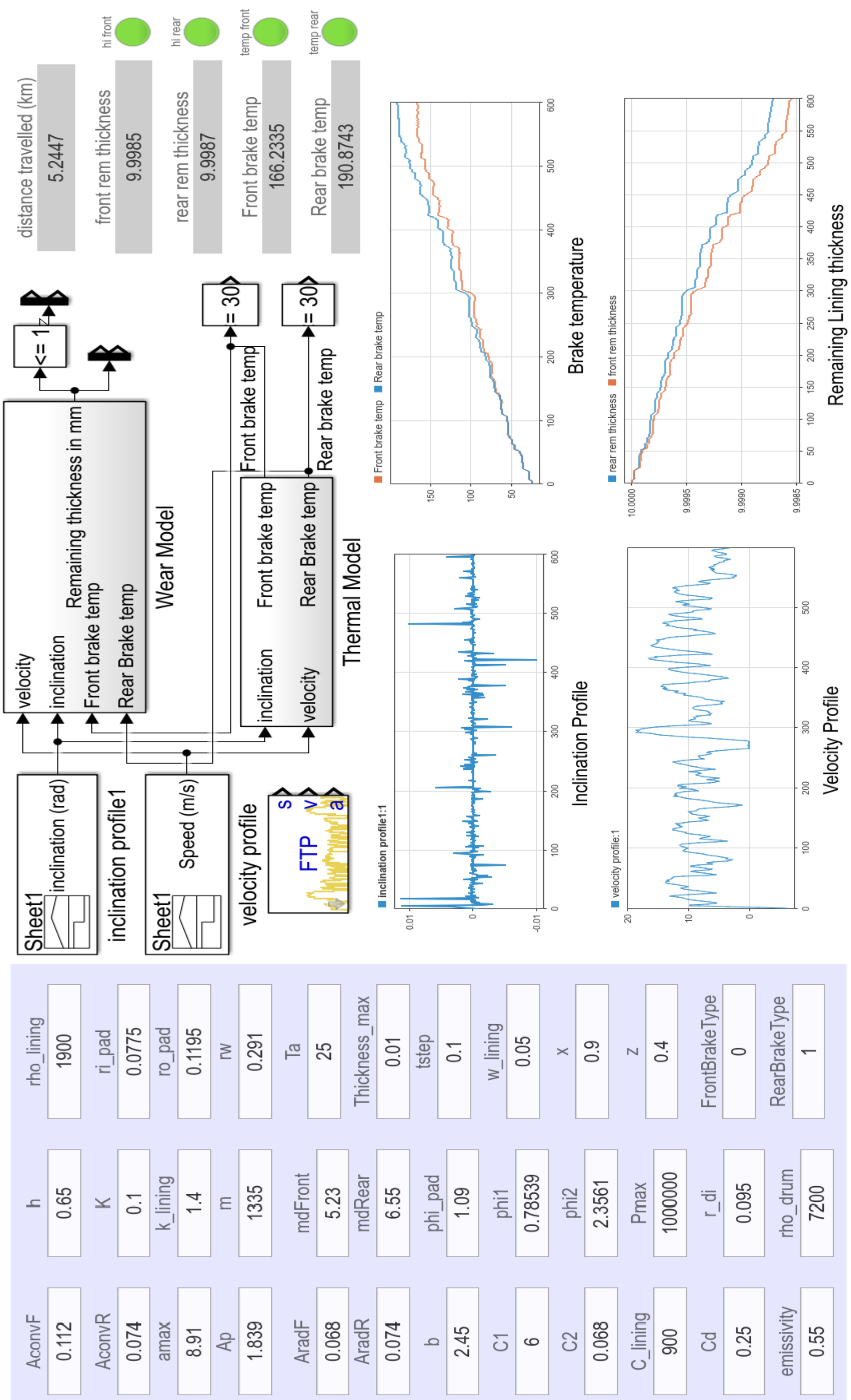


Figure 7. Simulink model

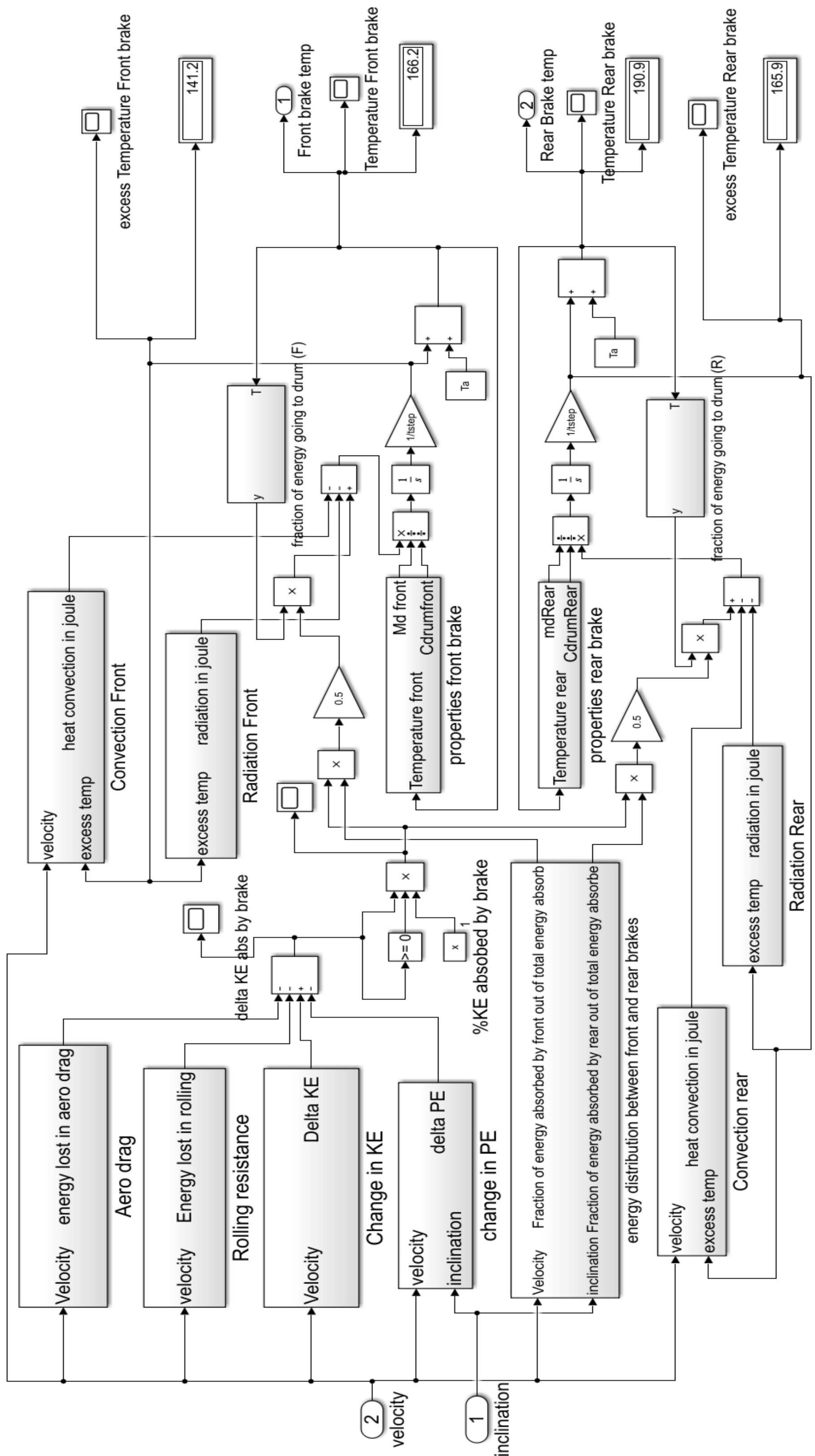


Figure 8. Thermal Model in Simulink

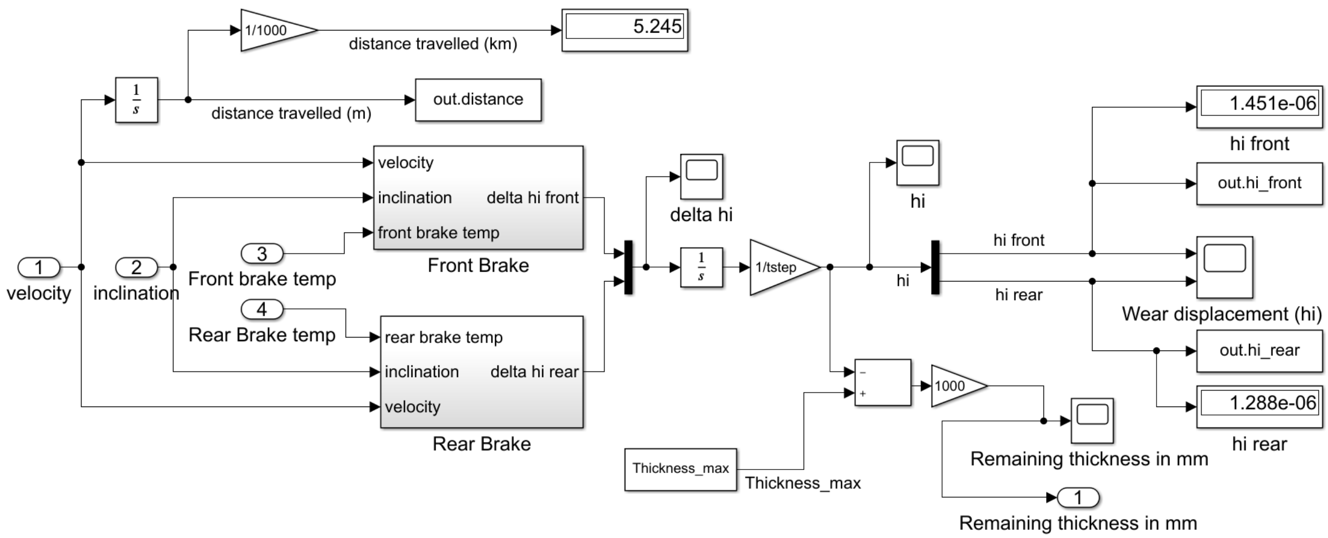


Figure 9. Wear model in Simulink

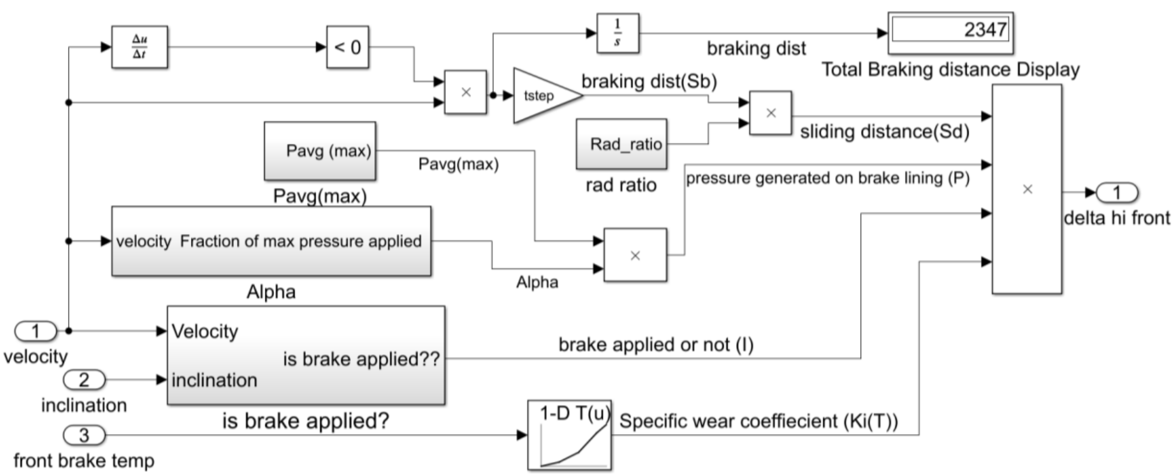


Figure 10. Front brake subsystem from wear model

Figure 9 shows the wear part of the Simulink model. As specific wear rate varies in accordance with disc/drum temperature, this model takes temperature of both front & rear disc/drum along with velocity and road inclination as inputs. This model calculates increase in wear displacement in given instant, then sums it up to that instant, to get the total wear displacement up to that instant.

Figure 10 is expanded view of front brake subsystem from wear model. Similar arrangement is used for rear brake subsystem also.

4. CASE STUDY

Model has to be tested for different velocity profiles and terrains to check its feasibility. Many standard velocity profiles like FTP, HWFET, ArtUrban, ArtRoad etc. were fed to the model along with different road inclination profiles to study and compare wear and temperature behavior of brakes. Actual readings of velocity and road inclination profiles were acquired by using Phyphox mobile application while driving Maruti Suzuki Swift Dzire (LDi, 4 cylinder manual transmission) vehicle. These obtained readings were also fed to the model. All the inputs to the model for this case study are given in Table 8 (Annexure A) & Figures 11-16 (Belhocine & Wan Omar, 2017; Öztürk et al., 2013; Yan & Jinliang, 2018). These values are either obtained from vehicle manual or calculated using software CATIA. The parameters stated in Table 8 (Annexure A) are considered for a specific vehicle and a specific journey. But these parameters can be changed as per vehicle and readings for velocity & road inclination can also be captured for any other journey, and results for corresponding vehicle and journey can be found out, proving this model a generalized one.

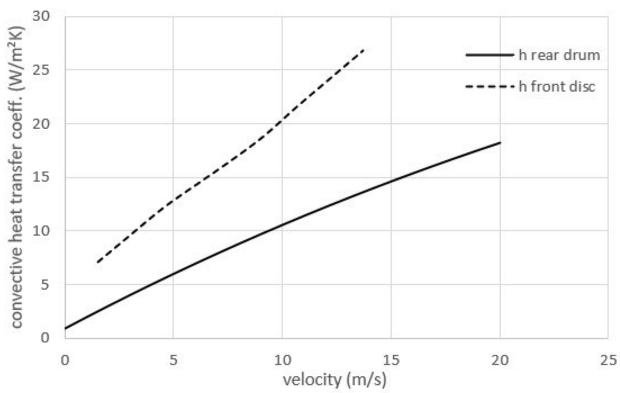


Figure 11. convection heat transfer coefficient

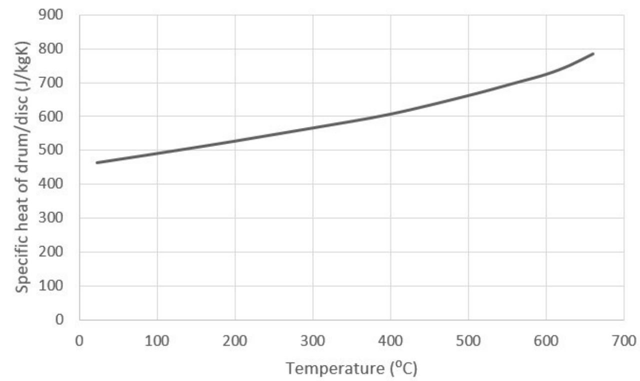


Figure 12. Specific heat of disc/drum

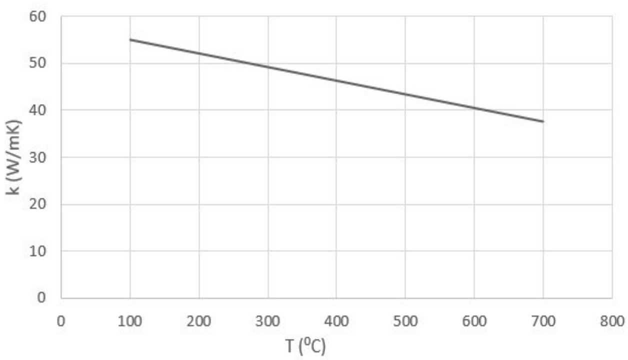


Figure 13. Thermal conductivity of drum/disc

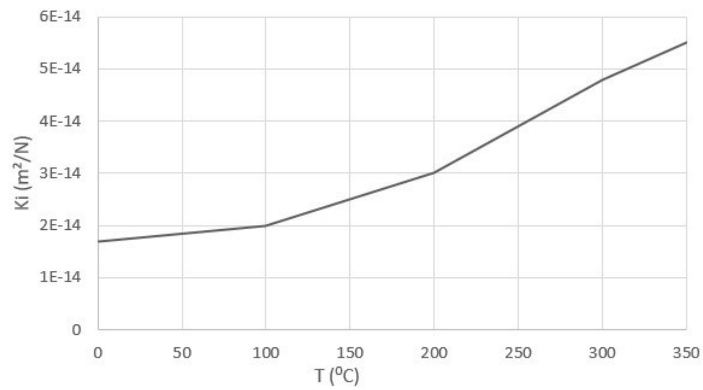


Figure 14. Specific wear rate coefficient

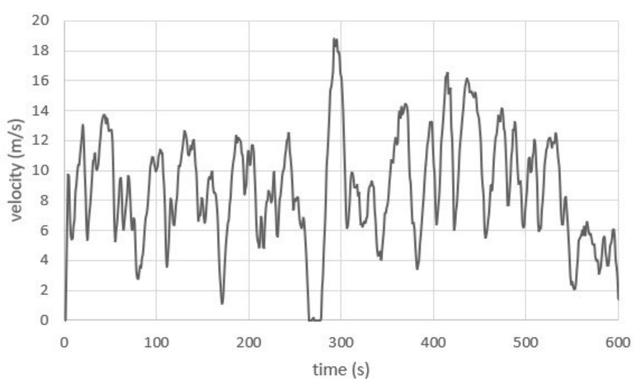


Figure 15. Velocity profile for the given case study

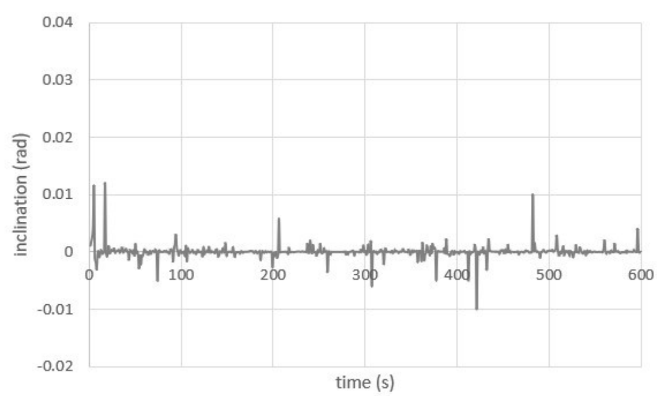


Figure 16. Road inclination profile for the given case study

5. RESULTS & DISCUSSIONS

5.1. Results for given case study

Remaining lining/pad thickness for front and rear brakes is shown in Figure 17. Temperature profiles for front disc and rear drum are shown in Figure 18. Figure 19 shows linear fitting for front pad wear displacement plotted against distance travelled. The slope of the linear fit is 3×10^{-10} which is equivalent to 3×10^{-4} mm/km, which indicates that on an average 3×10^{-4} mm of lining thickness got worn out per km of travelled distance.

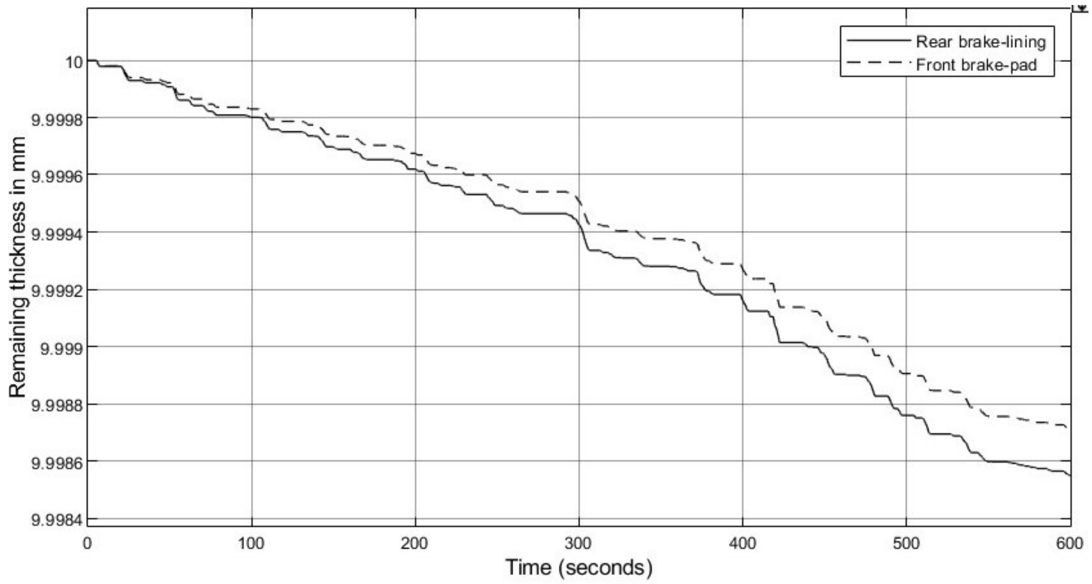


Figure 17. Remaining lining/pad thickness

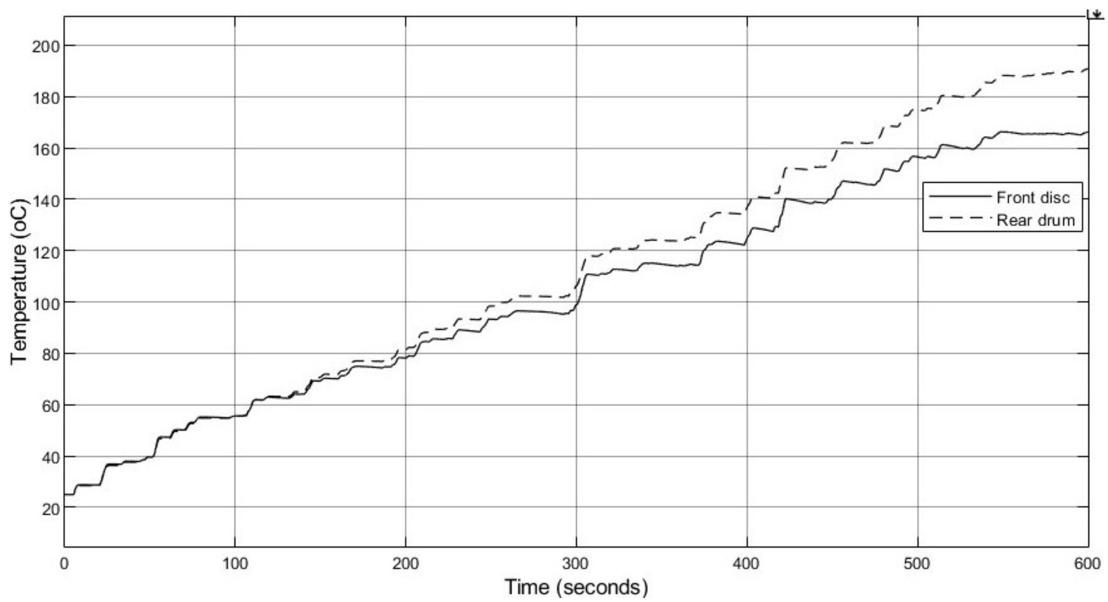


Figure 18. Temperature of drum & disc

5.2. Comparison of various driving behaviors

Model is experimented with various standard velocity profiles (Jafari, Gauchia, Zhang & Gauchia, 2015; Shabbir & Evangelou, 2014) and the results are obtained for those profiles in similar way as shown in Figure 19.

From Table 1, it can be concluded that the velocity profile- HWFET indicated the best driving behavior and the velocity profile- ART URBAN indicated the roughest/worst driving behavior for similar road conditions. In this way, the results obtained from the model can be used to decipher the driver performance.

Name of standard velocity profile	Slope (m/m)
ART MW 130	8.41043×10^{-11}
ART ROAD	1.52672×10^{-10}
FTP	1.40845×10^{-10}
HWFET	3.89105×10^{-11}
ART URBAN	3.20513×10^{-10}

Table 1. Comparison of various standard velocity profiles for lining/pad wear

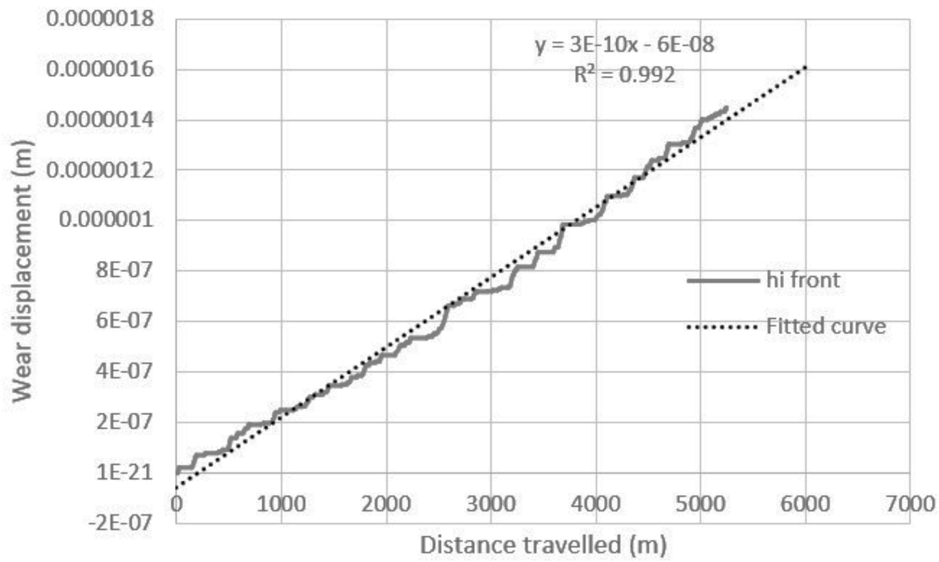


Figure 19. Linear fitting for front pad wear

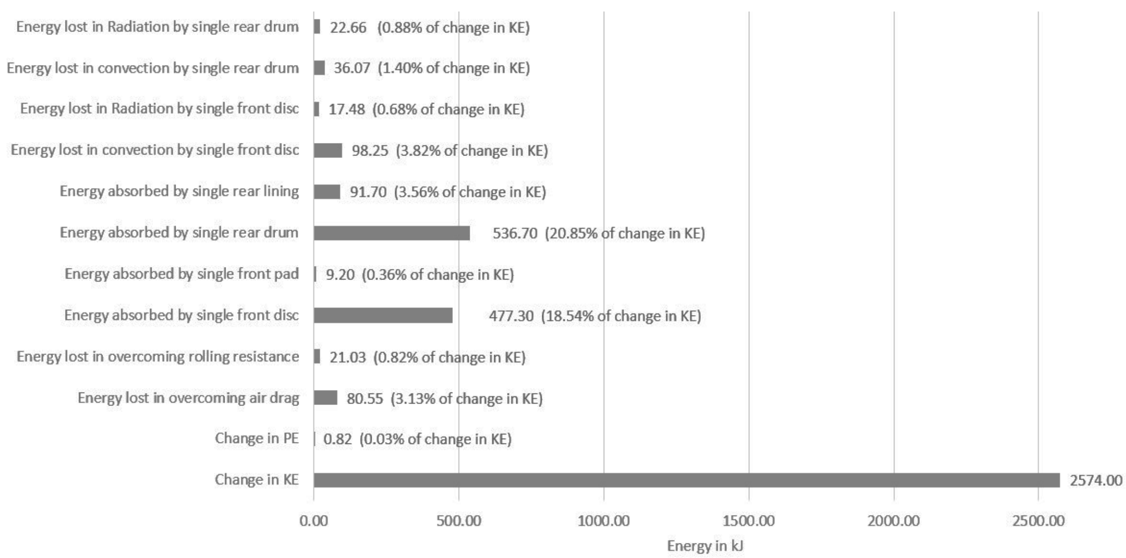


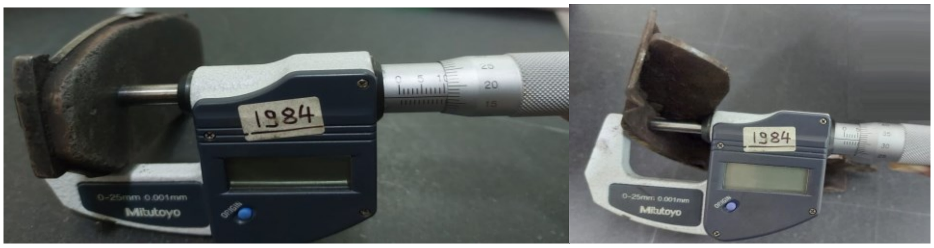
Figure 20. Energy distribution in the given case study

5.3. Effect of ascent and descent on temperature of front and rear brakes for same velocity profile

Model was fed with same velocity profile (as in case study) but different road profiles and results are compared in Table 2. From Table 2, we can conclude that vehicle travelling on a descent generally is prone to brake-failure, due to excessive increase of temperature of drum/disc, as compared to a car which is travelling on flat road or an ascent.

Factors of comparison	Flat road	Ascent (+7°)	Descent (-7°)	Case study
Front brake temperature (°C)	166	53	364	166
Rear brake temperature (°C)	190	60	386	190

Table 2. Effect of ascent and descent on temperature of drum/disc



(a)

(b)

(c)

Figure 21. Experimentation for wear and thermal model (a) measurement of wear (b) measurement of drum temperature (c) measurement of disc temperature

5.4. Energy distribution in the given case study

The objective of brakes is to reduce the velocity and in turn reduce the kinetic energy of the vehicle. Some part of this kinetic energy to be reduced is absorbed by resistive forces like aerodynamic drag, sliding friction at road tire interface and rolling resistance.

Remaining part of this kinetic energy is absorbed by drum/disc and lining/pad causing an increase in their temperatures. This excess temperature over environment causes heat loss via convection and radiation. Figure 20 shows such energy distribution for the given case study obtained from the proposed model.

Figure 20 is representative figure for the given case study only, but such energy distribution can be obtained for any journey & vehicle. It is useful for designers & research scholars to analyze and optimize brake system parameters.

6. EXPERIMENTAL VALIDATION

The experimentation was performed for validating the results obtained from Simulink study. At the end of each journey, wear and temperature were measured. During the experimentation, it was observed that, the wear is insignificant for smaller journey with less distance travelled. However on the other hand the temperature changes significantly even for smaller journey. Thus, total seven journeys were considered. The results are presented in subsequent sections.

6.1. Wear model

Three journeys (not necessarily continuous) were considered and at the end of the journey brake pad lining thickness from both disc & drum brakes has been measured. The observations were recorded with the help of micrometer screw gauge as shown in Figure 21(a). The experimental & analytical values are compared in Table 3 to 5.

Factors of comparison	Type of brake	Value (mm)
Initial thickness	Disc	5.504
	Drum	4.000

Table 3. Initial conditions

Journey No.	1	2	3
Distance travelled (Km)	53	29.7	39.4
Analytical (mm)	5.4978	5.4926	5.4887
Experimental (mm)	5.479	5.471	5.468
Error (%)	0.34	0.39	0.37
Average error (%)	**0.36**		

Table 4. Wear for Disc brake

Journey No.	1	2	3
Distance travelled (Km)	53	29.7	39.4
Analytical (mm)	3.9936	3.9886	3.9848
Experimental (mm)	3.979	3.975	3.969
Error (%)	0.37	0.34	0.40
Average error (%)	**0.37**		

Table 5. Wear for drum brake

6.2. Thermal model

Four separate journeys were considered, each starting from ambient temperature of brake pad. Temperature of brake is measured using STANLEY STHT0-77365 High Accuracy Industrial Digital Infrared Thermometer with -38°C to 520°C temperature range. Figure 21 depicts experimental setup for thermal model (b) measurement of drum temperature (c) measurement of disc temperature.

Journey No.	**1**	**2**	**3**	**4**
Distance travelled (Km)	14.5	7.7	0.9	5
Analytical (°C)	156.5	162.5	32.2	64
Experimental (°C)	149.7	155.1	30.3	60.5
Error (%)	4.5	4.79	6.27	5.83
Average error (%)	**5.43**			

Table 6. Temperature of disc

Journey No.	**1**	**2**	**3**	**4**
Distance travelled (Km)	14.5	7.7	0.9	5
Analytical (°C)	198.2	164.9	30.7	74.4
Experimental (°C)	189.4	156.3	29.2	69.8
Error (%)	4.66	5.5	5.17	6.59
Average error (%)	**5.48**			

Table 7. Temperature of drum

For wear model, the experimentation demonstrates that the error between analytical and experimental wear for disc brake is 0.36% and drum brake is 0.37%. For thermal model, the experimentation demonstrates that the error between analytical and experimental temperature for disc is 5.43% and drum is 5.48%. The emissivity of shiny surfaces is low hence accuracy of infrared thermometer gets affected, so the readings were taken on oxidized area of drum/disc having high emissivity. The cooling rate increases after disassembly of drum/disc; thus, readings were taken immediately. The accuracy of industry grade STANLEY STHT0-77365 infrared thermometer used in this experiment is ± 2^0C. This shows that experimental values are in accordance with the analytical values obtained by the model. Thus, deployment of this model would help in on-board estimation of wear and temperature.

7. CONCLUSION

A wear model, as demonstrated, can provide the thickness of lining at each instant without need of complicated assembly of sensors or wiring inside brakes. Drivers could also be classified based upon their wear rates. Indication to the driver about their driving can improve the driver's driving behavior, improving the vehicle's performance and fuel efficiency. This would also be of uttermost importance in the case of rental vehicles where drivers could be charged based upon their driving behavior. A thermal model can give us an early warning about the brake overheating, based upon which appropriate action can be taken. Although, the proposed model could be improved in the areas like considering transmission losses, tyre pressure losses & regenerative braking that are not considered currently. Also, this model does not consider sudden failures & ruptures, hence not applicable for brake systems in which elastomers are used. These improvements could make the model even more realistic. Further, the contribution of major or critical parameters in thermal and wear model can be examined using the parameter importance analysis in life consumption under the settings already experimented in near future. Nevertheless, proposed model is novel as it requires velocity and road inclination profiles as the only input variables, it could be directly implemented on any vehicle for the purpose of health monitoring and prognosis by using suitable hardware like Arduino/raspberry pi. Designers and researchers can also use this model to analyze, design and optimize the brake system.

ANNEXURE A

Parameter	**Abbreviation**	**Value**	
Convection area front brake	AconvF	0.112	m^2
Convection area rear brake	AconvR	0.074	m^2
Max deceleration	amax	8.910	m/s^2
Frontal area	Ap	1.839	m^2
Radiation area front brake	AradF	0.068	m^2
Radiation area rear brake	AradR	0.074	m^2
Wheelbase	b	2.450	m
Tire Constant1	C1	6	-
Tire Constant2	C2	0.068	-
Specific heat of lining	C_lining	900	J/kgK
Drag coefficient	Cd	0.250	-
Emissivity of drum/disc	emissivity	0.550	-
Type of front brake (disc=0)	FrontBrakeType	0	-
CG of vehicle	h	0.650	m
Rot. KE/ transl. KE	K	0.100	-
Thermal conductivity of lining/pad	k_lining	1.40	W/mK
Gross weight of vehicle	m	1335	kg
Front drum/disc mass	mdFront	5.23	kg
Rear drum/disc mass	mdRear	6.55	kg

Angles related to drum brake	phi1	0.7853	rad
	phi2	2.3561	rad
Angle inscribed by pad at center of disc	phi_pad	1.09	rad
Pressure sustaining Capacity of lining/pad	Pmax	10^6	Pa
Inner radius of drum	r_di	0.095	m
Type of front brake (drum=1)	RearBrakeType	1	-
Density of air	Rho	1.225	kg/m^3
Density of drum/disc	rho_drum	7200	kg/m^3
Density of lining/pad	rho_lining	1900	kg/m^3
Inner radius brakepad	ri_pad	0.0775	m
Outer radius brakepad	ro_pad	0.1195	m
Wheel radius	rw	0.291	m
Ambient temperature	Ta	25	ºC
Original lining thickness	Thickness_max	0.01	m
Incremental time	tstep	0.1	sec
Width of lining	w_lining	0.05	m
Fraction of energy absorbed by road tire interface	x	0.9	
Stationary weight distribution factor for front tires	z	0.4	-

Table 8. Inputs for the case study

ANNEXURE B

The equations for N_f & N_r as follows:

$$N_f = 0.5mg\left(z\cos i - \frac{h}{b}\sin i\right) - 0.5ma\frac{h}{b}$$

$$N_r = 0.5mg\left((1-z)\cos i + \frac{h}{b}\sin i\right) + 0.5ma\frac{h}{b}$$

z is weight distribution fraction on front tires when vehicle is stationary. It is the ratio of a_2 to *b*.

a_2 can be seen in figure 5 and *b* is the wheelbase.

$$\xi_d(T) = \sqrt{k_{drum/disc}(T) \times \rho_{drum/disc} \times C_{drum/disc}(T)}$$

$$\xi_l(T) = \sqrt{k_{lining/pad}(T) \times \rho_{lining/pad} \times C_{lining/pad}(T)}$$

$$A_{drum} = 2\pi r_{di} * w_{lining}$$

$$A_{disc} = \pi({r_o}^2 - {r_i}^2)$$

$$A_{lining} = 2 \times \frac{(\emptyset_2 - \emptyset_1)}{2\pi} A_{drum}$$

$$A_{pad} = \frac{\emptyset_{pad}}{2\pi} A_{disc}$$

k is thermal conductivity, *ρ* is density, *C* is specific heat capacity. *A* is frictional area of contact, $\emptyset_{pad}$ is angle inscribed by pad at the centre of wheel in radians.

Link for the Repository containing the

Simulink model for Case study (600 sec)

Simulink model for long run (5700 sec)

https://drive.google.com/drive/folders/1MDzJ8g7lI92xbhBn_UW-xXoilcMRTaYu?usp=sharing

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