U.S. Department
of Transportation
**Federal Railroad
Administration**

Conceptual Designs of a Passenger Rail Car Brake Shoe to Enhance Heat Dissipation

Working Paper
DOT-VNTSC-RR528-WP-96-1
April 1996

Approved for Distribution:

Division Chief

Research and Special Programs
Administration
John A. Volpe National
Transportation Systems Center
Cambridge MA 02142-1093

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PREFACE

The work presented in this report was performed by Technology & Management Systems, Inc. (TMS), under Contract No. DTRS-57-93-C-00040, Technical Task Directive No. 2, from the Volpe National Transportation Systems Center (Volpe Center), U.S. Department of Transportation, Cambridge, MA 02142.

Dr. Phani K. Raj was the Project Manager at TMS. The project team included Dr. Keith E. Crowe, who performed a significant part of the technical work, and Ms. Tamara L. DeGray, who helped in the organization and preparation of the report.

This report is the ninth in a series of engineering studies on rail vehicle wheel performance. The series was started in response to high rates of cracking observed in the wheels of certain multiple unit (MU) power cars used in commuter service. One of the MU fleets, owned by New Jersey Transit Rail Operations (NJTRO), was originally equipped with tread brakes only and was operated at speeds up to 100 mph. Although that fleet is now being retrofitted with blended dynamic brakes, there was initially a strong interest in making improvements to the brake shoe as a means of reducing wheel tread temperature. NJTRO worked with a brake shoe supplier to reformulate the material composition, and one of the modified compositions was found to provide some temperature reduction without any sacrifice of stopping distance.

This report summarizes the results of a conceptual design study that was conducted to assess the potential for improved cooling efficiency at the interface between the brake shoe and wheel tread. The principles of heat transfer were applied to evaluate several candidate design concepts involving changes of brake system configuration and/or brake shoe material to reduce the amount of heat conducted into the wheel rim. Three of the four concepts evaluated show some promise.

In view of the facts that the NJTRO fleet is being upgraded to blended dynamic brakes, and that the other MU fleets are already so equipped, there is no longer any practical reason to consider the design concepts presented here for the MU application. However, it was deemed worthwhile to document the concept study in order to provide a source of ideas, should there arise a need in the future to extend the safe range of tread brake operation for high speed rail applications.

The TMS team acknowledges with gratitude the illuminating technical discussions with Dr. Oscar Orringer of the Volpe Center. Dr. Orringer, who was the COTR on this project, provided valuable technical direction. We are also indebted to Mr. Jeffrey Gordon and Ms. Yim Har Tang of the Structures and Dynamics Division for their draft report reviews and informative comments.

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1. INTRODUCTION

1.1 BACKGROUND

The recent trend in the transit industry is to increase the speed of commuter and intra-city trains so that more trains can be run on the same track within a given period. The result of higher operating speeds is that a train's brake shoes wear out faster requiring more frequent maintenance and consequent higher costs. Therefore, any brake design that reduces the peak temperature attained by the brake shoe (without compromising its function) prolongs the shoe life. This will benefit the transit industry significantly. The study described in this report was undertaken to analyze various brake shoe design improvement options.

High temperature at the brake shoe-wheel rim interface results when a rapid reduction of the railcar velocity (from a relatively high velocity) is achieved by stop braking. The effects of high temperature are multifold. First, it creates severe thermal stresses in the wheel rim. Second, if the temperature is very high, it may affect wheel physical properties. Third, because of the potential softening of the brake shoe material at elevated temperatures and the abrasive action of the wheel, a higher wear rate of the shoe may result. It is therefore desirable to reduce the shoe-wheel interface temperature. One potential method of accomplishing this is to increase the heat dissipation rate through the brake shoe. That is, it is postulated that by proper design of the shoe (modifying both physical and material characteristics) it may be possible to extract heat out of the interface at a higher rate and facilitate its dissipation to the environment.

In this project, several different physical brake shoe design concepts were investigated, theoretically, to evaluate the improvement in heat transfer associated with each. Our analysis was limited to heat transfer improvements arising from physical changes to the brake shoe geometry only, and did not include any material property modifications. The designs considered are discussed in this report.

1.2 PROJECT OBJECTIVES

The project objectives were to develop and analyze different brake shoe heat transfer enhancement concepts without sacrificing the braking performance characteristics.

1.3 SCOPE OF WORK

The scope of work in this project included:

- developing several conceptual designs to enhance brake shoe heat dissipation;
- analyzing the magnitude of heat transfer effects in the new design(s); and
- evaluating the potential unknowns and developing approaches to conduct feasible experiments for more detailed evaluations.

1.4 REPORT ORGANIZATION

In this report, we discuss three conceptual brake shoe designs in section 2. After a brief discussion of the problem description and brake shoe material, in sections 2.1 and 2.2, respectively, the specific designs are presented. The first design, the groove fin, is discussed in section 2.3. The external fin design is evaluated in section 2.4. The water jacket is analyzed in section 2.5. Finally, a direct water spray cooling approach is considered in section 2.6. Simple experiments to test certain concepts are discussed in section 3 and conclusions are provided in section 4.

2. BRAKE DESIGNS TO ENHANCE HEAT DISSIPATION

2.1 DESCRIPTION OF THE HEAT TRANSFER PROBLEM

The stop braking action is initiated with the brake shoe contacting, under considerable normal force, the rim of the rolling wheel. The friction between the shoe material and the wheel retards the wheel rotation. The same friction also results in the generation of heat at the brake shoe-wheel rim interface. A portion of this heat is conducted into the wheel, and the remaining heat is conducted into the brake shoe.

The magnitude of the brake shoe temperature at the interface depends on the rate of heat input into the shoe and the thermal properties of the shoe material. For example, if the heat input to the shoe can be conducted away from the interface at a higher rate (with a conductive material) than for a given heat input rate, the interface temperature can be reduced. Alternatively, the temperature can be lowered by providing cooling at the interface. Lastly, the interface temperature can be reduced by providing a jacket of cooling fluid at the back face of the brake shoe with the intent of maintaining a relatively low brake shoe temperature. All of these approaches and associated conceptual designs are discussed in this chapter.

2.2 BRAKE SHOE MATERIAL PROPERTIES

The characteristics of a good brake shoe material include:

1. good frictional properties;
2. low wear for a given service and high mechanical robustness, even at elevated temperatures;
3. high thermal conductivity.

Until a decade ago, asbestos-based shoe materials were common because they had good wear resistance and mechanical strength. However, because of the prohibition on the use of asbestos, these brake shoes have been replaced by phenolic resin-based materials. The currently used materials are, generally, a mixture of phenolic resin and sand (75:25 percent ratio). These materials have poor wear resistance and low thermal conductivity, which leads to higher shoe temperature and faster wear-out.

Metals having high thermal conductivity capability are never used in pure form because they lack frictional properties. Sintered powder materials are used in specialized braking applications; these have high thermal conductivity values. However, because of the difficulty in forming these materials to different geometric shapes, they are used only in flat disk braking applications. Graphite-based composites are currently being investigated for brake shoe applications. These materials have good frictional characteristics. They also have high thermal conductivity (up to 1200 W/m K), but are difficult to form into different brake shoe shapes.

Because the combined attributes of high thermal conductivity, wear resistance, friction, and manufacturability are not currently available in a brake shoe, this report does not compare thermal performance of brake shoes using different materials. The emphasis is rather on heat transfer enhancements resulting from innovative geometric redesigns.

2.3 GROOVED BRAKE SHOE DESIGN

In this section, a scheme is presented to augment the heat transfer from the hot brake shoe to the ambient air in order to reduce the temperature of the shoe and also that of the wheel tread. This plan proposes to conduct heat from the brake shoe within the time scales over which it is generated. To accomplish this, the heat transfer to the ambient air must occur in the region where the heat generation takes place, i.e., the interface. It is proposed to provide extended surface near the interface by cutting shallow grooves into the face of the brake shoe, thereby creating "fins" with extended surface for convective heat transfer. Figure 2-1 depicts the proposed modification. The sides of these groove fins provide surface area for heat transfer that did not exist before, while the groove provides a channel for air flow. Air will be swept into the passage by viscous drag as the wheel rotates and tread slides beneath the shoe. This air flow allows cooling of the fin sides.

2.3.1 Model Details

Each portion of the brake shoe material in contact with the wheel will be modeled as a one-dimensional fin with base temperature equal to the wheel-shoe interface temperature. We now develop a simple model to calculate the magnitude of the steady state heat transfer in order to evaluate whether design optimization is justified. The following assumptions are made:[1]

- The total shoe-wheel contact area is fixed. The brake shoe must be made longer (along the circumference) to compensate for the contact area lost to the grooves. An upper limit of twice the present shoe length (0.35 m) is imposed.
- The interface temperature is fixed at a value corresponding to a typical maximum rim temperature: $T_{max} = 425$ °C.
- The ambient temperature is $T_a = 0$ °C.
- The fins and grooves will have a rectangular cross section, although in a real design, triangular grooves might be more practical.
- The ambient temperature is fixed, i.e., no "air heat-up" within the grooves as it flows through the channel.
- Additional assumptions will be invoked during the analysis.

1. The assumptions may not be conservative (for example, air temperature is generally 20 °C). However, our aim in these assumptions is to see what is the maximum cooling effect that can be achieved.

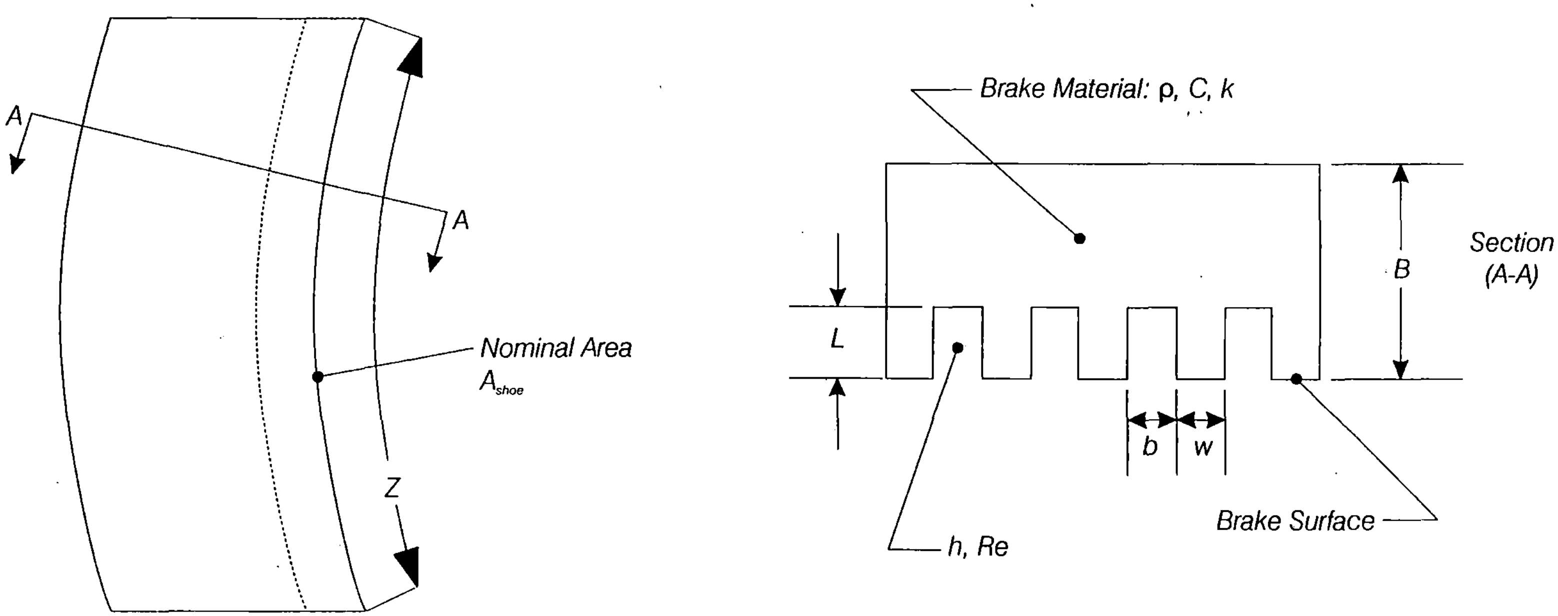

Figure 2-1. Sketch of Shoe Redesign with Fins/Grooves

For a single one-dimensional fin, the steady-state heat transfer rate is given by:[2]

$$\dot{Q}_{fin} = k\ m\ w\ Z\ \tanh\left(mL\right)\left(T_{\max}-T_{a}\right) \tag{2-1}$$

The fin dimensions are chosen based upon maximizing heat transfer while staying within the geometric constraints and producing a practical design. The dimensions of the fin will dictate the number of fins present and, thus, the total heat transfer rate. Longer fins (measured along the heat flow direction, i.e., groove depth) will transfer more heat due to the larger surface area. However, this effect is limited because the fin effectiveness decreases with increasing length. The fin effectiveness, η, is defined as ratio of the fin effective area (area acting at the fin base temperature) to actual surface area. It is not advantageous to use fins longer than the distance that heat will conduct over the characteristic time scale. The fin length (groove depth) will be taken equal to the heat penetration depth over the braking duration:

$$L = \delta \approx \sqrt{\alpha t_{stop}} \tag{2-2}$$

Using this length results in a relatively large fin effectiveness.

A number of factors will influence the choice of fin (and groove) width. The fin width will be based upon maximizing heat transfer while maintaining adequate strength. The groove width affects the forced air convection heat transfer coefficient and the number of fins present. For simplicity, the following is assumed: (1) fin length as given above, (2) groove width equal to fin width (not to be less than 1 mm for practical reasons). This will force a total shoe length of double that presently used (our limiting assumption) in order to maintain the contact area.

To estimate the heat transfer coefficient within the grooves, we will invoke further assumptions. Essentially, the groove is a square channel carrying a forced convective flow driven by the longitudinal motion of one surface and resembles the well known *Couette flow*. However, the flow field is three-dimensional and the velocity distribution is difficult to calculate. Obviously, the average velocity in the channel is bounded by the rim velocity and zero. It will be assumed that the average velocity is one-half of the wheel tread speed. We now calculate the Reynolds number based on this average flow velocity, the hydraulic diameter of the groove, and the kinematic viscosity of air at ambient temperature.

$$Re = \frac{V}{2}\ \frac{D_h}{\nu} \tag{2-3}$$

2. The symbols are defined in the Nomenclature.

The heat transfer coefficient is found from the well known McAdams correlation for turbulent flow inside channels (Chapman 1960):

$$\frac{hD_h}{k_{air}} = 0.023\ Re^{0.8}\ Pr^{0.4} \tag{2-4}$$

2.3.2 Results

Although it is obviously not suitable for a brake material, we employ copper as the shoe material to achieve maximum thermal performance. Table 2-1 shows the calculated results using the inputs given above and copper as the brake material. It was found that the best performance was attained using the smallest allowable fin/groove width (1 mm). As can be seen, only a small fraction of the generated heat can be conducted away into the air stream. The major shortcoming lies in the fact that to get a large solid/air heat exchange area, the passage dimension becomes small, giving rise to a small heat transfer coefficient. It should be pointed out that this represents an upper limit, since a highly conductive material was used, air heat-up within the passage was neglected, and the heat transfer was maximized by using the maximum driving temperature (T_{max} = 425 °C). It is therefore concluded that under no circumstance can a significant amount of heat be rejected to the environment using a grooved brake shoe.

2.4 BRAKE SHOE WITH EXTERNAL FINS

In this section, a design is proposed whereby fins are integrated onto the brake shoe external surfaces so that as the shoe heats up, free stream air will transfer heat from the fins. This design is shown schematically in figure 2-2. The objective of this design is to maximize both the heat transfer area and the convective transfer coefficient by providing external fins. The major shortcomings in the (previous) groove/fin design were the limited heat transfer area, and small heat transfer coefficient, *h*.

This heat transfer strategy will require that heat be conducted deeper into the brake shoe over the time scale of the braking event (t_{stop}) and that the brake shoe function as a sink for heat. As will be seen, the effectiveness of the strategy relies heavily upon the brake shoe's ability to store thermal energy, i.e., the thermal capacity. The next section addresses this.

2.4.1 Thermal Capacity of Brake Shoes

The performance limit of this design is given by the total energy storage capacity of the shoe. If one assumes a brake shoe geometry (size), material, and a temperature, the thermal energy storage can be estimated. The energy stored upon bringing the entire shoe mass from ambient to maximum temperature is given by

$$Q_{shoe} = \rho\ c\ B\ A\ \Delta T \tag{2-5}$$

Table 2-1. Input Parameters and Heat Transfer Results for Grooved Brake Shoe Example (Material: Copper)

Parameter	Value
Velocity, V_i	35.8 m/s (80 mph)
Deceleration, a_d	0.894 m/s^2 (2 mph/s)
Car weight, W	623 kN (140 kips)
Stopping time, t_{stop}	40 s
Power dissipation, Q_{wheel}	254 kW/wheel
Maximum rim temperature, T_{max}	425 °C
Ambient temperature, T_a	0 °C
Fin length, L (depth of grooves)	0.068 m
Fin width, w	0.001 m
Fin spacing, b (width of groove)	0.001 m
Shoe length, Z (modified value)	0.7 m
Channel Reynolds number, Re	1800
Heat transfer coefficient, h	3.5 W/m^2 K
Heat transfer per fin, Q_{fin}	140 W
Number of fins	30
Total heat transfer, Q_{total}	4000 W
Enhanced Heat Transfer Factor (Percent of braking power) Q_{fins}/Q_{wheel}	1.6%

Using a maximum brake shoe temperature of 425 °C and ambient temperature $T_a = 0$ °C (as in the previous section), the present brake shoe geometry, and the thermal properties, copper gives an energy storage of roughly 1.1 MJ per shoe. The initial kinetic energy of the railcar is found from

$$KE_i = \frac{WV_i^2}{2g} \tag{2-6}$$

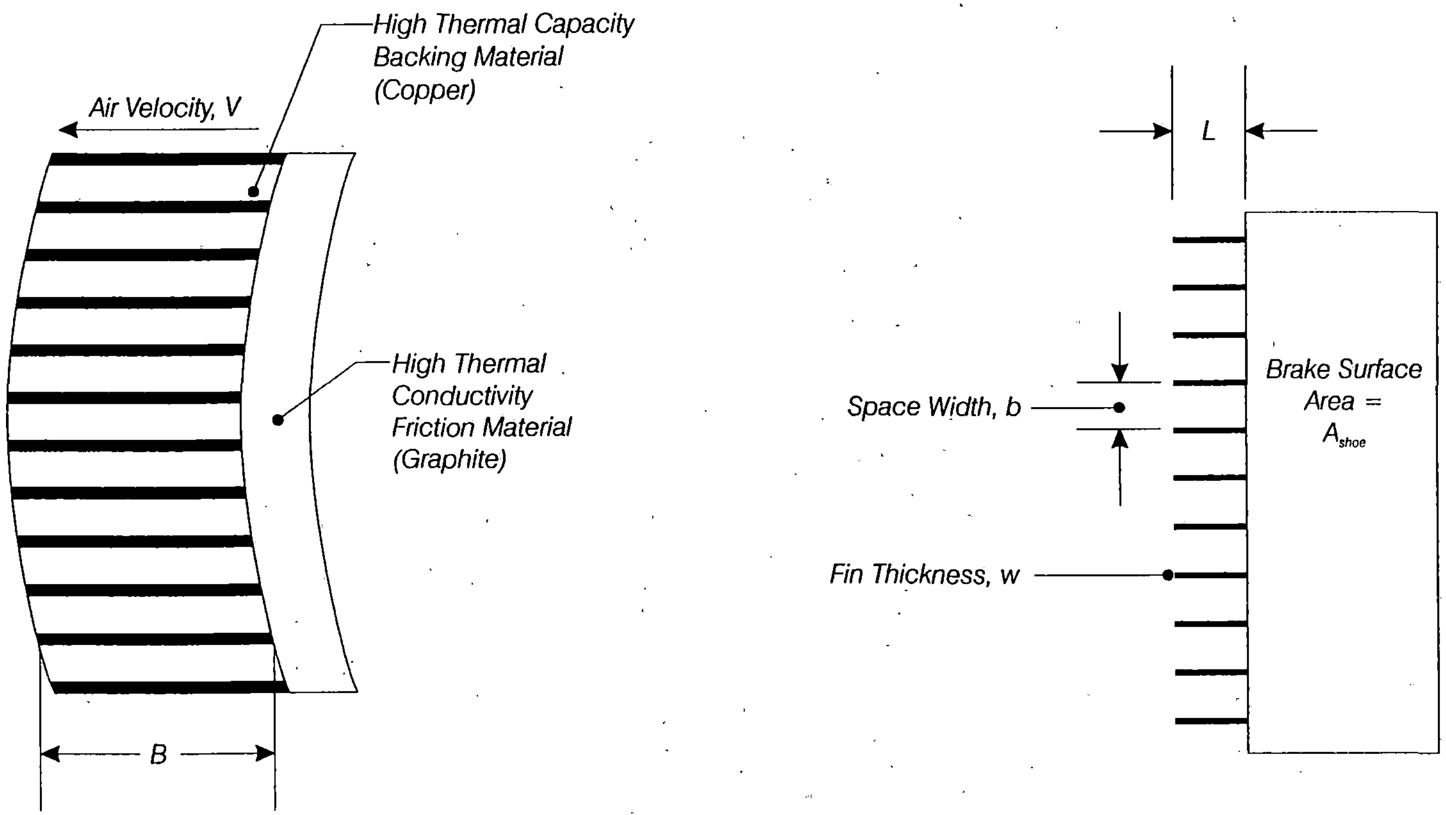

Figure 2-2. Sketch of Brake Shoe with External Fins

Thus, for a 623 kN (140 kip) car traveling at 36 m/s (~80 mph), each of eight brake systems must absorb thermal energy, E, of about 5.1 MJ to bring the vehicle to rest. Therefore, the shoes can absorb about 20% of this energy. Several effects have been neglected. The spatial average brake shoe temperature will be less than the maximum value that was used to compute the energy storage; therefore, this calculation is an overestimate. However, there will be some heat transfer to the environment during the braking period; this heat transfer has been neglected. In addition, this calculation assumes a low ambient temperature (or initial brake shoe temperature). In the event the ambient temperature (initial shoe) is higher, or the brake shoe temperature has ratcheted upward from successive closely spaced train stops, the fraction of the generated heat going to the brake shoe will be lower.

A different way of looking at the result in the case where there are external fins is to say that the peak temperature rise of the wheel rim/brake shoe will be lower than in the case where there are no fins on the brake shoe. The reduction in peak temperature can be considered as significant. Hence, additional work (both theoretical and experimental) may be justified. We therefore analyze in the following sections the heat dissipation rates from external fins and their sizes.

Since we need the entire brake shoe mass to attain a uniformly high temperature as quickly as possible, we assume the material to be copper (high thermal conductivity); however, copper does not have a high specific heat.

2.4.2 Convective Heat Transfer

In this section, the heat transfer from the brake shoe to a proposed fin structure is estimated. Figure 2-2 is a sketch of the proposed brake shoe design incorporating copper fins. Unfortunately, the presence of the flange on the inner side of the wheel limits the use of fins to only one side of the shoe. The fins are oriented horizontally to facilitate forced air convection (parallel to the ground) as the car moves along at full speed. Additional fins could be mounted vertically along the back face. However, it is not expected that back mounted fins can make a strong contribution for three reasons: (1) the back surface temperature is the lowest, (2) the natural convection, h, is typically much smaller than that for forced convection, and (3) "air heat-up" within these longer fin passages will impede heat transfer. Using this design plan, the following assumptions are employed:

- The brake shoe is spatially isothermal.
- There is negligible "air heat-up" as air convects through the space.
- The fin heat transfer is one-dimensional.
- The value of the heat transfer coefficient between the fins and air is 20 W/m^2 K.

The heat transfer per fin is evaluated using (Chapman 1960):

$$\dot{Q}_{fin} = k\ m\ w\ B\ \tanh\left(mL\right)\left(T_{\max} - T_a\right) \tag{2-7}$$

The number of fins is dictated by the fin spacing, thickness, and the shoe length. A means by which the fin design can be evaluated is through measurement of a characteristic period of time required to cool off the brake shoe from its maximum temperature. It can be shown that the temperature will decrease according to

$$T(t) = T_{\max} e^{-t/t_c} \tag{2-8}$$

where the time constant, t_c, is defined by

$$t_c = -\frac{T_{\max} - T_a}{\left.\frac{dT}{dt}\right|_{t=0}} \tag{2-9}$$

This may be expressed equivalently as

$$t_c = \frac{Q_{shoe}}{\left.\dot{Q}_{shoe}\right|_{t=0}} \tag{2-10}$$

that is to say, time to dissipate all stored energy in a shoe if the cooling rate is always equal to the initial rate of cooling.

The above equations are used with certain specified conditions to obtain the effective cooling rate. These input values and condition parameters are summarized in table 2-2.

The cooling time constant for the conditions indicated in table 2-2 is about 7 minutes. After one time constant has elapsed, the temperature difference between the brake shoe and the ambient air will be only about 37% of the initial difference. Cooling to near ambient temperature will require upward of three time constants (21 minutes). For example, if the train stops every 7 minutes, then the brake shoe temperature will reduce from 425 °C to 156 °C when the ambient temperature is at 0 °C. Then, only about 63% of the stored thermal energy (1.1 MJ) has been dissipated. Recall that this was about 20% of the total kinetic energy (divided by eight wheels). So, for 7-minute stop intervals, only about 13% of the train's kinetic energy will ultimately be dissipated in the form of heat to the environment through the fins.

Table 2-2. Input Parameters and Heat Transfer Results for Finned Copper Brake Shoe/Heat Sink Example

Parameter	Value
Initial Velocity, V_i	35.8 m/s (80 mph)
Car weight, W	623 kN (140 kips)
Initial kinetic energy, KE_i	41 MJ
Maximum shoe temperature, T_{max}	425 °C
Ambient temperature, T_a	0 °C
Shoe width	0.060 m
Shoe thickness, B	0.038 m
Shoe arc length	0.35 m
Fin length, L	25 mm
Fin thickness, w	0.5 mm
Fin spacing, b	1.58 mm (12 fins per inch)
Total number of fins	168
Heat transfer coefficient, h	20 W/m^2 K
Fin effectiveness, η	96%
Heat transfer rate per fin (maximum)	15.3 W
Total heat transfer rate, Q_{total}	2580 W
Thermal energy stored, Q_{shoe}	1.1 MJ
Cooling time constant	7.1 min.

The fin dimensions chosen for this calculation were not strictly based on optimization. The length, thickness, and spacing used represent commonly employed values for fin structures. In this example, for the chosen h, longer, thinner, more closely spaced fins would provide higher heat transfer, but would be impractical. Longer, thinner fins would damage more easily, and closer spacing degrades the heat transfer coefficient and facilitates clogging with debris.

Thermal radiation heat transfer has been neglected in this analysis. Obviously, its effects will only be beneficial, reducing the characteristic cooling time, but short lived, since it will diminish rapidly with decreasing temperature.

2.4.3 Other Design Considerations

Copper cannot practically be used as a frictional material; however, graphite, which has the required high thermal conductivity but a lower heat capacity, potentially can be. We propose a composite brake shoe employing the graphite brake surface backed by a copper sink. Clearly, good thermal contact between the graphite and copper is required. The fabrication details associated with forming this composite design into the proper shape and ensuring good thermal contact between materials is beyond the scope of this study.

It was shown using the thermal capacity of copper that 20% of the total energy generated per stop can be absorbed into a shoe of the present geometry. If the shoe size is expanded, then the total capacity is further augmented. This can be accomplished by increasing the shoe length and thickness. The shoe length should be maximized and the thickness increased to the thermal boundary layer thickness developed over the time t_{stop}. For the properties of copper and a 40 sec stop time the thermal boundary layer is about 2.5 inches thick. Fortunately, the available fin area will scale with these dimensions, thus providing the necessary heat removal to the environment.

This example was used to illustrate the ability of extended surface to remove significant energy absorbed by a high thermal capacity brake shoe. The fin structure could potentially be optimized, providing much better performance. This is beyond the scope of this preliminary study since there are numerous technical/manufacturing difficulties to overcome to implement this design.

2.5 BRAKE SHOE WITH INTEGRATED WATER JACKET

The design described in the previous sections had the advantage of simplicity, however, the performance was somewhat limited. In addition, the performance hinges on proper maintenance, i.e., keeping the fin structure free of dirt and debris. A different plan is proposed in this section whereby heat is removed from the brake shoe not by solid-to-air conduction but by the vaporization of a liquid. It is expected that higher performance is attainable with less maintenance.

2.5.1 Liquid Jacket Conceptual Design

Since the latent heat of vaporization is large for water, an abundant substance, it is often an attractive medium for cooling. A conceptual design is presented that allows brake shoe cooling by phase change through direct contact with water. Figure 2-3 shows the design schematically. The brake shoe is modified to have an integrated water jacket on the back side. As the brake shoe heats up, the water evaporates and the steam is vented. Water lines running to each shoe refill the tanks on demand and are controlled by some type of float valve. This section sets out to address the feasibility of this design from the heat transfer standpoint. The amount of water consumed in each brake shoe tank and the average heat transfer rate will be estimated.

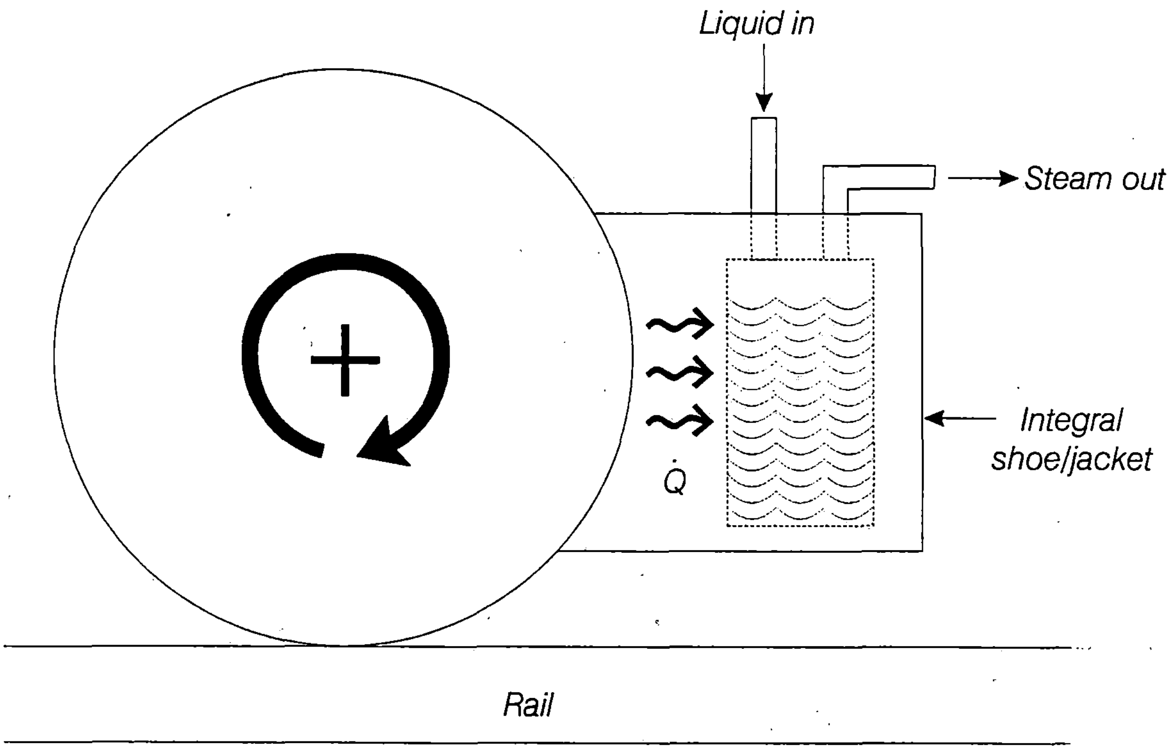

Figure 2-3. Sketch of Brake Shoe with Integrated Liquid Jacket

2.5.2 Model

We make a first order calculation of the water mass requirement and heat transfer rates for this design. The following assumptions are made:

- All of the kinetic energy of the railcar is converted to thermal energy at the brake and none of this is "lost" to the surroundings.
- Perfect conduction through the brake shoe, which implies that the wheel-brake interface surface temperature is at water boiling temperature.
- All heat conducted through the shoe goes to evaporate the water, i.e., there is no storage in the shoe material. Sensible heating of the water is neglected in comparison with total heat of evaporation.

It was found in the last section that each of eight brake systems on a railcar must absorb 5.1 MJ of energy when a 140 kip car is brought to rest from 36 m/s (80 mph) initial speed.

The mass of water that would be vaporized is found from the following equation:

$$m_{water} = \frac{E}{h_{fg}} \tag{2-11}$$

For water at atmospheric pressure, h_{fg} = 2257 kJ/kg (970.4 Btu/lbm), and the mass vaporized per wheel per stop is 2.3 kg (2.3 liters volume). Thus, for each railcar, about 18 kg would be required for each braking event from 36 m/s speed. This is not an unreasonably large amount, but enough to require a liquid delivery system of some type to keep each jacket full. (A system capable of delivering water at the rate of about 4ℓ/min or 1 gpm per shoe will be sufficient.) We will next address the expected performance by looking at the heat transfer issue.

2.5.3 Heat Transfer Considerations

The previous section dealt with the thermodynamics of the energy exchange. The effectiveness of this energy transfer requires a look at the character of boiling heat transfer. The energy transfer to each brake shoe is done in a short period of time. This time is found from

$$t_{stop} = \frac{V_i}{a_d} \tag{2-12}$$

Thus, the average power delivered to the brake shoe (and to the water) is

$$\bar{P} = \frac{E}{t_{stop}} \tag{2-13}$$

The heat flux at the liquid-solid interface is computed from

$$q'' = \frac{\bar{P}}{A_{shoe}} \tag{2-14}$$

For the standard size brake shoe and a deceleration, $a_d = 0.89\ m/s^2$ (2 mph/s), this average heat flux exceeds 6 MW/m^2. This is beyond the critical heat flux (CHF) seen in most pool boiling data (Rohsenow 1973). Therefore, the pool boiling of water cannot be expected to pass this level of heat flux. What will result, as the heat flux is raised, is a regime change from nucleate boiling to film boiling; a regime with much less effective heat transfer. Continued energy transfer to the shoe will result in very high shoe temperatures. High shoe temperature will lower the fraction of the generated heat that the shoe can absorb, the balance being transferred to the wheel.

2.5.4 Discussion

From the thermodynamic standpoint, this method of dissipating heat is attractive since water is abundant, inexpensive, and has a high latent heat. The functionality again depends on the requirement of a high conductivity brake shoe material (we assumed perfect conduction in this model). However, even in this case there is a fundamental limit to the heat transfer rate dictated by CHF.

It is conceivable that better performance can be achieved if the brake shoe is not perfectly conducting since this will allow the heat transfer to the liquid to take place over a longer time period, lowering the average power and possibly avoiding the occurrence of CHF. The penalty for this is higher wheel-brake shoe interface temperature. It is this counterintuitive behavior that can motivate certain experiments to assess the feasibility of the design. This is discussed further in section 3.

2.6 DIRECT WATER SPRAY COOLING OF THE WHEEL RIM

2.6.1 Introduction

The conceptual redesigns discussed thus far have focused on cooling of the brake shoe to keep both the brake shoe and the wheel tread cooler. These schemes operate in an indirect way since wheel cooling is achieved by cooling of the shoe and therefore will have limited success. In this section we investigate a more direct approach, namely, actively cooling the wheel tread by water spray impingement. Though the primary concern is to maintain a low wheel temperature, keeping the brake shoe cool is also important. If a significant quantity of heat can be removed from the wheel, the tread temperature will be significantly reduced. Since continuity of temperature at the brake-wheel interface is to be maintained, the brake shoe temperature will be reduced as well. Though it would seem that liquid applied to tread would reduce friction, it may be acceptable since wheels, brake shoes, and rails all get wet when it rains. In the same spirit as the other conceptual designs, we will not address the problems of implementing the design, but rather the thermal effectiveness.

Figure 2-4 illustrates the present concept. Water spray heads are mounted around the wheel periphery at some close spacing to the tread. They direct ambient temperature water in the form of a fine spray (with some air entrainment) at the tread over the duration of the braking event (and perhaps longer). The exact location of the spray heads is not addressed here, but will be dictated more by practical considerations than by thermal performance. We proceed in the following sections to formulate the heat transfer model for evaluating this method.

2.6.2 Spray Cooling Model

The goal of this work is to reduce the temperature attained by the rim during a braking event. The model developed in this section will calculate the amount of heat extracted from the wheel by spray cooling and estimate the resulting rim temperature. The input parameters for this model will be taken from an example scenario that was numerically modeled to estimate the temperature history of the rim (Tang 1993).

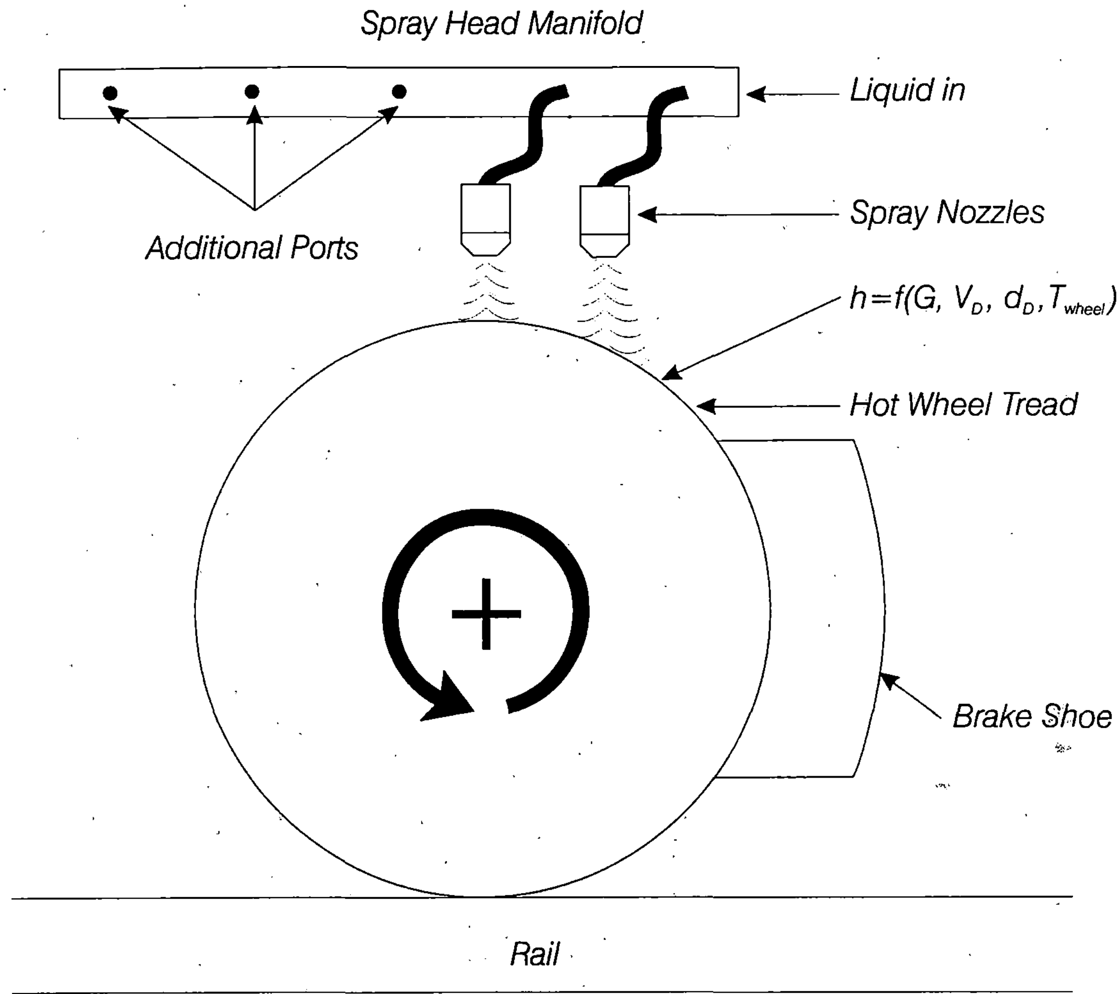

Figure 2-4. Sketch of Wheel Tread Spray Cooling Scheme

2.6.2.1 Modeling Assumptions. The following simplifying assumptions are made. Additional assumptions are invoked as the analysis proceeds.

- The wheel is initially at ambient temperature, $T_a = 23$ °C
- The tread temperature history (without active cooling) follows a linear ramp from ambient temperature to the maximum value, T_{max}, which occurs at a time, t_{max}.
- The spray cooling commences upon application of the brake.
- The spray cooled area, A_{spray}, is some fraction (A_*) of the wheel circumference multiplied by the brake shoe width.
- As the vehicle slows, all the kinetic energy change is converted at the wheel to thermal energy. That is, heat transfer to the shoe and rail, and aerodynamic drag on the vehicle, will all be neglected.

- The temperature rise ($T_{wheel} - T_a$) at any time t will scale in direct proportion to the net energy flow into the wheel over the time t.

- The moving tread surface does not affect the spray cooling heat transfer coefficient. Heat transfer coefficients measured on a stationary surface will employed.

2.6.2.2 Modeling Details. The power input to one wheel at any time t is found from

$$\dot{Q}_{wheel} = \frac{W\, a_d\, V(t)}{8g} \tag{2-15}$$

The velocity is found from kinematics:

$$V(t) = V_i - a_d\, t \tag{2-16}$$

The wheel temperature history (uncooled) described in the list of model assumptions is represented by

$$T_{wheel} = T_{\max} \frac{t}{t_{\max}} \tag{2-17}$$

The heat removal rate by spray cooling can be written

$$\dot{Q}_{spray} = hA_{spray} \left(T_{wheel} - T_a\right) \tag{2-18}$$

where h depends on the surface temperature; $h = h(T_{wheel})$. More will be said on this in a later section. The value of A_{spray} will depend on the fraction of wheel circumference (A_*) that is acted on by the spray cooling and will be treated as a parameter in this problem. The net heat transfer rate to the wheel is thus

$$\dot{Q}_{net} = \dot{Q}_{wheel} - \dot{Q}_{spray} \tag{2-19}$$

One of the modeling assumptions stated that the rim temperature will scale with total net heat transferred to the wheel up to any time t. This may be expressed by the following:

$$\left(T_{wheel} - T_a\right)_{cooling} = \frac{Q_{net}}{Q_{wheel}} \left(T_{wheel} - T_a\right)_{no\ cooling} \tag{2-20}$$

2.6.3 Spray Heat Transfer Coefficient

The heat transfer coefficient obtained when an aerated or atomized spray impacts a hot surface depends on numerous factors, including: the surface temperature, liquid mass flux, the drop velocity, and the drop size. It is beyond the scope of this work to present a detailed study of how these parameters affect h. Instead, we will refer to one of the more comprehensive works on spray cooling, where the effects of the parameters given above were studied in several heat transfer regimes.

Mudawar (1994) has published data for spray cooling on heated surfaces and gives correlations for single phase convection, nucleate boiling, and transitional boiling regimes, the presence of which depends on the surface temperature. The temperature range experienced by railcar wheels during braking is large enough to encompass all these regimes. The heat transfer coefficient varies dramatically in these different regimes, so it is not sufficient to use one fixed value of h.

Up to temperatures slightly in excess of the normal boiling point, single phase convective heat transfer to the impinging spray will prevail. Mudawar (1994) has correlated the data in this regime with the spray parameters by

$$Nu = \frac{h d_D}{k_f} = 2.5\ Re_D^{0.76}\ Pr^{0.56} \tag{2-21}$$

where the fluid properties refer to the liquid phase. The Reynolds number here is defined by

$$Re_D = \frac{G d_D}{\rho_f\ \nu_f} \tag{2-22}$$

This correlation will apply for surface temperatures below:

$$T_s = 13.4\ Re_D^{0.167}\ Pr^{0.123} \left(\frac{k_f}{d_D}\right)^{0.22} + T_f \tag{2-23}$$

Above this temperature nucleate boiling will begin, accompanied by a dramatic increase in the heat transfer coefficient. Mudawar (1994) reports that the nucleate boiling h does not depend on the spray parameters or liquid flux but rather only the surface temperature:

$$h = 1.87 \times 10^{-5} \left(T_s - T_f\right)^{4.55} \tag{2-24}$$

The nucleate boiling will prevail for heat flux up to the CHF when significant portions of the surface cannot stay wet and the heat transfer effectiveness begins to diminish. The temperature at which nucleate boiling ceases and transitional boiling begins is correlated by

$$T_s = 18\left[\frac{\rho_g\, h_{fg}\, G}{\rho_f}\left(\frac{\rho_f\, \sigma}{G^2\, d_D}\right)^{0.198}\right]^{0.18} + T_f \tag{2-25}$$

Above this temperature, transitional (or unstable film boiling) ensues. Mudawar (1994) gives a very lengthy (cumbersome) correlation for the heat flux in the region. However, the heat transfer coefficient may be expressed approximately by a simpler expression:

$$Nu = 1.3\,\frac{Re^{0.75}}{\left(\dfrac{c_p\left(T_s - T_f\right)}{h_{fg}}\right)^2} \tag{2-26}$$

This correlation applies up to about T_s = 500 °C.

2.6.4 Model Implementation

The objective of the model is to predict the tread temperature when spray cooling is employed. The model equations were implemented in the following way. An area was chosen for spray cooling. The wheel temperature was increased at each time step according to equation 2-17. Using this surface temperature, the appropriate heat transfer regime was identified, and h for that regime was calculated using equations 2-21, 2-24, or 2-26. The spray cooling rate is found from equation 2-18. The net heat transfer rate to the wheel is found from equation 2-19. This is integrated at each time step and used in equation 2-20 to estimate the actual wheel temperature. The calculations are run up to time t_{max}, the time at which the maximum temperature *without cooling* occurs, which is determined from the results of numerical simulations for several braking scenarios (Tang 1993).

2.6.5 Results

The model above was employed for an example case to estimate the reduction in the wheel rim maximum temperature. The wheel temperature during braking was calculated (Tang 1993) using finite element simulations. Table 2-3 shows the input parameters used. The spray parameters chosen were those that gave the highest heat transfer performance in the experiments of Mudawar (1994).

Figure 2-5 shows the input power (curve A), cooling rate (curve B), tread temperature with cooling (curve C), and tread temperature without cooling (curve D), over the first 23 seconds of the braking event for a normalized spray area of 10%. The three heat transfer regimes encountered are labeled. The final temperature (with cooling) of 385 °C is considerably lower than the predicted temperature without active cooling (514 °C). As seen, the temperature traces with and without active cooling deviate substantially when the nucleate boiling heat transfer regime is entered. As time continues, the difference in temperature becomes larger, owing to the significant cooling rate in the transition boiling region.

The spray area is a parameter in this problem, and its importance is demonstrated in figure 2-6. For practical reasons the area fraction A_* has been limited to 50%. The plotted temperature is the wheel temperature after 23 seconds of braking. Up to about 20% sprayed area the tread temperature drops drastically with increases in area. Beyond that point an increase in area is almost ineffective. This curious result highlights a fundamental difference between the nucleate boiling regime and the unstable film (transition) boiling regime. Below surface temperatures of about 140 °C, nucleate boiling with very high h prevails. Increasing the sprayed area will result in lower surface temperature, which will dramatically lower the heat transfer coefficient (see equation 2-24). Thus, there is a tradeoff between area and h, resulting in very small decreases in surface temperature. The opposite is true above 140 °C. Increasing spray area decreases T_s, which increases h. Thus, dramatic changes in the surface temperature are possible when the cooled area fraction is increased, if the surface temperature is higher than the CHF surface temperature.

2.6.6 Discussion

The results indicate that direct, active cooling of the wheel can result in substantially lower temperatures. Practical considerations may dictate the amount of wheel periphery that can be cooled and the mass flux, G, that can be supplied. This method will consume more water than the liquid jacket method analyzed earlier because only a fraction of the sprayed water will vaporize. For example, a mass flux $G = 5$ kg/m^2 s acting on 10% of the tread area for a 40-second stop accounts for 2.9 kg of water per wheel per stop compared to 2.3 kg per wheel per stop calculated in the last section for water jacket cooling. It is perhaps possible to attain higher heat transfer coefficients by optimizing the spray conditions. However, it may be possible to accept inferior heat transfer in lieu of conserving the water supply. Optimization of this method requires more knowledge of target operating conditions and is beyond the scope of this work.

This plan would require major equipment modification, including carrying an inventory of water on board. It may also be unacceptable to have wet wheel treads as a standard operating condition. The maintenance costs may increase since each spray head (one for each wheel) will require periodic checks and perhaps will become clogged with debris. Finally, it may not be acceptable to perform a rapid cool down of the rim from a high temperature on a periodic basis. If enough spray heads are used on each wheel, it is possible to maintain the tread at a low temperature, however system malfunction could result in spraying the rim after it has been heated to a high temperature; an undesirable occurrence.

The first order model presented here indicates that, from the thermal standpoint, direct spray cooling of the tread is a potentially attractive means to keep the wheel (and brake shoe) temperature to acceptably low levels. If the difficulties cited above are not pivotal, experiments to measure the performance of this concept should be performed. This is discussed in the next section.

Table 2-3. Input Parameters Used in Direct Spray Cooling Model Example

Parameter	Value
Railcar weight	543 kN (122 kips)
Initial velocity, V_i	44.7 m/s (100 mph)
Deceleration, a_d	0.89 m/s^2 (2 mph/s)
Ambient temperature, T_a	23 °C (73 °F)
Maximum tread temp. (no cooling) T_{max}	514 °C (958 °F)
Time to reach T_{max}, (t_{max})	23 sec.
Liquid/Water Spray mass flux, G	5 kg/m^2 s
Drop diameter, d_D	0.5 mm
Drop velocity, V_D	18.0 m/s
Spray cooled area relative to tread surface, A.	10%

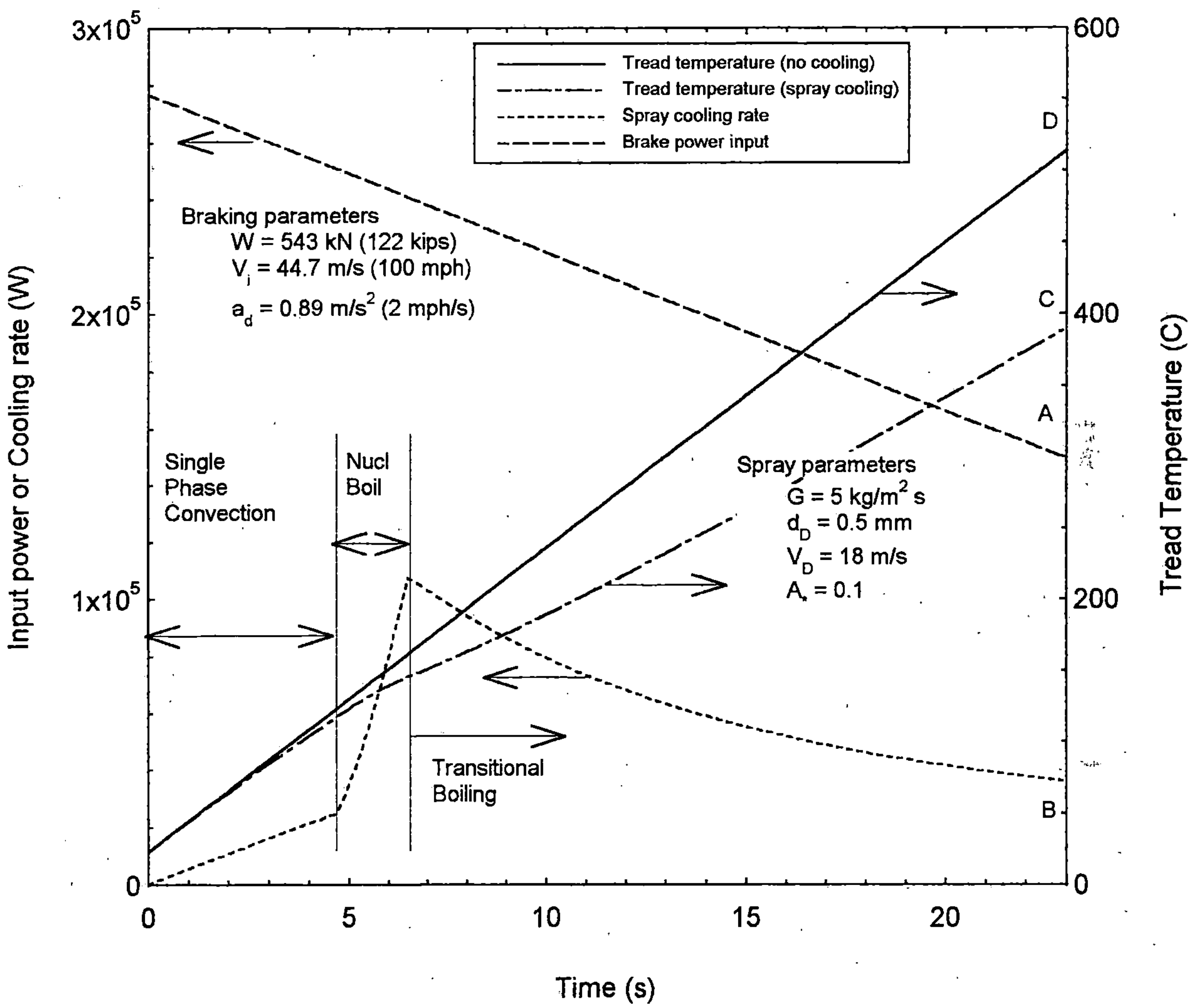

Figure 2-5. Brake Power Input, Heat Dissipation, and Rim Temperature

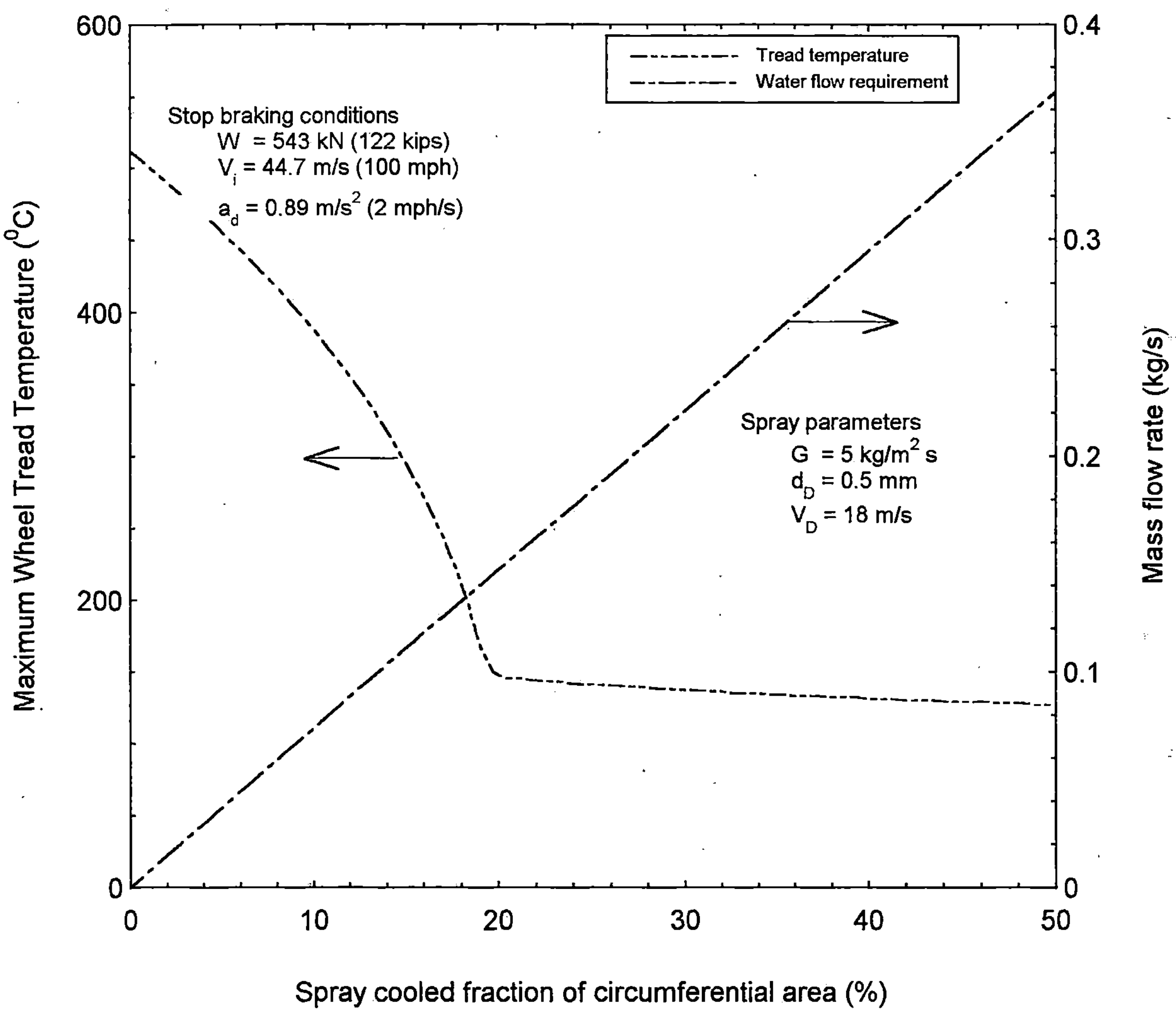

Figure 2-6. Dependence of Maximum Tread Temperature and Coolant Required on Spray Area

3. RECOMMENDED EXPERIMENTS TO TEST PROPOSED DESIGNS

The designs put forth in this report are very much conceptual. The emphasis in design was based on heat transfer performance and to a lesser extent on operational or mechanical feasibility. In addition, the modeling was limited to first order estimates, since these are often sufficient to evaluate whether an idea is fundamentally flawed or requires a closer look with more detailed calculations, experiments, and prototypes. The results of analysis of these designs suggest that further work (besides evaluating the practicality of implementing a design) be done in the form of simple experiments to verify the predicted thermal performance. Three experiments are outlined below.

3.1 EVALUATION OF EXTERNAL FIN DESIGN

It was proposed in section 2.4 to use a composite structure employing a graphite braking surface in good thermal contact with a high thermal capacity material like copper. The integrated copper fin structure transfers heat to the environment. The requirements of good friction and wear resistance of graphite, good thermal contact between copper and graphite, and efficient removal of energy from the shoe motivates three areas of experimental work:

- ♦ Measurements of the friction and wear characteristics of graphite that has been processed to be a brake material.
- ♦ Thermal evaluation of contact resistance of bonding between the friction material (graphite) and the heat sink material (copper).
- ♦ Measurements of the temperature history of a prototype brake shoe using power input, stop time, and air speed characteristic of an actual braking event to evaluate the effectiveness of the shoe as a heat sink and the effectiveness of the fin structure. Optimization of the fin structure should follow.

3.2 EVALUATION OF THE INTEGRATED BRAKE SHOE-WATER JACKET DESIGN

It was shown in section 2.5 that the brake shoe-water jacket scheme had potential to keep the brake shoe cool with a modest amount of water consumption per stop. Clearly, the heat transfer rate to the liquid will depend strongly on the thermal resistance of the brake shoe material. A brake shoe that is too thick or too thermally resistive will give poor performance. It was also shown that the rate of heat transfer to the liquid was bounded due to the occurrence of critical heat flux (CHF). This limit could apply if the brake shoe is too conductive or too thin. The "perfect conduction" assumption of section 2.5 led to heat flux rates well in excess of the expected CHF. Some thermal resistance in the brake shoe is not only inevitable but may be beneficial. In this regard, the following experiments are recommended:

- ♦ Fabricate a prototype brake shoe with integrated water tank. Conduct steady-state heat transfer measurements to determine the value of CHF in this geometry.

- Analytically and experimentally determine the optimum brake shoe thickness to give highest power dissipation for a specified thermal conductivity of the candidate brake material.

- Build a prototype and simulate a braking event to measure the energy transferred to the water in the jacket by measuring the liquid consumption.

3.3 EVALUATION OF DIRECT SPRAY COOLING OF THE WHEEL TREAD

In terms of actual performance, the most promising method for cooling the wheel tread is the direct liquid spray scheme described in section 2.6. The first order model developed earlier gave rough estimates of the reduction in tread temperature when spray cooling is used. Experimental comparison to validate the model is essential since there is often a large discrepancy between two-phase flow/heat transfer models and experimental results. Another important issue in implementing this design centers on the liquid requirement. A tradeoff will exist between limitations on water supply and heat transfer performance, forming the basis for characterization experiments. Recommendations for further experimental investigation into the effects of direct liquid spray cooling are indicated in section 4.

- Measurements of the heat transfer characteristics in simple steady-state spray cooling experiments with the following three important parameters studied: the surface temperature, T_s, the mass flux, G, and the nozzle type. The nozzle type and mass flux chosen for further experiments should give the highest heat transfer for a given liquid consumption.

- On a single wheel subject to drag braking, similar to experiments run at AAR in Chicago (Stone 1994), the tread temperature could be measured with a sliding contact thermocouple with and without cooling for a known, fixed drag brake power. The effect of the mass flux, G, and fraction of area sprayed, A_*, should be characterized.

4. CONCLUSIONS AND RECOMMENDATIONS

4.1 CONCLUSIONS

Several conceptual redesigns for a railcar brake have been reported to address the problem of wheel tread and shoe overheating during stop braking. The models developed to predict performance of each design show varying degrees of success in terms of thermal performance. It must be emphasized that these schemes are conceptual because their successful application requires that close attention be paid to mechanical and material designs in the context of a brake system. In addition, some designs will require substantial investment in a maintenance/construction "infrastructure" to operate a fleet of railcars. The following conclusions have been developed based on the study results:

1. The simplest of all redesigns gave poor thermal performance according to the model developed. The grooved brake shoe would represent a simple change but gave negligible heat transfer augmentation to the shoe (even with a high thermal conductivity material) and would still require maintenance to keep the fin passages clean.

2. Significantly better performance is obtained using a high thermal capacity brake shoe with integral fins. This design relies on a high thermal conductivity brake material that can also act as a friction element. Experiments to evaluate this material and the thermal contact between it and a higher capacity sink were discussed. This design also requires much maintenance to keep the fins free from debris.

3. Using the brake shoe-water jacket design, significant heat can be removed from the brake shoe during a stop cycle with modest amounts of water consumption. Heat transfer performance was not quantitatively evaluated but was shown to be controlled by the potential occurrence of critical heat flux (CHF). This design would require investment in a liquid supply system serving each brake shoe.

4. Direct cooling of the wheel tread by impingement spray cooling (without shoe redesign) was investigated and found to have great potential for keeping the wheel cool. Experiments were outlined to further investigate this idea. A liquid distribution system is needed and may require much maintenance to keep the spray nozzles from being clogged. However, the degradation in braking efficiency, if any, from water acting as a lubricant needs to be investigated.

Of the four possible brake shoe designs considered in this report, the direct liquid spray cooling of the wheel rim and the brake shoe holds a high potential for success in reducing the peak temperature attained by the brake shoe. However, very little experimental data exist on the "boiling heat transfer" of liquids on a moving surface (wheel rim). It is not known what the optimal characteristics of the spray are for maximum heat transfer. The spray characteristics include the liquid drop size distribution, mean drop size, velocity of the spray, size of spray relative to the width of the weel rim, angle of spray relative to the vertical direction. Since the goal is to minimize the amount of water needed for a given cooling load (i.e., reduction of wheel rim peak temperature to a desired lower temperature), it is essential to obtain heat transfer data from a test with a railcar wheel sprayed with water when the wheel is rotating and is hot. In addition, the reduction in the coefficient of friction between the brake shoe and the wheel rim wetted by water can also be determined.

4.2 RECOMMENDATIONS

It is recommended that a series of wheel rim cooling tests be performed to:

1. Obtain data on the heat transfer characteristics in a simple steady state spray cooling of a rotating, hot wheel/rim. Both the wheel surface temperature and the spray liquid nozzle characteristics (i.e., droplet distribution, size of mean drop, and overall area of wheel rim coverage) must be measured. The angle of the spray centerline with regard to the vertical must also be varied (one with negative angle of spray attack, one normal to the wheel, i.e., in the radial direction, and one trailing spray angle).

   This series of tests will provide valuable data that are not currently available in the literature. This would include the characterization, in terms of dimensionless parameters, of the boiling heat transfer process on the hot moving surface (such as the rim of a rotating wheel). The boiling Nusselt number may be correlatable with the initial temperature difference, wheel rotational speed, and a non-dimensional number, including the liquid properties and mean droplet size.

2. Evaluate the peak wheel rim temperature under the conditions of drag braking and intermittent braking and compare the test results for the peak wheel rim temperatures with model results indicated in this (and other related) report(s).

   A series of tests similar to the one conducted by AAR (Stone 1994) should be conducted. The tread temperature could be measured with a sliding contact thermocouple with and without cooling for a known, fixed drag brake power. Two or three thermocouples may be located at several depths within the tread (e.g., 1 mm, 5 mm, and 10 mm below the tread surface), onset at each of the opposite ends of the wheel diameter. Thermocouples should also be provided on the brake shoe surface (one at the leading edge of the shoe — relative to the direction of rotation of the wheel — and the other at the trailing edge). These thermocouples should be capable of measuring the peak temperature attained by the brake shoe (i.e., as close to the brake shoe-wheel rim interface as possible).

The tests by Stone (1994) included a track wheel in contact with the wheel rim to simulate the wheel-rail contact heat transfer. However, since the aim of the above recommended tests is to obtain certain important heat transfer data on liquid spray heat transfer characteristics on a rotating wheel, the provision of a track wheel may not be useful — it may add complications to the determination of effectiveness of the spray by providing another cooling path.

NOMENCLATURE

A	area	cm^2
A_*	area fraction	(%)
a_d	deceleration	m/s^2 or mph/s
b	fin spacing	m
B	Brake shoe thickness	m
c	solid specific heat	J/kg K
c_p	liquid specific heat	J/kg K
D,d	diameter	m
E	energy	J
g	gravitational acceleration	m/s^2
G	liquid spray mass flux rate	kg/m^2 s
h	heat transfer coefficient	W/m^2 K
h_{fg}	enthalpy of phase change	J/kg
k	thermal conductivity	W/m K
KE	kinetic energy	J
L	fin length	m
m	fin parameter, $(2h/kw)^{1/2}$	m^{-1}
m	mass	kg
Nu	Nusselt number	—
P	power	W
Pr	Prandtl number, ν/α	—
q''	heat flux	W/m^2
Q	heat transfer	J
$\dot{Q}$	heat transfer rate	W
Re	Reynolds number	—
t	time	s

NOMENCLATURE (continued)

t_c	time constant	s
T	temperature	K
ΔT	temperature difference	K
V	velocity	m/s *or* mph
w	fin width	m
W	railcar weight	N *or* lbs.
Z	brake shoe length	m

Greek

α	thermal diffusivity	m^2/s
δ	thermal penetration depth	m
η	fin effectiveness	—
ν	kinematic viscosity	m^2/s
ρ	density	kg/m^3
σ	surface tension	N/m

Subscripts

a	ambient
D	drop
f	liquid
g	vapor
h	hydraulic
i	initial
max	referring to a maximum condition
s	surface

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