

High Temperature Brake Cooling - Characterization for Brake System Modeling in Race Track and High Energy Driving Conditions

2011-01-0566

Published
04/12/2011

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doi:10.4271/2011-01-0566

ABSTRACT

At elevated temperatures, such as those encountered under race track or fade test conditions, the closed-form solution to the lumped capacitance model for characterizing brake cooling (fitted to a standard cooling test temperature range) tends to break down and provide an inaccurate representation of brake rotor cooling behavior. Accurate prediction of cooling is fundamental to brake system component sizing and selection of materials at the early stages of a vehicle program; this is especially true of a high performance vehicle with track performance requirements. To this end, alternative approaches to characterizing brake cooling have been examined to determine their suitability for use in measurement and simulation of brake performance.

In this paper, three methods of representing in-vehicle measured brake cooling behavior are reviewed and explained (standard temperature range closed-form solution lumped capacitance model, high temperature range closed form solution lumped capacitance model, and numerically solved linearly varying specific heat capacity lumped capacitance model). The efficacy of each model when employed in brake system thermal models is examined via case studies, in which modeled rotor temperatures are compared to in-vehicle measured high energy fade test results. The influence of radial temperature gradients developed in the rotor on cooling behavior representation and on model to vehicle test correlation is also considered.

INTRODUCTION

Brake cooling is a primary consideration in the application of a brake system to a vehicle. For most applications, the cooling is a major factor in the selection of brake system components to achieve the lining life and quality goals for the vehicle. In high performance vehicles, where thermal management of the brakes can be the difference between achieving required performance and incurring issues such as severe fade and lining wear, this is of particular interest. Cooling of a brake system is concerned mostly with the cooling of the rotor, as it is the primary storage for thermal energy and its temperature influences the temperature of the nearby components.

Cooling of a brake rotor has traditionally been evaluated and characterized with a simple lumped capacitance model of the rotor following a cooling coefficient representation of the heat transfer out of the rotor by convection to the surrounding air. This model was described by Sheridan, Kutchev and Samie in reference (4). This model of the brake thermal performance has performed well for operating conditions that can be described as 'normal', with relatively modest energy inputs and generally restricted to a narrow temperature range. This model does not function as well when it is pushed beyond its intended limits, or beyond its assumed operating conditions. Several factors influence the decreased function of the model outside its intended operating range; material changes with temperature and heat transfer changes with temperature are notable contributors. The work discussed in this paper centers on modeling the cooling of the brake at elevated temperatures and under the extreme conditions imposed by high performance or track driving.

The theoretical basis for rotor cooling performance is addressed and correlation of the models with measured cooling performance is demonstrated. Some practical issues of rotor temperature measurements and the relationship to the theoretical assumptions are discussed. The influence of rotor / caliper / lining interaction on thermal gradients within the rotor is also explored. Finally, the vehicle level performance of the rotor under a specific fade schedule is examined and compared to measured results.

The paper is divided in the following sections:

1. Representation of Rotor Cooling Data
2. Influence of Temperature Gradients on Embedded Thermocouple Readings
3. Simulation and Correlation of Brake Thermal Performance
4. Conclusions

Throughout this paper, reference will be made to two case study vehicles. Vehicle 'A' is a high performance, race track capable sedan with a 19" wheel envelope brake system. Vehicle 'B' is a race track capable sports car with an 18" wheel envelope brake system. Both vehicles are rated in excess of 500 net horsepower, and both used opposed piston brake calipers and high performance low-metallic brake linings front and rear.

1. REPRESENTATION OF MEASURED ROTOR COOLING DATA

It is typical practice to measure brake rotor cooling behavior over a set temperature range, fit a model of cooling behavior to this range, and then use this model to predict brake cooling behavior over a wide range of operating temperatures. It has been well noted in the literature [1,4] that the cooling behavior of a cast iron brake rotor is well represented in most cases by a 'lumped capacitance' convective heat transfer model. The central assumption of this model is that conduction of heat energy within the brake rotor occurs at a high enough rate relative to convection of heat energy from the rotor into the cooling air flow, that the assumption of a uniform temperature within the rotor during cooling becomes reasonable [5]. Indeed, calculations of the Biot number relative to guidelines in literature show the brake rotor to fit well within the range of this model's validity. For reference, the Biot number is the ratio of the convective heat transfer coefficient and the conductive heat transfer coefficient (divided by the dimension over which the conduction occurs), as shown in [equation \(1\)](#) below:

$$Bi = \frac{hr_0}{k} \quad (1)$$

Where Bi is the Biot number, h is the convective heat transfer coefficient, k is the conductive heat transfer coefficient, and r_0 is the critical dimension over which conduction occurs. For a brake rotor, the highest temperature gradients within the brake rotor material generally occur in the radial direction, so the radial height of the friction surface is considered the critical dimension. For a typical passenger car brake rotor at 50 kph vehicle speed, a Biot number of 0.08 may be calculated (16" wheel envelope size), which compares favorably with the guideline of $Bi < 0.1$ for the lumped capacitance model to be valid. For larger rotor sizes, the Biot number may be above this guideline; for example, the front rotor of vehicle B is as high as 0.24, which means that temperature gradients within the rotor may influence the heat transfer behavior somewhat. This will be re-visited later in this section.

The simple differential equation that represents the lumped capacitance model for a brake rotor is shown as [equation \(2\)](#) below:

$$-\frac{hA}{mc_p}(T - T_\infty) = \frac{dT}{dt} \quad (2)$$

Where h is the convective heat transfer coefficient, A is the working area, m is the working mass of the rotor, c_p is the specific heat capacity of the rotor material, T is the bulk temperature of the rotor, T_∞ is the stabilization temperature of the rotor, and t is time.

However, in order to solve [equation \(1\)](#), another assumption is generally applied - one of a having a constant rotor material specific heat capacity versus the material temperature. When this assumption is applied, the solution emerges as shown in [equation \(3\)](#) below:

$$T(t) = e^{-bt}(T - T_\infty) + T_\infty \quad (3)$$

Where $b = \frac{hA}{mc_p}$ is the "cooling coefficient" and the balance of the terms are as defined previously above.

While extensive use of this model with this assumption has generally resulted in good correlation to measured data [2], when considering operating conditions that involve very high temperatures and significant changes in temperature, some error in the model representing the cooling behavior of measured rotor data becomes evident. Examining the specific heat capacity versus temperature curve for cast iron (figure 1

below), one may note immediately that significant change occurs across the typical operating temperature range of the material:

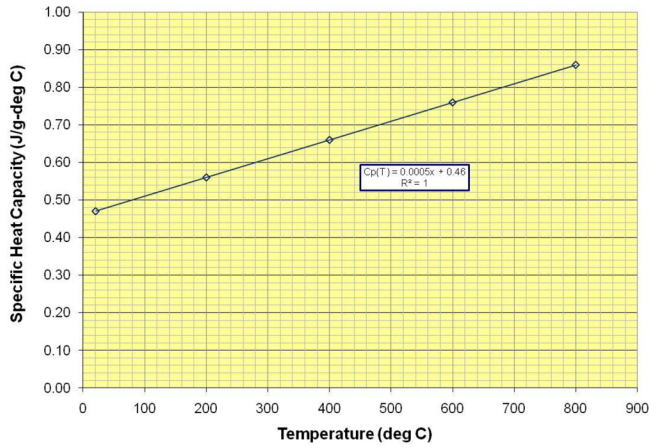


Figure 1. Brake Rotor Cast Iron Specific Heat Capacity versus Temperature

A common process for representing brake rotor cooling behavior for brake system performance and thermal modeling is to fit a regression equation of the form of [equation \(3\)](#) (the solution to the lumped capacitance model representation for heat transfer from the rotor). In so doing, the model is able to accommodate - to some degree - other heat transfer mechanisms, including radiation and conduction into the rotor hub and wheel, and error in the model, including temperature gradients within the rotor, and the changing rotor specific heat capacity with temperature. The rotor cooling model must still conform to the range of curves defined by [equation \(3\)](#), and when the other cooling effects and the sources of error become too great, the model is not able to represent the measured cooling data and error in its use for future brake temperature predictions increases.

A step towards reducing the error in the cooling model's ability to represent measured cooling data can be achieved through solving [equation \(2\)](#) with rotor specific heat capacity varying with temperature. As shown in [figure \(1\)](#), the relationship between rotor material specific heat capacity and rotor temperature is well represented as a linear relationship of the form:

$$c_p = a_1 T + a_2 \quad (4)$$

Where T is temperature, and a1 and a2 are constant coefficients.

With this modification, [equation \(2\)](#) takes on the form of [equation \(5\)](#) below:

$$-\frac{hA}{m(a_1 T + a_2)} (T - T_\infty) = \frac{dT}{dt} \quad (5)$$

Unlike [equation \(2\)](#), this equation does not have a readily accessible closed-form solution. A convenient means of solving [equation \(5\)](#) to get Temperature (T) as a function of time (t) is to use Maclaurin series expansion. The basic form of the Maclaurin series is given by [equation \(6\)](#):

$$F(x) = F(0) + \frac{dF}{dx}(0)x + \frac{\left[\frac{d^2 F}{dx^2}(0)\right]}{2!}x^2 + \frac{\left[\frac{d^3 F}{dx^3}(0)\right]}{3!}x^3 + \frac{\left[\frac{d^4 F}{dx^4}(0)\right]}{4!}x^4 + \dots \quad (6)$$

Starting with [equation \(5\)](#), the following forms of the 1st through 5th derivatives may be found (as with many series expansion representations, the more terms used in the series, the closer the approximation of the intended function):

$$\frac{dT}{dt} = -\frac{hA(T - T_\infty)}{m(a_1 T + a_2)} \quad (7)$$

$$\frac{d^2 T}{dt^2} = \frac{\left[-hA\left(\frac{dT}{dt}\right) - ma_1\left(\frac{dT}{dt}\right)^2\right]}{2ma_1 T} \quad (8)$$

$$\frac{d^3 T}{dt^3} = \frac{\left[-hA\left(\frac{d^2 T}{dt^2}\right) - 3ma_1\left(\frac{dT}{dt}\right)\left(\frac{d^2 T}{dt^2}\right)\right]}{2ma_1 T} \quad (9)$$

$$\frac{d^4 T}{dt^4} = \frac{\left[-hA\left(\frac{d^3 T}{dt^3}\right) - 3ma_1\left(\frac{d^2 T}{dt^2}\right)^2 - 5ma_1\left(\frac{dT}{dt}\right)\left(\frac{d^3 T}{dt^3}\right)\right]}{2ma_1 T} \quad (10)$$

$$\frac{d^5 T}{dt^5} = \frac{\left[-hA\left(\frac{d^4 T}{dt^4}\right) - 7ma_1\left(\frac{dT}{dt}\right)\left(\frac{d^4 T}{dt^4}\right) - 5ma_1\left(\frac{d^2 T}{dt^2}\right)\left(\frac{d^3 T}{dt^3}\right)\right]}{2ma_1 T} \quad (11)$$

The terms in these equations are as defined earlier in this section. Substitution of [equations \(7\), \(8\), \(9\), \(10\), \(11\)](#) into [equation \(6\)](#) gives the 5-term Maclaurin series expansion representation of rotor temperature as a function of time. This results in an equation that may be fitted to the measured data, solving for the value of h (for a given working area, A) or solved for the term hA together to get the best fit.

As noted earlier, it is typical practice to measure brake cooling behavior over a set temperature range, fit a model to this range, and then use this model to represent brake cooling behavior over a wide range of operating temperatures. Unlike regression models that are not physics-based, the physics-based lumped capacitance model may be fit to measured cooling data and then used to represent cooling behavior in temperature ranges outside of the fitted range. However, it follows intuitively that cooling behavior with the range of the initial fit would be represented more accurately by the fitted model than behavior outside of the fit range - due in no small part to the influence of sources of error such as changing rotor material specific heat capacity with temperature. There are some tradeoffs to consider in selecting the temperature range to measure, including the following considerations:

- lower temperatures to start a cooling measurement may be achieved through more gentle heating, invoking less error due to temperature gradients in the rotor
- lower temperatures can represent most customer usage type operating temperatures
- lower temperatures are easier on the friction material and can extend its usefulness for additional testing
- higher temperatures will allow the cooling model, in whatever form it takes, to capture and predict the high temperature behavior more accurately

A typical temperature range to measure rotor cooling behavior for use in brake system modeling is the 400 deg C and below range, which captures most customer usage type operating temperatures. To improve rotor cooling behavior predictions for higher temperature work, the range may be extended; for the purpose of this study the target initial temperature prior to recording cooling behavior was extended to at least 600 deg C.

The success of the several methods of modeling brake rotor cooling behavior from measured data is examined via a case study involving the front brake rotors of two different high performance brake systems. Rotor temperature is represented for these comparisons by two thermocouples, one each embedded in the radial center of each brake rotor friction plate. The quality of the cooling model fit to the data is represented through a Mean Squared Error (MSE) term. The reason for this metric as opposed to a correlation coefficient

is that the relationship between rotor temperature and time is extremely strong and correlation coefficients are generally above 0.99 regardless of the model used, but still allow errors of greater than 20 degrees C in model prediction of measured data.

[Table 1](#) and [figures 2 a, b, c, d](#) below summarizes a comparison of all three methods of representing brake rotor cooling behavior in the case study vehicle A. To calculate equivalent hA values for the cooling coefficient solution to the lumped capacitance model, it was necessary to assume a single rotor material specific heat capacity value, so the value at 400 deg C was selected (0.66 J/gram-deg C).

Table 1. Summary of High Temp Lumped Capacitance Model vs. Linear Cp vs. T Model Performance for Case Study Vehicle A

Speed kph	Slide	400 deg C IBT lumped cap cooling					600 deg C IBT lumped cap cooling					hA (Linear Cp vs T model)				
		Axle Average Cooling Coef. 1/sec	Equiv. hA W-deg C	Max Error deg C	MSE deg C ²		Axle Average Cooling Coef. 1/sec	Equiv. hA W-deg C	Max Error deg C	MSE deg C ²		Axle Avg hA W-deg C	Max Error deg C	MSE deg C ²		
50kph	LF	0.002871	19.7	11.57	22.99	0.003007	0.002974	20.6	11.19	68.70	22.8	22.7	12.04	27.18		
	RF	0.002943	20.2	11.13	65.93	0.002940	0.002974	20.2	11.18	63.85	22.5		14.33	43.21		
110kph	LF	0.005058	34.7	46.07	1,547.87	0.005014	0.005045	34.4	44.07	1,415.84	32.2	32.5	6.44	8.84		
	RF	0.005364	36.8	58.32	2,385.90	0.005076	0.005045	34.8	46.00	1,428.88	32.8		3.58	2.56		
160kph	LF	--	--	--	--	0.006290	0.006430	43.2	39.18	813.85	42.3	43.2	3.92	3.61		
	RF	--	--	--	--	0.006569	0.006430	45.1	38.16	797.51	44.2		4.79	5.88		

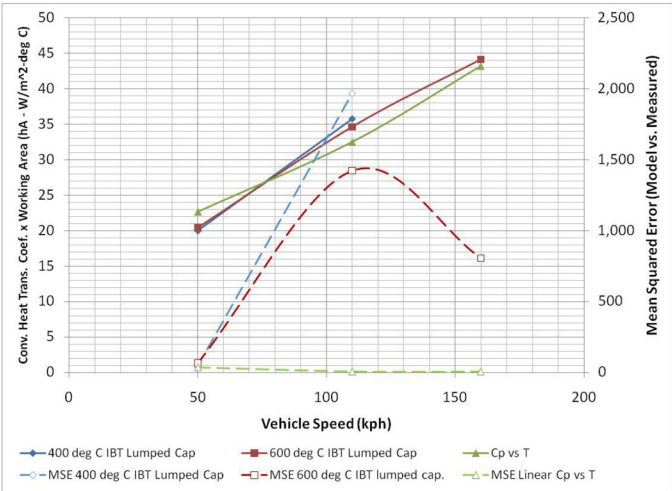


Figure Set 2a. Summary of Standard and High Temp Lumped Capacitance Model vs. Linear Cp vs. T Model Performance for Case Study Vehicle A (Specific Heat Capacity at 400 deg C value for Lumped Capacitance Models)

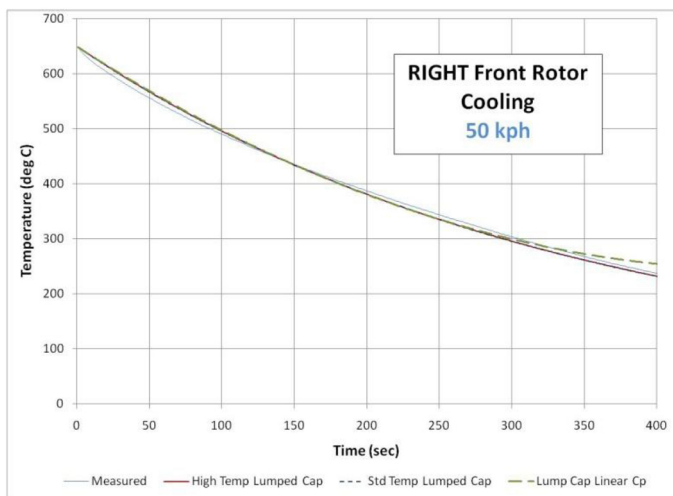
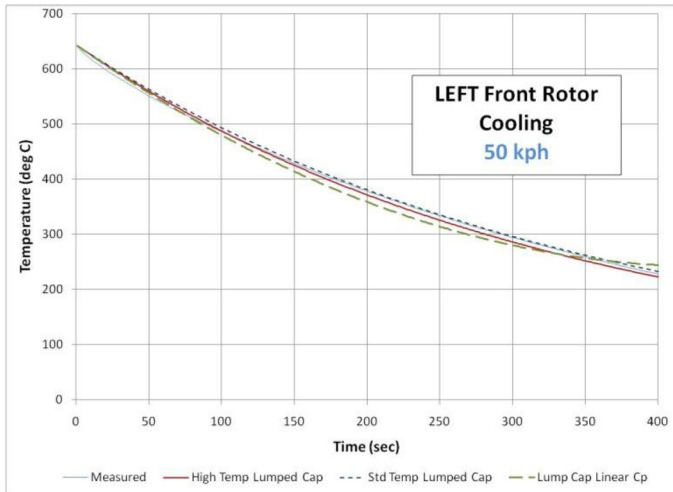


Figure Set 2b. Plots of Measured and Fitted Rotor Cooling Data for Case Study Vehicle A (Left Front Rotor at 50 kph on LEFT; Right Front Rotor at 50 kph on RIGHT)

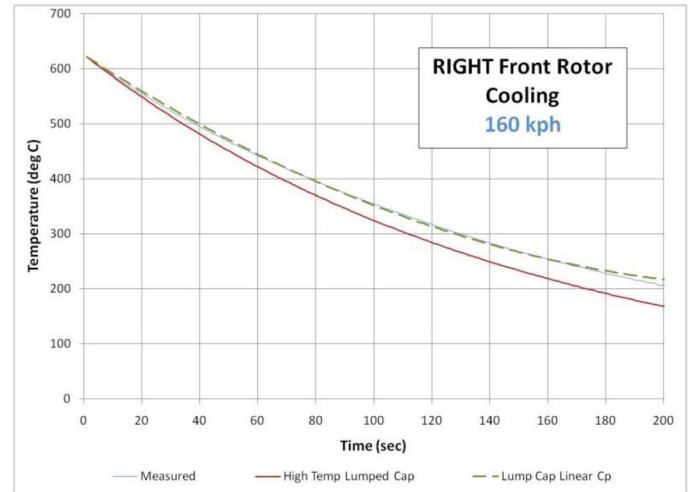
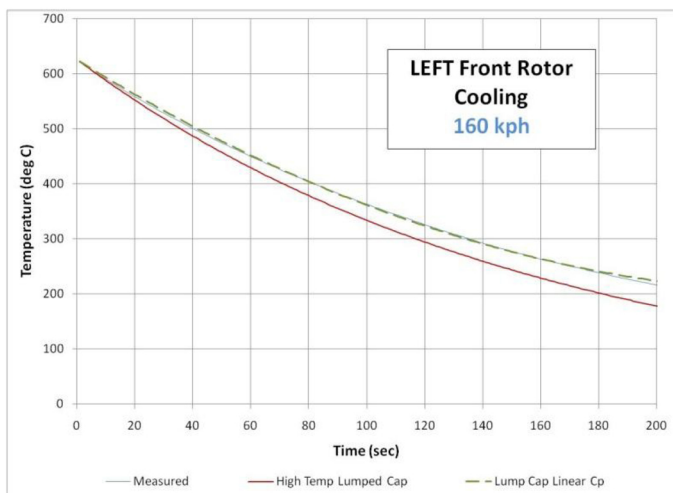


Figure Set 2c. Plots of Measured and Fitted Rotor Cooling Data for Case Study Vehicle A (Left Front Rotor at 160 kph on LEFT; Right Front Rotor at 160 kph on RIGHT)

At 50 kph vehicle speed (for vehicle A) the error in all of the models were very similar and the fits were generally quite good. It can be seen that the high temperature fitted (600 deg C initial brake temperature) cooling coefficient model improved the fit slightly at 110 kph relative to the lower temperature fitted cooling coefficient model. At 110 and 160 kph, the linear specific heat capacity versus temperature model fitted with the 5-term Maclaurin series expansion resulted in improved fits and reduced model error (maximum temperature error of around 5 deg C at 160 kph versus around 40 deg C for the high temperature fitted cooling coefficient model).

As noted earlier in this section, the Biot number for large, performance car rotors can come in somewhat higher than the typical guideline of $Bi < 0.1$ for good representation by the lumped capacitance model. Vehicle A, which had a Biot number of around 0.24 for the front rotors, was equipped with three thermocouples on the outboard cheek of the left front rotor for the cooling measurements, one each at 10mm radially inward of the outer edge, the center of the friction surface, and 10mm radially outward of the friction surface inner edge. Examination of the thermocouple signals from these three positions can give a sense for how quickly the temperature gradients subside. [Figures set 2d](#) below show the temperature versus time signals for a low test speed (80 kph) and the highest (160 kph):