U.S. Department
of Transportation
Federal Railroad Administration

Investigation of the Effects of Braking System Configurations on Thermal Input to Commuter Car Wheels

Office of Research and
Development
Washington, DC 20590

Research and Special Programs Administration
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Cambridge, MA 02142-1093

DOT/FRA/ORD-96/06
DOT-VNTSC-FRA-95-7

Final Report
March 1996

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REPORT DOCUMENTATION PAGE	Form Approved OMB No. 0704-0188

1. AGENCY USE ONLY (Leave blank)	2. REPORT DATE	3. REPORT TYPE AND DATES COVERED
	March 1996	Final Report February 1994 - May 1995

4. TITLE AND SUBTITLE Investigation of the Effects of Braking System Configurations on Thermal Input to Commuter Car Wheels	5. FUNDING NUMBERS R5005/RR528
6. AUTHOR(S) Jeffrey E. Gordon and Oscar Orringer	
7. PERFORMING ORGANIZATION NAME(S) AND ADDRESS(ES) U.S. Department of Transportation Research and Special Programs Administration Volpe National Transportation Systems Center Kendall Square, Cambridge, MA 02142-1093	8. PERFORMING ORGANIZATION REPORT NUMBER DOT-VNTSC-FRA-95-7
9. SPONSORING/MONITORING AGENCY NAME(S) AND ADDRESS(ES) U.S. Department of Transportation Federal Railroad Administration Office of Research and Development Washington, DC 20590	10. SPONSORING/MONITORING AGENCY REPORT NUMBER DOT/FRA/ORD-96/06
11. SUPPLEMENTARY NOTES	
12a. DISTRIBUTION/AVAILABILITY STATEMENT This document is available to the public through the National Technical Information Service, Springfield, VA 22161	12b. DISTRIBUTION CODE

13. ABSTRACT (Maximum 200 words)

A heat transfer model, previously developed to estimate wheel rim temperatures during tread braking of MU power cars and validated by comparison with operational test results, is extended and applied to cases involving several different blended brake system configurations. Preliminary and final system configurations defined by the car owner/operator are evaluated. The ability of the selected option to maintain wheel rim temperatures at safe levels is demonstrated.

14. SUBJECT TERMS Braking; Heat transfer analysis; Rail vehicles; Wheels	15. NUMBER OF PAGES 68
	16. PRICE CODE

17. SECURITY CLASSIFICATION OF REPORT	18. SECURITY CLASSIFICATION OF THIS PAGE	19. SECURITY CLASSIFICATION OF ABSTRACT	20. LIMITATION OF ABSTRACT
Unclassified	Unclassified	Unclassified	Unlimited

NSN 7540-01-280-5500

Standard Form 298 (Rev. 2-89)
Prescribed by ANSI Std. 239-18
298-102

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PREFACE

This report is the sixth in a series of engineering studies on railroad vehicle wheel performance. The work was performed by the Volpe National Transportation Systems Center (Volpe Center), in support of the Federal Railroad Administration (FRA), Office of Research and Development.

Preliminary studies involving evaluation of actions taken to respond to high rates of crack occurrence observed in the wheels of certain multiple unit (MU) power cars used in commuter service were summarized in the first report. The second report documented an operational test that was conducted to determine the effects of high-performance stop braking on temperature distributions in the MU wheels. In the third report, heat transfer and stress finite element models of the MU wheels are documented. The heat transfer models were validated by comparison of calculated temperature distributions with temperature measurements taken during the operational test. The fourth report describes the application of moiré interferometry to the measurement of crack opening during wheel saw-cutting, and companion finite element analyses conducted to reconstruct the stress distributions in the wheel rim from these measurements. The fifth report summarizes metallurgical evaluation of the microstructure produced by the combination of wheel/rail contact stress and high temperature found near the tread surfaces of wheels subjected to high-performance stop braking.

This report extends the application of the heat transfer model described in the third report for the purpose of investigating the effects on wheel temperature of modifications to the braking profile and redistribution of the braking effort between the tread and dynamic brake systems. Results are presented for nominal profiles and adjusted braking profiles corresponding to several alternative system configurations. The alternatives reflect the evolution of the system design for the Arrow-III MU cars owned and run by New Jersey Transit Rail Operations.

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METRIC/ENGLISH CONVERSION FACTORS

ENGLISH TO METRIC

LENGTH (APPROXIMATE)

1 inch (in)	= 2.5 centimeters (cm)
1 foot (ft)	= 30 centimeters (cm)
1 yard (yd)	= 0.9 meter (m)
1 mile (mi)	= 1.6 kilometers (km)

AREA (APPROXIMATE)

1 square inch (sq in, in^2)	= 6.5 square centimeters (cm^2)
1 square foot (sq ft, ft^2)	= 0.09 square meter (m^2)
1 square yard (sq yd, yd^2)	= 0.8 square meter (m^2)
1 square mile (sq mi, mi^2)	= 2.6 square kilometers (km^2)
1 acre = 0.4 hectare (he)	= 4,000 square meters (m^2)

MASS - WEIGHT (APPROXIMATE)

1 ounce (oz)	= 28 grams (gm)
1 pound (lb)	= 0.45 kilogram (kg)
1 short ton = 2,000 pounds (lb)	= 0.9 tonne (t)

VOLUME (APPROXIMATE)

1 teaspoon (tsp)	= 5 milliliters (ml)
1 tablespoon (tbsp)	= 15 milliliters (ml)
1 fluid ounce (fl oz)	= 30 milliliters (ml)
1 cup (c)	= 0.24 liter (l)
1 pint (pt)	= 0.47 liter (l)
1 quart (qt)	= 0.96 liter (l)
1 gallon (gal)	= 3.8 liters (l)
1 cubic foot (cu ft, ft^3)	= 0.03 cubic meter (m^3)
1 cubic yard (cu yd, yd^3)	= 0.76 cubic meter (m^3)

TEMPERATURE (EXACT)

[(x-32)(5/9)] °F = y °C

METRIC TO ENGLISH

LENGTH (APPROXIMATE)

1 millimeter (mm)	= 0.04 inch (in)
1 centimeter (cm)	= 0.4 inch (in)
1 meter (m)	= 3.3 feet (ft)
1 meter (m)	= 1.1 yards (yd)
1 kilometer (k)	= 0.6 mile (mi)

AREA (APPROXIMATE)

1 square centimeter (cm^2)	= 0.16 square inch (sq in, in^2)
1 square meter (m^2)	= 1.2 square yards (sq yd, yd^2)
1 square kilometer (km^2)	= 0.4 square mile (sq mi, mi^2)
10,000 square meters (m^2)	= 1 hectare (he) = 2.5 acres

MASS - WEIGHT (APPROXIMATE)

1 gram (gm)	= 0.036 ounce (oz)
1 kilogram (kg)	= 2.2 pounds (lb)
1 tonne (t) = 1,000 kilograms (kg)	= 1.1 short tons

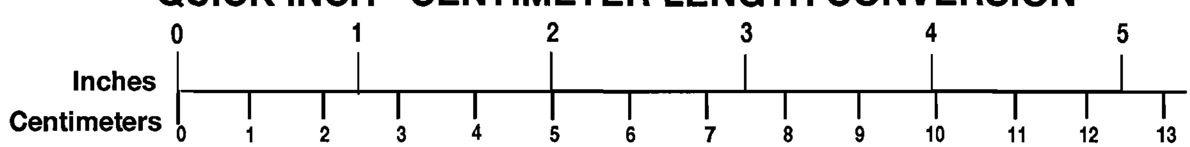
VOLUME (APPROXIMATE)

1 milliliter (ml)	= 0.03 fluid ounce (fl oz)
1 liter (l)	= 2.1 pints (pt)
1 liter (l)	= 1.06 quarts (qt)
1 liter (l)	= 0.26 gallon (gal)
1 cubic meter (m^3)	= 36 cubic feet (cu ft, ft^3)
1 cubic meter (m^3)	= 1.3 cubic yards (cu yd, yd^3)

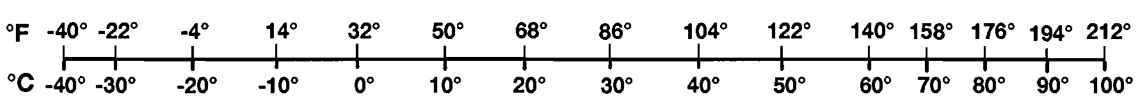
TEMPERATURE (EXACT)

[(9/5) y + 32] °C = x °F

QUICK INCH - CENTIMETER LENGTH CONVERSION



QUICK FAHRENHEIT - CELSIUS TEMPERATURE CONVERSION



For more exact and or other conversion factors, see NBS Miscellaneous Publication 286, Units of Weights and Measures. Price $2.50 SD Catalog No. C13 10286

Updated 1/23/95

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1. INTRODUCTION

This report is the sixth in a series on the results of an engineering study of the effects of service loads on railroad vehicle wheels. The study was initiated in September 1991, in response to a request for assessment of contributing factors and corrective actions taken regarding high rates of crack occurrence in certain multiple unit (MU) power cars used in commuter service. The ultimate goal of the study is the evaluation of safe limits on performance demand (weight carried per wheel, maximum speed, vehicle deceleration rate) as a function of wheel design, material selection, and manufacture, as well as percentage of braking effort absorbed through the wheel tread in service. The models developed in the study are intended to provide the capability for similar engineering design analyses of other railroad vehicle wheels besides the types used on MU cars.

1.1 Background

Special inspections of commuter rail vehicles conducted by the Federal Railroad Administration (FRA) Office of Safety in 1991 revealed chronic problems of cracking in the wheels of MU cars operated by three railroads serving the Greater New York area. Daily inspections were immediately undertaken to assure continuing operational safety, while options for lasting solutions were proposed and studied.

One of the affected groups of cars was the New Jersey Transit Rail Operations (NJTRO) Arrow-III fleet. The Arrow-III is a moderate weight vehicle, originally equipped solely with tread brakes, operated at speeds up to 100 mph (160 km/h). The wheel cracks in this fleet, of thermal origin in the center tread position, were found to have been caused by the demand for heat absorption through the tread imposed by high-speed operation without auxiliary brakes.

A finite element heat transfer model was developed to simulate the transient heating effects of braking, in order to provide a means of evaluating long-term options. An instrumented operational test was conducted to obtain actual histories of temperature near the wheel tread versus time, and the test results were used to calibrate the finite element model. Both the simulation and test showed that slowing or stopping from high speed can produce tread temperatures on the order of 1000 °F (800 °C) if no auxiliary brakes are available to absorb some of the vehicle kinetic energy.

1.2 Evaluation of Alternative Braking Options

This report summarizes the results of several follow-on analyses that were made with the finite element heat transfer model. The objective was to project the effects of alternative options for equipping the Arrow-III MU fleet with auxiliary brakes.

In 1993, NJTRO began a previously scheduled program to refurbish and upgrade the Arrow-III fleet. Based on the recent experience with wheel thermal cracking, the railroad decided to add auxiliary brakes to the upgraded equipment list and requested that the Volpe Center assist in the evaluation of alternative options.

Options for auxiliary braking were based on the previously planned motive power upgrade, which involved replacement of the existing DC traction motors with AC motors. Since the new AC motor was more powerful than the existing DC motor, cost and weight savings were achieved at no sacrifice of performance by equipping each married pair of MU cars with AC motors on only three of the four trucks. This unusual configuration gave rise to various possible blended brake arrangements.

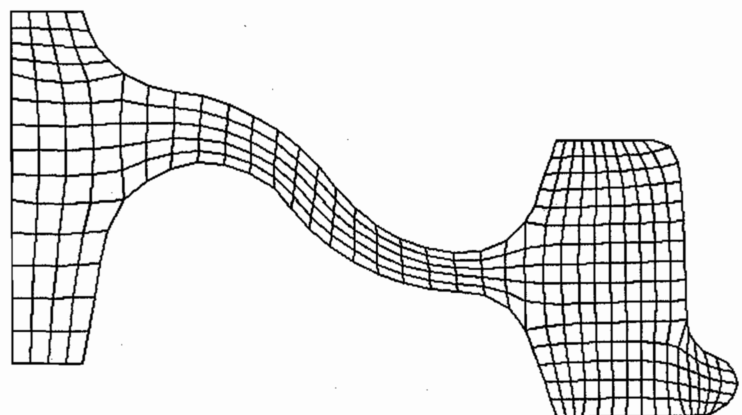
With the addition of rheostatic energy dissipation equipment, the three powered trucks could be set up for dynamic braking. This auxiliary braking effort could be either: (1) rated for the existing DC motor capability to take advantage of existing rheostatic equipment designed for the other fleets; or (2) augmented with a new rheostatic design to take advantage of the AC motor capability. Also, the tread braking effort could be reduced either by distributing the trade-off, with some of the benefit obtained from dynamic braking going to the unpowered truck, or by reducing the tread brake effort only on the powered trucks.

The initial configuration was based on the existing dynamic capability. Blended brakes on the three powered trucks were combined with straight tread braking, through an 80% brake pipe pressure reduction valve, on the unpowered truck. When this condition was found to provide insufficient relief, NJTRO defined additional options, under the assumption of a 30% dynamic brake augmentation in the 50 to 100 mph (80 to 160 km/h) speed range. These options were also analyzed under the assumption of a 90 mph (144 km/h) maximum operating speed. Analyses of wheel temperature effects suggested that one of the options, involving auxiliary disc brakes on the unpowered truck, did not offer any significant performance gain, relative to distributed augmented dynamic braking. NJTRO then abandoned the disc brake option, after further consideration, to avoid unnecessary complication of the fleet maintenance program.

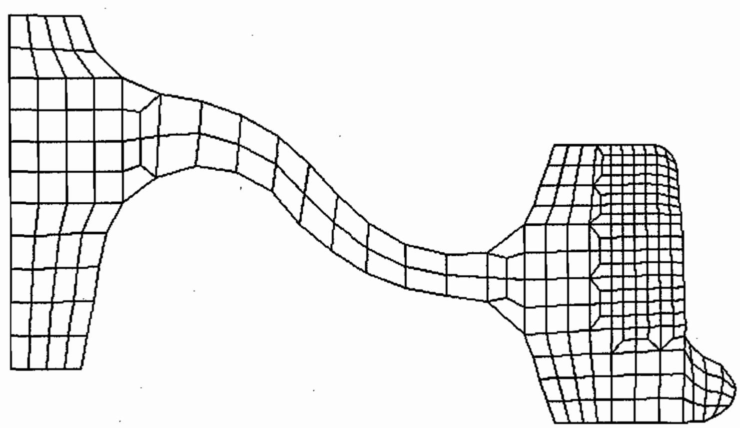
Based on extensive development tests, the actual dynamic augmentation was verified as 26%, averaged over the 50 to 100 mph (80 to 160 km/h) speed range. NJTRO set up the final configuration with balanced braking, i.e., average retardation force equal for all trucks in the 50 to 100 mph (80 to 160 km/h) range as well as the 0 to 50 mph (0 to 80 km/h) range. This configuration was analyzed for a maximum operating speed of 100 mph (160 km/h).

2. MODELING

Finite element programs obtained from the Lawrence Livermore National Laboratory (LLNL) were used to conduct the thermal analyses on the S-plate wheel, as shown in figure 1. This geometry corresponds to a 32-inch (813 mm) wrought, reverse-dish (S-plate) wheel with 1:40 taper. As the wheel is an axisymmetric body, a two-dimensional representation of the geometry of the cross-section is sufficient to describe the three-dimensional structure. The axisymmetry reasonably represents the distribution of the heat input by rolling of the wheel. The computed temperatures are thus averaged around the circumference. Asymmetric variations (e.g., local temperature increase at the brake shoe) are not modeled. Also, it is assumed that all tread brake units and brake shoes are uniformly functional.



(a)



(b)

Figure 1. Finite Element Meshes Used for Braking Studies: (a) Model Used for Preliminary Analyses (377 nodes/316 elements); (b) Refined Model Used in Revised Analyses (320 nodes/268 elements)

The mesh used in this study, depicted in figure 1, was generated using *MAZE* [1], a member of the LLNL suite of codes, which is a two-dimensional pre-processor. Thermal calculations are accomplished using *TOPAZ2D* [2], which calculates the temperatures at nodes due to heat flux applied at the brake shoe/wheel tread interface during the course of a braking event.

2.1 Development of Matrix of Cases for Preliminary Assessment

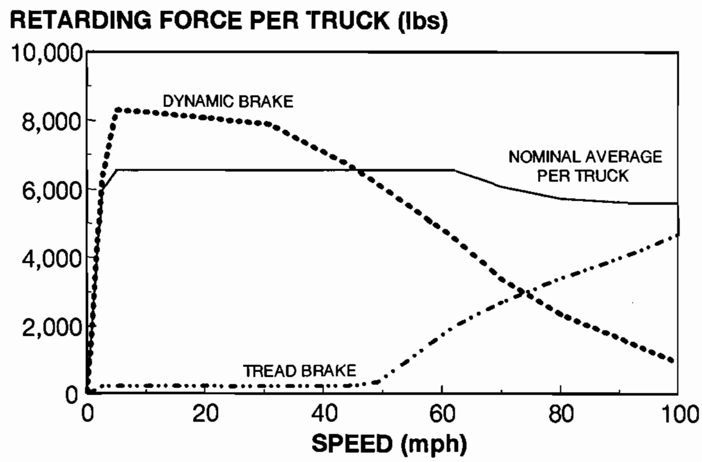
The braking system performance diagrams for the original Arrow-III modification are presented in figure 2. The curves in these figures represent retardation force per brake system component as a function of speed on a per-truck basis, and correspond to the car configuration after the retrofit from DC to AC motors, but before the further modifications to take advantage of the augmented dynamic brake.

Figure 2(a) shows how the control system blends the tread and dynamic braking efforts on a powered truck to maintain an effective deceleration rate over the entire operating speed range. Also shown for reference is a curve of the average retarding force per truck, which reflects the tread braking effort from the fourth (unpowered) truck as well as the blended efforts from the three powered trucks. Strictly speaking, the nominal average is the car rebuilder's design curve and is slightly different from the true average. However, the differences and their effects on calculations are insignificant.

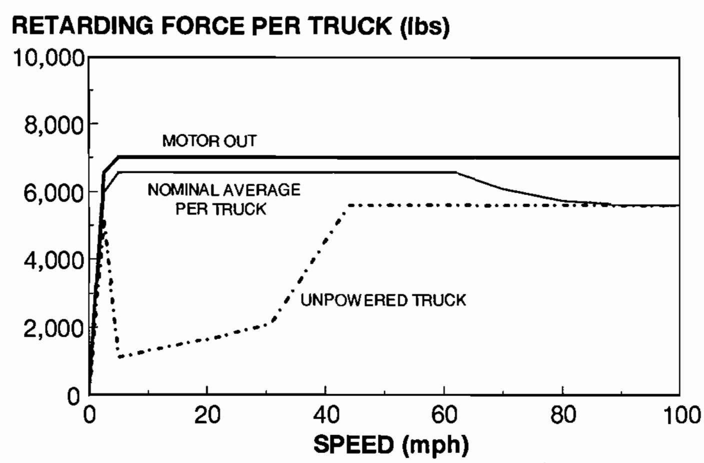
Figure 2(b) shows the retarding forces developed when dynamic braking is not available. The tread brakes on the fourth (unpowered) truck are operated at 80% of full brake cylinder pressure (BCP), via a reduction valve, to develop 5,600 lb/truck (25 kN/truck) retarding force between 100 and 44 mph (160 and 70 km/h). Below 44 mph (70 km/h) the control system further reduces BCP and retarding force in order to compensate for the above-nominal dynamic braking effort from the powered trucks. Also, if dynamic braking becomes unavailable due to motor outage on a powered truck, the control system resets the BCP to 100% (7,000 lb [31 kN] retarding force) on that truck.

The contributions from the braking components are blended to maintain a deceleration rate of 1.7 mph/sec (0.08 g) from 100 mph to 60 mph (160 km/h to 96 km/h) and 2.0 mph/sec (0.09 g) from 60 mph (96 km/h) to a standstill. These rates are minima (nominal case) with an allowable overshoot of 15% (limit case), which corresponds to 1.96 mph/sec (0.092 g) and 2.3 mph/sec (0.10 g), respectively, for the two speed regimes. These limits are imposed by the constraints of signal spacing and required stopping distances, as well as passenger ride comfort.

The temperature distributions over time caused by the thermal input from the tread brakes during a stop from 100 mph (160 km/h) were computed for the preliminary assessment. Table 1 summarizes the case matrix.



(a) Powered Truck



(b) Unpowered Truck

Figure 2. Retardation Force Versus Speed

Table 1. Matrix of Test Cases for Preliminary Investigation

CONDITION	DECELERATION RATE (mph/sec)	BRAKING SCENARIO		
		BLENDED BRAKE	UNPOWERED TRUCK	MOTOR OUT
NOMINAL	1.7 from 100 mph to 60 mph 2.0 from 60 to 0 mph	N1	N2	N3
LIMIT	1.96 from 100 mph to 60 mph 2.3 from 60 mph to 0 mph	L1	L2	L3

2.2 Translation of Braking Profiles into Thermal Input

In order to develop the thermal input to the wheel rim, the braking (deceleration) profile must be known in some form. For the analyses reported here, the retardation force versus speed curves are converted into deceleration rates by dividing by the vehicle mass. These decelerations are then integrated over the speed range to obtain similar plots of speed versus time for the stop. In calculating the stop time, the retardation force is adjusted by adding the contribution from train resistance (the "Davis resistance") according to equation (1) from [4]:

$$R(V) = 1.3 + \left(\frac{29}{W}\right) - 0.045 * V + \left(\frac{0.07 * V^2}{W * N}\right) \qquad (1)$$

where W is the axle load (in tons), N is the number of axles in the train, V is the train velocity in mph, and $R(V)$ is the Davis resistance in lbs/ton for the train as a function of speed. These values are converted into retarding force in lbs/truck by multiplying by the vehicle weight in tons and dividing by two. The Davis resistance reduces the work required by the braking system, therefore it is added to the retarding force per truck for the purposes of calculating the time required for the train to stop from a given speed. The Davis resistance amounts to approximately 130 lbs (0.6 kN) per truck at 100 mph (160 km/h) and drops to about 105 lbs (0.47 kN) per truck at 0 mph.

Knowing the speed and the tread braking component of the retardation force permits calculation of the instantaneous power input to the wheel rim as the product of these quantities. The thermal input to the wheel rim is obtained by dividing the instantaneous power by the brake shoe contact area. It is assumed, for these calculations, that the shoe is 2.5 inches (6.35 cm) wide and the contact area is, therefore, a strip of this width around the wheel circumference. The heat flux is applied uniformly over the width of the contact strip. This heat flux (per wheel) is plotted against time in figure 3 for the powered truck, unpowered truck, and motor out cases for the nominal deceleration profile. The same curves appear in figure 4 for the emergency (limit) deceleration rate. These curves represent 100% of the flux as calculated by the above procedure. Appendix B contains a sample of the calculations necessary for translation of the deceleration profile into heat flux in MATHCAD 5.0 PLUS© format.

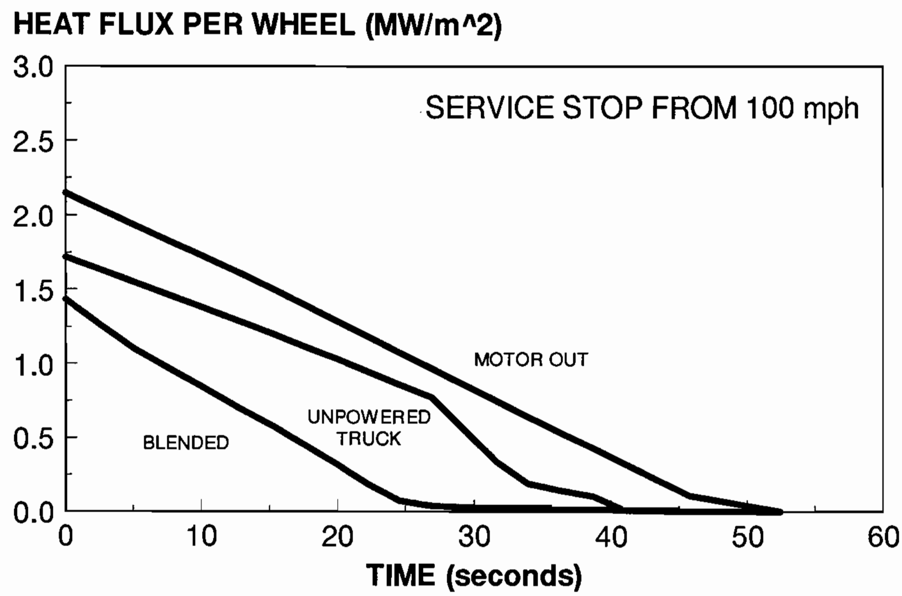


Figure 3. **Heat Flux Applied through Brake Shoe during Stop from 100 mph (160 km/h) – NOMINAL CASE**

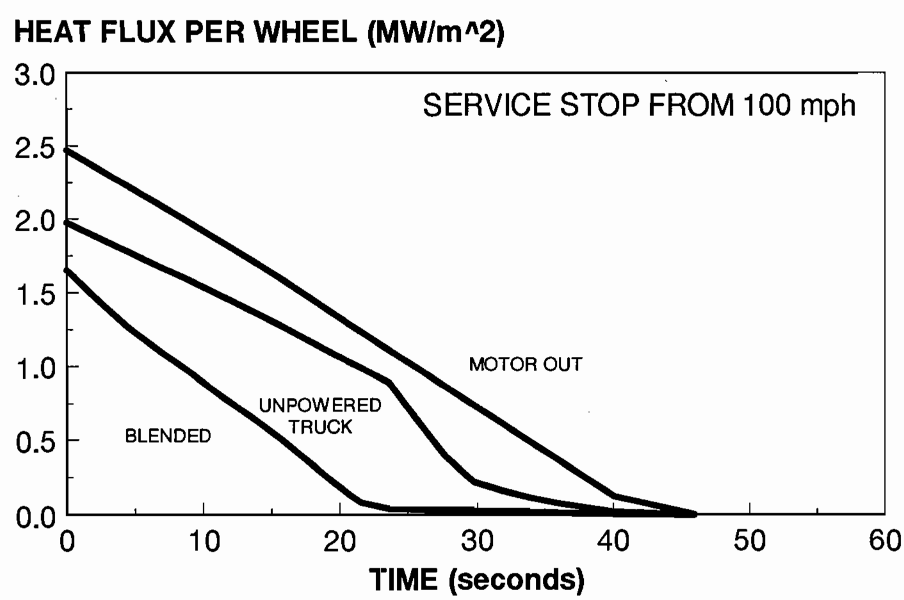


Figure 4. **Heat Flux Applied through Brake Shoe during Stop from 100 mph (160 km/h) – LIMIT CASE**

It is reported in the literature [5] that a considerable portion of the thermal load applied to the wheel tread is absorbed by the cooler running rail, through the wheel/rail contact patch. The so-called "rail chill" effect has been estimated to account for approximately 18% of the applied heat flux. Based on the good agreement between previously reported results of this model and the field test, it is believed that this effect represents no more than about 5% of the thermal load. Therefore, these thermal analyses are conducted by reducing the applied heat flux (as shown in figures 3 and 4) by 5% to account for losses into the rail.

2.3 Material Properties

Material properties for wheel steel must be specified over the expected temperature range in order to execute the thermal analysis. Temperature-dependent values for the thermal conductivity (k) and the specific heat (c_p) are tabulated in appendix A [3].

2.4 Results of Thermal Analyses

Figure 5 illustrates the results obtained for the nominal blended braking profile. Temperature-time histories for a point on the tread surface (node 110) and two points immediately beneath it (nodes 109 and 108) are shown. Presenting the results in this way sheds some light on the temperature gradient through the rim. These plots represent temperatures at three discrete points. Node 110 is on the surface at the center of the brake shoe contact zone, node 109 is immediately beneath it at a depth of 0.29 in. (7.4 mm), and node 108 is below node 109 at a depth of 0.58 in. (14.7 mm) from the tread. Appendix C contains the complete set of results.

The rim temperatures for the blended braking case do not exceed 600 °F (316 °C) for either the nominal or the limit deceleration rates (cases N1 and L1). The unpowered truck case predicts a layer approximately one element thick for the nominal and limit cases (N2 and L2) in which the temperature exceeds 600 °F. For the motor out cases (N3 and L3), the temperature exceeds 600 °F in a layer approximately two elements thick, i.e., 0.58 in. (14.7 mm).

Appendix D contains temperature contour plots for these six cases. The contour plots provide a better view of the depth of penetration of the high-temperature layer. It is perhaps simpler to evaluate these results quantitatively by examining the data in a reduced form, as presented in table 2. The maximum temperature at the tread surface is tabulated, along with the time at which this temperature was attained. The duration of the simulated braking event is approximately 52 seconds for the nominal deceleration rate and 46 seconds for the emergency (limit) rate. Note that the maximum temperature occurs well before the conclusion of the braking event.

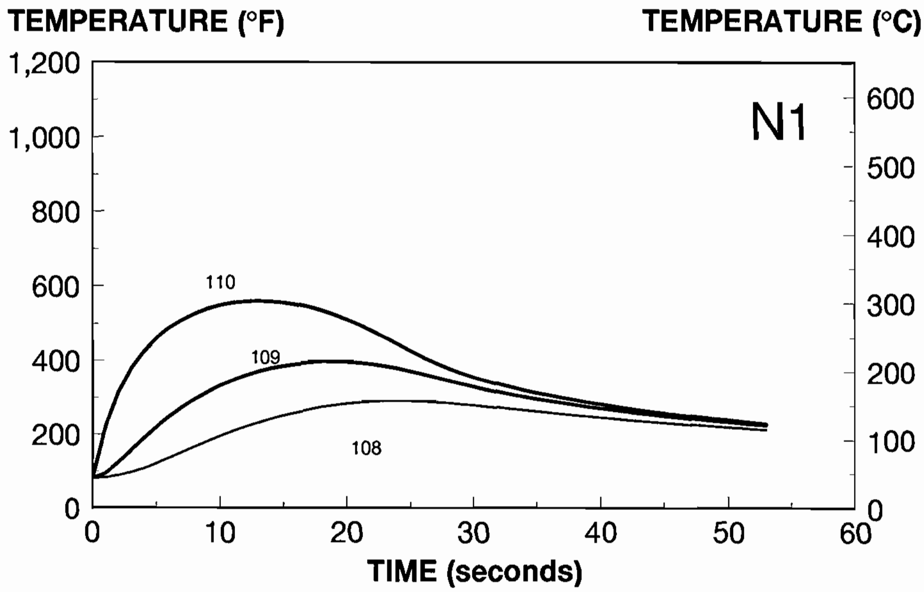


Figure 5. Temperature-Time History of Three Nodes Near Tread Surface for Blended Braking Case at Nominal Deceleration Rate

Table 2. Results of Preliminary Thermal Analyses

CASE	DESCRIPTION	TIME (seconds)	TEMPERATURE	
			°F	(°C)
N1	Blended braking (nominal)	13	559	(293)
L1	Blended braking (limit)	12	598	(314)
N2	Unpowered truck (nominal)	22	877	(469)
L2	Unpowered truck (limit)	19	945	(507)
N3	Motor out (nominal)	22	1072	(578)
L3	Motor out (limit)	19	1157	(625)

3. MODIFICATIONS TO BRAKING SYSTEM

Based on the preliminary results summarized in section 2.4, it became apparent that something more than the dynamic brake retrofit would be required in order to address the thermal cracking problem for the wheels on unpowered trucks and for the motor out situation. In addition, some corrected information was received that relates to the vehicle weight. A second series of analyses was conducted, based on the new information and proposed changes to the braking system that were being evaluated by the railroad.

3.1 Revised Baseline Thermal Analyses

The revised A and B car weights for a married pair were adjusted as follows:

A-car light weight.........................139,500 lbs
B-car light weight.........................117,100 lbs
Passenger weight per car................30,000 lbs

For the married pair, the average weights are thus 128,300 lbs (568 kN) for a light car and 158,300 lbs (701 kN) for a loaded car, and an average vehicle weight of 143,300 lbs (635 kN), whereas the analyses described earlier were based on a vehicle weight of 139,000 lbs (616 kN). Following the preliminary assessment, it was also proposed that the operating speed be limited to 90 mph (144 km/h) instead of the previous 100 mph (160 km/h) limit, at least as a temporary measure, until experience with the modified fleet was gained.

Further, there was under consideration an idea of limiting the speed to 70 mph (112 km/h) in the event of a traction motor failure, which would be conveyed to the train operator via a lighted indicator on his console. The effects of the reduced operating speed for the normal operation and the motor out cases were investigated using the corrected data for the vehicle weights. These cases essentially repeat the three discussed in the preliminary analysis, but will be included here for completeness.

Figures 6, 7, 8, and 9 contain the heat flux versus time plots for the reduced operating speeds for vehicles in the light and loaded configurations, as described above. The procedure for converting the braking profile into heat flux is identical to that described above in section 2.2.

3.2 Results of Revised Baseline Thermal Analyses

The revised heat inputs were applied to the wheel rim via the finite element model as described in section 2. The maximum tread temperature for each of the cases appears in table 3. Each case was run in the light and loaded configurations to establish bounds. These results show that wheel rim temperatures on the unpowered truck can still exceed 800 °F (431 °C).

As a consequence of these results, NJTRO embarked on a search for ways to reduce the wheel thermal loads on the unpowered trucks. Several options were identified as candidates. The wheel model was applied to assess the effectiveness of each option at lowering the rim temperature. Also, the idea of a 70 mph speed restriction in the event of a motor outage was abandoned at this point as impractical, and the railroad instead embarked upon a quality improvement campaign to reduce motor outages.

Table 3. Results of Baseline Scenarios at Reduced Operating Speeds

		MAXIMUM TEMPERATURE °F (°C)	
CASE DESCRIPTION	**MAXIMUM SPEED (mph)**	**LIGHT WEIGHT**	**LOADED WEIGHT**
Powered Truck	90	421 (216)	454 (234)
Unpowered Truck	90	748 (398)	808 (431)
Motor Out	70	647 (342)	700 (371)

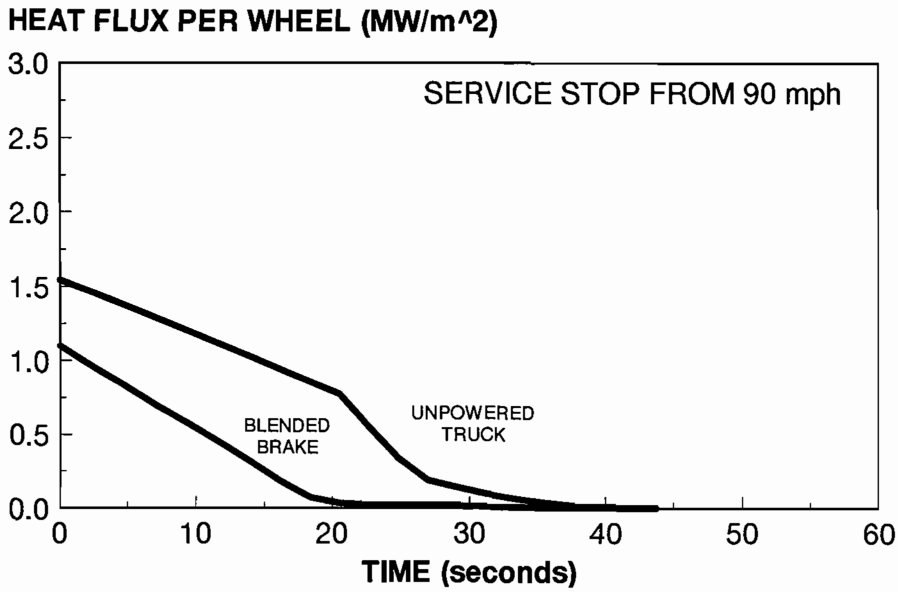


Figure 6. Heat Flux Applied to Wheel Tread on Powered and Unpowered Trucks – LIGHT

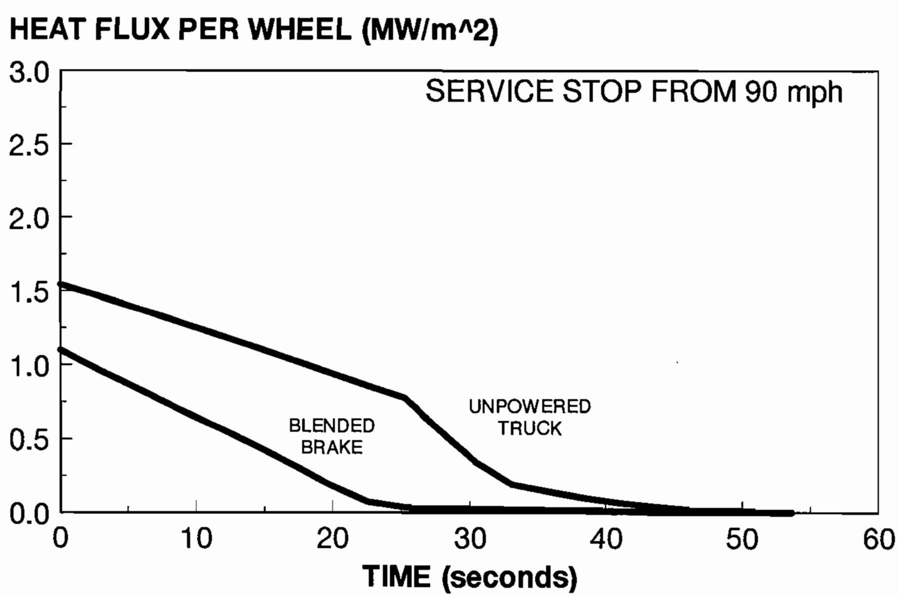


Figure 7. Heat Flux Applied to Wheel Tread on Powered and Unpowered Trucks – LOADED

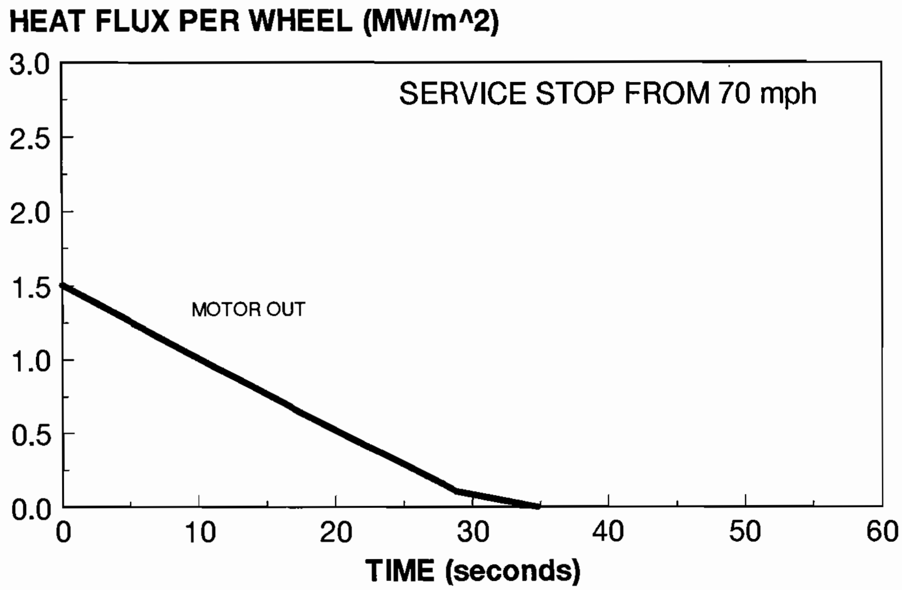


Figure 8. Heat Flux Applied to Wheel Tread When Traction Motor Fails – LIGHT

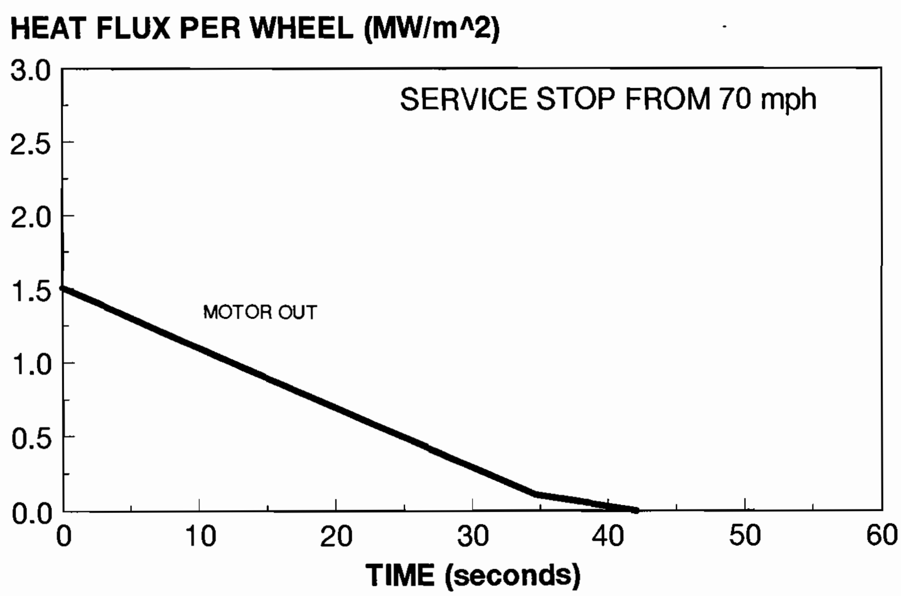


Figure 9. Heat Flux Applied to Wheel Tread When Traction Motor Fails – LOADED

3.3 Augmented Dynamic Braking

The car rebuilder initially estimated that a 30% increase in dynamic braking capability could be obtained via modification of the existing rheostatic grid design. Two alternative options for allocation of the benefit from augmented dynamic braking were investigated.

The first option uses the dynamic brake augmentation to unload the tread brakes on the powered trucks. Figure 10 shows the change in the blending profile, as assumed for the purposes of estimating wheel temperatures. The shaded areas depict the redistribution of effort from the tread to the augmented dynamic braking in the high speed regime. The nominal average retarding force per truck is unchanged, as can be seen by comparing with figure 2(a). Thus, the heat flux at the wheel tread is determined from the power-versus-speed curve for the reduced tread braking contribution (lower boldface curve in figure 10) over the same speed-versus-time profiles calculated for the revised baseline car weights.

Since the above modification does not relieve the wheels on the unpowered truck, this option also provided for auxiliary disc brakes on the unpowered axles. For the purpose of the evaluation, it was assumed that the braking effort on the unpowered truck would be 60% disc and 40% tread.

The second option uses the dynamic augmentation from the three powered trucks to relieve the tread braking effort on the unpowered truck. The relief is effected by a straight BCP reduction valve, rather than a subsystem blending control, such that the unpowered truck retardation force is reduced by the average of the dynamic augmentation forces over the relief speed range.

The tread braking effort on a powered truck is the same as specified for the revised baseline cases. On the unpowered truck, the vehicle retarding force profile depends on the vehicle speed at which the main braking controller begins to regulate the tread brake units on that truck. Two versions were assumed for the purpose of estimating wheel temperatures, as shown in figures 11 and 12.

A new matrix of cases was developed in order to assess the wheel temperatures that would result from the proposed braking system modifications. The modified deceleration profiles were converted into heat flux following the same procedure as outlined during the preliminary investigation, and appear below as figures 13, 14, and 15.

3.4 Results of Thermal Analyses for Augmented Dynamic Braking

Thermal analyses were conducted using the modified heat flux plots as described above. Wheel temperature contour plots for these cases appear in appendix E. The tread surface temperature maxima are summarized in table 4. These data indicate that wheel temperatures do not exceed 600 °F (316 °C) for any of the cases considered.

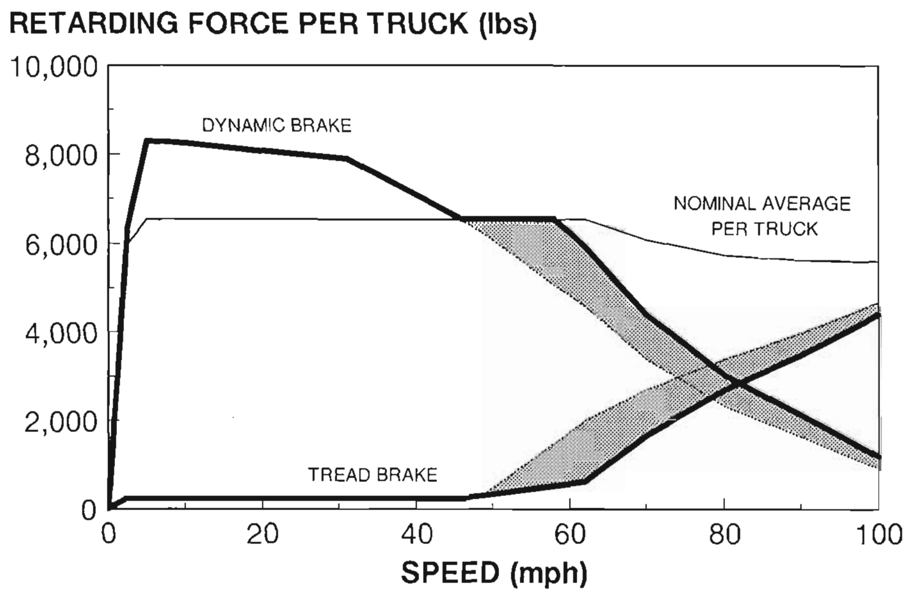


Figure 10. Redistribution of Dynamic Brake Effort to Powered Trucks

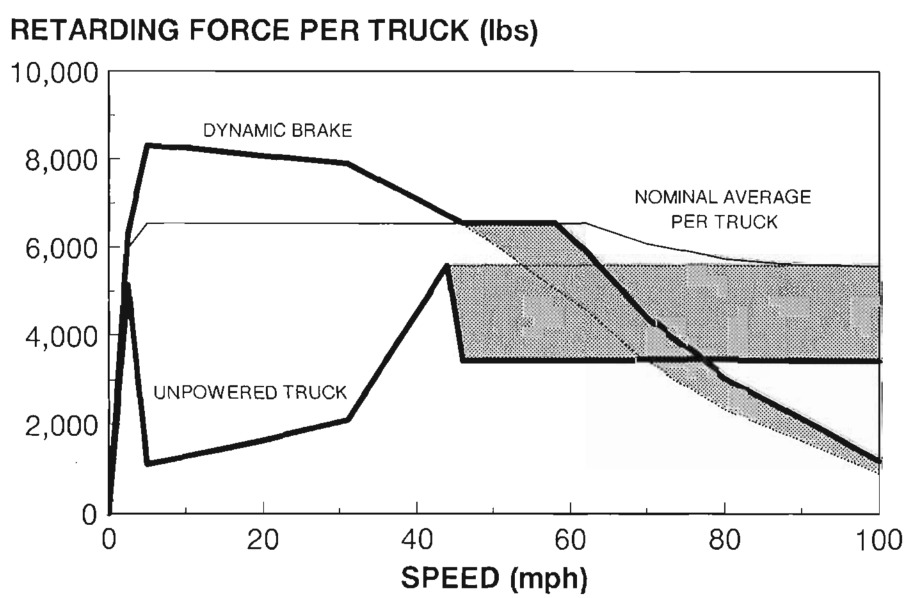


Figure 11. Redistribution of Dynamic Brake Effort to Unpowered Trucks (Version A)

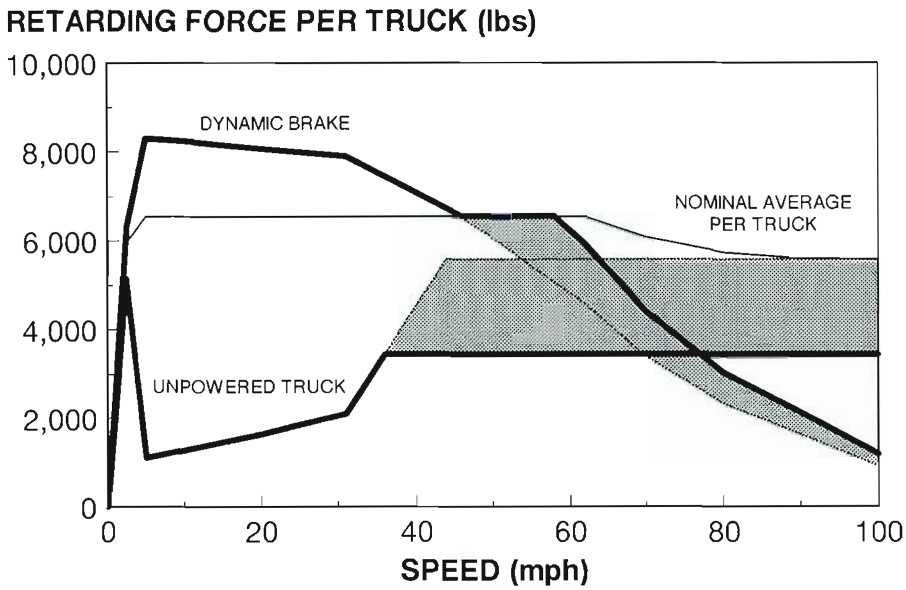


Figure 12. Redistribution of Dynamic Brake Effort to Unpowered Trucks (Version B)

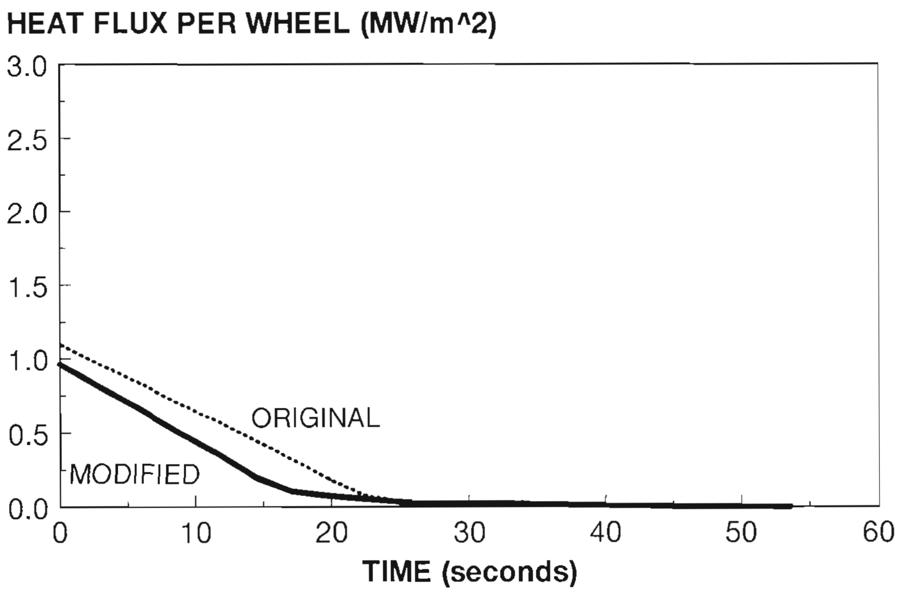


Figure 13. Heat Flux Applied at Wheel Tread for Modified Tread Brake Effort on Powered Trucks (Option 1)

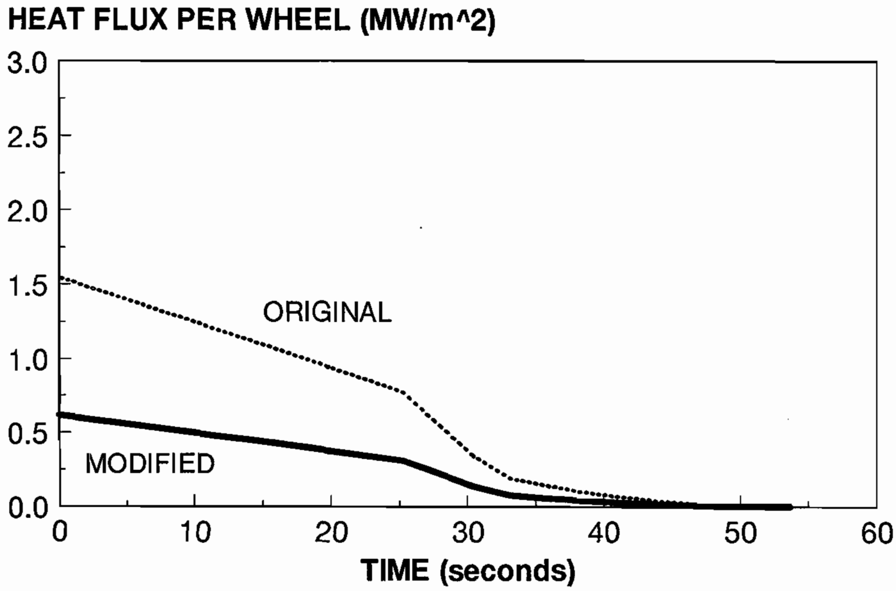


Figure 14. Heat Flux Applied at Wheel Tread for Unpowered Trucks with Disc Brakes (Option 1)

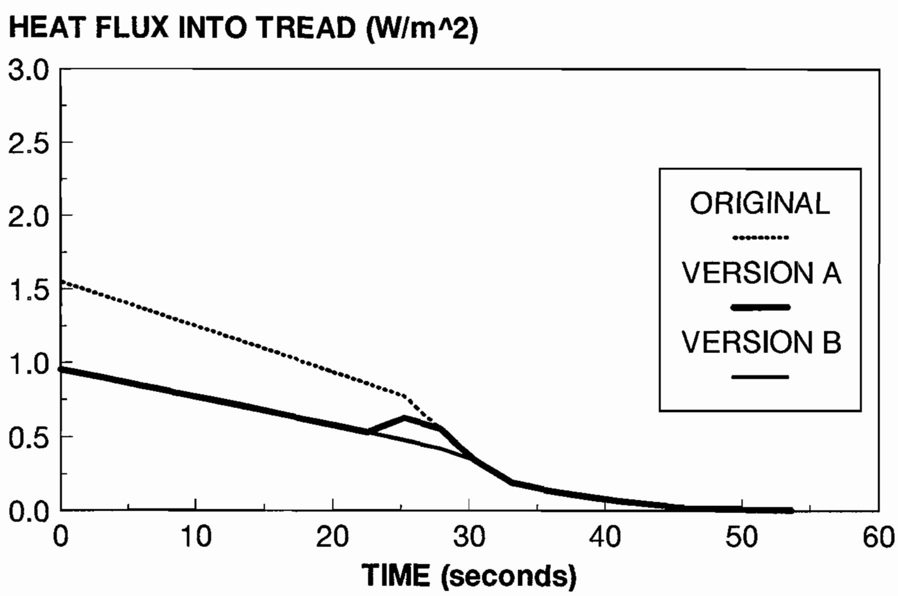


Figure 15. Heat Flux Applied at Wheel Tread for Modified Tread Brake Effort on Unpowered Trucks (Option 2)

Table 4. Results of Thermal Analyses of Brake Effort Redistribution for Loaded Vehicle Configuration

CASE DESCRIPTION	MAXIMUM SPEED	MAX. TEMP. °F	(°C)
Powered truck: tread brakes relieved by dynamic augment between 100 mph and 60 mph	90 mph	369	(187)
Unpowered truck with auxiliary discs that pick up 60% of retarding force	90 mph	370	(188)
Unpowered truck: tread brakes relieved by dynamic augment (Version A)	90 mph	552	(289)
Unpowered truck: tread brakes relieved by dynamic augment (Version B)	90 mph	525	(274)

3.5 Final Configuration and Thermal Analysis

After reviewing analysis and operational test results, NJTRO chose to proceed with Option 2, but the final configuration differed from that described earlier in two respects. First, the car rebuilder's modified rheostatic system design provided 26%, rather than the initially estimated 30%, dynamic braking augmentation. Second, the braking effort was balanced for all four trucks with a common brake cylinder pressure, i.e., no reduction valve for the unpowered truck. The solid curve in figure 16 shows the finally selected dynamic brake retarding force profile. The dashed curve is the original profile from figure 2 shown for comparison. Figure 17 illustrates the same data for the powered and unpowered trucks, with the original profiles shown as dashed lines.

The final configuration was analyzed for a full stop from 100 mph (160 km/h). Figure 18 shows a plot of the heat flux input to the wheel rim (on all trucks) versus time. For the final configuration, the applied flux at the start of the braking event (time = 0 seconds) is 1.84 MW/m^2 for the powered truck, and 1.35 MW/m^2 for the unpowered truck. The calculated maximum wheel tread temperatures are 545 °F (285 °C) for the powered truck, and 777 °F (414 °C) for the unpowered truck.

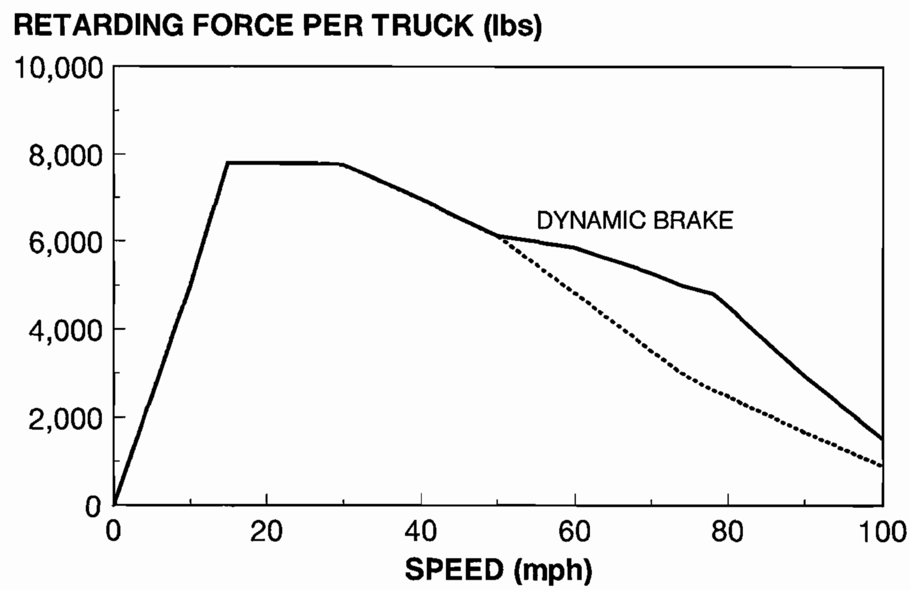


Figure 16. Dynamic Brake Retarding Force per Truck Corresponding to 26% Increase in Dynamic Capability

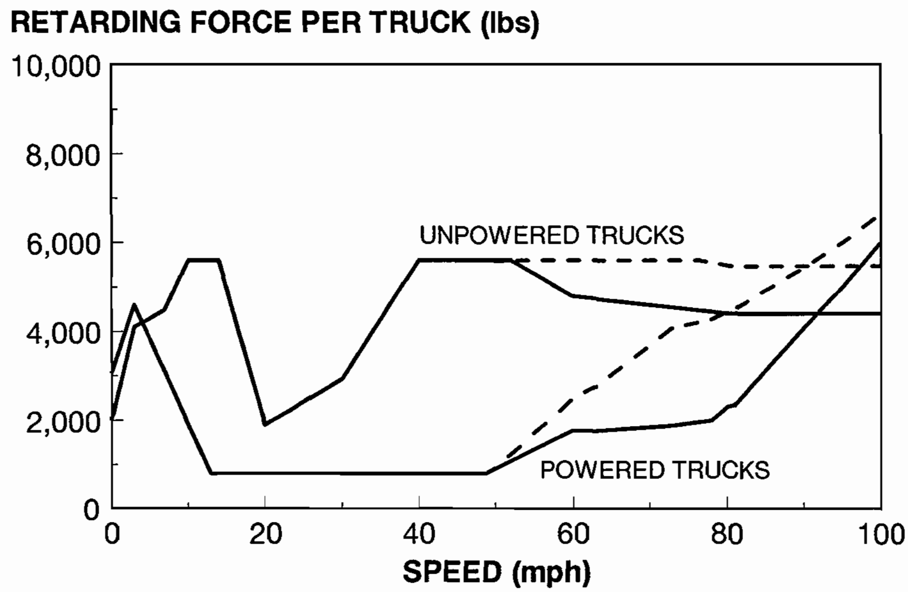


Figure 17. Tread Brake Retarding Force per Truck after Redistribution of Dynamic Augment to Powered and Unpowered Trucks

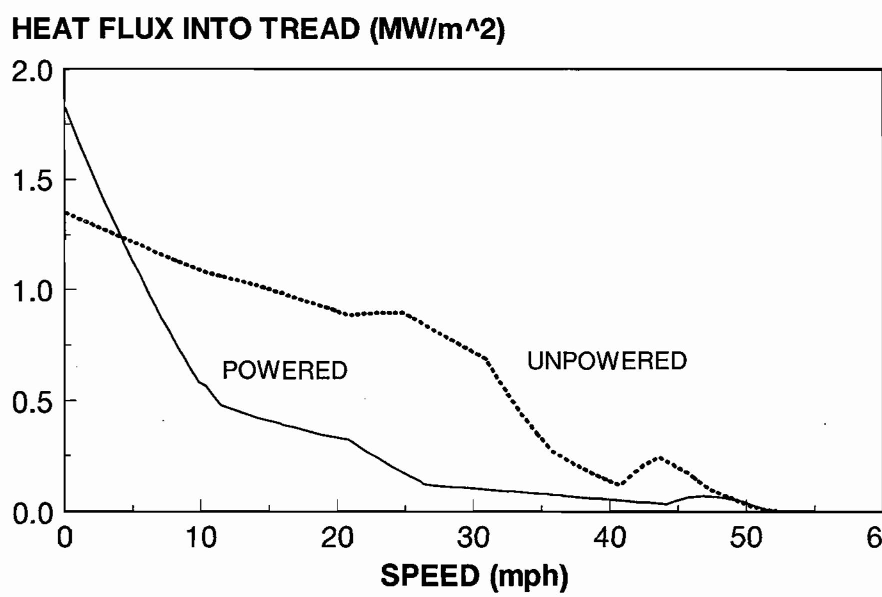


Figure 18. Heat Flux Input into Wheel Rim for Service Stop from 100 mph (160 km/h) for Powered and Unpowered Trucks

4. CONCLUSIONS

Several alternative options were evaluated for equipping a fleet of multiple unit (MU) power cars with auxiliary brakes in conjunction with a planned upgrade from DC to AC motors. Because the new motors are more powerful than the old motors, the owning railroad decided to reconfigure each married pair of cars to have one unpowered and three powered trucks. Therefore, the options under consideration involved dynamic braking on the powered trucks, disc brakes on the unpowered truck, and the distribution of tread braking relief among the four trucks.

The purpose of the evaluation was to assess the effects of the various options on wheel temperatures produced by stop braking. The objective was to find the simplest configuration that would allow 100 mph operation without overheating the wheels, in order to avoid thermal cracking problems, which were encountered in 100 mph operation with unassisted tread brakes. The evaluation was conducted by means of heat transfer analyses, using a model previously developed for this wheel and calibrated by correlation with measurements from an instrumented operational test.

The final configuration, which is being retrofitted into the fleet, consists of auxiliary dynamic brakes on the three powered trucks supplementing the tread brakes on all four trucks. A resistor grid design originally developed for use with the less powerful DC motors was upgraded to obtain 26% augmentation of the dynamic braking effort in the upper half of the operating speed range. The tread brakes are blended on the powered trucks and set on the unpowered truck to provide balanced braking for the married pair of cars.

The heat transfer analysis results show that, for a full stop from 100 mph (160 km/h), the expected maximum wheel rim temperatures are below 600 °F (316 °C) on the powered trucks and below 800 °F (427 °C) on the unpowered truck. These improvements are sufficient to warrant 100 mph (160 km/h) operation under normal conditions.

APPENDIX A.

Temperature-Dependent Material Properties from [3] and Other Sources

MATERIAL PROPERTY			TEMPERATURE (°C)							
SYMBOL	NAME	UNITS	0	350	703	704	710	800	950	1200
k	Thermal conductivity	W/m °C	51.46	40.88	30.21	30.18	30.00	25.00	27.05	30.46
c_p	Specific heat	J/kg °C	511.5	629.5	744.5	652.9	653.2	657.7	665.2	677.3

MATERIAL PROPERTY			TEMPERATURE (°C)							
SYMBOL	NAME	UNITS		20	268	400	500	700	900	
E	Young's modulus	GPa		202.	194.	190.	172.	117.	43.	
υ	Poisson's ratio	------		0.295	0.309	0.316	0.322	0.333	0.345	
σ_{ys}	Yield strength	MPa		400.	386.	345.	263.	100.	60.	
E_k	Plastic modulus	GPa		15.0	22.2	18.5	14.0	5.1	0.9	

MATERIAL PROPERTY			TEMPERATURE (°C)							
SYMBOL	NAME	UNITS	0	703	720	776	832	888	944	1000
α	Coeff. thermal expansion	10E-06/°C	9.70	13.2	13.35	13.97	14.51	14.98	15.40	15.77

33
34X

APPENDIX B.

Sample Calculation Using MATHCAD PLUS 5.0©
to Convert Braking Retardation Force into Heat Flux

Dynamic brake power calculations including train resistance effects

POWERED TRUCK / NOMINAL CASE / [BLENDED BRAKING]

ORIGIN := 1
TOL := 0.0001

Initialize variables

car weight	$W := 139000$
number of axles per car	$nax := 4$
weight per axle (tons)	$Wax := \frac{W}{(nax \cdot 2000)}$
accel. in mph/sec	$g := 32.2 \cdot \left(\frac{3600}{5280}\right)$
half car mass	$C := 2 \cdot \frac{g}{W}$
Davis scale factor equals 1 to include Davis resistance, 0 to exclude	$davis\_fac := 1.0$
number of wheels per truck	$nwheels := 4$
Nominal/limit scale factor used for scaling for emergency brake rate	$nomlim\_fac := 1.00$

Calculate effective shoe contact area

shoe width (inches)	$w := 2.5$
wheel diameter (inches)	$d := 32$
contact area (meters**2)	$area := \frac{(\pi \cdot d \cdot w)}{39.370079^2}$

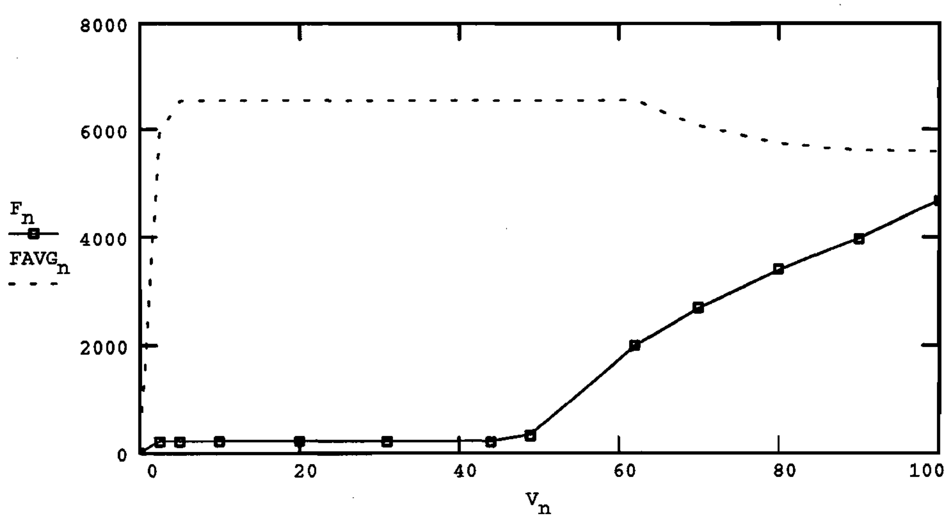
Tabulate data from retarding force versus speed plot

initialize counter $\quad n := 1 .. 13$

retarding force per powered truck (lbs)	speed (mph)	average retarding force for all trucks (lbs)

$$F := \begin{bmatrix} 0 \\ 234 \\ 234 \\ 234 \\ 234 \\ 234 \\ 234 \\ 351 \\ 1989 \\ 2691 \\ 3393 \\ 3978 \\ 4680 \end{bmatrix} \qquad V := \begin{bmatrix} 0 \\ 2.5 \\ 5 \\ 10 \\ 20 \\ 31 \\ 44 \\ 49 \\ 62 \\ 70 \\ 80 \\ 90 \\ 100 \end{bmatrix} \qquad FAVG := \begin{bmatrix} 0 \\ 5967 \\ 6552 \\ 6552 \\ 6552 \\ 6552 \\ 6552 \\ 6552 \\ 6552 \\ 6084 \\ 5733 \\ 5616 \\ 5600 \end{bmatrix}$$

Plot of FAVG and F as function of speed



Initialize counters

$$i := 0 .. 100$$

$$j := 100, 95 .. 0$$

$$speed(i) := i$$

Interpolate continuous values of friction force, and adjust by nomlim_fac if necessary

$$fric\_brake\_force(v) := linterp(V, F, v) \cdot nomlim\_fac$$

Calculate Davis resistance as a function of speed. (R(v) in lbs/ton)

$$R(v) := 1.3 + \left(\frac{29}{Wax}\right) + 0.045 \cdot v + \frac{\left(0.07 \cdot v^2\right)}{(Wax \cdot nax)}$$

Scale R(v) so that r(v) is Davis train resistance in lbs per truck

$$r(v) := R(v) \cdot \left(\frac{W}{2000 \cdot 2}\right) \cdot davis\_fac$$

Interpolate continuous values of FAVG and adjust FAVG to include contribution of Davis resistance in deceleration and stop time calculations

$$favgtrk(v) := linterp(V, FAVG, v) \cdot nomlim\_fac$$

$$decel(v) := (favgtrk(v) + r(v)) \cdot C$$

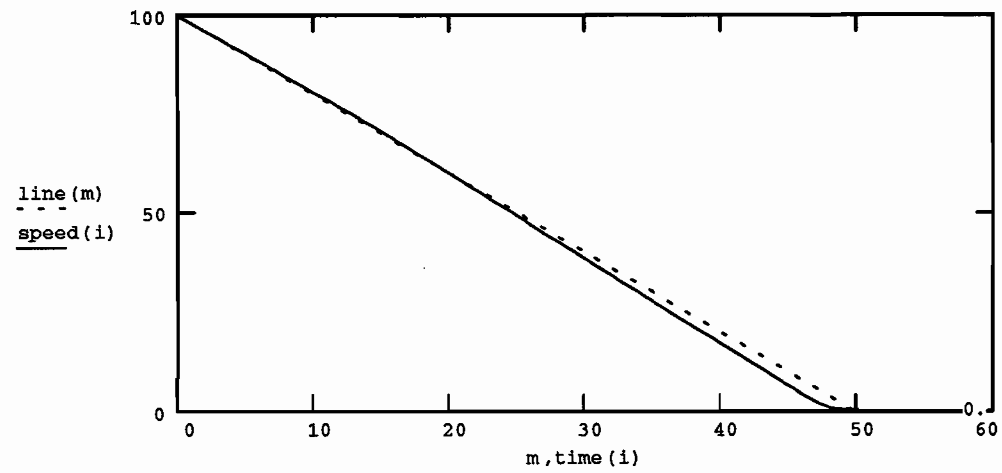
$$time(i) := \int_{speed(i)}^{100} \frac{1}{decel(v)} \, dv$$

Draw a straight line for comparison with deceleration trace

$$m := 0 .. 50$$

$$line(m) := 100 - (2 \cdot m)$$

Plot deceleration as speed versus time:



stop time is time(0): time(0) = 52.4153

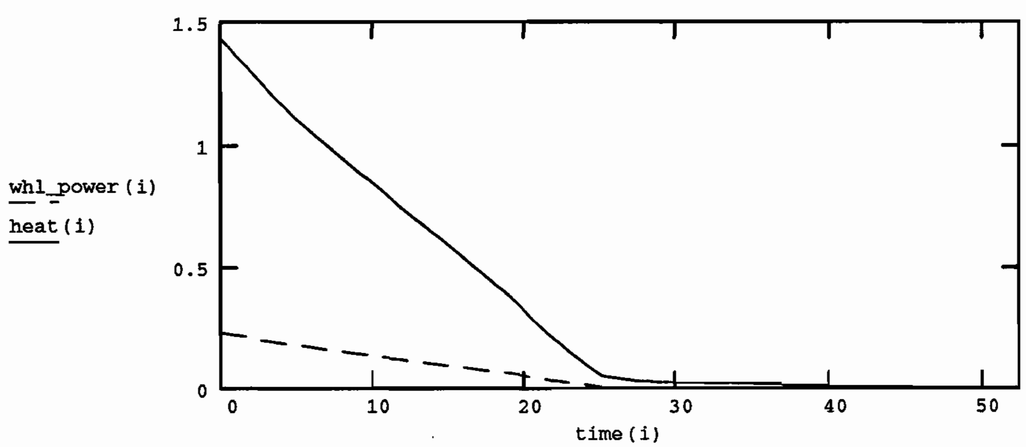
Calculate instantaneous power in MWatts

$$whl\_power(i) := \left[\frac{fric\_brake\_force(i)}{nwheels}\cdot speed(i)\cdot\left(\frac{5280}{3600}\right)\cdot\frac{746}{550}\cdot 10^{-6}\right]$$

Convert power to heat flux in MWatts/m**2

$$heat(i) := \frac{whl\_power(i)}{area}$$

Plot variables as functions of time:



Display data every 5 mph

speed(j)	time(j)	whl_power(j)	heat(j)
100	0	0.2328	1.4354
95	2.5561	0.2045	1.2614
90	5.1258	0.1781	1.0981
85	7.6981	0.1558	0.9609
80	10.2617	0.135	0.8326
75	12.7921	0.1135	0.6998
70	15.2666	0.0937	0.5778
65	17.6656	0.0728	0.449
60	19.9809	0.0518	0.3197
55	22.2879	0.0303	0.1867
50	24.6039	0.0119	0.0732
45	26.9288	0.0058	0.0355
40	29.2615	0.0047	0.0287
35	31.602	0.0041	0.0251
30	33.9491	0.0035	0.0215
25	36.303	0.0029	0.0179
20	38.6631	0.0023	0.0144
15	41.0274	0.0017	0.0108
10	43.3968	0.0012	0.0072
5	45.7701	0.0006	0.0036
0	52.4153	0	0

APPENDIX C.

Temperature-Time Histories of Three Nodes Near Tread Surface for Preliminary Cases

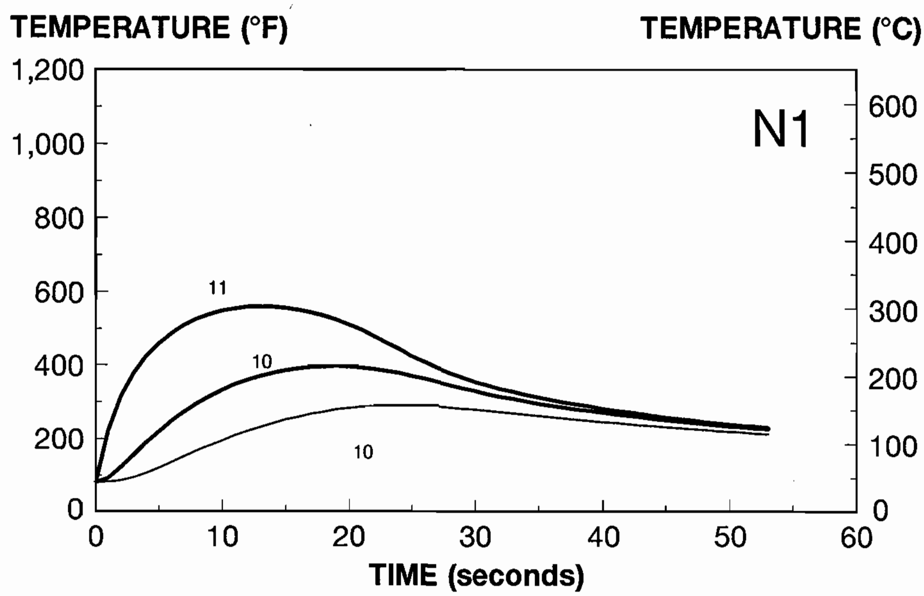


Figure C-1. Temperature-Time History of Three Nodes Near Tread Surface for Blended Braking Case at Nominal Deceleration Rate

42

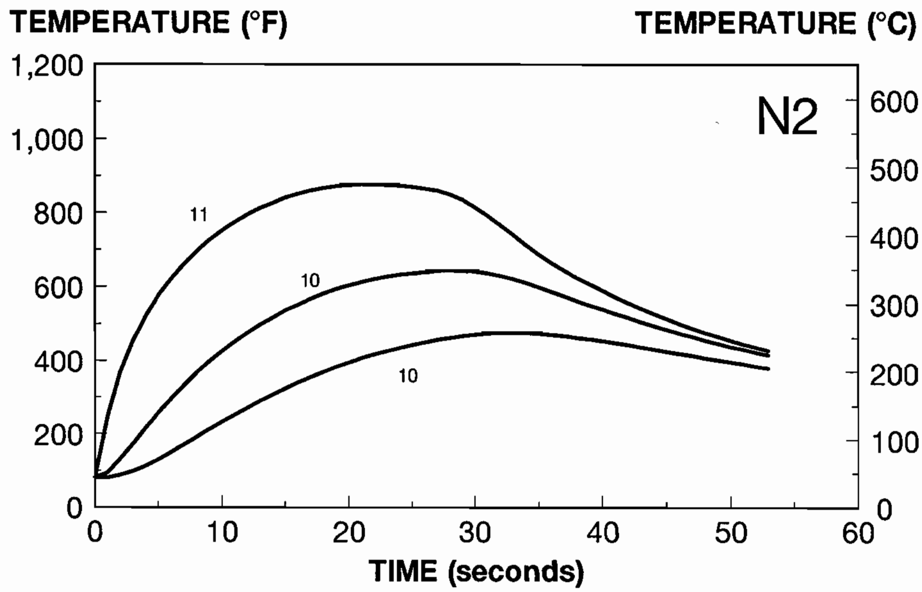


Figure C-2. Temperature-Time History of Three Nodes Near Tread Surface for Unpowered Truck Case at Nominal Deceleration Rate

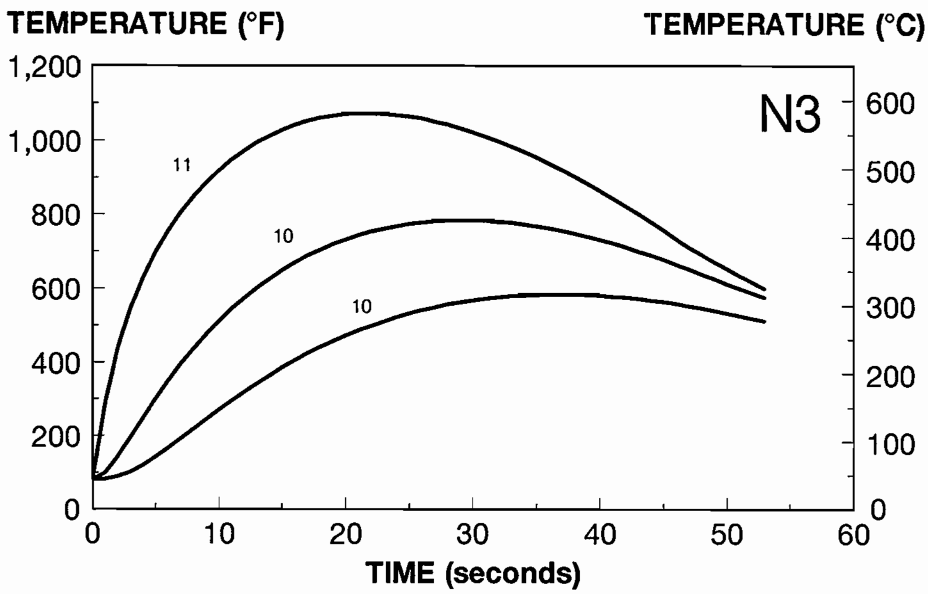


Figure C-3. Temperature-Time History of Three Nodes Near Tread Surface for Motor Out Case at Nominal Deceleration Rate

44

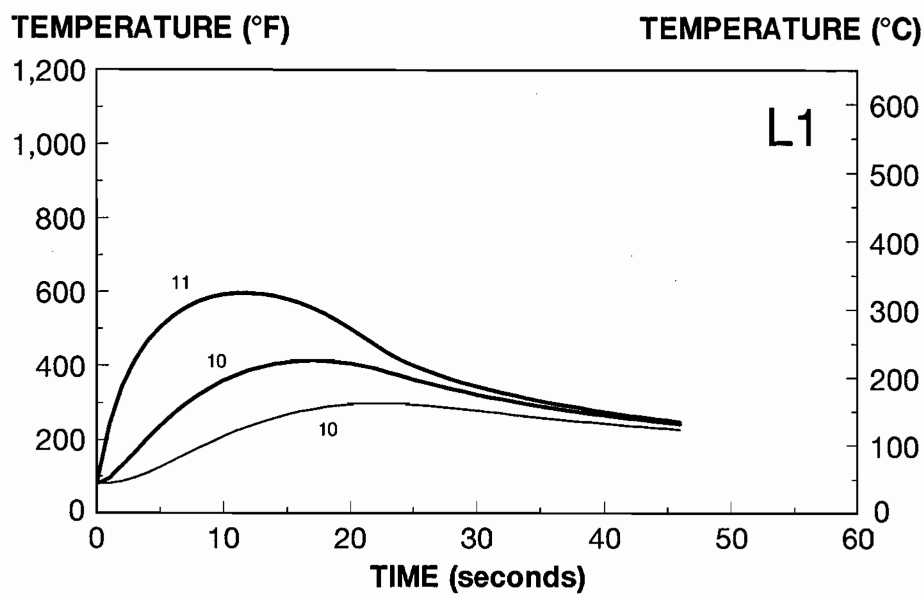


Figure C-4. Temperature-Time History of Three Nodes Near Tread Surface for Blended Braking Case at Limit Deceleration Rate

45

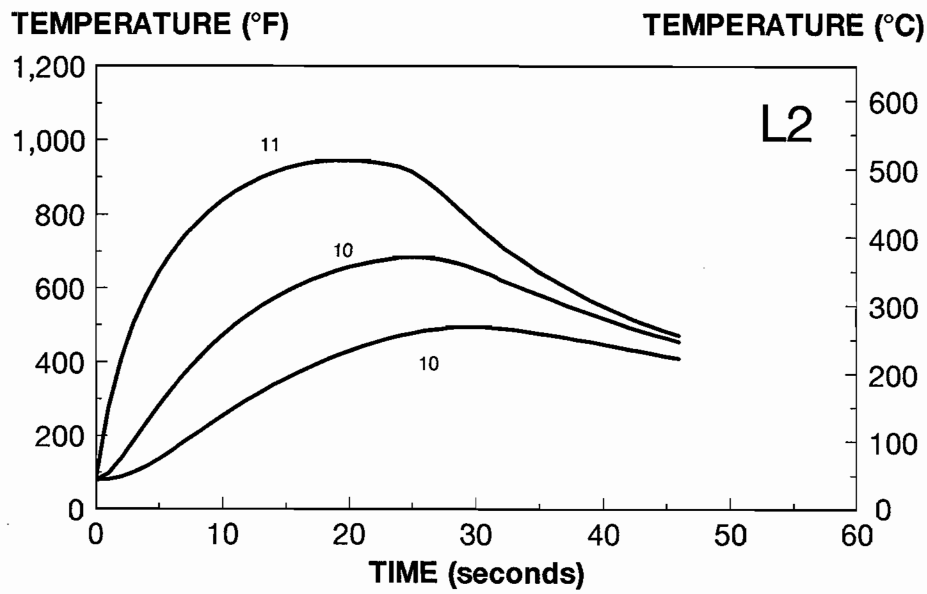


Figure C-5. Temperature-Time History of Three Nodes Near Tread Surface for Unpowered Truck Case at Limit Deceleration Rate

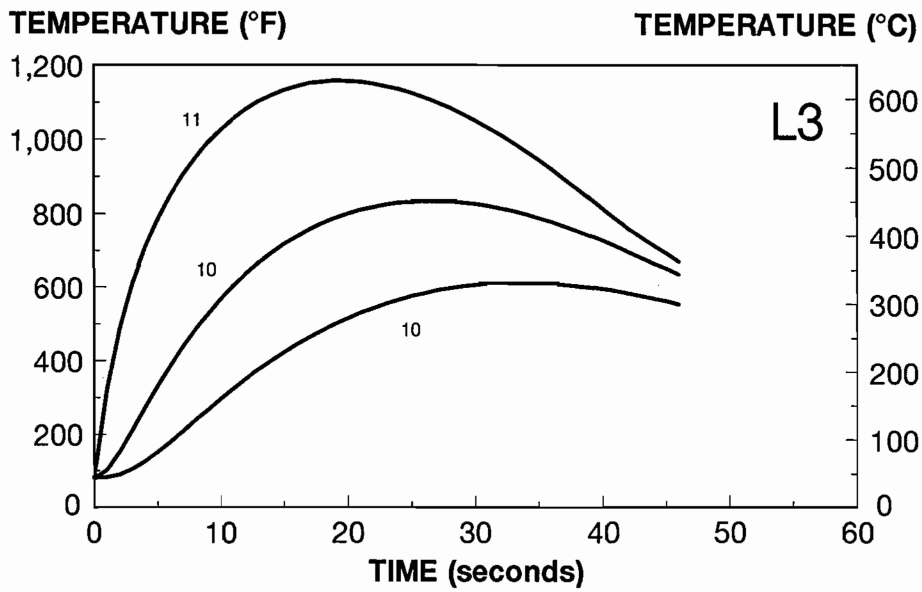


Figure C-6. Temperature-Time History of Three Nodes Near Tread Surface for Motor Out Case at Limit Deceleration Rate

APPENDIX D.

Contour Plots of Temperature in Wheel When Maximum Temperature on Tread Is Attained for Preliminary Analyses

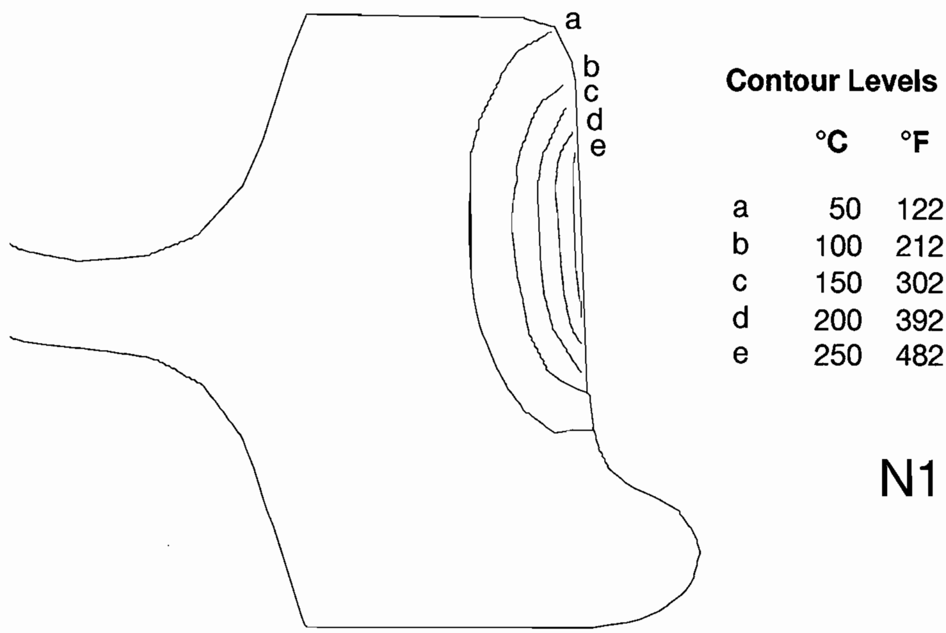


Figure D-1. Contour Plot of Temperature When Maximum Is Attained for Blended Braking Case at Nominal Deceleration Rate

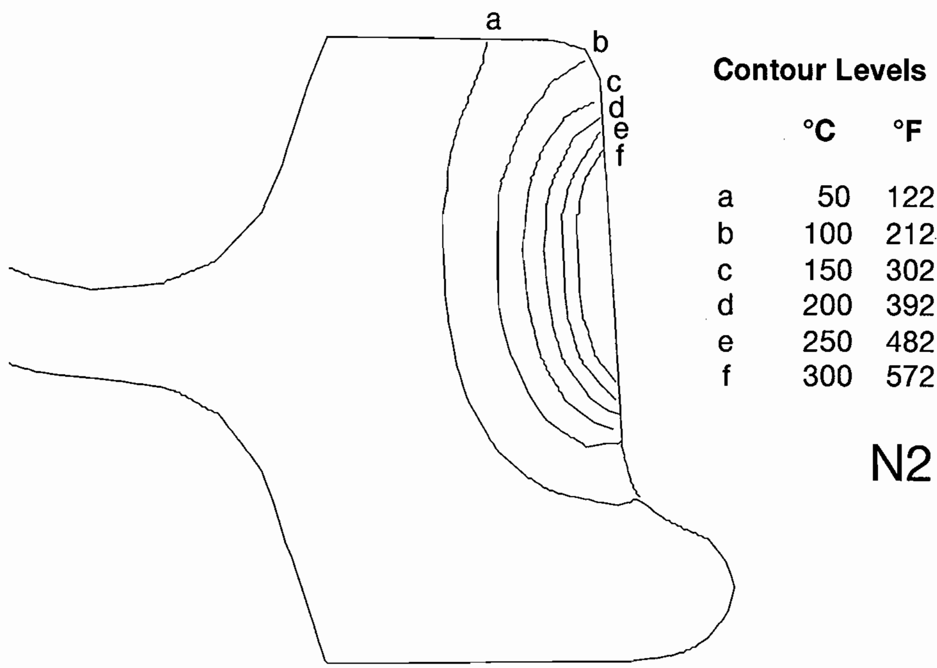


Figure D-2. Contour Plot of Temperature When Maximum Is Attained for Unpowered Truck Case at Nominal Deceleration Rate

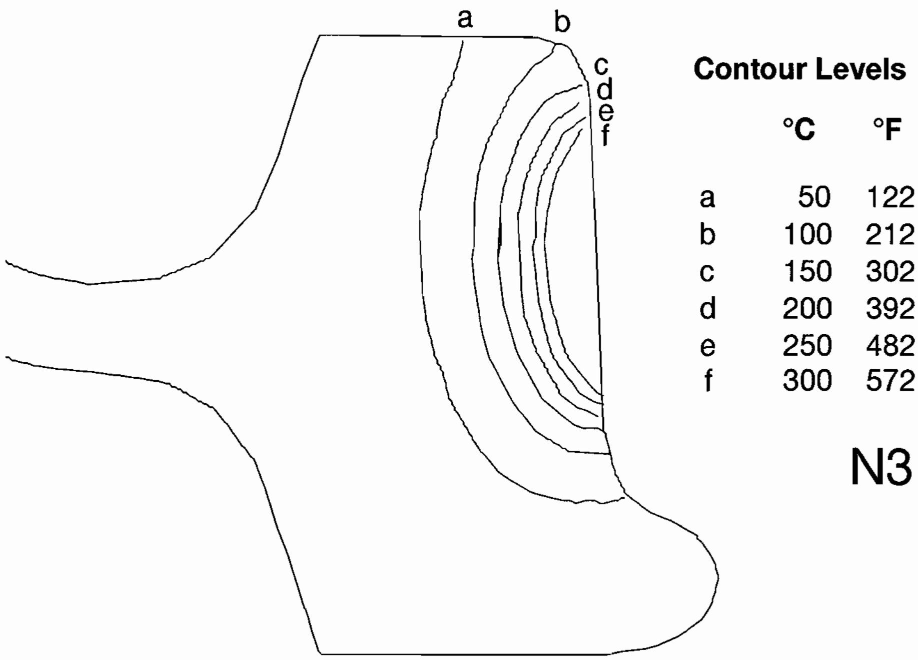


Figure D-3. Contour Plot of Temperature When Maximum Is Attained for Motor Out Case at Nominal Deceleration Rate

52

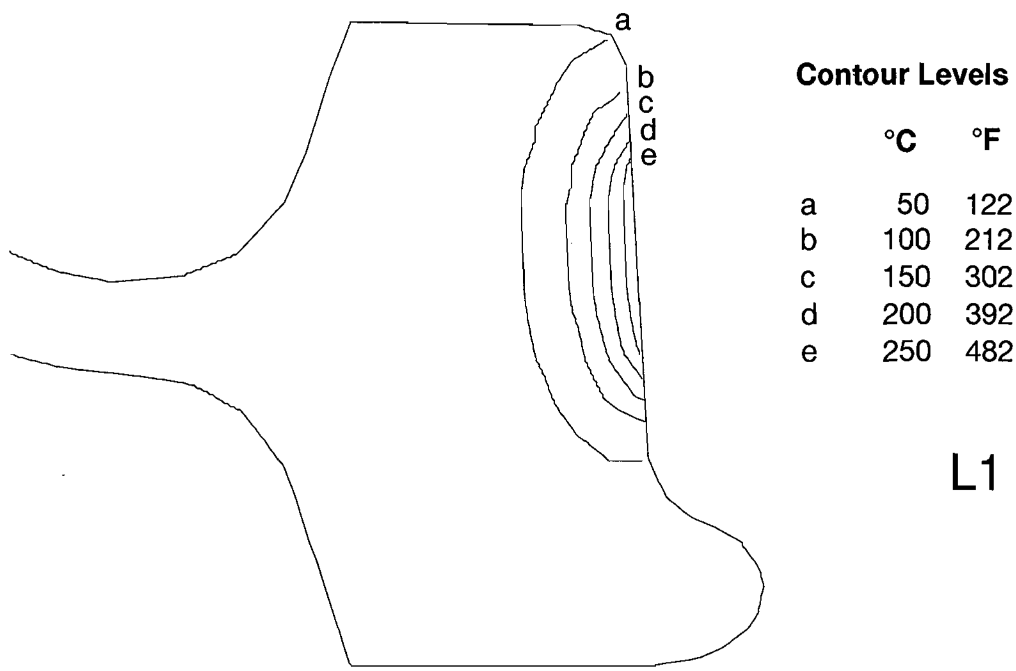


Figure D-4. Contour Plot of Temperature When Maximum Is Attained for Blended Braking Case at Limit Deceleration Rate

53

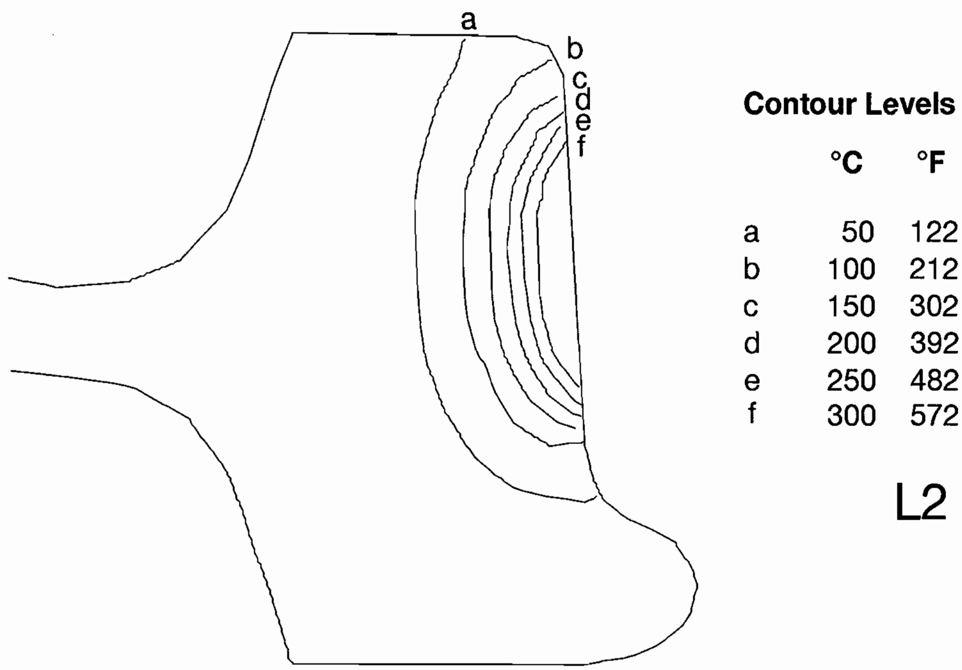


Figure D-5. Contour Plot of Temperature When Maximum Is Attained for Unpowered Truck Case at Limit Deceleration Rate

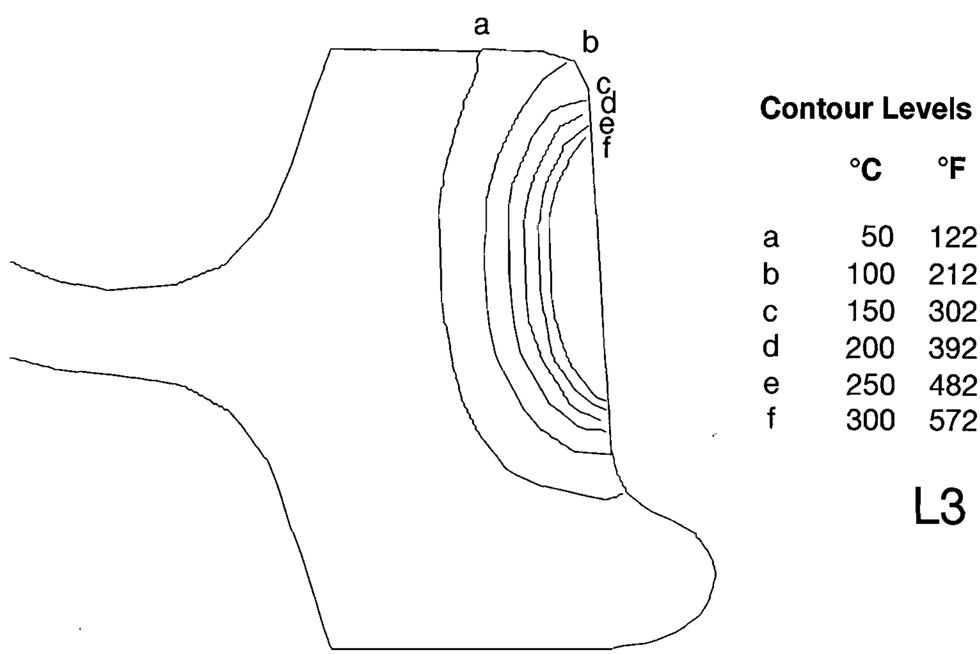


Figure D-6. Contour Plot of Temperature When Maximum Is Attained for Motor Out Case at Limit Deceleration Rate

APPENDIX E.

Contour Plots of Temperature in Wheel When Maximum Temperature on Tread Is Attained for Modified Braking System

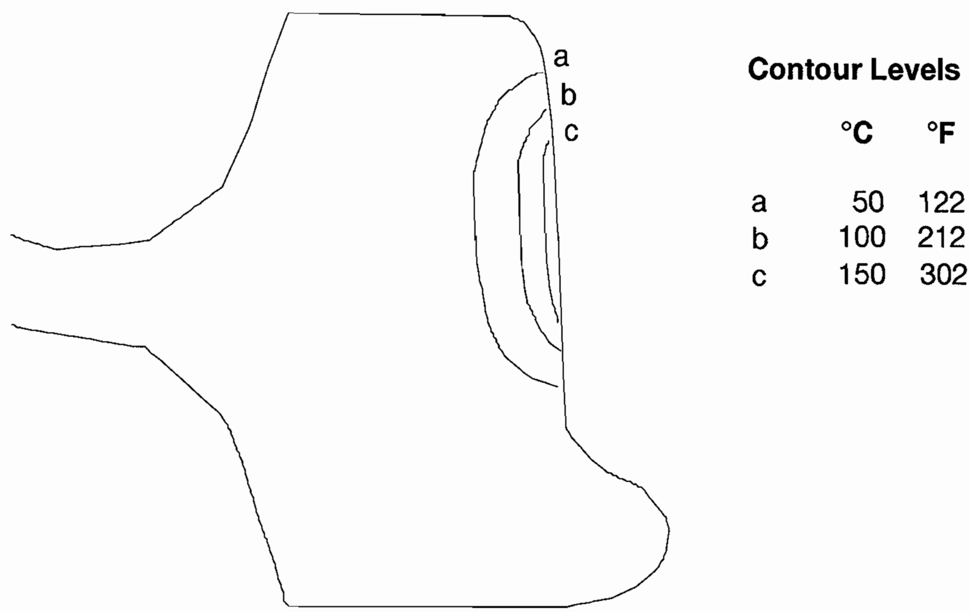
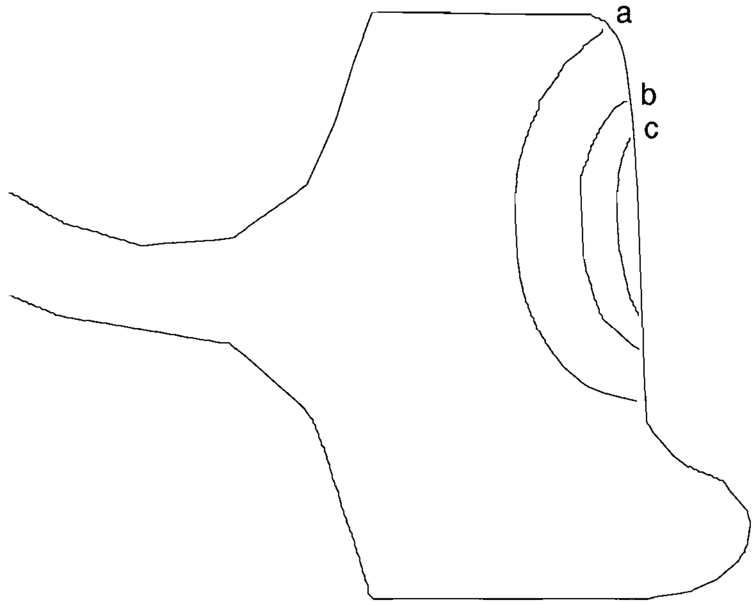


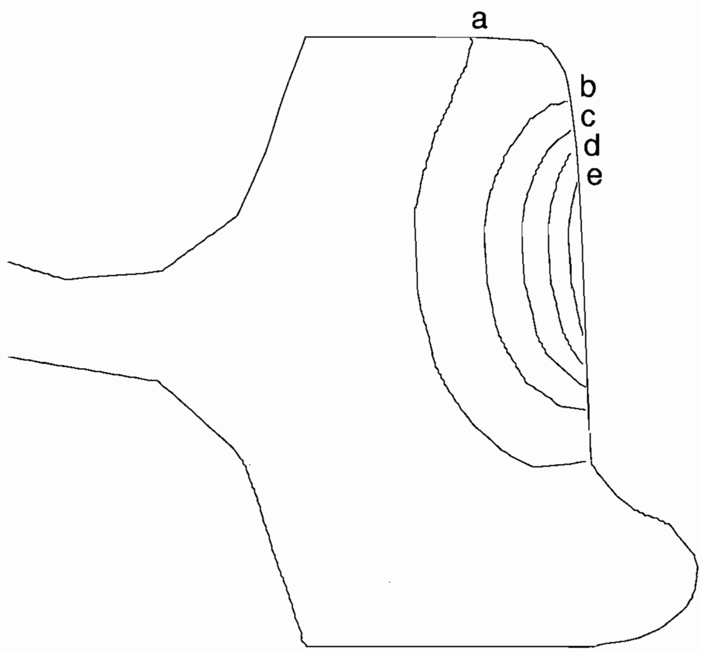
Figure E-1. Powered Truck: Tread Brakes Relieved by Dynamic Augment between 100 mph (160 km/h) and 60 mph (96 km/h)



Contour Levels

	°C	°F
a	50	122
b	100	212
c	150	302

Figure E-2. Unpowered Truck with Auxiliary Discs That Pick Up 60% of Retarding Force

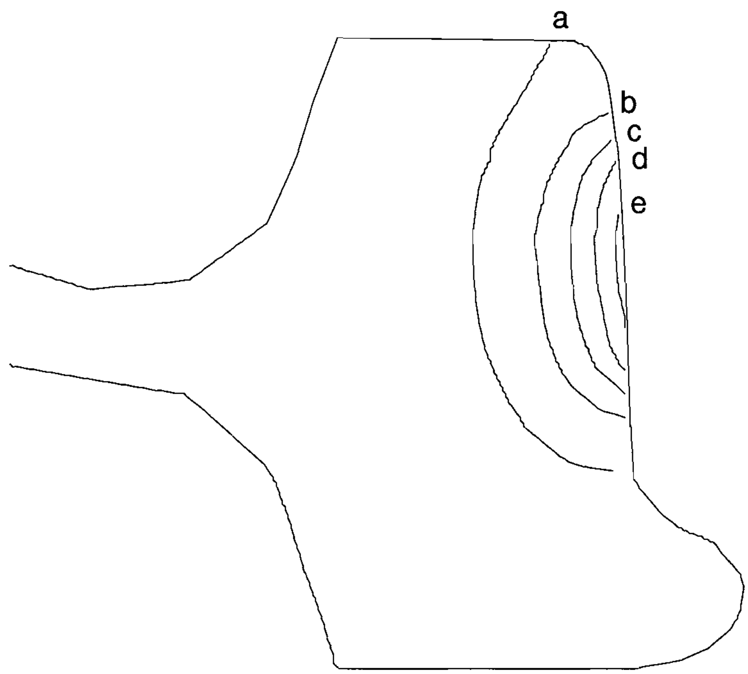


Contour Levels

	°C	°F
a	50	122
b	100	212
c	150	302
d	200	392
e	250	482

Figure E-3. Unpowered Truck: Tread Brakes Relieved by Dynamic Augment (Version A)

60



Contour Levels

	°C	°F
a	50	122
b	100	212
c	150	302
d	200	392
e	250	482

Figure E-4. Unpowered Truck: Tread Brakes Relieved by Dynamic Augment (Version B)

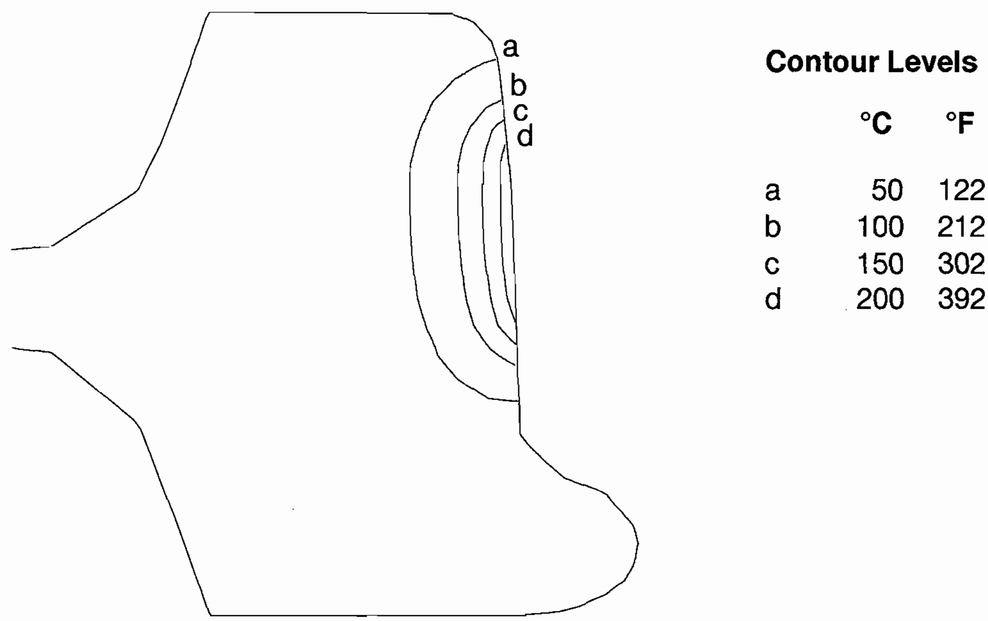


Figure E-5. **Twenty-Six Percent Dynamic Augment Distributed to Powered and Unpowered Trucks, Temperature Distribution in Powered Truck Wheel after Service Stop from 100 mph (160 km/h)**

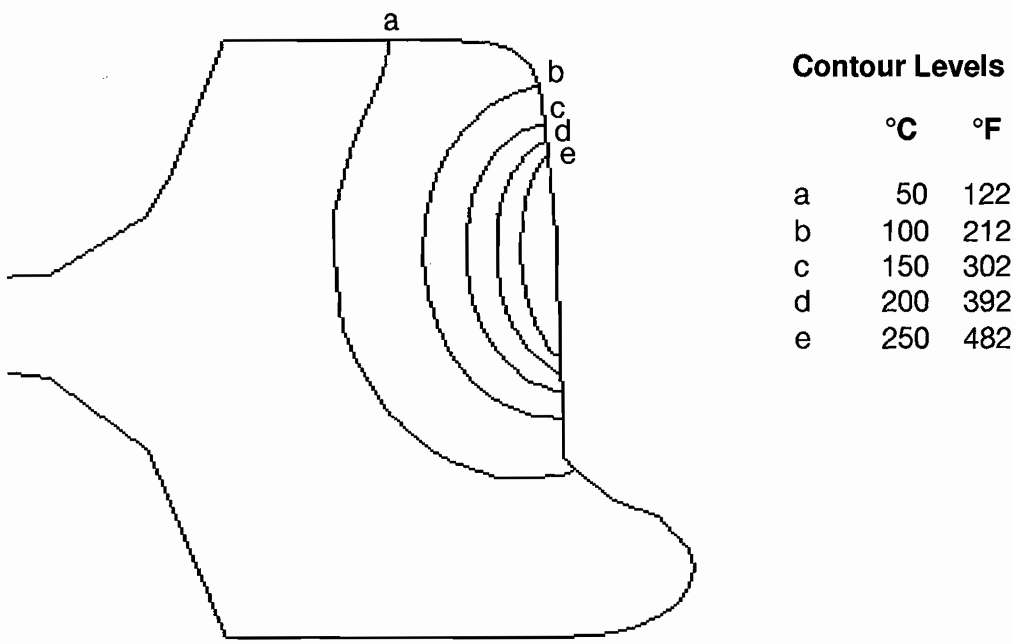


Figure E-6. Twenty-Six Percent Dynamic Augment Distributed to Powered and Unpowered Trucks, Temperature Distribution in Unpowered Truck Wheel after Service Stop from 100 mph (160 km/h)

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67x
68x