EXPERIMENTAL STUDY OF AUTOMOTIVE BRAKE SYSTEM TEMPERATURES

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Modern automotive disk brake systems can generate extremely high temperatures under high but short duration braking loads or under relatively light but continuous braking. One consequence of the high disk and pad temperatures is a gradual heating of the brake hydraulic fluid, which can lead to boiling of absorbed water and loss of braking.

This paper describes an experiment in which fluid and braking system temperatures, pressures, and operating conditions (vehicle speed, braking energy dissipation) were measured in three different classes of operating vehicles during brake application and subsequent brake release. Brake failure was observed and correlated to moisture content of the fluid, severity of brake application, and application time. In addition, a finite element and lumped capacity analysis of the system was conducted. The analytically predicted temperature histories agreed well with the measured temperatures and can be used to estimate the time at which the fluid will boil.

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EXPERIMENTAL STUDY OF AUTOMOTIVE BRAKE SYSTEM TEMPERATURES

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DISCLAIMER

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SUMMARY

Modern automotive disk brake systems can generate extremely high temperatures under high but short duration braking loads or under relatively light but continuous braking. One consequence of the high disk and pad temperatures is a gradual heating of the brake hydraulic fluid, which can lead to boiling of absorbed water and loss of braking. This paper describes an experiment in which fluid and braking system temperatures, pressures, and operating conditions (vehicle speed, braking energy dissipation, etc) were measured in three different classes of operating vehicles during brake application and subsequent brake release. Brake failure was observed and correlated to moisture content of the fluid, severity of brake application, and application time. In addition, a finite element and lumped capacity analysis of the system was conducted. The analytically predicted temperature histories agreed well with the measured temperatures and can be used to estimate the time at which the fluid will boil.

boiling point. New fluid is generally moisture free, but moisture is thought to diffuse over time through the flexible hoses or to enter when the system is opened, and the moisture content increases approximately 1 to 2 percent (Halderman 1996) per year. Given the data in Table 1, a minimum acceptable boiling point is 275°F. A study that surveyed a number of vehicles showed that approximately 25 percent of the vehicles examined had fluid that boiled at less than 275°F and thus were at risk for brake failure (NHTSA 1973). Figure 1 shows how sensitive the boiling point is to small amounts of moisture.

Table 1
Boiling Points

Fluid	DOT 3	DOT 4	DOT 5
Dry	401	446	500
Wet (3.7% moisture)	284	311	356

(Federal Motor Vehicle Safety Standards 1991)

Although many drivers may never experience brake failure due to boiling, the possibility of brake overheating is always present, particularly in mountainous terrain or with overloaded vehicles (Burgess 1974, Wollenweber 1993). Although it is uneconomical to design standard braking systems for unusual situations, it is worthwhile for vehicle operators and fleet schedulers to be able to estimate the maximum component temperatures to avoid operating failure or damage to their brake systems. One approach is to develop a simulation method that can predict brake fluid temperatures as a function of braking energy dissipation rates and vehicle operation. Unfortunately, brake systems are too complex to make such predictions on the basis of theory only, and some empiricism must be employed. This report describes a test program that measured the time histories of the temperatures of brake components in three typical vehicles. These experimental data were then used to test a thermal model of the braking system to validate its use in predicting brake fluid boiling.

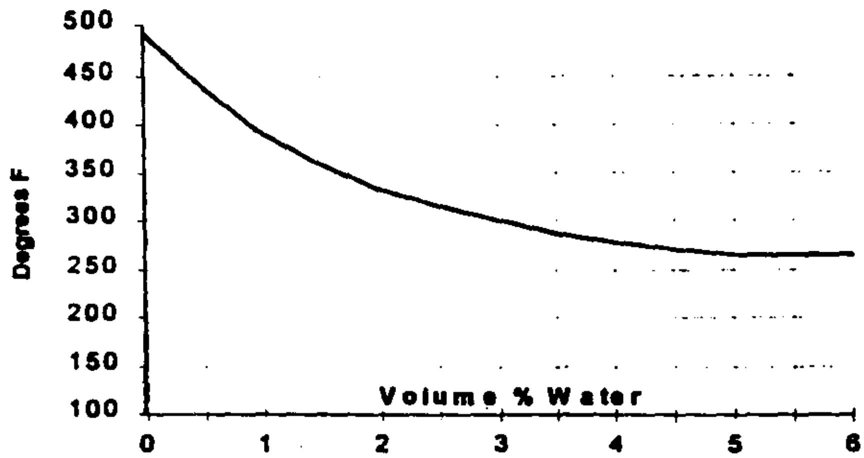

Figure 1 Effect of Moisture on the Boiling Point of DOT3 Fluid

EXPERIMENTAL TESTS

Three representative vehicles were instrumented and tested. Details of the vehicles are given in Table 2. Full details of the instrumentation and the testing procedure are described in a report by Kumar (1997). The vehicles were towed at a constant speed of 30 mph with their brakes applied sufficiently to produce a force equivalent to braking on a downgrade of approximately 5 percent. Briefly, the test procedure was as follows.

1. The vehicle was weighed and then driven to observe the behavior of the brakes. During this drive, the maximum deceleration was measured with a G-analyst. (G-analyst readings of the maximum vehicle deceleration were taken to confirm that the braking system functioned consistently before and after the test. Good agreement was assumed to mean that unacceptable pad or rotor degradation had not occurred.) The instrumentation was then installed, ambient conditions were recorded, the brake fluid was replaced with a fluid containing 5 percent moisture, and the brake boost system was disabled. Brake fluid boiling temperatures were measured at the master cylinder and at each of the wheel cylinders with a device specifically designed for these measurements.[1]
2. The vehicle engine was started to provide fairly normal air flow within the engine and suspension space. Then the brakes were released.
3. The vehicle was then towed in a straight line until it reached a speed of 30 mph.
4. The brakes were then applied by actuating a ram (an air pressurized piston/cylinder), which pressed against the brake pedal with a constant force, and the vehicle was towed in a straight line for approximately 2 miles.
5. A U-turn was made and then the vehicle was stopped. The brakes were left applied for 2 minutes and then released for 1 minute. During this time, tire pressures were

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[1]The Tech+Plus, Mark 2 tester manufactured by ALBA Tools Ltd of England.

measured and rotor temperatures were measured with a hand-held infrared sensor to compare them with the IR sensors mounted on the brakes.

6. Steps 1 through 5 were repeated until brake failure, as indicated by a loss of brake pressure in one or more of the wheel cylinders, was observed.

7. After brake failure, the vehicle was allowed to rest and the brakes to cool. Intermittently, the brake pedal was depressed (either by the air cylinder or by foot) and the hydraulic pressures were recorded. This was continued until the hydraulic pressure recovered, indicating that the vapor had condensed and the brakes were again functioning.

8. The power boost was reactivated and the brake performance was tested with the G-analyst.

9. Steps 1 through 8 were then repeated using a different brake fluid, first with 3 percent moisture and then with 0 percent moisture. Each time the fluid was changed, the entire system was flushed with about 5 times its fluid capacity, and the boiling point was measured at each wheel cylinder. Flushing continued until the fluid boiling points matched that of the unused fluid.

Table 2
Vehicle Specifications

Vehicle		Truck	Car	Van
Weight	Total	6020 lb	2812	3900
	Front	2880	1743	2145
	Rear	3140	1069	1755
Hydraulic System Type		Front/Rear	Diagonal	Front/Rear
Brakes	Front	Disk	Disk	Disk
	Rotor Weight	18.15 lb	10.4	15.36
	Calipers Weight	9.90	7.9	6.38
	Piston Material	Plastic	Plastic	Steel
	Piston Number	1	1	2
Model	Piston/Cylinder Wt	2.16		0.86
Brakes	Rear	Drum	Drum	Drum

The instrumentation consisted of the following:

- strain gauge pressure transducers mounted at the master cylinder, each wheel cylinder, and on the air piston/cylinder ram
- a strain gauge load cell to measure the towing force
- thermocouples inserted through the bleed screws into each wheel cylinder as close to the back face of the piston as possible. Although the SAE (Standard J291 1992) specifies that the measurement be taken at the tip of the bleed screw, these thermocouples were inserted as far as possible into the fluid chamber to measure the fluid temperature at the back of the piston
- thermocouples inserted into each brake pad or shoe according to SAE specifications (Standard J212 1992)
- a speed measuring device
- IR sensors to measure the rotor or drum temperatures and a hand-held IR sensor to measure the temperatures of the visible portions of the brakes after the vehicle had stopped
- anemometers to record the speed of air flowing over the brake components.

The raw data were read at a rate of 20 per second, recorded on hard disk, and displayed on a computer screen during the test to monitor the test.

TEST RESULTS

Figure 2a shows the results for the truck with 5 percent fluid[2]; these results are typical of all the tests. The figure shows the towing force, the speed, the front and rear wheel cylinder hydraulic pressures (both left and right pressures are shown, but they were almost always equal), and the pressure applied to the ram to indicate when the brakes were applied. The time history shows the following.

1. The vehicle quickly reached a towed speed of 30 mph and when the ram was actuated the towing force jumped up to about 300 lbf and then settled down to approximately 200 lbf.
2. At the point labeled H, the truck reached the end of the straight section and was quickly towed through a U-turn with a radius of approximately 75 ft; it then traversed the return straight section and came to a halt. After the vehicle stopped, the conduction of heat into the components increased the fluid temperature and thus the pressure in both front and rear brakes, point A. The brakes were then released for 1 minute.
3. The towing resumed, the brakes were reapplied, and the truck was towed down the straight section, made another U-turn, and stopped, points I to J'. The brake application continued for 2 minutes after stopping and then the brakes were released for 1 minute. Again, heating increased the brake pressure after the truck had stopped.
4. Another straight section was traversed, points K to L'. This time the truck was stopped for 10 minutes.

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[2]The different fluids will be termed 5 percent, 3 percent, and 0 percent fluids, referring to the moisture content. The fluids were supplied by the Union Carbide Corporation.

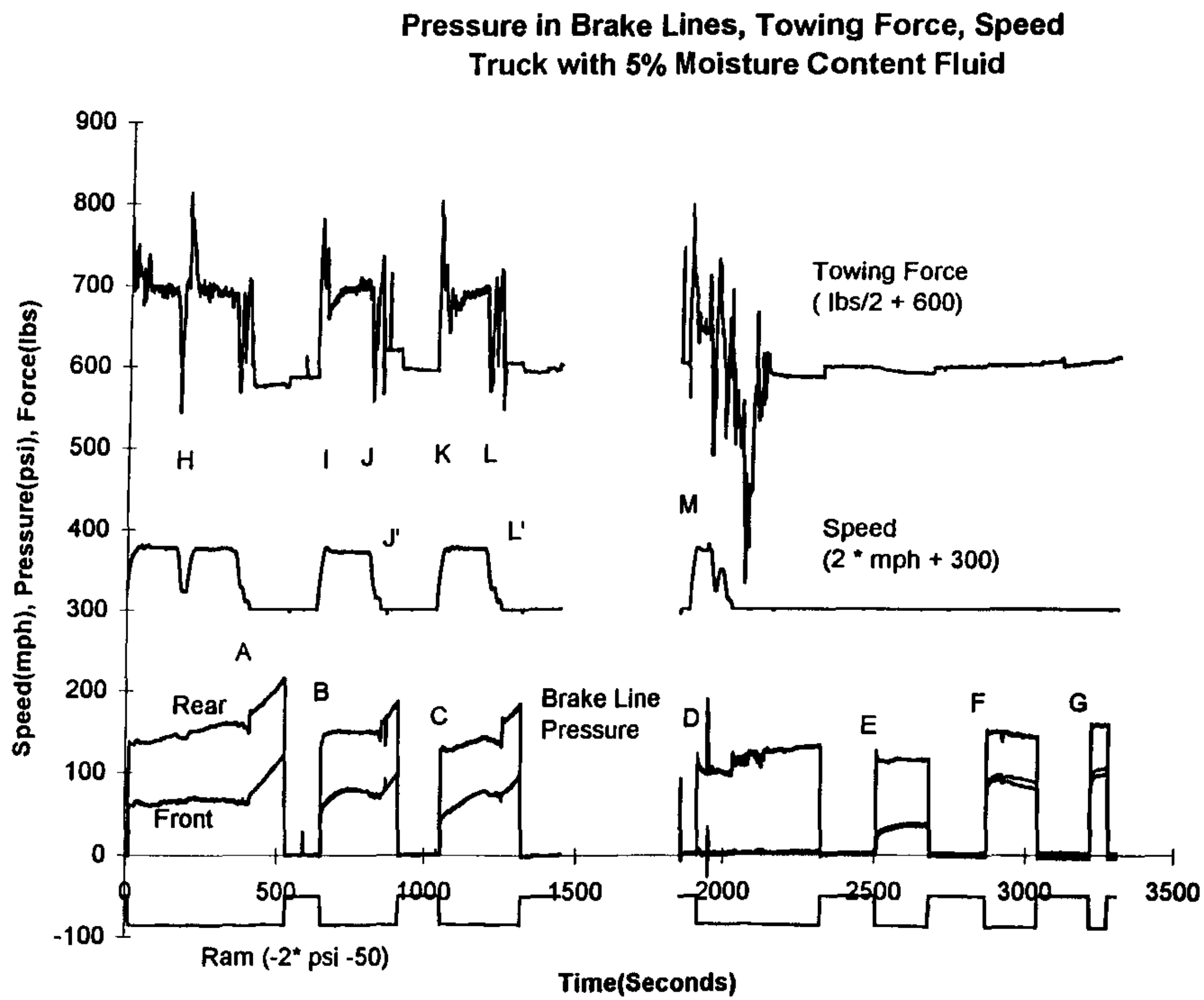

Figure 2a Experimental Pressures, Forces, Speed and Temperatures for the Truck with 0% Fluid

5. Upon towing through another straight section, application of the brakes produced no pressure in the front brakes, point D.

6. Towing then ceased, but the brakes were reapplied at points E, F, and G to determine if the vapor had condensed and the pressure had recovered. As the fluid cooled, the pressure in the front gradually recovered until, at G, the brakes functioned normally.

Figure 2b shows the temperature history of the front brake components. As expected, each time the vehicle was towed with the brakes applied, the pad and rotor temperatures showed a marked increase. When the towing stopped, the pad temperatures decreased precipitously, points L to L', as the rotor cooled rapidly. At point L', the brakes were released, and although the pad was no longer in contact with the rotor, the rate of temperature decrease diminished only slightly because the pad was still very close to the rotor and the high rate of heat transfer to the rotor continued. On the last towing, the failure of the front brakes resulted in basically no braking force on the front wheels, and thus the pad temperatures continued to fall, point D. Even though the pad temperature dropped when towing ceased, the continued conduction of heat into the wheel cylinder caused the fluid temperatures to rise. The onset of boiling was easily seen at about 1750 seconds, when the temperature rise was arrested and the fluid temperatures reached a maximum. The left front (LF) fluid temperature then slowly dropped until about 2500 seconds, when the vapor began to condense and the temperatures remained essentially constant.

Figure 2c shows the history of the rear brakes. These brakes were characterized by gentler rises and falls and a lack of boiling. Although the shoe temperatures were high, the low conductive path from the shoe to the wheel cylinder effectively insulated the fluid from the high temperatures.

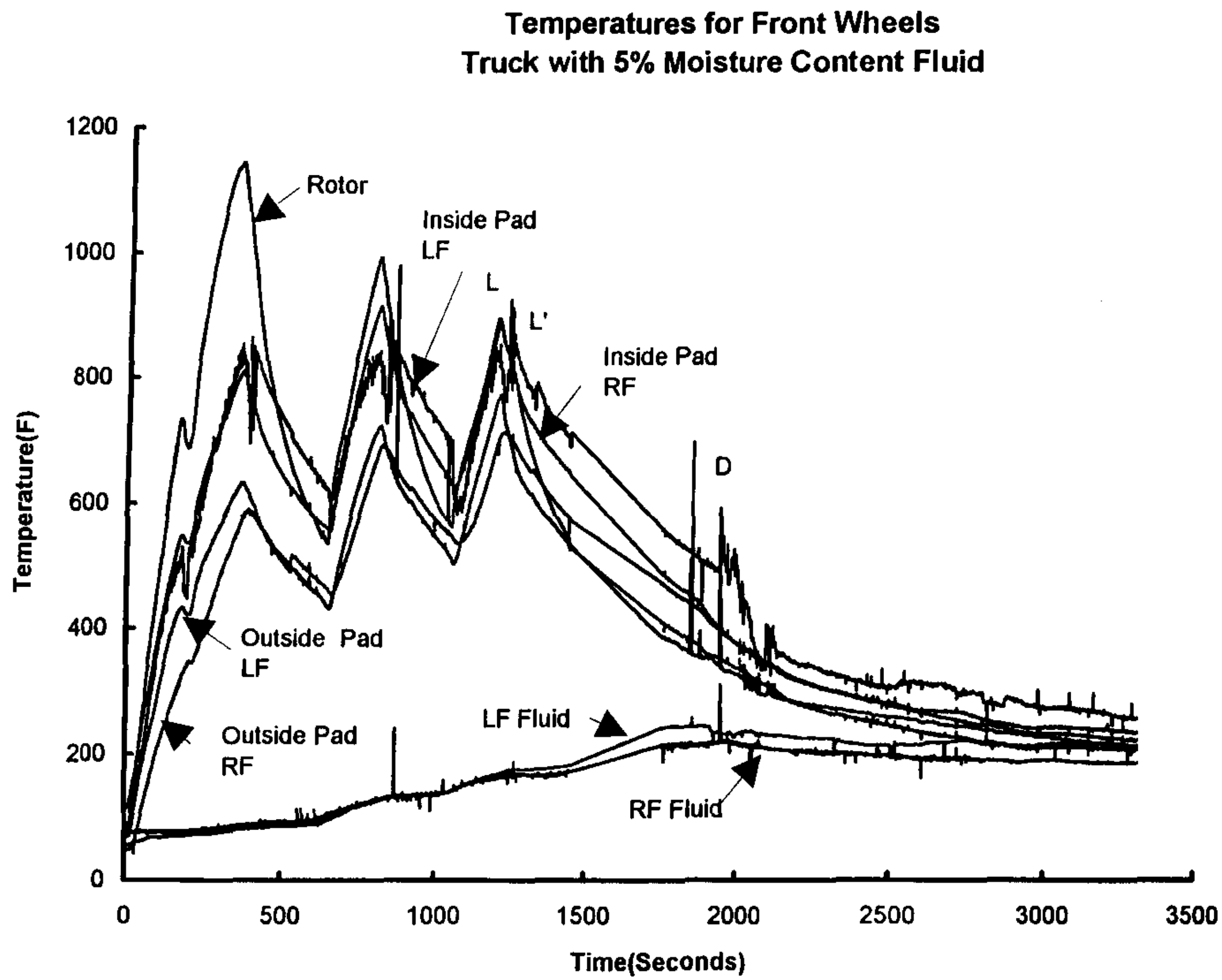

Figure 2b Front Wheel Temperatures for the Truck with 0% Fluid

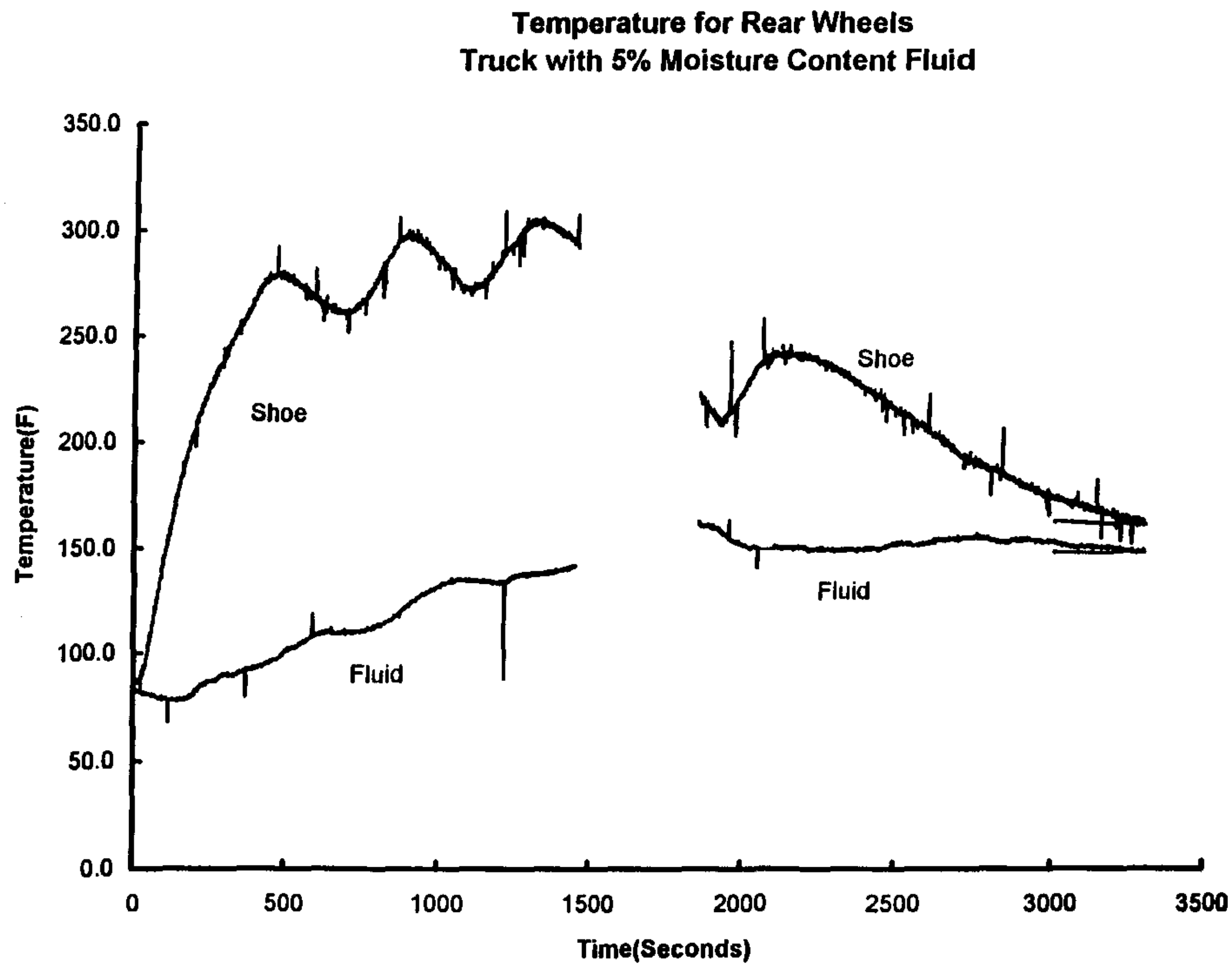

Figure 2c Rear Wheel Temperatures for the Truck with 0% Fluid

Figure 3a is for a car with a diagonal braking system in which the left front and right rear are on the same system. In this case, boiling occurred at about 1700 seconds in the right front system and then in the left front system at about 2200 seconds. The left rear-right front pressure recovered at about 3400 seconds, and both systems had recovered at about 3700 seconds. The failure of the right front system is clearly shown in Figure 3b at 1700 seconds when the application of the brakes during the towing period did not generate a braking force at the right front, and the pad temperatures continued to decrease as the ambient air cooled the brakes. During the next period of towing, because boiling caused the failure of all the brakes, all the rotors and pads continued to cool. Even in the absence of heat production at the brakes, and thus a consequent continual cooling of the rotor and pad, the conduction of heat from the hot pad and rotor to the cylinder caused a continual increase in fluid temperature, with the result that both fluid temperatures experienced almost identical histories.

Table 3 lists the boiling points of the different fluids and the measured temperatures at which boiling was evident. The boiling points were measured with a commercial tester before and after each test. In the front brakes, the boiling points of the 5 percent fluid showed no change, but the 3 percent and particularly the 0 percent fluid showed substantial decreases in boiling point with use. We suspect that the high temperatures that these latter two fluids were exposed to caused irreversible chemical changes. The rear brake fluids, which never experienced high temperatures, showed little change. It is interesting to note the difference between the peak measured temperatures during the tests and the before and after boiling points. Good agreement was found for the 5 percent fluid, but the measured peak temperatures were low by about 60 to 80 degrees for the 3 and 0 percent fluid. There were substantial temperature gradients across the components and through the fluid, so it is not surprising that the experimental temperatures, being measured slightly behind the back face of the piston, were lower than the measured boiling points.

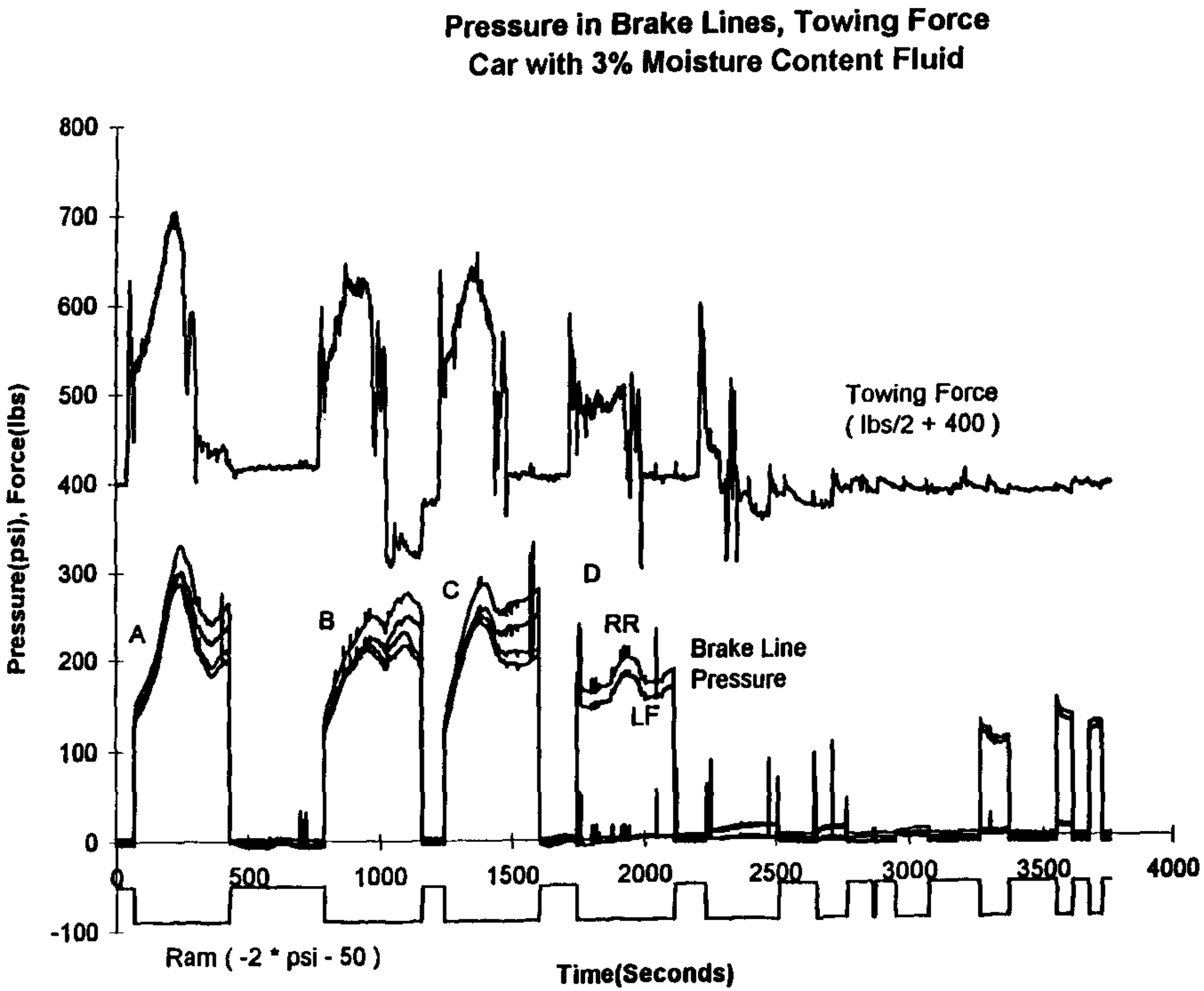

Figure 3a Experimental Pressures, Forces, Speed and Temperatures for the Car with 3% Fluid

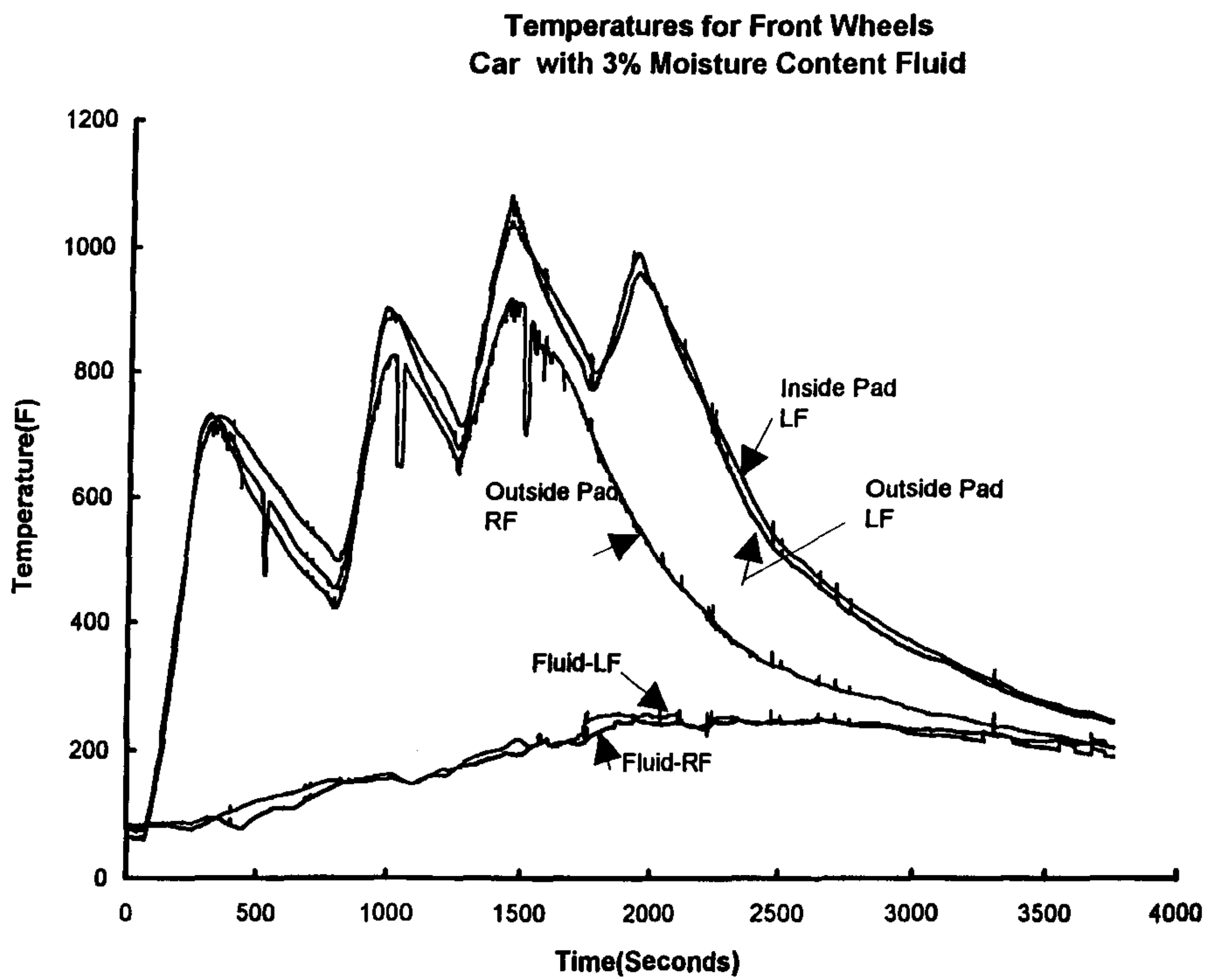

Figure 3b Front Wheel Temperatures for the Car with 3% Fluid

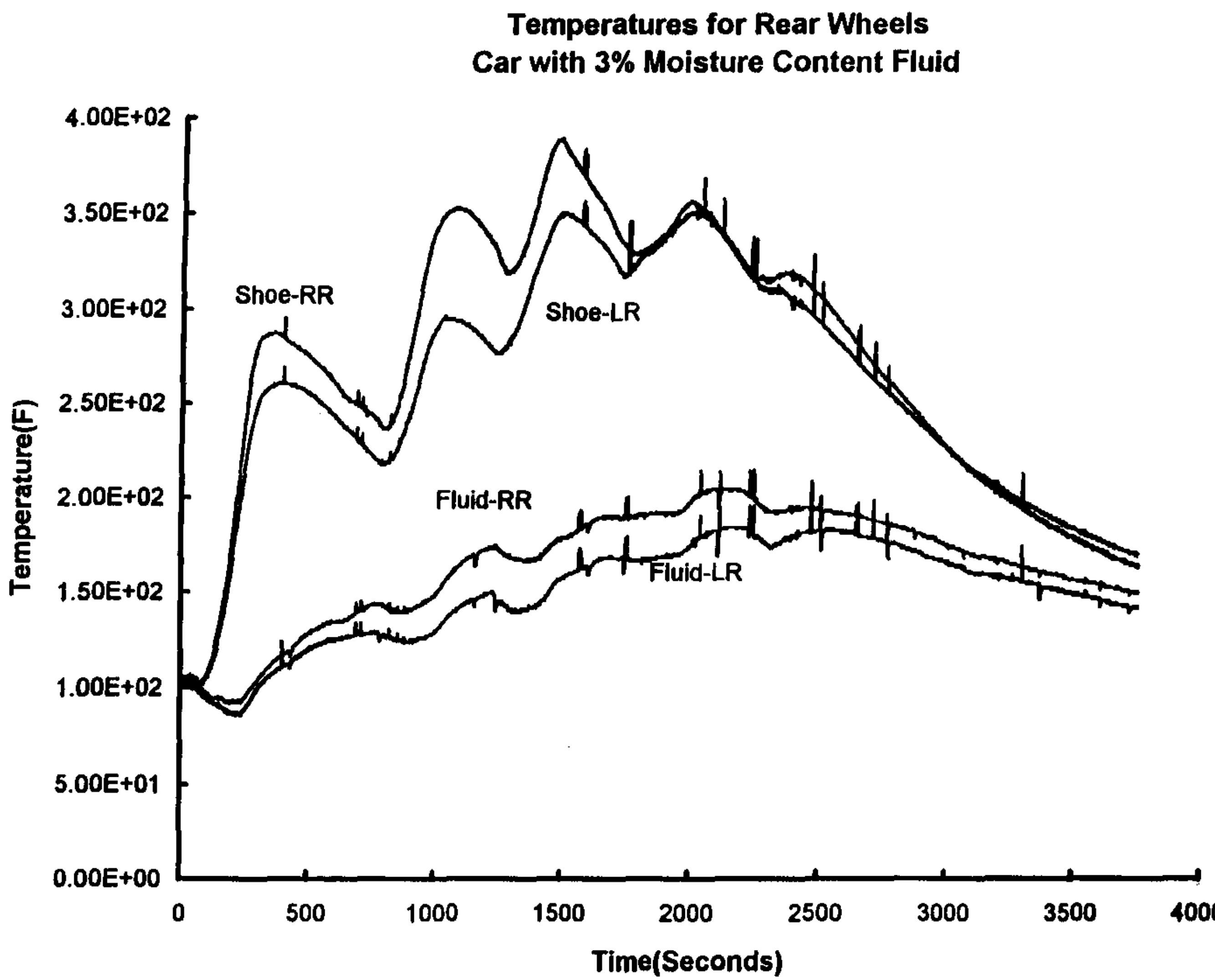

Figure 3c Rear Wheel Temperatures for the Car with 3% Fluid

Table 3 also lists the approximate temperatures at which the brake pressures recovered. In all cases, this appeared to occur at about 250°F. We do not know what constituents boil when the fluid is overheated, nor what the chemical compositions and latent heat of these constituents are, so it is not possible to predict the brake recovery on the basis of a thermal model. However, it does appear safe to suggest that the brake fluid must cool to at least 250°F before the brakes safely recover.

Table 3
Measured Boiling Points (°F)

Vehicle	% Moisture	bp before Front	bp after Front	Towing Test Peak T	Recovery Temp	bp before Rear	bp after Rear
Truck	0	>446*	372±2			446	401
	3	341±1	325±1	340	275	341±1	332±1
	5	281±1	279±1	244	220	285	282±2
Car	0	>446	365±0	321	<250	446	385±8
	3	334±1	301±8	257	210	335	290
	5	280	285±1	288	220	280	284±1
Van	0	>446	439±1	375	<280	446	438±3
	3	322±7	305±1	289	260	324±6	299
	5	284±3	287±2	300	<250	282±2	282±7

*the maximum reading of the instrument was 446°F

MODELING

One of the major goals of this study was to develop a thermal model that could predict the temperature history of the brake fluids under a variety of braking scenarios. Figure 4 shows a typical disk brake rotor and caliper assembly. Clearly, a precise thermal simulation would require a complex 3D model that could represent both the stationary caliper assembly and the rotating rotor. Instead, our goal was to determine whether a simpler model could suffice, given the many uncertainties in braking forces, the variable boundary conditions, for braking that produced significant temperature rises.

In developing the thermal models, besides the usual uncertainties about the thermal properties and their variation with temperature, there are four major uncertainties:

- What fraction of the braking energy is dissipated at each wheel?
- What is the state of the fluid in the cylinder, i.e., must we account for convection and change of phase in the brake fluid?
- Can the complex shape of the caliper assembly be modeled in a reasonable way, and what is the effect of the contact resistance between the piston and the steel backing of the brake pad?
- What portion of the energy is absorbed by the rotor and the caliper assembly?

The braking energy that was dissipated at each wheel was determined by assuming that the braking force was proportional to the pressure in the brake line. For the truck and the van, the brake failure was recorded in the front brakes, so the constant of proportionality could be determined for the rear wheels by using the measured braking energy after failure of the front brakes. With this constant known, it was possible to determine the fraction of the braking energy that was dissipated by the front brakes when all brakes were functioning.

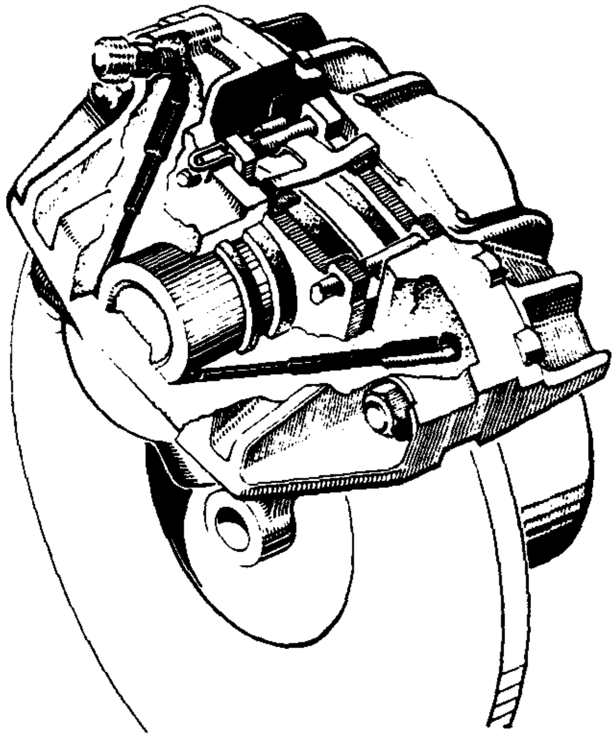

Figure 4 Typical Front Wheel Disk Brake System (from Limpert)

The most difficult question to answer concerned the behavior of the fluid. It is expected that the vibrations induced by a moving vehicle cause a stochastic state of convection in the fluid. Unfortunately, it is virtually impossible to account for these effects, first because the input data are not available, second because the computation would be prohibitively expensive, if at all possible. In addition, overheating leads to a phase change in a material whose chemical composition and thermal properties in a change of phase are unknown. From a practical point of view, we were primarily interested in predicting when the brake fluid would boil and not the thermal behavior after boiling. Therefore, we chose to model the fluid simply as a quiescent layer.

The system was modeled in two ways:

1. a separate model of the rotor and the caliper
2. a combined model of both.

In each case, the caliper was modeled by an axisymmetric piston/cylinder with the mass of the assembly distributed evenly around the cylinder (Figure 5). Although the model was not geometrically exact, it was judged that the thermal response would be adequately simulated. This was confirmed by the good agreement described below. The thermal properties were taken from Limpert (1992) and Incropera and DeWitt (1985). The contact resistance and its effects were unknown. Calculations performed with a contact resistance varying from 0 to 220 Btu/hr-ft^2 -F (Incropera and DeWitt 1985) produced a difference in peak fluid temperature of only 15°F; therefore, its effect was judged to be small. Previous models (Burgess 1974; Wolenweber 1993; Day, Harding and Newcomb 1999) have concentrated on modeling the rotor, pad, and average fluid temperatures. In our case, the goal was for the model to estimate the maximum fluid temperatures.

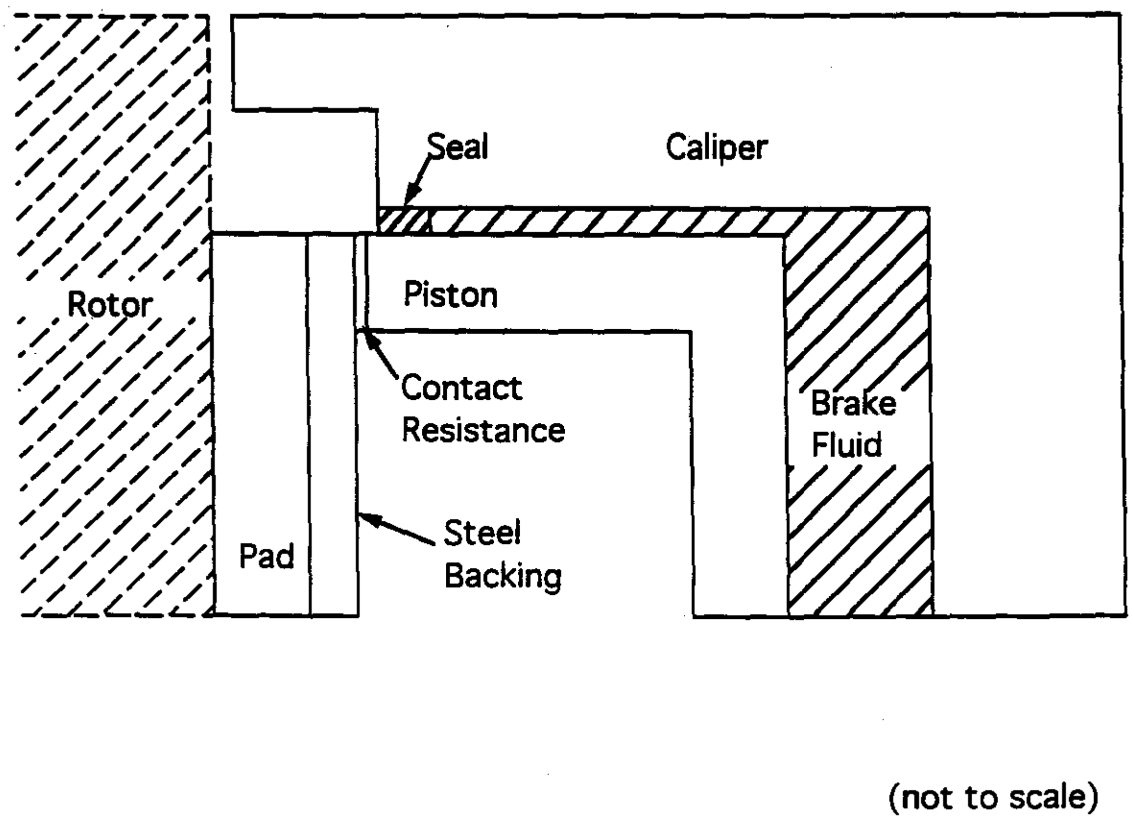

Figure 5 Schematic of the Finite Element Model

THE MODELING PROCESS

First the rotor and the pad/piston/caliper were modeled separately. The finite element mesh shown in Figure 5, but without the rotor, was used for the pad/piston. A control volume approach was used to calculate the rotor temperatures. This model considered the heat lost by convection and radiation from the rotor hub, the ventilated disk, the rotor rim, and the flat sides of the rotor. All of the convection coefficients were computed from the correlations given by Limpert (1992) that use the measured vehicle speed to compute the rotor rotational speed. As noted by Limpert the convection coefficients are difficult to estimate accurately, particularly for such a complex shape. The adjustments that were found to suffice for the truck brakes, namely increasing the coefficients by 150 percent when the vehicle was in motion and using a convection coefficient of 6 Btu/Ft^2-Hr-F when it was stationary, were found to apply equally well to the van brakes.

A fraction, F, of the heat generated at the rotor/pad interface was input to the rotor uniformly over the pad area. The remaining fraction, 1-F, was input to the FEM model. The thermal histories of the rotor and the pad/piston were computed with different values of F. The values of F and the convective heat transfer coefficients were varied by trial and error until the interface temperatures predicted by the two models were equal and matched the measured temperatures. This procedure was used successfully by Wolak (1985) in modeling the abrasion of a jet engine caused by the tips of the turbine blades and by Lavine (1991) in modeling cutting tools. An initial estimate of F can be obtained by considering the rotor and the pad to be one-dimensional, semi-infinite solids with a heat input at the interface and by requiring that the interface temperatures be equal. This model gives

$$F = \frac{\sqrt{(k\rho c)_{rotor}}}{\sqrt{(k\rho c)_{rotor}} + \sqrt{(k\rho c)_{pad}}} \qquad \text{Equation 1}$$

Table 4 lists the fraction of braking energy at the wheel that was applied to the pad in the finite element model.

Table 4
Fraction of Total Braking Energy Assigned to One Pad/Piston FEM Model

	Truck	Van
From Equation 1	9.0%	3.9
Separate Models	5.6	4.5
Combined Model	6.4	5.1

Equation 1 assumes that the heat is conducted through a semi-infinite homogeneous material. In the truck the heat quickly conducted through the pad and reached the low capacitance plastic piston. The effect of the plastic piston was to cause the interface temperature to rise above that predicted by using the energy computed from Equation 1. Thus the heat input determined from the FEM was substantially less (Table 4). The van's piston was steel, and the effect of this highly conductive backing was to reduce the predicted temperature; thus matching the experimental temperature required a slightly greater heat input.

The separate rotor and pad/piston analysis revealed the following:

- During the entire test, braking and stationary, there was negligible spatial variation of temperature in the rotor.
- The same convection coefficient adjusted to produce good agreement for the truck produced good agreement for the van.
- An effective heat transfer coefficient that combined convection and radiation was nearly constant at one value throughout the braking period. The same was true for the stationary cooling period.
- With the braking energy distributed uniformly over the pad surface area, the average of the interface temperatures predicted for the rotor and the pad/piston agreed well with the measured values.

Given these observations, it was possible to

- replace the separate models for the rotor and the pad/piston with the combined model shown in Figure 4
- use constant convection coefficients regardless of the variations in vehicle speed.

In this combined model, the braking energy was applied as a source to the interface nodes, and the model automatically apportioned the heat to the rotor and the pad. However, a major question remained. In this model, some of the heat generated at the rotor/pad interface would be conducted through the pad to the piston, and some would be lost from the back, unsupported, side of the pad. We assumed, subject to later examination, that the fraction of the total braking energy that should be used in the combined model was equal to the fraction of the pad area that was covered by the piston.

Figure 6a compares the predicted and measured temperatures for the truck. Given the variations in wind speed, vehicle speed, surface temperatures, radiation, and convection, the agreement was quite satisfactory. The fraction of the braking energy input to the model was 0.64, and the piston/pad area fraction was 0.69. Figure 6b shows the predicted and measured fluid temperatures. Because of the low conductivity of the piston, the peak fluid temperatures were found in the gap between the piston and the cylinder, not in the chamber behind the piston. Because of the low conductivity of the piston, little heat reached the rear surface of the piston, and the temperature was relatively isothermal in the chamber. On the basis of brake failure, boiling was estimated to occur between 1500 and 2000 seconds, which agrees well with the measured boiling point of the 5 percent fluid.

Figure 7a compares the predicted and measured temperatures for the van based upon an input energy fraction equal to the piston/pad area fraction. In these tests, the pad thermocouple was recessed slightly below the pad surface, and the peak pad surface temperatures were considerably higher, as indicated by the curve marked Peak Pad. Figure 7b shows the measured and predicted fluid temperatures. Because of the highly conducting

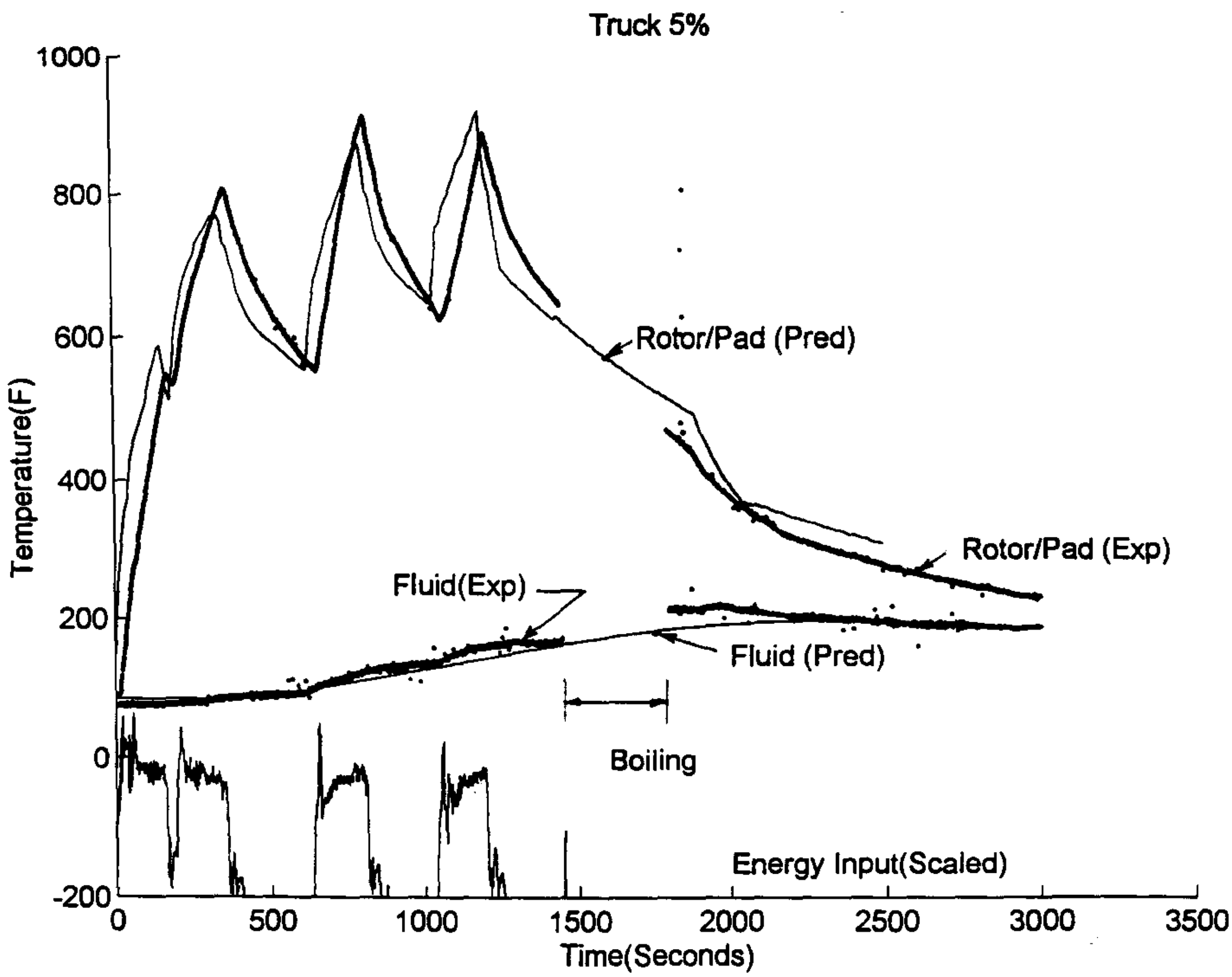

Figure 6a Predicted Temperatures for the Truck with 0% Fluid

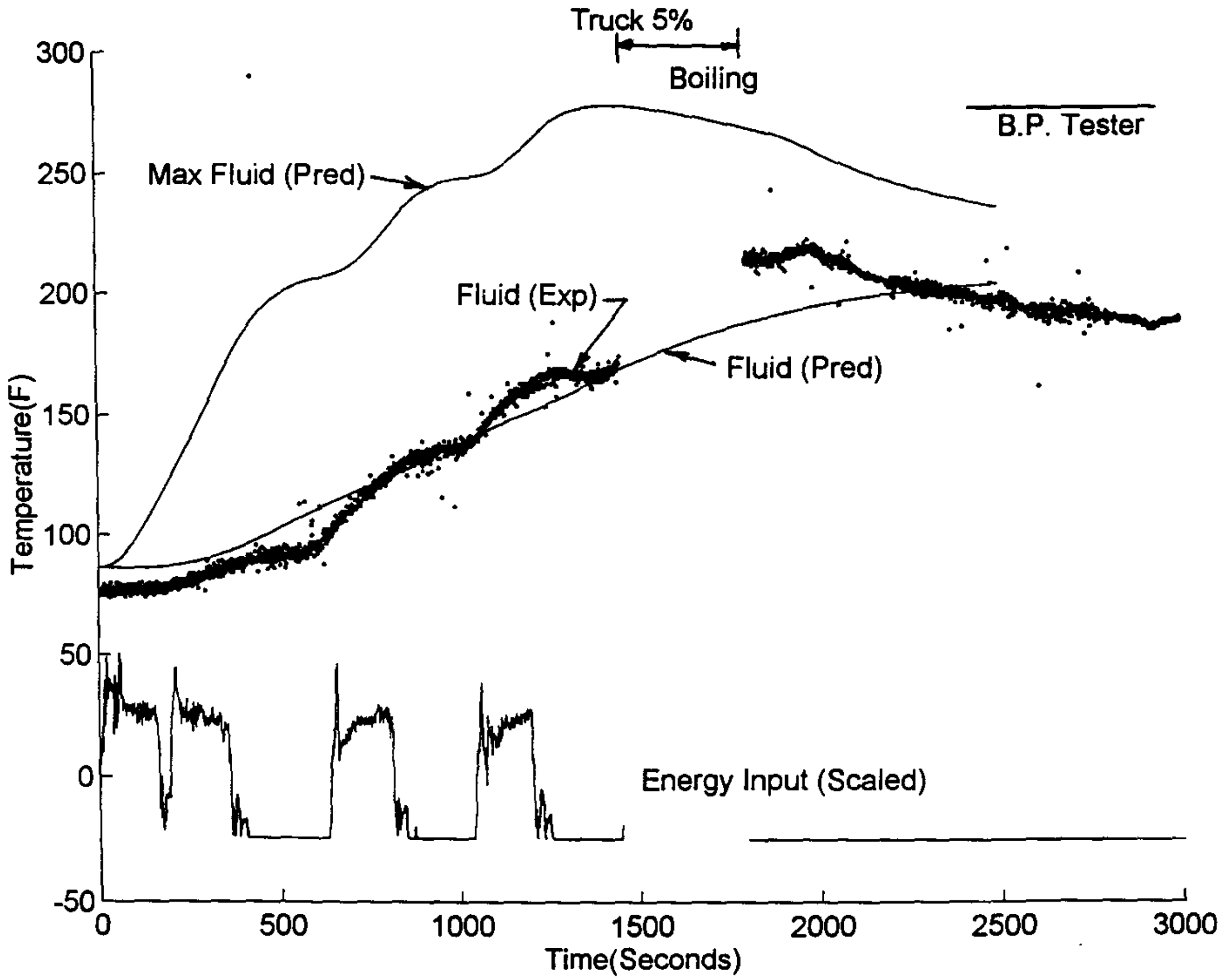

Figure 6b Predicted Fluid Temperatures for the Truck with 0% Fluid

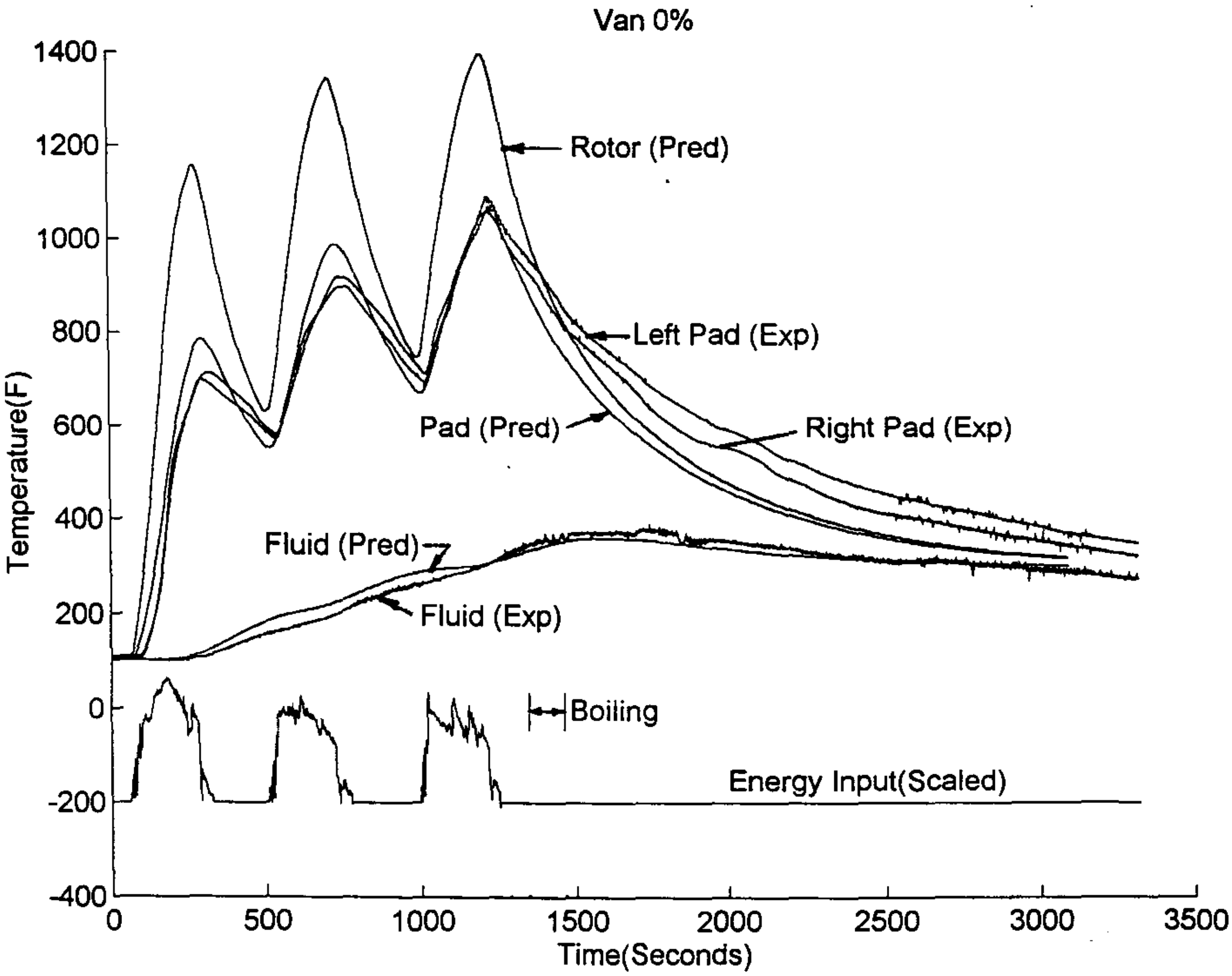

Figure 7a Predicted Temperatures for the Van with 0% Fluid

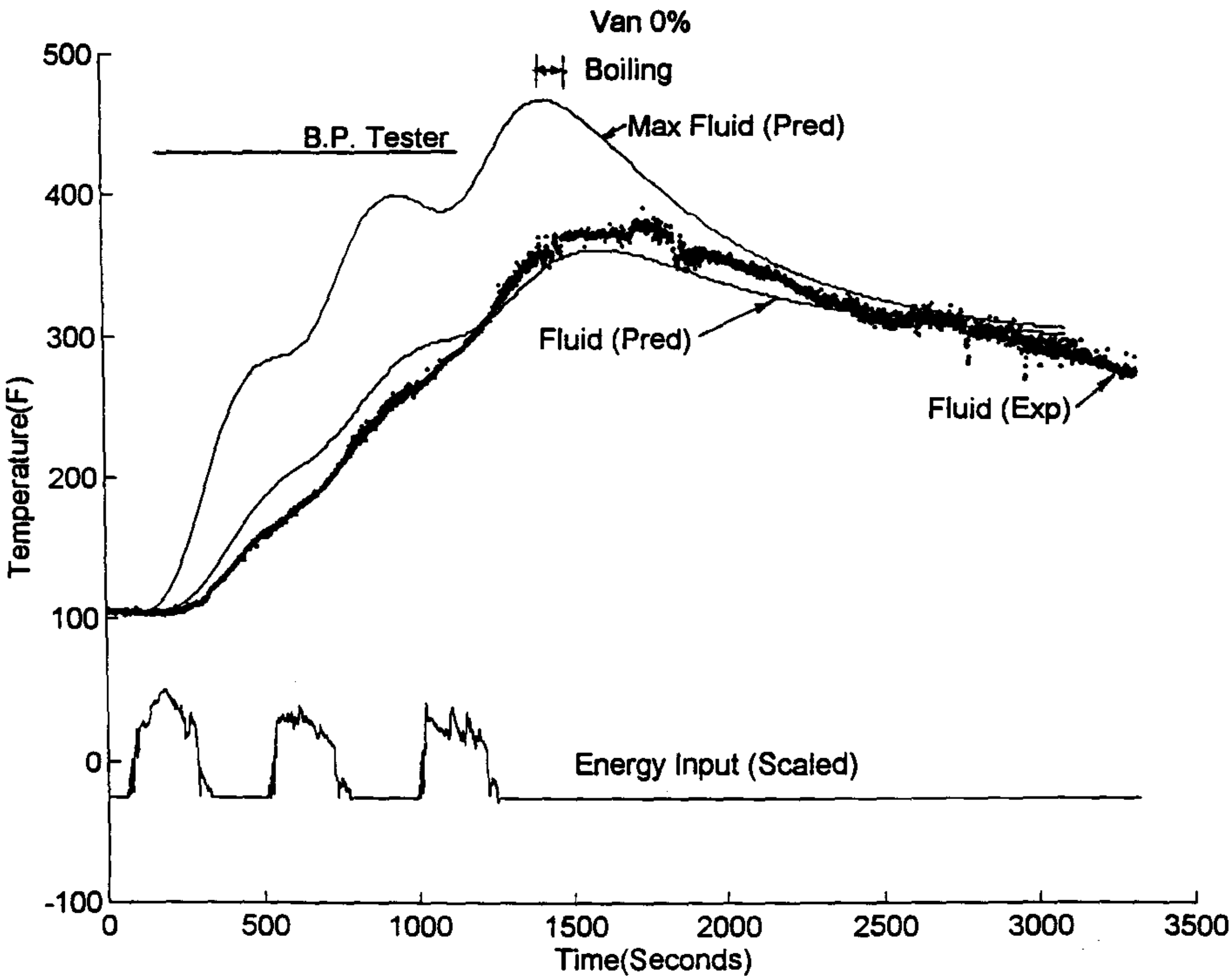

Figure 7b Predicted Fluid Temperatures for the Van with 0% Fluid

steel piston, the fluid temperatures were greatest in the gap between the piston and the cylinder and at the rear piston face. Substantial variations in fluid temperature were calculated through the thickness of the fluid layer in the chamber. In the experiments it was not possible to determine the location of the thermocouple, so the curve marked Fluid (Predicted) represents the nodal temperature that best matched the measured temperature. This node was used to represent the fluid temperatures in the other van prediction. Brake failure occurred in the interval marked "Boiling," which agrees well with the time of predicted maximum fluid temperature.

Because of the uncertainty about the fluid properties during a phase change, the model did not account for boiling. Therefore, the predicted temperatures are valid only until the boiling point is reached. In Figure 7b, when the maximum fluid temperature reaches the boiling point, a phase change should occur. However, the total energy associated with the phase change is quite small in this case, and the boiling can be neglected in predicting the temperature history. This is borne out by the good agreement between the measured and predicted fluid temperatures.

Figures 7c and d are for tests in which substantial amounts of boiling did occur. Here ignoring the phase change energy resulted in over-prediction of the system temperatures. The effect of the latent heat of evaporation was clearly shown by the almost constant experimental temperature when the brakes were no longer applied. In addition, the reverse heat liberation when the vapor condensed kept the measured temperature high, whereas the overestimated model temperatures predicted a high heat loss and thus a too rapid drop in temperature.

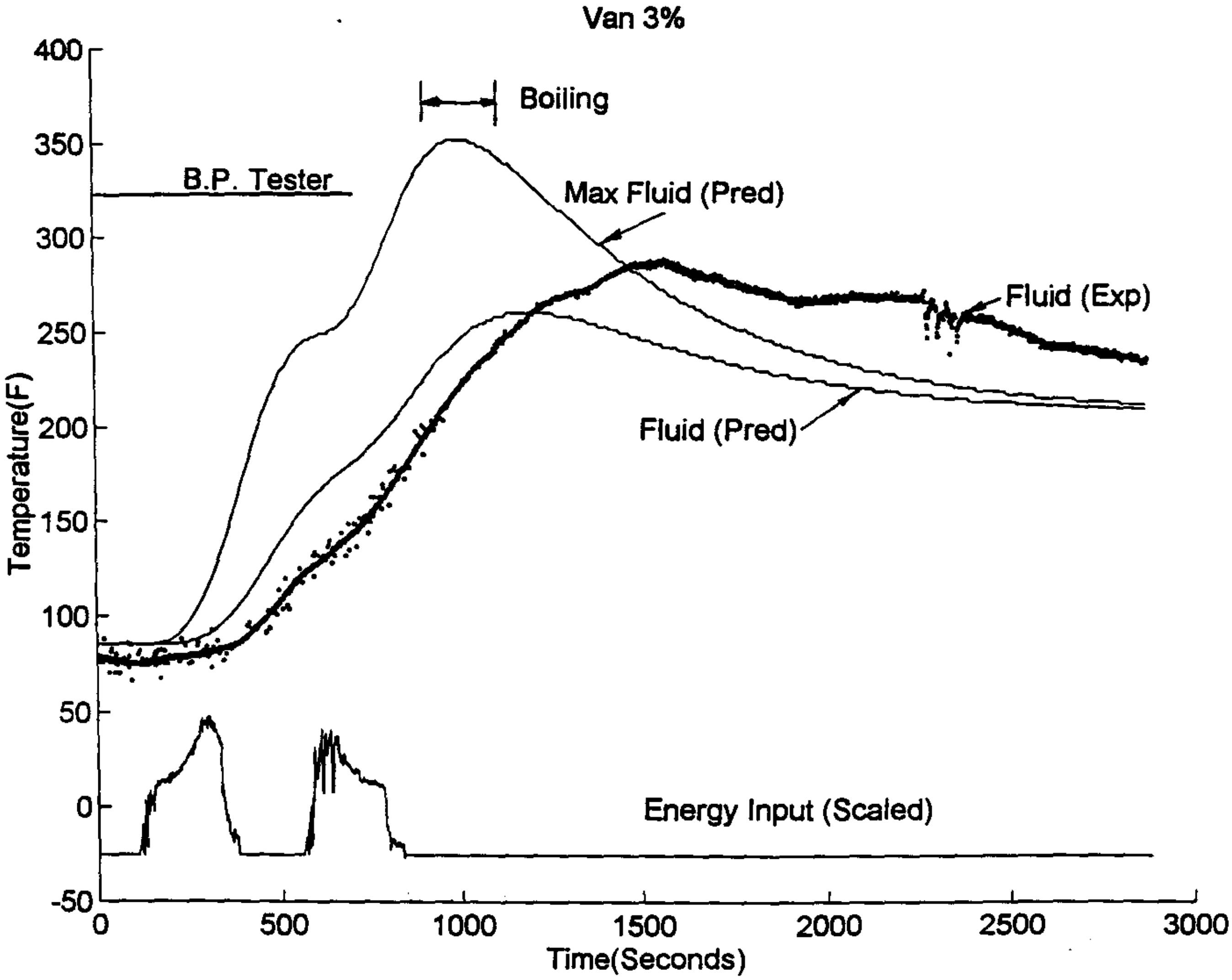

Figure 7c Predicted Fluid Temperatures for the Van with 3% Fluid

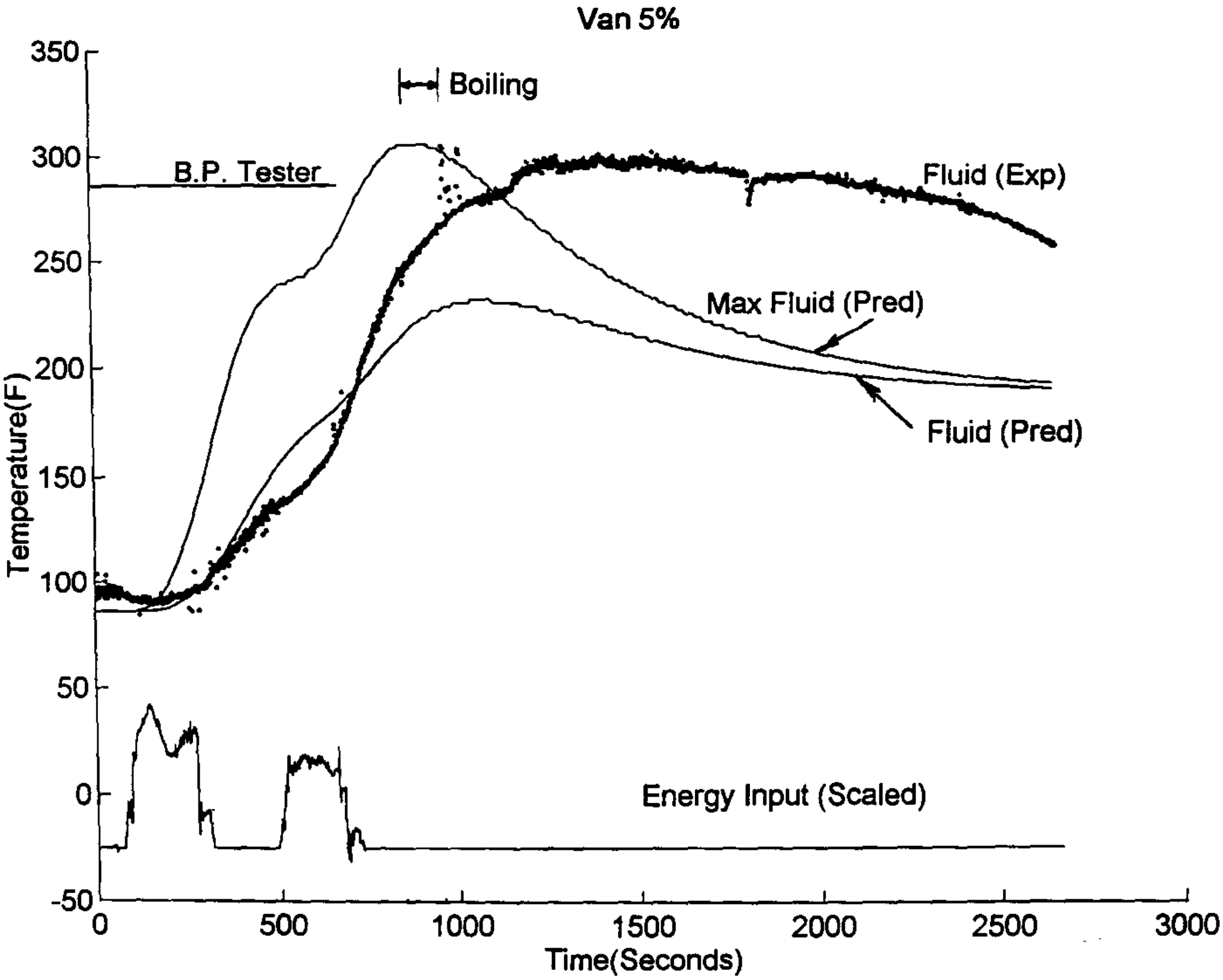

Figure 7d Predicted Fluid Temperatures for the Van with 5% Fluid

CONCLUSIONS

In addition to the results presented here, we developed a companion lumped mass model that predicted the average fluid temperature with reasonable accuracy. However, because of the large temperature variations that exist through the pad/fluid/caliper assembly, a knowledge of the average fluid temperature is not sufficient to estimate whether and when boiling will occur. The combined rotor-pad/piston/caliper finite element model can be used to predict when brake fluid boiling is likely to occur under different braking conditions if reasonable estimates of convection coefficients and the fraction of braking energy supplied at the interface are known.

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