NATIONAL COOPERATIVE TRANSIT RESEARCH & DEVELOPMENT PROGRAM

Report 14

Improved Service Life of Urban Transit Coach Brakes

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NATIONAL COOPERATIVE TRANSIT RESEARCH & DEVELOPMENT PROGRAM

Report 14

Improved Service Life of Urban Transit Coach Brakes

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Columbus, Ohio

AREAS OF INTEREST

Construction and Maintenance Equipment
Vehicle Characteristics
(Public Transit)

RESEARCH SPONSORED BY THE URBAN MASS TRANSPORTATION ADMINISTRATION OF THE U.S. DEPARTMENT OF TRANSPORTATION

TRANSPORTATION RESEARCH BOARD
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WASHINGTON, D.C. DECEMBER 1986

NATIONAL COOPERATIVE TRANSIT RESEARCH & DEVELOPMENT PROGRAM

Administrators, engineers, and many others in the transit industry are faced with a multitude of complex problems that range between local, regional, and national in their prevalence. How they might be solved is open to a variety of approaches; however, it is an established fact that a highly effective approach to problems of widespread commonality is one is which operating agencies join cooperatively to support, both in financial and other participatory respects, systematic research that is well designed, practically oriented, and carried out by highly competent researchers. As problems grow rapidly in number and escalate in complexity, the value of an orderly, high-quality cooperative endeavor likewise escalates.

Recognizing this in light of the many needs of the transit industry at large, the Urban Mass Transportation Administration, U.S. Department of Transportation, got under way in 1980 the National Cooperative Transit Research & Development Program (NCTRP). This is an objective national program that provides a mechanism by which UMTA's principal client groups across the nation can join cooperatively in an attempt to solve near-term public transportation problems through applied research, development, test, and evaluation. The client groups thereby have a channel through which they can directly influence a portion of UMTA's annual activities in transit technology development and deployment. Although present funding of the NCTRP is entirely from UMTA's Section 6 funds, the planning leading to inception of the Program envisioned that UMTA's client groups would join ultimately in providing additional support, thereby enabling the Program to address a large number of problems each year.

The NCTRP operates by means of agreements between UMTA as the sponsor and (1) the National Research Council as the Primary Technical Contractor (PTC) responsible for administrative and technical services, (2) the American Public Transit Association, responsible for operation of a Technical Steering Group (TSG) comprised of representatives of transit operators, local government officials, State DOT officials, and officials from UMTA's Office of Technical Assistance.

Research Programs for the NCTRP are developed annually by the Technical Steering Group, which identifies key problems, ranks them in order of priority, and establishes programs of projects for UMTA approval. Once approved, they are referred to the National Research Council for acceptance and administration through the Transportation Research Board.

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The needs for transit research are many, and the National Cooperative Transit Research & Development Program is a mechanism for deriving timely solutions for transportation problems of mutual concern to many responsible groups. In doing so, the Program operates complementary to, rather than as a substitute for or duplicate of, other transit research programs.

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The members of the technical committee selected to monitor this project and to review this report were chosen for recognized scholarly competence and with due consideration for the balance of disciplines appropriate to the project. The opinions and conclusions expressed or implied are those of the research agency that performed the research, and, while they have been accepted as appropriate by the technical committee, they are not necessarily those of the Transportation Research Board, the National Research Council, or the Urban Mass Transportation Administration, U.S. Department of Transportation.

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FOREWORD

By Staff
Transportation Research Board

The research described in this report advances the state of knowledge concerning factors that influence the service life of urban transit coach brakes and suggests procedures and materials for improving brake lining life. A major activity of the research was the field documentation of in-service temperatures of brake systems for several bus types. The higher temperatures developed in the brake systems of the Advanced Design Buses (ADB) were determined to be sufficient to initiate higher brake lining wear rates. The findings will be of particular interest and value to urban transit system administration and bus maintenance personnel.

The operation and maintenance history of "ADB" transit coaches shows a rather dramatic decline in brake life compared with earlier "New Look" coaches. Some of the factors associated with this decline in brake life appear to be (a) increased gross vehicle weight and (b) increased operating speed. The increased brake temperatures caused by these factors are believed to be the primary influence on reduced brake life.

The objectives of the research conducted by Battelle Memorial Institute were to (a) document the in-service operating temperatures of brake systems for "ADB" and "New Look" buses and (b) develop practical methods for improving service life of brake systems by reduction of operating temperatures and identification of brake lining materials with improved resistance to high temperatures.

Conduct of the study involved the instrumentation of various components of brake systems on three buses of the Southern California Rapid Transit District (Los Angeles) and the collection of temperature data during in-service operation of these buses. The data provide documentation of brake drum and brake lining temperatures for front and rear wheels of the different buses under several different types of operating conditions. The general findings indicate that the in-service brake lining temperatures of some buses often exceed the "threshold" temperatures for conventional asbestos-phenolic lining materials resulting in excessive wear.

During the study, attempts were made to reduce brake temperatures during operation by increasing the air flow over the brakes. Electric blowers to increase convective cooling of the brakes were helpful in reducing temperatures, but their application to existing buses was not considered cost effective.

Evidence was obtained indicating that driving style could have a substantial influence on brake temperatures. Consequently, education and training in driving style could be helpful in improving brake life. Because of possible health hazards, there is a general shift from asbestos brake linings to linings containing nonasbestos fibers or to semimetallic materials. Preliminary studies suggest that these materials may be more resistant to higher temperatures.

CONTENTS

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The research report herein was performed under NCTRP Project 47-1 by Battelle-Columbus Laboratories. The program manager was Dr. Allen T. Hopper, and the principal investigator was Mr. Scott A. Barber. Other contributors include Mr. Joseph Hoess, who conducted the economic analysis and was responsible for collecting data from transit authorities, and Dr. Richard Razgaitis, who formulated the finite element model and conducted the thermal modeling exercise. Mr. Dean Stockwell was instrumental in designing and fabricating the instrumental hardware associated with data collection, while Mr. James Tuten was responsible for assembling the necessary instrumentation and conducting the necessary field measurements.

Various non-Battelle staff were large contributors to this effort. Mr. Wilbur "Buck" Ewing and George Hallenbrook, both of Central Ohio Transit Authority (COTA), assisted in the initial stages of developing suitable onboard instrumentation; their help was very much appreciated. Likewise, Mr. Frank Kirshner and Mr. James Wagner of Southern California Rapid Transit District (SCRTD) provided all the assistance needed to conduct in-service experiments with a minimum of problems. The experimental efforts of the project could not have been successfully conducted without the assistance of COTA and SCRTD personnel.

Organization of the final report was the responsibility of Mr. Scott Barber and Mr. Joseph Hoess.

IMPROVED SERVICE LIFE OF URBAN TRANSIT COACH BRAKES

SUMMARY

This research was concerned with investigating reported reductions in brake lining life in Advanced Design Buses manufactured by General Motors and Flxible. According to a survey of six transit authorities, the average lining life exhibited by the Advanced Design Buses (ADB) manufactured by General Motors and Flxible was 22,000 miles, compared with an average of 40,000 miles exhibited by the older "New Look" bus manufactured by General Motors. The objectives of the research were to determine the causes of increased lining wear and to develop pragmatic, "bolt-on" solutions for improving lining wear.

In-service measurements of brake drum temperatures were made on both Advanced Design Buses and on an older "New Look" bus. These measurements indicated that the temperatures of the brake drums of the GM RTS-ADB were slightly higher than the temperatures measured for the "New Look" bus. Because of the relationship between wear rate and temperature for asbestos-phenolic friction materials, the slight temperature increase observed in the GM ADB was determined to be sufficient to initiate higher wear rates in the brakes. The most likely culprit responsible for increased drum temperatures was an increase in stopping velocity resulting from the slightly more powerful engine and drive train in the Advanced Design Buses.

Analysis of the in-service data showed that although the convective conditions around the brake drums of the ADBs were similar to those observed for the GM "New Look" bus, more convective cooling was necessary to reduce the drum temperature of the GM ADB. To improve convective conditions, an air scoop, brake shoe retractors, and electrical blowers were evaluated under inservice experiments to determine the technical feasibility of these schemes. Only the electrical blowers were capable of providing the convective cooling necessary to reduce brake drum temperatures to optimal levels.

Economic analysis of the electric blower retrofit scheme showed that although an improvement in lining life was forecast, the costs savings were balanced out by the estimated costs of implementing and maintaining the units. This retrofit technique is not recommended.

The most realistic technique of improving lining life appears to be the use of lining materials having better temperature resistance. Preliminary studies suggest that asbestos-free friction materials, or semimetallic materials, may have these desired properties.

CHAPTER ONE

INTRODUCTION AND RESEARCH APPROACH

BACKGROUND

Various metropolitan transit systems have reported a reduction in brake lining life in the Advanced Design Buses (ADB) manufactured by General Motors and Flxible. In some instances, the brake life has been reduced by a factor of two compared with the early "New Look" coaches that are still in use. This difference could be a result of a number of factors, including (1) increased gross vehicle weight; (2) increased load factor; (3) changes in brake actuation; (4) changes in federal regulations concerning brake performance; (5) changes in body configuration; and (6) improved drive train efficiency. These and other factors could impose severe operating conditions on the brakes, leading to either an increase in the amount of energy dissipated during braking or a reduction in brake cooling.

Table 1 shows the bus designs under examination. The design factors most related to brake actuation and brake lining life are gross vehicle weight, method of brake actuation, and changes in drive train efficiency. While initially designed for more efficient operation, the Advanced Design Buses are slightly heavier than the "New Look" buses they were designed to replace. The increase in curb weight represents an approximate 15 to 20 percent increase in the amount of energy that must be dissipated through the brakes during stops. In addition, the ADB manufactured by General Motors (GM RTS-II) employs a floating wedge brake design purported to be more effective in braking than the S-cam system used on the older "New Look" buses. This brake design has certain features which may promote drag and cause excessive wear rates. However, inasmuch as the Flxible ADB employs the S-cam design and reportedly also experiences excessive brake lining wear, the method of brake actuation may not account for reportedly higher brake lining wear rates. Finally, changes in the engine and transmission in the ADBs may allow drivers to accelerate to higher speeds between stops, leading to an increase in the stopping velocity. Because the energy dissipated during braking is proportional to the square of the stopping velocity, this factor alone could have a large effect on lining life, depending on how this energy is dissipated during braking.

The increase in the amount of energy dissipated during braking could lead to an increase in the operating temperature of the brakes. Most composite brake materials, with and without asbestos, contain a variety of organic and inorganic compounds that are held together in a matrix by a phenolic resin. At operating temperatures that are below the degradation temperature of the resin, the composite material may act as a continuous solid during sliding. At elevated temperatures, however, the binder resin may break down and cause the other matrix constituents to be freed from the material matrix at much higher rates than before.

The purpose of this program was first to determine whether excessive brake system temperatures in the Advanced Design Buses could account for observed decreases in brake lining life and second to design retrofit schemes to reduce system temperatures and prolong lining life. Although alternative systems, such as disc brakes and vehicle retarders, have demonstrated the ability to improve lining life, these systems require extensive modifications and are therefore not economical alternatives for most transit systems.

Battelle's Columbus Laboratories, under NCTRP Project 47-1, conducted a program to investigate plausible techniques for improving transit coach brake life. The purpose of this report is to present the findings and related recommendations of this study.

PROGRAM OBJECTIVES

The objectives of this program were (1) to verify or refute the premise that elevated brake temperatures in Advanced Design Buses (ADBs) lead to reduced lining life; (2) to determine cooling trends for the brake systems of transit coaches; and (3) to design and evaluate, from a technical and economic viewpoint, schemes to reduce brake drum temperatures and therefore improve brake lining life.

Specific experimental and analytical efforts directed toward these objectives are described in this report. Specific tasks included: (1) an analysis of current costs of brake maintenance; (2) establishment of temperature–wear properties of asbestos-phenolic friction materials; (3) establishment of current temperature and cooling trends in actual transit buses; (4) design of retrofit techniques for improving lining life; and (5) technical and economic analysis of retrofit techniques. The organization of these tasks is described in the next section.

PROGRAM DESIGN

The efforts briefly described in the previous section were directed toward resolving the following questions:

1. *Do Advanced Design Buses (ADBs) exhibit reduced brake lining life compared with the older GM "New Look" buses used as standard equipment with many bus transit authorities?* To resolve this question, Battelle contacted various transit maintenance officials by mail and by phone and asked them to provide data on frequency of reline and cost of reline for an older, well-established coach design (GM 5307 "New Look") as well as

Table 1. Buses selected for brake temperature study.

Bus	Approximate Weight lb	Engine	Transmission	Brake Actuation	Brake Width, in. F	R	Slack Adjustment	Shoe Manufacturer
GMC 5307 (New Look)	24,000	8V71	VS2 (3-sp)	S-Cam	5*	10*	Manual	Carlisle
GM RTS II-4	26,500	6V92 Turbo	730 (3-sp)	10° Double Wedge	6**	10**	Auto	Carlisle
Flxible 870	26,500	8V71 Turbo	730 (3-sp)	S-Cam	6*	10*	Auto	Carlisle

* 14-1/2 in. diameter drums.

** 15 in. diameter drums.

for the two Advanced Design Buses (ADBs) manufactured by General Motors and Flxible (GM RTS-II and Flxible 870). On the basis of these data the researchers sought to verify that the newer buses were actually getting fewer miles before brake reline than were obtained by the older GM 5307 coach.

2. *Do the brake shoes of the ADBs encounter higher temperatures during service than do the older GM "New Look" buses?* To resolve this question, a series of in-service experiments were conducted in which brake drum and brake shoe temperatures were measured and recorded for the following three bus designs: GM 5307 "New Look", GM RTS-II ADB, and Grumman Flxible 870 ADB. On the basis of these data the researchers determined: (a) the average temperature levels for the brake systems of the three buses while in-service, and (b) the cooling modes experienced by the brake drums of each bus design.

3. *Are the temperatures encountered by the brake shoes of the ADBs high enough to influence brake lining life?* To resolve this question, a literature search on the wear performance of asbestos-phenolic brake lining materials at various temperatures was conducted. With these data the researchers determined an optimal temperature range for this class of materials.

4. *If brake system temperatures are found to be high enough to cause accelerated brake shoe wear in the ADBs, are there any economical methods of reducing brake system temperatures and prolonging brake shoe life?* To resolve this question, techniques for altering the cooling rates of the brake drums to reduce temperatures to optimal levels were designed using data collected in the in-service experiments. These retrofit schemes were evaluated from both a technical and an economic standpoint to determine the feasibility of using the technique to prolong brake lining life.

Figure 1 shows a flow chart of the program. A number of experimental and analytical activities were conducted in combination with data collected from transit companies and published sources. Phase I activities were directed toward analyzing brake system thermal characteristics and designing techniques for reducing brake system temperatures, and Phase II was directed toward evaluating the technical and economic credibility of the techniques developed in Phase I.

Table 2 describes the type of activities in each task. In-service measurements of transit bus brake system temperatures required the designing and assembling of specialized hardware to perform this task. Data acquisition equipment was mounted on transit buses in Los Angeles and used to collect data during regular passenger service under urban traffic conditions. Analysis of the collected data was performed using a finite element model formulated for this study.

The program organization consisted of various types of investigative activities. Detailed descriptions of these activities and results are presented in the next chapter of the report.

Table 2. Description of program tasks.

Task	Method of Investigation
Measure current brake system operating temperatures	In-service measurements of brake lining and brake drum temperatures
Determine temperature-wear properties of friction materials	Literature search of published papers
Determine system cooling characteristics	Finite-element modeling of brake drums
Design schemes for temperature reduction	Convective heat transfer analysis of brake systems
Measure system temperature after retrofit	In-service measurements of brake lining and brake drum temperature
Determine reline frequency for brakes	Data collection through telephone and letter solicitation of transit maintenance directors
Calculate cost of brake service for life of bus	Examine analysis through constructed model

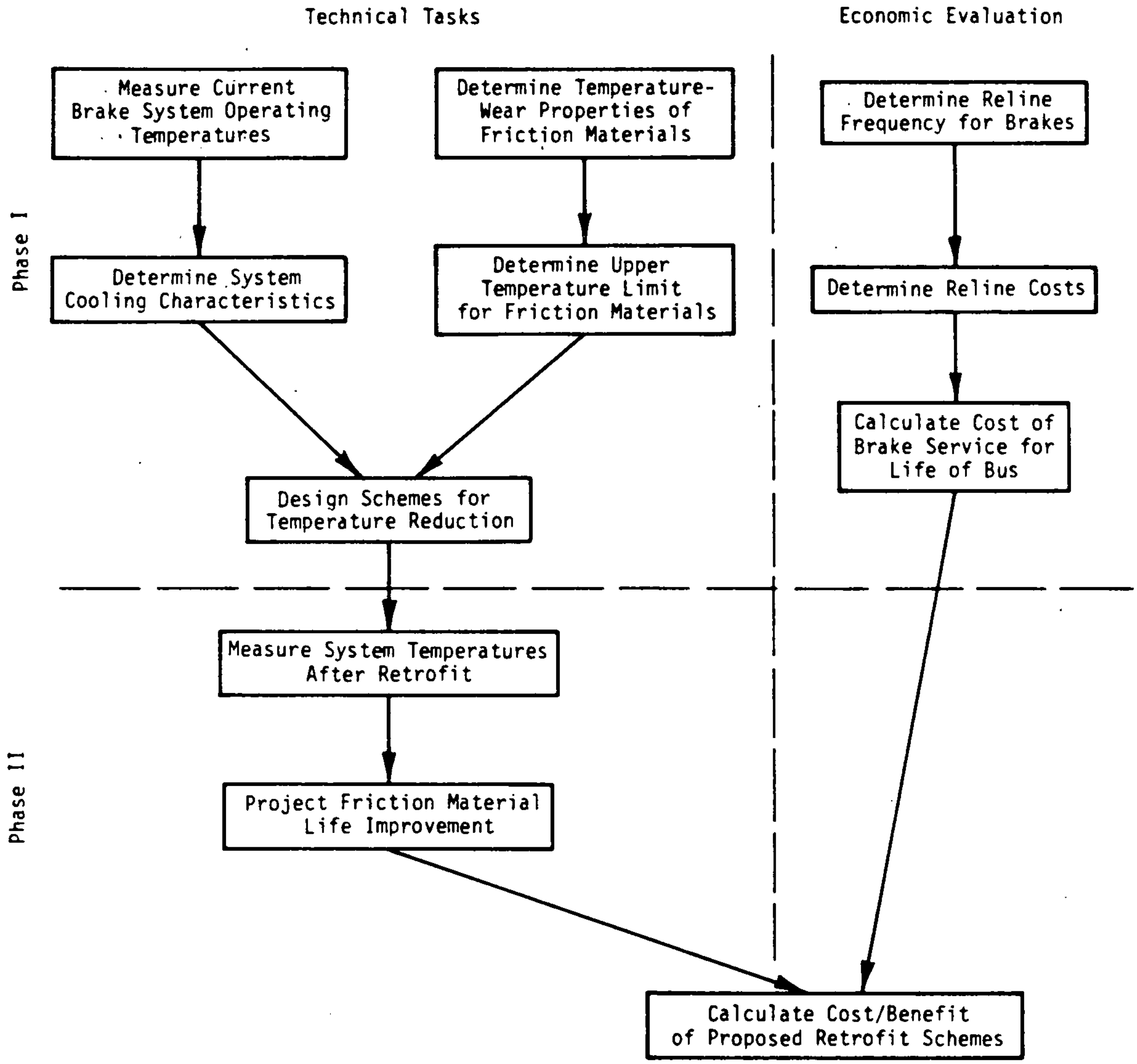


Figure 1. Flow chart of program activities.

CHAPTER TWO

FINDINGS

Because of the variety of tasks involved, this chapter is intended to describe the program activities depicted in Figure 1 in more detail. Technical activities of Phase I will be described first, followed by the technical activities of Phase II. The economic analysis of both phases will be described last.

EVALUATION OF TEMPERATURE-WEAR PROPERTIES OF FRICTION MATERIALS

In order to assess the possibility of thermally induced brake lining wear, a literature search of composite-brake-lining-wear data versus temperature was conducted. This activity was di-

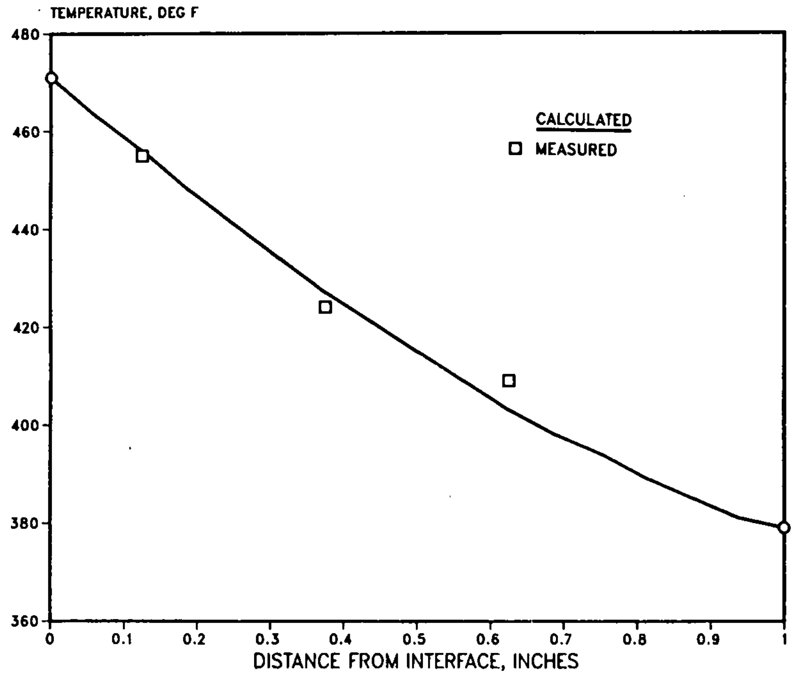


Figure 2. Drum temperature vs distance from drum/shoe interface during nonbraking time segment.

rected toward determining operating temperature limits for typical materials presently used in vehicle brakes.

The temperature–wear characteristics of the brake materials, as determined through the literature search, depict qualitative rather than quantitative trends. These data exhibit considerable scatter because of differences in test procedure and materials from one study to the next, although they do indicate that for most phenolic-asbestos brake lining compositions there exists a threshold temperature above which lining wear increases exponentially. The researchers attempted to quantify this threshold temperature and model composite wear above the threshold temperature.

Temperature Term Definitions

Several temperature-related terms are used in the discussion of the experimental results, and these terms are defined in this section. The terms refer to temperatures measured at different locations in the brake systems at different times during the braking process. Terms related to a brake drumshoe system as opposed to a brake disc-pad system are discussed.

Near surface steady-state temperature refers to the brake drum temperature measured by a thermocouple placed in the brake drum close (~ 0.125 in.) to the brake drum-brake lining interface. Figure 2 shows an example of brake drum temperature data collected during a nonbraking portion of bus operation in the field experiments. The data have been extrapolated to estimate the interface temperature. At this data segment, the near-surface temperature is 455 F, and the interface temperature is estimated to be 470 F. Under conditions of braking, the near-surface thermocouple measurement was about 15 F or 20 F lower than the interface temperature.

Near-surface transient temperature refers to the temperature near the braking interface during braking as measured by a thermocouple close (~ 0.125 in.) to the interface. Figure 3 shows temperatures both before and during braking for a bus stopping from 27 mph. Prior to braking, the near-surface temperature was 460 F. During braking, the near-surface temperature rose

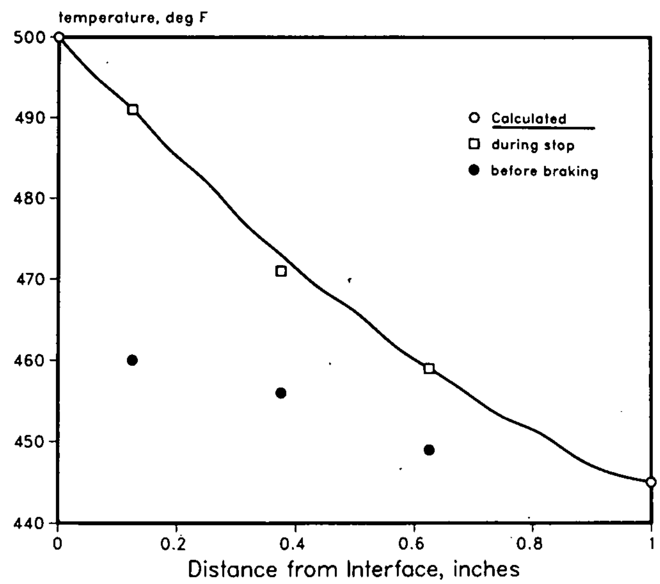


Figure 3. Brake drum temperatures measured before and during braking.

to 492 F, with an extrapolated interface temperature of 500 F. Under conditions of braking, the near surface thermocouple measurement was about 8 F lower than the interface temperature.

Literature references describing studies conducted on the temperature–wear properties of friction materials use a number of terms. *Brake soak temperature* refers to steady-state temperatures that exist in brake drums (or discs) and brake linings. These soak temperatures are usually measured by thermocouples and generally reflect an average or lumped temperature of the components. With respect to brake lining materials, however, there generally is no good correlation between brake soak temperature as measured in experiments and brake lining wear (*1*). One reason for the poor correlation is the large temperature gradient, which typically exists across the lining, that is due to the low thermal conductivity of typical lining materials (*2*). Because of the large temperature gradient, the temperature throughout the lining may vary widely so that slight differences in thermocouple placement will record wide fluctuations in the measured temperature. For this reason, studies of wear versus lining soak temperature represent fairly qualitative studies.

Factors Influencing Drum-Shoe Interface Temperatures

For this program, efforts were directed toward determining the temperature of the interface between the rotating brake drum and the brake shoe. Heat is generated at the interface and dissipated into both the drum and the shoe. Because most composite friction materials exhibit low thermal conductivity, steep thermal gradients exist in the shoe materials. Under excessive braking conditions, a thin heat-affected zone of degraded material may exist on the surface of the shoe. Thermal degradation of the shoe was assumed to occur when the interface temperature was higher than the threshold failure temperature of the composite.

The interface temperature during braking is the sum of the steady-state drum temperature achieved through previous multiple brake applications and the transient temperature at the surface caused by the unsteady heat transmission through the drum. The temperature profiles at various depths below the interface temperature both before braking (steady-state) and during braking (transient) are illustrated in Figures 2 and 3. Estimates of interface temperatures from brake shoe temperature measurements were considered invalid because of the material inconsistencies often found in composite materials and because of the low thermal conductivity of the material.

An analysis of the steady-state temperatures achieved by a brake drum after multiple brake applications was conducted by Fazekas (*2*). Assuming the drum to be lumped, one may obtain the drum temperature by the following equation:

$$T_{ss} = T_o + Q\left(\frac{1}{hAt_o} + \frac{1}{2\,WC}\right) \qquad (1)$$

where the quantity *WC* refers to the product of the drum weight and the heat capacity of the drum material. This product describes temperature excursions experienced by the drum during braking, which was considered to be negligible for the lumped mass. The equation was reduced to:

$$T_{ss} = T_o + Q\left(\frac{1}{hAt_o}\right) \qquad (2)$$

where T_o = ambient air temperature; Q = heat input; A = brake drum surface area; h = convection heat transfer coefficient; and t_o = time between braking.

With the further assumption that $Q = \frac{1}{2}mv^2$, the equation becomes

$$T_{ss} = T_o + \frac{mv^2}{2hAt_o} \qquad (3)$$

where m = bus mass, and v = bus stopping velocity.

Equation 3 shows that as the time between stops and the amount of convective cooling increases, the average temperature of the drum during service decreases. Likewise, an increase in braking velocity will cause an increase in steady-state temperature proportional to the velocity squared.

An examination of the transient interface temperature use during braking was conducted by using solutions derived by Carslaw and Jaeger (*3*). Modeling the drum as a semi-infinite solid and assuming the heat input from braking to be constant, an expression for the transient temperature at the surface is:

$$T_t = T_{ss} + \frac{2Fo}{K}\left(\frac{\alpha t}{\pi}\right)^{1/2}\left(1 - \frac{2}{3}\frac{t}{T}\right) \qquad (4)$$

where T_t = transient interface temperature; Fo = heating rate/area; t = duration of heat application; K = thermal conductivity; α = thermal diffusivity; T_{ss} = initial steady-state temperature; and T = stopping time.

Figure 4 shows the temperature profile and interface temperature history for sliding contact conditions similar to those at the brake drum-brake shoe interface of a transit bus during

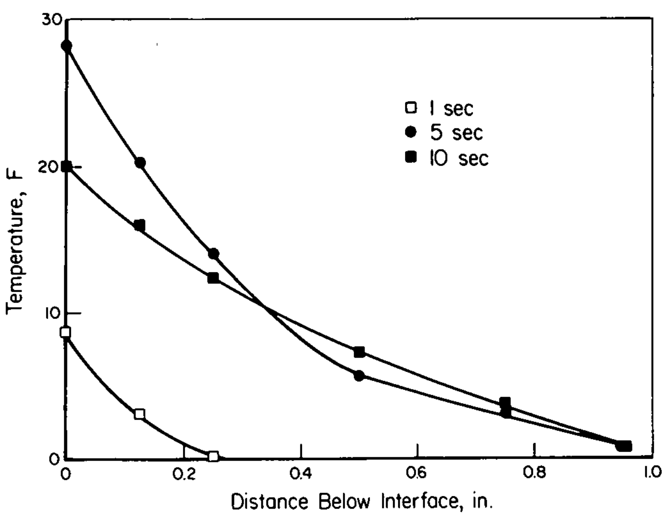


Figure 4. Temperature profiles in semiinfinite solid with constant heat flux (initial temperature = 0°F).

braking. The heat input was chosen to replicate the amount of energy dissipated into four brake drums by a 24,000-lb bus stopping from 25 mph in 10 sec. After 5 sec of application, the interface temperature has increased to 28 F over the initial steady-state drum temperature. This compares favorably with general interface temperatures extrapolated from data, as shown in Figure 3.

Equations 3 and 4 show that the transient interface temperature is a function of the stopping energy and the steady-state temperature. The steady-state temperature is, in turn, a function of the stopping energy, the convective cooling coefficient, the drum convective area, the time between stops, and the ambient temperature. Practical techniques for reducing the interface temperature seen during braking can only be directed at reducing the steady-state temperature, since the transient temperature rise is material-dependent and operator-dependent. Reducing the steady-state temperature without redesigning the brake components or regulating operators' conduct can best be accomplished by increasing convective or conductive cooling of the drums.

In summary, under typical stopping conditions (25 to 27 mph initial velocity), interface temperatures seen by the composite brake lining were about 30 to 40 F higher than the temperature before braking. This temperature level was confirmed by analytical studies of the transient temperatures. Practical techniques for reducing the net interface temperature seen by friction material hinge on reducing steady-state temperatures of the brake drum.

Temperature–Wear Properties for Asbestos-Phenolic Friction Materials

Phenolic resin-asbestos linings typically exhibit constant wear at relatively low temperatures until a threshold temperature is reached. Above this threshold temperature, lining wear increases exponentially with further increases in temperature.

Newcomb and Spurr (*4*) indicate that most molded phenolic resin-asbestos linings have a maximum operating temperature ranging from about 350 F to 440 F, depending on the intended duty and composition. At temperatures above this threshold temperature, the phenolic resin binder begins to burn away and the composite structure deteriorates rapidly. The exact temperature of decomposition is affected by the remaining organic and inorganic constituents that make up the lining composition.

Table 3 gives estimates provided by Ref. *4* of the temperature limits of some phenolic-asbestos molded brake lining materials. Most automotive applications use either light duty or medium duty brake linings which usually demonstrate a maximum operating temperature of between 350 F and 400 F.

A number of experimental studies support these observations. Bush, Rowan, and Warren (*5*) used an inertial dynamometer and a disc brake assembly to investigate the wear of phenolic-based friction materials. During these experiments, an appreciable increase in brake pad wear occurred when the pad temperature increased from 350 F to 480 F. Spurgeon et al. (*6*) and Jacko (*7*) reported fairly constant brake lining wear up to a drum temperature of about 450 F, after which lining wear increased exponentially. Similar results were reported by Sinclair (*8*), with the "knee" of the temperature-wear phenomenon for the phenolic resin-asbestos material at about 450 F. Kwolek (*9*) examined experimentally the temperature–wear properties of several types of automotive brake lining materials for use in a solid brake disc rotor assembly. The normal organic resin lining showed an appreciable increase in wear above 450 F, while another organic resin material incorporating a higher percentage of inorganics and abrasives demonstrated constant, low wear up to rotor temperatures of 500 F and above. A semimetallic lining demonstrated very low wear up to rotor temperatures greater than 550 F. This particular study shows how higher operating temperatures can be endured by different, although more expensive, brake linings.

Table 3. Estimated temperature ranges for phenolic resin-asbestos brake materials (from Ref. *4*).

Molded Lining	f	Temperatures: Maximum F (C)		Operating Temperature F (C)	
Light Duty (flexible)	.40	660	(350)	350	(175)
Medium duty (semi-flexible)	.40	750	(400)	400	(200)
Heavy duty (rigid)	.35	930	(500)	440	(225)

A number of studies to express brake material wear mathematically as a function of temperature and load have been conducted. Rhee (*10, 11*) evaluated the temperature–wear characteristics of an asbestos-reinforced friction material and determined that at temperatures below 450 F at the dynamometer rotor, the wear rate increases linearly with temperature. Above 450 F, the wear rate increases exponentially with temperature. Using wear data generated at two different loads and three different temperatures, Rhee plotted an Arrhenius plot and determined that the average activation energy of the high-temperature wear process was similar to the energy associated with the thermal decomposition of phenolic binders, which suggests that the primary mode of wear at elevated temperature is degradation of the phenolic binder.

Lui and Rhee (*12*) demonstrated that below 450 F, asbestos brake lining wear generally follows the form:

$$w = \alpha \; P^a V^b t \tag{5}$$

where α = wear constant; w = wear rate; P = load; V = velocity; and t = sliding time.

Above 450 F, decomposition and pyrolysis of the binder leads to an exponential increase in brake lining wear. In this regime, the wear equation becomes:

$$W = \beta \; P^a V^b t \; \ell^{\left(-\frac{E}{RT}\right)} \tag{6}$$

where β = wear constant; E = activation energy; R = universal gas constant = 1.99 cal/mole K; and T = temperature (absolute).

For most asbestos linings with a phenolic binder, the activation energy is about 9.6 cal/mole K. To see the effect of temperature on wear at elevated temperatures, the ratio of wear at two temperatures, 500 F (533 K) and 450 F (505 K), can be estimated from Eq. 6:

$$\frac{W(500\text{ F})}{W(450\text{ F})} = \frac{e^{\frac{-E}{RT_2}}}{e^{\frac{-E}{RT_1}}}$$

$$= \exp\left[-4.82 \times 10^3 \left(\frac{1}{533} - \frac{1}{505}\right)\right] \quad (7)$$

$$= e^{0.501} \cong 1.65$$

At elevated temperatures, a 50 F temperature difference can account for an approximate 60 percent increase in brake lining wear. Therefore, a brake lining that endures 35,000 miles at 450 F might be expected to last only about 22,000 miles at 500 F, assuming load conditions remain fairly constant. This model also predicts that the negative effect of temperature on brake material wear can be alleviated somewhat by selecting a brake material having a higher activation energy, such as a composition having more filler material or a semimetallic sintered material.

Investigations of asbestos-free and semimetallic brake pad materials have also been conducted. Loken (*13*) investigated a Kevlar-reinforced brake material using a dynamometer with a constant drag force in which disc temperatures were monitored.

Kevlar-reinforced components performed substantially better than conventional asbestos materials, as did a semimetallic brake pad material used for comparison. Halberstadt and Rhee (*14*) evaluated a brake material composition in which 50 percent of the asbestos fiber had been replaced with potassium titanate fiber in an attempt to formulate a brake material composition having higher temperature stability. Experiments were conducted using a full-scale commercial dynamometer. Brake composition wear above 400 F was substantially improved (40 percent) using the new composition, in comparison with a standard commercial asbestos lining material. In these experiments, a lower threshold temperature (approximately 350 to 400 F) for the initiation of high-temperature wear was observed for the asbestos fiber linings. The addition of potassium titanate appeared to improve wear resistance above 450 F.

Semimetallics are currently under serious consideration for elevated temperature braking conditions. Lui and Rhee (*15*) evaluated three different commercial friction materials having various percentages of metal fillers using a simple dynamometer (Chase machine). They concluded that at elevated temperatures, the semimetallics wear by means of pyrolysis of the binder, although this degradation appeared to occur over a higher temperature range (500 to 625 F) than the organic asbestos composites (350 to 450 F). Loken (*13*) also presented data showing low semimetallic wear up to temperatures as high as 900 F. Kwokek (*9*) conducted a study of brake lining wear versus temperature for semimetallics being formulated for small car solid rotor use. These semimetallics exhibited steady, low wear up to temperatures of 550 F. From a standpoint of heat-induced lining wear, the semimetallics appear to be more wear-resistant at higher temperatures than the conventional organic resin-asbestos linings.

A summary of pertinent temperature–wear studies is presented in Table 4. Because exact formulations for the numerous compositions are not available, phenolic resin-asbestos composition is referred to in a general sense. Although Table 4 presents a somewhat broad range of operating temperatures for the various phenolic compositions, those studies in which alternative materials were evaluated are also identified.

Table 4. Summary of relevant wear-temperature brake lining studies.

Author	Ref. No.	Units for Wear Rates	Material and Threshold	Temperature, F
Bush, et al.	5	Wear per stop (mg) vs sliding distance (m)	Phenolic resin-asbestos	250
Spurgeon, et al.	6	Lining wear (gms) vs drum temp., F	Phenolic resin-asbestos	450
		Lining wear (mg/stop) vs lining temp., F	Phenolic resin-asbestos	450
Sinclair	8	Wear factor (in^3/hp-hr) vs temp.	Phenolic resin-asbestos	450
Halberstadt, et al.	14	Thickness change vs temperature	Phenolic-resin, modified with potassium titanite	400-450
Loken	13	Wear rate (gms/hp-hr) vs drum temperature	Phenolic-asbestos	400-450
Kwolek	9	Wear (in.) vs rotor temperature	Phenolic-asbestos Semi-metallic	450 550
Weintraub, et al.	21	Wear vs cast iron bulk temp.	Phenolic-asbestos	450
Anderson	1	Wear rate (cm^3/GJ) vs interface temperature	Phenolic-asbestos Semi-metallic	550 870
Jacko	7	Average wear (mm) vs max. temperature	Phenolic-asbestos	660

Table 4 illustrates two points. First, the temperature at which normal phenolic-asbestos linings begin to undergo pyrolysis and experience a subsequent increase in wear is not definite and varies from 400 F to 550 F, although 450 F was the value most consistently reported. Second, little comparison between studies with respect to wear rate of the lining materials is possible because of the variety of experimental procedures used in the various investigations. Although lining wear is frequently reported in terms of thickness change or weight loss, the procedures used to induce this wear vary. Only a few of the references contain sufficient information to calculate a comparable value of material loss as a function of both temperature and horsepower dissipated.

Figure 5 shows wear in the form of cubic inches of material lost per horsepower-hour as a function of temperature for several references. In general, the two data sets for the commercial phenolic resin-asbestos samples showed a wear transition at a temperature of around 400 to 425 F. The semimetallic data are considered to be typical in that the wear rate appears to be relatively constant over a wide range of temperatures. Some wear data for a heavy-duty asbestos-phenolic and truck brake lining are also shown.

The commercially available phenolic-asbestos lining materials demonstrated relatively constant wear up to an elevated temperature that ranged typically from 400 to 450 F. At temperatures above this point, pyrolysis of the phenolic binder is believed to take place, with an exponential increase in wear resulting. If one assumes a 40 to 50 F temperature rise at the interface during braking, a good upper limit for the steady-state temperature as measured by near-surface thermocouples would be 350 F, which would extrapolate to an interface temperature of around 400 F.

The next section describes the experimental activities conducted to measure brake system operating temperatures in-service.

MEASUREMENT OF SYSTEM TEMPERATURES

In-service brake drum and shoe temperatures were measured on the three coach designs described in Table 1 in Phase 1. Data from the GM 5307 "New Look" coach, which gives acceptable brake lining life, was intended to be used to establish acceptable brake system operating temperatures. Temperature data from

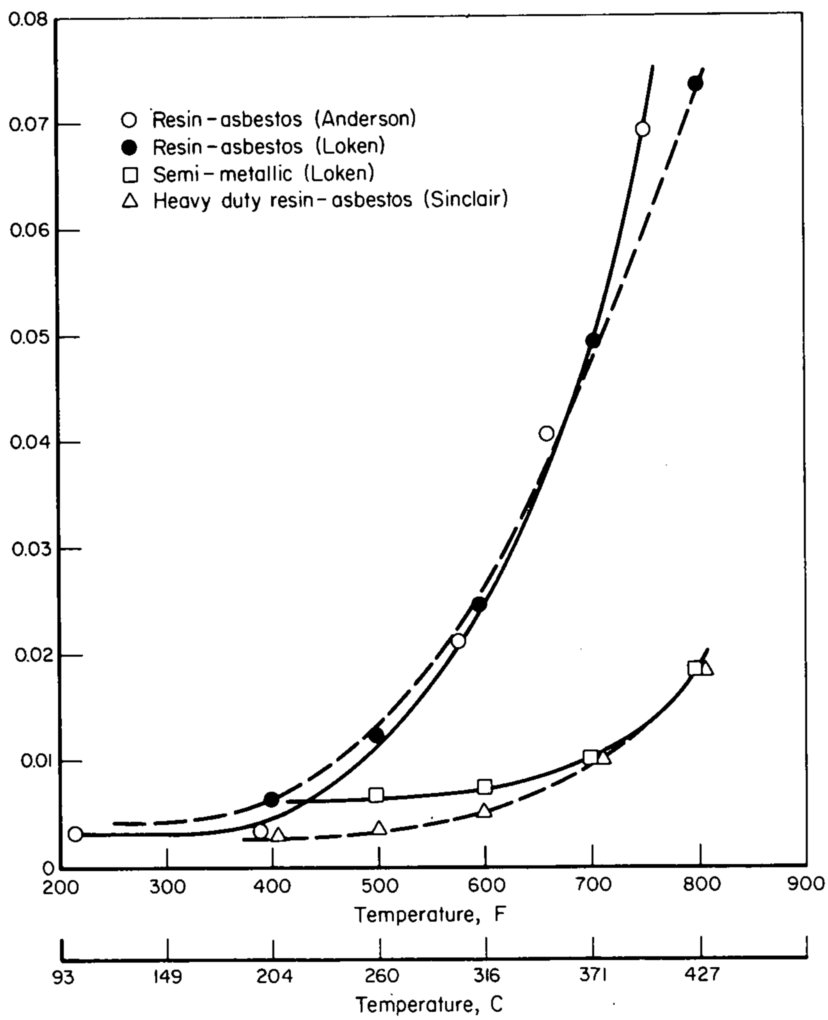


Figure 5. Wear-temperature relationship for selected brake material studies.

the two ADB buses were collected and compared to the baseline data provided by the GM "New Look" coach. In addition, temperature data provided by these experiments were used to calculate convective and conductive cooling characteristics for the various bus designs.

The following discussion describes the instrumentation used for the on-site experiments, as well as the routes selected for data collection. Results of these temperature measurements are also given.

Description of Instrumentation

The hardware required to measure bus brake temperature, bus speed, and brake air pressure was designed and fabricated at Battelle. The entire system is shown schematically in Figure 6. It consisted of thermocouples mounted in the front and rear brake drums and shoes on the street side of the bus, pressure transducers, a speed encoder, and the instrumentation required to record the data on magnetic tape. Measuring the bus speed required the design and fabrication of a frequency-to-voltage converter, which was designed to be connected directly to the transmission in the GM 5307 "New Look" bus or to the front wheel of the two ADBs.

Two types of thermocouple fittings were selected to measure brake lining and brake shoe temperatures. For the linings, type K thermocouples were silver-soldered into threaded copper tubing, which was later positioned into threaded holes in the lining itself. For the brake drum, chromel-alumel thermocouples were silver-soldered into small cast iron plugs that were shrunk-fit into counter-bored holes in the brake drum. These thermocouples were positioned in the drum so that conduction and convective heat losses could be calculated using the measured temperature profiles. Because of the low thermal conductivity of the composition shoe, extrapolation of temperatures from a variety of depths to the brake shoe interface was considered not to be feasible. However, in order to observe any bulk temperature gradients within the shoes themselves, thermocouples were po-

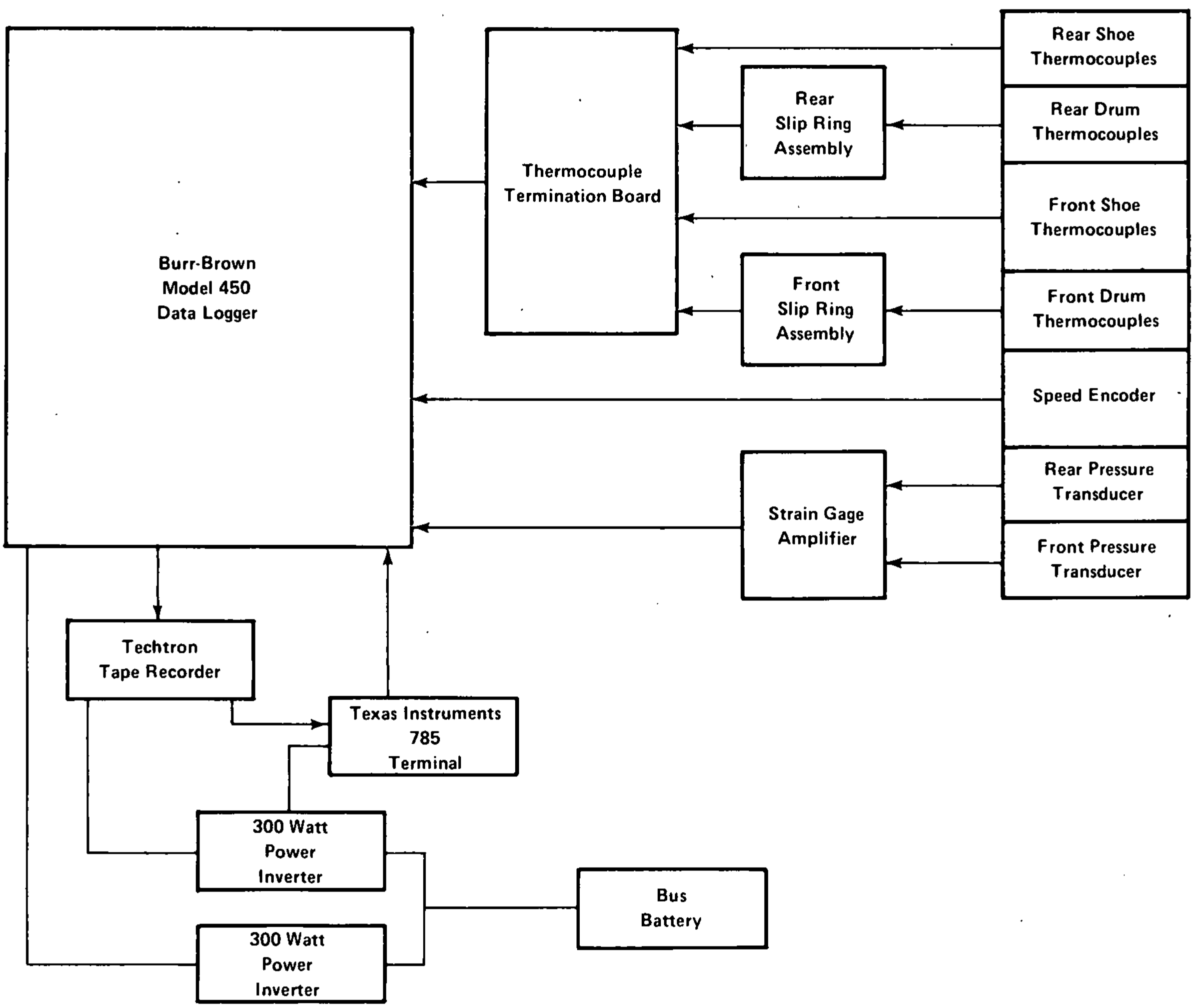


Figure 6. Block diagram of instrumentation.

sitioned periodically at two different positions: slightly recessed (~ $\frac{1}{16}$ in.) from the contact surface of the brake shoe, and slightly recessed into the back of the brake shoes behind the shoe backing plate.

Figure 7 shows the typical positions of the thermocouples in the brake shoes and drums of the three bus designs. The numbers beside each thermocouple position indicate the channel number on the data logger used to record the data. A thermocouple was placed between the brake drum mounting hub and the wheel rim of the tire in order to measure the temperature gradient in this region so that possible heat conduction paths from the brake assembly could be monitored. The interior ambient temperature of the brake drum was measured periodically so that an estimate of the convective heat transfer coefficient could be made.

Prior to installing the brake drums, brake shoes, and associated temperature-measuring instrumentation on the buses, the brake drums and brake shoes were cut to conformity using a Star Brake Drum Lathe. Careful measurement of the drum size was necessary so that the precise location of the drum thermocouples with respect to the drum braking surface could be determined. The distances of the thermocouples from the brake drum interface are indicated on the drawing. Since the temperature-time-distance data from the center plug thermocouples was used to calculate the interface temperature and other heat quantities, these distances were determined more accurately than the other three drum thermocouples.

Because the brake drum was rotating, slip-ring devices were fabricated at Battelle and used to transfer temperature signals

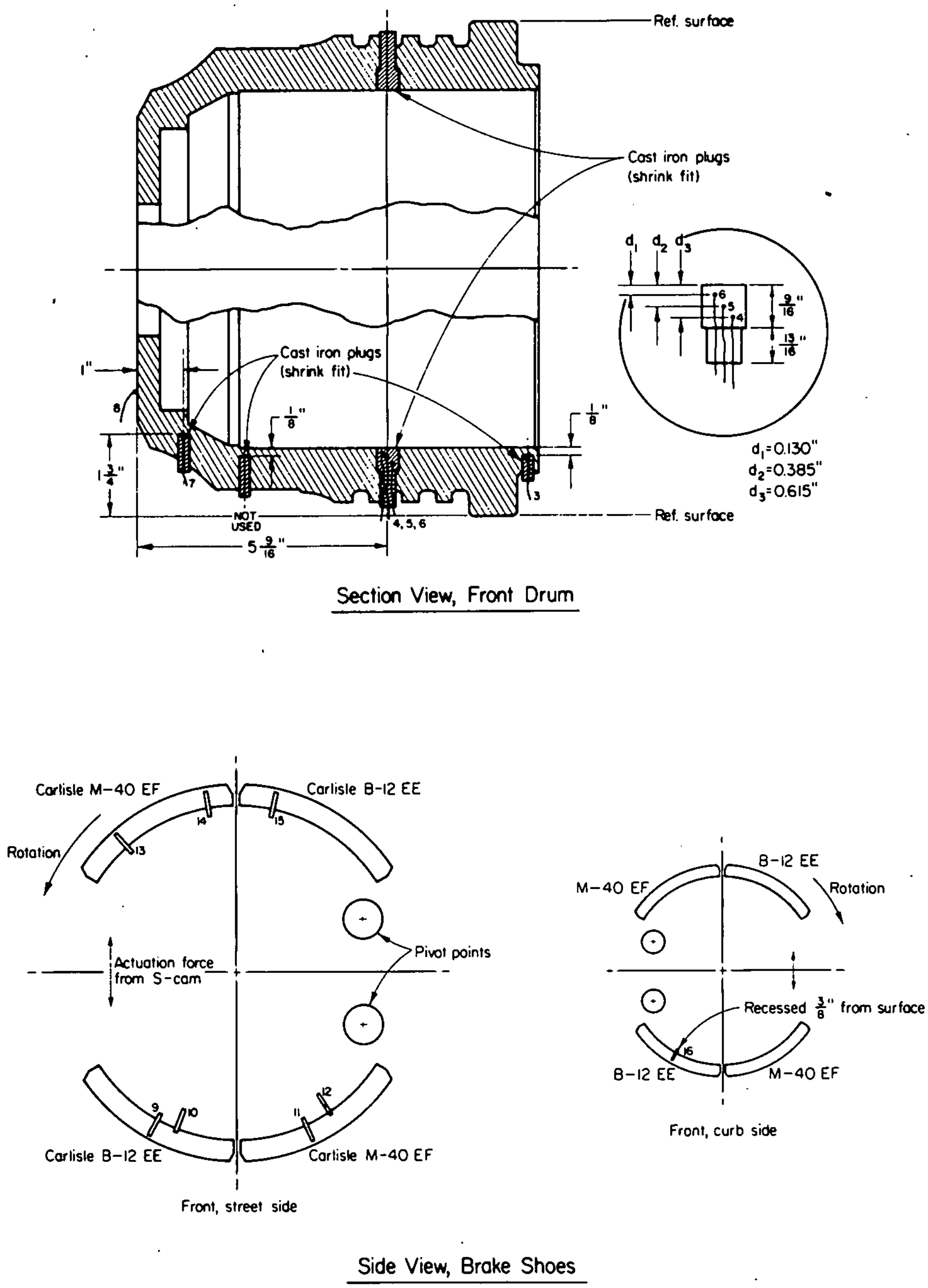


Figure 7. Location of front brake drum and brake shoe thermocouples, GM 5307 "New Look" bus, Phase I experiments.

from the rotating reference frame to the data logger. Two slip-ring assemblies, one for the front and one for the rear, were fabricated. Figure 8 shows a slip-ring assembly fixed to the front street-side wheel.

In order to check the temperature balance between the street-side and curb-side brake shoes, a single thermocouple was positioned in a segment of the curb-side shoe on both the front and rear of each bus. The position of this thermocouple is also indicated in the diagrams.

The brake drum and shoe thermocouples, plus the bus speed sensor and brake pressure sensors, were monitored and the signals recorded using a multichannel data logger. The sample rate of the data logger was primarily restricted by the time needed to sample through all 33 channels, which was approximately 6 sec. For the purpose of collecting general temperature history data, this sample rate was adequate. However, for the purpose of examining temperature histories during individual stops, which lasted from between 3 and 15 sec, this scanning rate was not rapid enough. For this reason, the temperature signals from the front brake assembly or the rear brake assembly only were monitored periodically at full scan rates. By reducing the total number of channels scanned, the number of temperature signals recorded per unit time was increased and more digital data per stop were recorded. Figure 9 shows the instrumentation used to collect the temperature data. This instrumentation was used to collect in-service temperature data for the three bus designs. The next section describes the experimental procedure and the routes selected.

Description of Experimental Procedure and Routes Selected for Study

During the month of June 1982, experiments were conducted on transit buses in Los Angeles to determine whether temperature differences exist between the various bus designs under comparable operating conditions. Instrumented brake drums for three different buses of three different bus designs (GM 5307 "New Look", GM RTS-II ADB, and Grumman Flxible 870) were prepared at Battelle and sent to Los Angeles for these experiments.

For these Phase I experiments, asbestos-phenolic friction materials were used. The same brake materials were used for all three bus designs. The leading edge material was Carlisle B-12 EE material, while the trailing edge was Carlisle M-40 EE material. These two materials exhibit different friction characteristics and contain different constituents (B-12 EE contains small brass fillings in the matrix, and the M-40 EE material was a straight brass-free composition).

After installation of brake shoes and drums on the bus, the bus was run under normal service conditions in order to wear-in the drums and linings. After wear-in, data were taken on each bus for two days under a variety of urban routes and conditions, and then the instrumentation was removed from the bus and installed on the next bus for temperature recording.

The bus operating line chosen for the experiments in Los Angeles was the Wilshire Blvd. route, designated as the 20 line according to SCRTD scheduling procedures. In order to keep passenger load, road grades, and the number of stops fairly constant for all three bus designs, the buses were run on the same routes during the tests. The 20 line designation refers to a series of separate sublines which service areas of Wilshire Blvd., the major urban route in Los Angeles. Each subline had a number of scheduled routes which operated under a variety of passenger and stop conditions. In all cases, urban routes were used for the test, and all tests were performed on weekdays.

The experiments were conducted on two sublines, designated as the 83/20 and 82/20, and each of these lines had a number of routes assigned to it. Appendix A describes the routes and schedules for these two lines.

Figure 8. Slip ring assembly on front wheel of COTA bus.

Figure 9. Data acquisition package in-place on COTA bus.

The 83/20 line consisted of five routes running between downtown Los Angeles and Ocean Boulevard in Santa Monica, which were designated by the numbers 308, 21, 22, 20, and 22B. The routes designated 308 were "limited" routes characterized by infrequent stops and long running intervals between stops, and this route was considered to resemble most closely the "moderate" route described previously. The remaining routes were curb-to-curb urban routes which spanned various passenger service loads during the day. The 83/20 run was selected because the routes spanned a long test period (from 6 a.m. to 10 p.m. continuous).

The 82/20 line also consisted of a number of urban routes, but the total duration of the run was somewhat less than that of the 83/20 line (116 miles vs 176 miles). Since the 82/20 run yielded a great deal of temperature data in itself, the data from the 83/20 run were considered to be "back up" data to be used to verify the validity of the 83/20 data.

Appendix A also indicates the average stopping speed and the number of stops. The number of stops refers to the number of "high energy stops" originating at a relatively high speed (greater than 15 mph) and does not take into account the frequent stops encountered when the bus moved slowly in inner city traffic. These data were compiled for the analytical model of the brake system.

For the experiments on the 83/20 routes, two drivers were used for the duration of service from 6 a.m. to 10 p.m. The researchers were able to minimize possible differences in brake system temperature as affected by driver selection by retaining the same two drivers for all the experiments on the 83/20 line. The first driver was characterized as being fairly aggressive in his driving style; rapid acceleration and sudden stops were the norm, and he followed closely behind urban traffic. The second driver was characterized as being fairly passive in his driving style. He regularly maintained a distance of four or five car lengths behind the front vehicle, and his stops were not particularly severe compared with those of the first driver.

Continuous temperature data for all three bus designs were recorded through the service day, including the stop-over points at the end of scheduled runs in which significant cool-down of the brakes was observed.

Results of Phase I Experiments

The data collected in-service were examined from two aspects: (1) the steady-state near-surface temperature history, which approximates the steady-state interface temperature; and (2) the convective cooling characteristics as determined by the temperature distribution within the brake drum. The wear-surface temperature history is discussed in the following paragraphs. The technique used to determine the convective cooling conditions is discussed in more detail in a separate section.

The near-surface temperatures for the brake drums of the GM 5307 and GM RTS-II were plotted against time to observe general brake system performance so that extreme temperature differences between each design could be examined.

Temperatures in the Flxible 870 were not plotted because of problems encountered with the particular bus selected for this study. During the experiments, temperatures in excess of 600 F were measured in the rear drums of bus number 7616, which was a Flxible 870 design. At the same time temperatures in the front drums remained fairly low at around 200 F. The brakes were checked and found to be properly adjusted, which suggested at the time that other system components could be at fault. Subsequent efforts by Flxible have indicated that on some early Flxible 870 buses, the brake treadle pressure valve, which directs compressed air from a main supply to the front and rear, could move out of adjustment. The maladjustment could cause substantial unbalance in the level of breaking, resulting in high rear drum temperatures. To remedy this potential problem, Flxible engineers recommend the replacement of faulty treadle assemblies with newer assemblies less prone to maladjustment. These changes have helped to lower component temperatures and improve lining life on these buses.

In order to observe the temperature history of each bus during normal service, the near-surface drum temperature was plotted for the duration of the 83/20 line on Wilshire Blvd. These plots were used to observe the brake system temperatures so that any extreme temperature differences between the bus designs could be determined. Temperatures for the Flxible 870 were not plotted because of the abnormally high temperatures measured in the rear drums.

Figure 10 shows the temperature history for the front and rear brake drums of two buses during the experiments in Los Angeles. They have been presented together to illustrate the temperature differences exhibited by the different drums. Several features can be seen. The front drum of the RTS-II consistently demonstrated higher bulk temperatures than the other three drum temperatures. The rear drums of the RTS and the 5307 demonstrated somewhat lower temperatures, although the general operating temperature for the rear drums of the RTS-II was slightly higher than that of the drums of the GM 5307. To find out how much higher the temperature of the drums of the GM RTS were, the temperature plot of Figure 10 was divided into multiple 12-min-long segments, and the temperature range for the front and rear brake drums of each bus was determined in each segment. Table 5 gives the results.

On the average, the temperature of the front brake drum of the RTS-II was about 110 F hotter than that of the GM 5307, while the rear drums of the RTS-II were about 30 F hotter than those of the GM 5307, for these experiments at SCRTD. SCRTD provided rough estimates of brake lining life for the GM 5307 and the GM RTS-II. The average lining life for the front axle of the GM RTS-II was around 16,500 miles, while the rear axle was around 25,000 miles. Mileage figures for the GM 5307 model were higher: 30,000 miles for the front, and 25,000 miles for the rear. Because SCRTD has reported similar lining lives for rear brake linings of both of these buses, the upper threshold temperature for the phenolic lining material was probably not

Table 5. Average drum temperature for two buses tested on route 83/20 in Los Angeles.

	Average Temp., F	
	Front Drum	Rear Drum
GM RTS-II	390	330
GM 5307	280	300

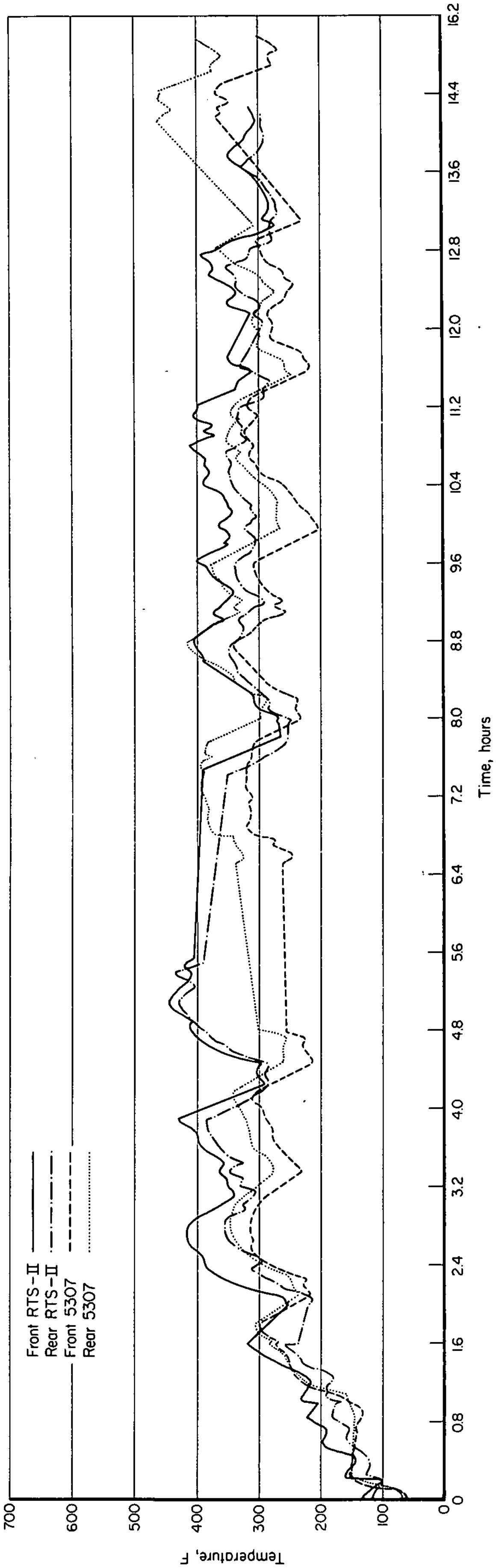


Figure 10. Temperature history for brake drums of GM RTS-II and GM 5307 buses during service in Los Angeles.

encountered for much of the service life of the rear linings of the RTS-II. Assuming the lining is exposed to temperatures 50 F higher than the steady-state during braking, the linings were exposed to 380 F at the rear drum and 440 F at the front drum. This suggests that an estimate of the lower limit for the threshold severe temperature is about 400 F at the interface. Linings which contact the drum at temperatures above this point may be expected to undergo high temperature pyrolysis. Assuming a 50 F interface temperature rise, an upper limit steady-state brake drum temperature to prevent excessive lining wear was estimated to be 350 F.

Various transit authorities were asked to provide information concerning brake lining life for various bus designs. These data were used to estimate the degree of brake lining life reduction in the Advanced Design Bus.

Table 6 gives the data provided by six transit systems. The two ADBs exhibited lining lives on both the front and rear linings that were about 40 percent less than the lives of the older "New Look" buses. This further supports the premise that the higher temperatures measured in the ADBs could contribute to reduced lining life.

The data collected in-service were also used in conjunction with a finite-element thermal model of the brake drum to estimate current convective and conductive cooling trends. The model and results are described in the following section.

DETERMINATION OF BRAKE SYSTEM COOLING CHARACTERISTICS

The purpose of this task was to determine the cooling characteristics that currently exist on the transit bus brake system. This information was used in conjunction with the temperature–wear properties of the asbestos-phenolic materials to estimate how large an increase in cooling was necessary to reduce the steady-state brake drum temperature.

A finite element model based on a widely used computer code called TRUMP was used to model the thermal conditions in the brake drums of the GM RTS-II and GM 5307 buses. The thermal modeling was constructed around brake temperature field data collected during in-service tests in Los Angeles. Since the Grumman Flxible 870 exhibited a large temperature difference between the front and rear brakes, which was most likely due to conditions other than thermal-cooling, thermal modeling of this bus was not conducted.

The primary intent of the thermal modeling was to identify possible differences in thermal behavior which could account for the observed differences in brake drum temperature between the GM 5307 and the GM TS-II designs. The in-service experiments identified the front drum of the RTS-II as having both the highest service temperatures and also the highest frequency of change for the transit authority investigated (SCRTD).

The following sections of the report describe the model, the model tuning procedures, and the results of the thermal analysis.

Description of the Finite Element Model

The model consisted of a series of thermal masses connected by thermal resistances and bounded by selected convective and conductive conditions. Figure 11 shows a cross section of the front drum of the GM 5307 model. A total of 17 nodal points

Table 6. Lining life reported by selected transit systems.*

Transit Property	Lining Life, vehicle miles: GMC 5307 Front Axle	GMC 5307 Rear Axle	GMC RTS Front Axle	GMC RTS Rear Axle	Flxible 870 Front Axle	Flxible 870 Rear Axle
A	48,000	42,000	X	X	22,000**	22,000
B	30-35,000	20,000	15-18,000	25,000	15-18,000	25,000
C	20-22,000	20-22,000	X	X	14-16,000	14-16,000
D	85,000	45,000	30,000	20,000	X	X
E	42,000	43,000	20,000	20,000	X	X
F	38,000	38,000	X	X	X	X
Representative Single-Point Value	42,000	38,000	22,000	22,000	21,000	21,000

* Brake lining life data were collected via telephone and letter solicitation of transit authority personnel.

** Property A has not yet had to reline any front axle brakes on their Flxible 870 coaches. An average life of 22,000 miles is assumed.

Model of OLDGMF

- Drum
  - Properties: c_p = 0.12 Btu/lbm-F (constant)
    ρ = 450 lbm/ft^3 (constant)
    k = 25.65 Btu/hr-ft-F @ 212 F
    = 28.07 Btu/hr-ft-F @ 752 F
    (2-point curve-fit conductivity)
  - Nodes: 16 Nodes (11 through 62)
    Total Mass = 122.2 lbm
- Boundary Conditions
  - 1 Conduction Boundary: Node 71

t (sec) =	0	8.3	410	57.5
T (F) =	315	300	289	285

  - 4 Convection Boundaries: Nodes 101-104
    101, 102, and 103 = 100 F
    104 = 85 F

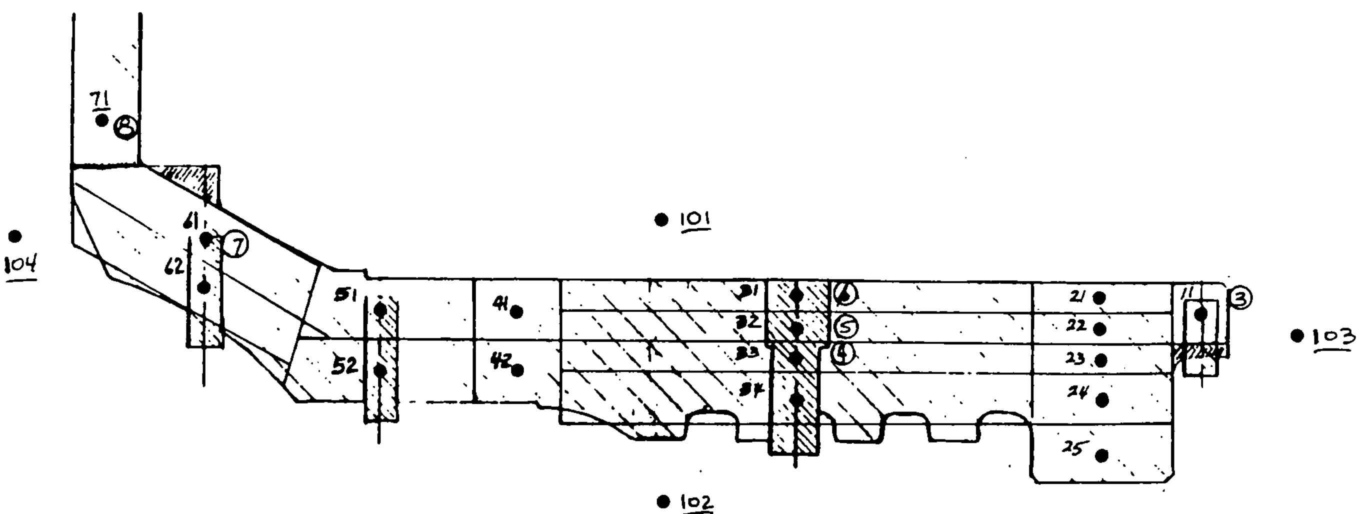


Figure 11. Finite element grid and boundary conditions for thermal model, GM 5307.

and discrete masses have been used to represent the brake drum. The circled numbers indicate the position of nodes in which temperatures were recorded during the field experiments. The entire model is axisymmetric about a central axis running along the length of the drum.

The amount of heat lost from the drum by convection and conduction was determined by recording temperature as a function of time at the various nodal positions. Thermocouple locations 7 and 8 (nodes 61 and 71) were used to track the amount of conduction from the drum to the wheel rim and bearing hub, while thermocouples 4, 5, and 6 (nodes 33, 32, and 31) were useful in determining the convective heat transfer losses from the drum.

To determine the current convective and conductive conditions in each drum, a "tuning" procedure was used. A segment of temperature–time data for each of the two buses was selected from the temperature data collected on the 83/20 run during service experiments in Los Angeles. Because it was of interest to determine cooling trends for the buses in the absence of braking, segments of data in which the buses were moving along the route but not applying brakes were selected. In order to duplicate all other operating conditions for the time segment in which the blocks of data were taken, the data segments for each bus were taken from periods of time in which each bus was fully loaded with passengers and operating on the same route location during service.

The external convective coefficients and the conductive heat losses for each of the buses were determined by using the segments of cooling data for the various thermocouple positions. At the initiation of cooling, the nodal temperatures in the model were set to match those of the data. The external convective coefficients were then adjusted until the cooling trends exhibited by the model matched those measured in service. Ambient air conditions inside the drum at nodal point 101 were based on measured air temperatures collected during testing in Los Angeles. The conduction boundary conditions at node 71 were based on measured values obtained in Los Angeles by positioning a thermocouple between the wheel rim and the brake drum.

Results of Finite Element Modeling

The iterative tuning process previously described was continued until a reasonable correlation between the actual data and the predicted values was achieved. Figures 12 and 13 summarize the results of this iteration. In both cases, matching of the temperature histories for the two brake drums has been successful to within several degrees of the actual measured temperature values. Table 7 summarizes the quantitative results of this thermal modeling.

In both buses, the heat transfer coefficient on the inside surfaces of the drum facing the brake linings is larger than the convective coefficient on the outside surface of the drum facing the wheel rim. This is probably because the annulus region between the brake drum and wheel rim is relatively small with little provision for air circulation. In addition, the wheel rim and the brake drum are rotating together with no relative motion between them to initiate air flow across the exterior of the brake drum. The interior surface of the brake drum, however, is in close proximity to a stationary body, the brake linings themselves. After braking, the linings are retracted somewhat from the rotating surface and an air inlet region between the lining at the drum is created. The relative motion between the lining and drum initiates air flow across the interior of the drum surface, causing a higher convective heat transfer to occur.

The difference between the interior convective coefficient for the GM 5307 and the GM RTS-II can be explained in part by the brake mechanism in use on both buses. The S-cam mechanism in use on the GM 5307 ensures consistent retraction of the linings from the drum surface, while the wedge mechanism in use in the GM RTS-II may be susceptible to periodic brake drag. Since any form of brake drag would prevent air circulation between the brake linings and the brake drum, this situation would result in a slightly lower convective coefficient at the inside of the brake drum. A later section in Chapter Three of this report, entitled "Influence of Brake Actuation Mechanism," describes differences between the S-cam and wedge actuation mechanisms.

The difference between the exterior convective coefficient for these drums may be explained in part by the drum design itself. The GM 5307 brake drums are banded circumferentially by fins. Because most of the air flow across the exterior of the drum comes from air slots in the wheel ring, these circumferential fins act to restrict air flow along the length of the drum. The GM RTS-II, however, has fins that are positioned longitudinally as seen in the cross section of Figure 13. Air flow initiated through the slots in the wheel rim is directed across the exterior of the brake drum by these fins, and a slightly higher exterior convective coefficient results.

Figures 14 and 15 show how the different heat loss modes in the two buses affect the temperature distribution in the brake drums during cooling. Because the dominant heat flow path in the drums of the GM 5307 is convective to the inside, the lowest temperature during cooling will be the nodal position closest to the inside surface, as seen in Figure 14. As cooling continues, the temperature gradient in the thickness of the drum is maintained; the exterior of the drum remains at a higher temperature with respect to the inside surface. In the GM RTS-II, the heat flow to the outside is significant, and the highest temperatures during cooling exist towards the interior node rather than the exterior node.

DESIGN OF SCHEMES FOR BRAKE DRUM TEMPERATURE REDUCTION

The purpose of this task was to design practical retrofit techniques for improving brake lining life through a reduction in brake drum operating temperatures. First, estimates of the de-

Table 7. Heat transfer coefficients calculated from thermal model for front drums of GM 5307 and GM RTS-II.

	Convection Coefficient to inside of drum (h_{31})	Convection Coefficient, outside of drum
GM 5307	21.8 BTu/hr-ft^2-F	2.9 BTu/hr-ft^2-F
GM RTS-II	14	7.7

	Data		Model	
Location	Symbol	T/C	Symbol	Node
Corner	○	7	●	61
Inside-Center	△	6	▲	31
Nub	□	3	■	11

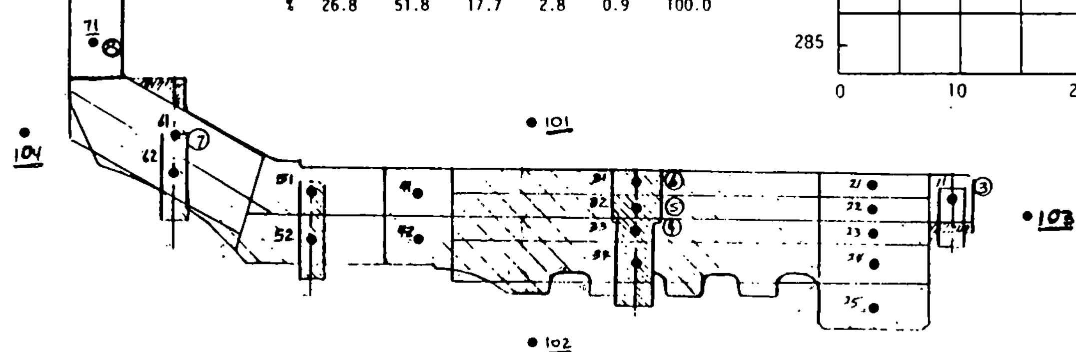
Model: $h_{Node\ 31}$ = 21.8 Btu/hr-ft^2-F; $h_{all\ other\ Nodes}$ = 2.9 Btu/hr-ft^{2}F

$\bar{T}$ = 302.24 F @ t = 0; = 289.79 F @ t = 60 sec.

Total energy lost from all 16 Nodes = 182.7 Btu

(Equivalent to stopping 1/4 of 25000 lbm from 26.1 mph)

Energy rejected to:	Node	71	101	102	103	104	Total
	Btu	49.01	94.67	32.38	5.03	1.61	182.7
	%	26.8	51.8	17.7	2.8	0.9	100.0



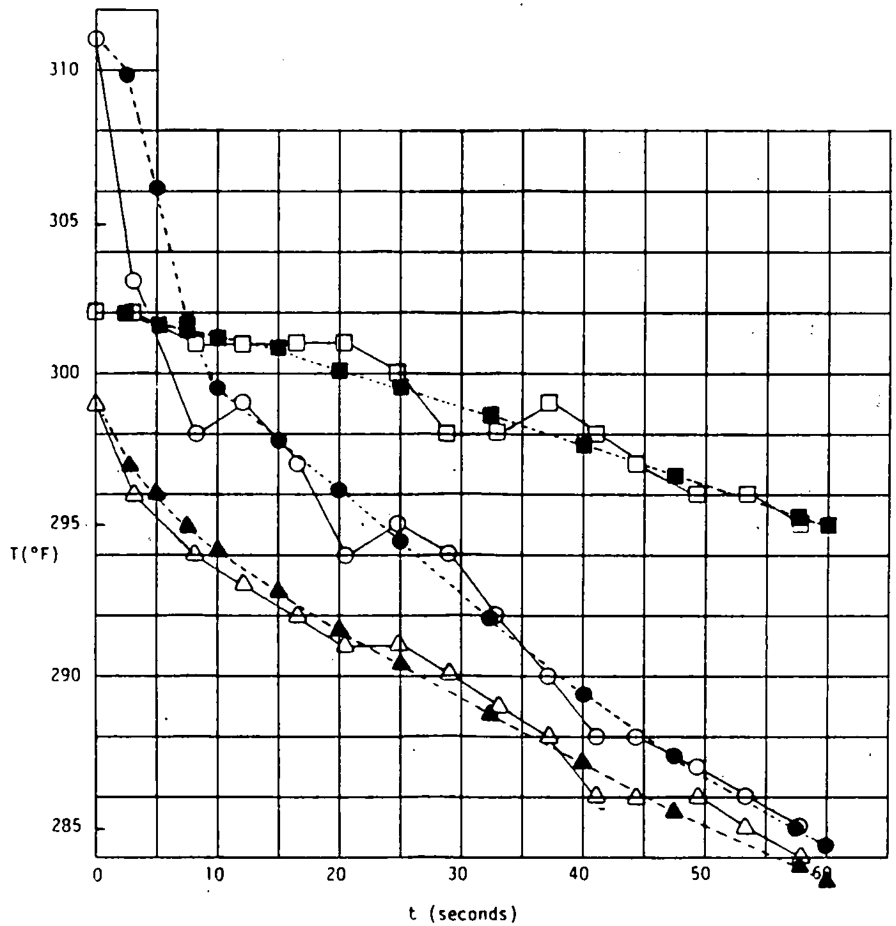


Figure 12. Final matched thermal conditions in front drum of GM 5307 bus.

Location	Data: Symbol	Data: T/C	Model: Symbol	Model: Node
Corner	○	7	●	61
Inside-Center	△	6	▲	31
Middle-Center	◇	5	◆	32
Outside-Center	⬡	4	⬢	33
Nub	□	3	■	11

Model: $h_{Node\ 31}$ = 14 Btu/hr-ft^2-F

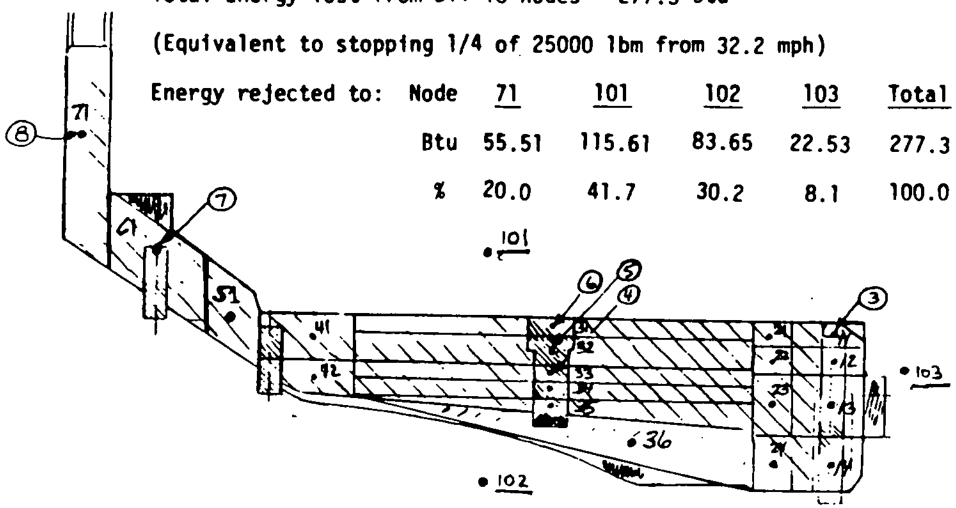
$h_{all\ other\ nodes}$ = 7.7 Btu/hr-ft^2-F

$\bar{T}$ = 365.85 @ t = 0; = 345.56 @ t = 60 sec.

Total energy lost from all 18 nodes = 277.3 Btu

(Equivalent to stopping 1/4 of 25000 lbm from 32.2 mph)

Energy rejected to: Node	71	101	102	103	Total
Btu	55.51	115.61	83.65	22.53	277.3
%	20.0	41.7	30.2	8.1	100.0



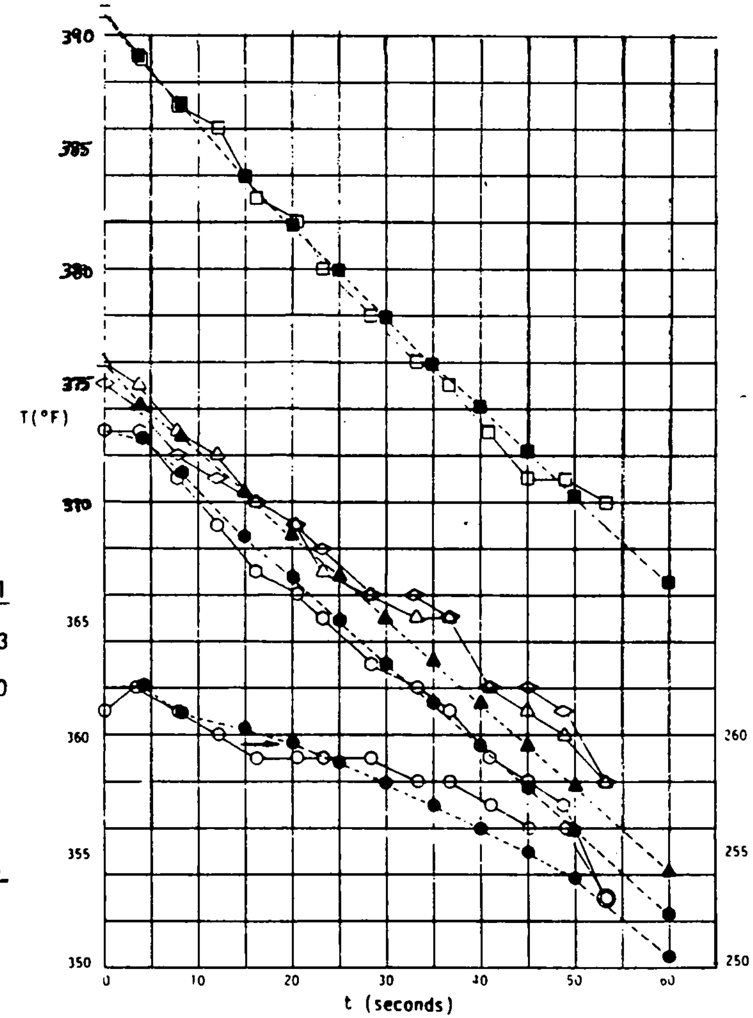


Figure 13. Final matched thermal conditions in front drum of GM RTS-II.

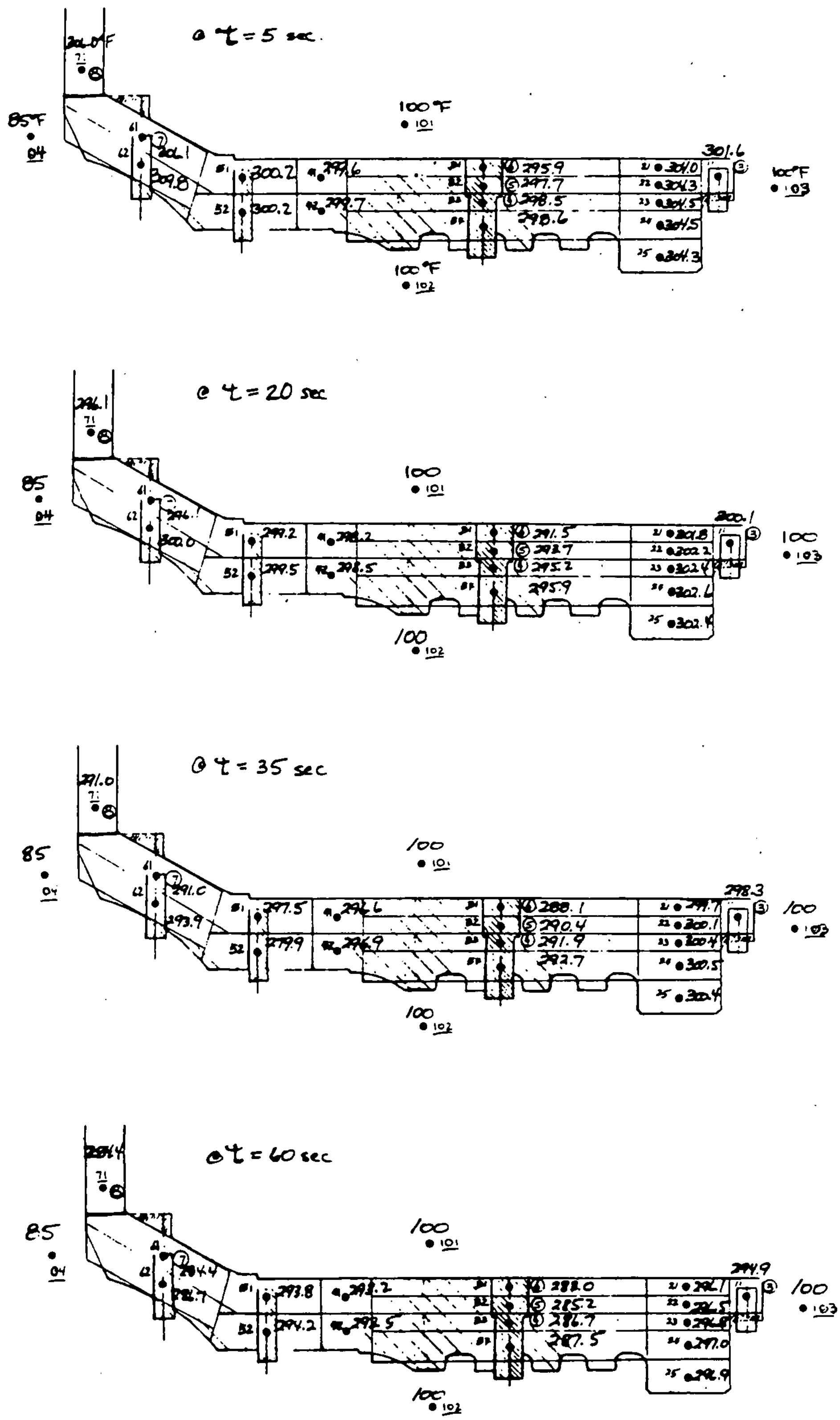


Figure 14. Temperature profile during cooling in front brake drum, GM 5307 model.

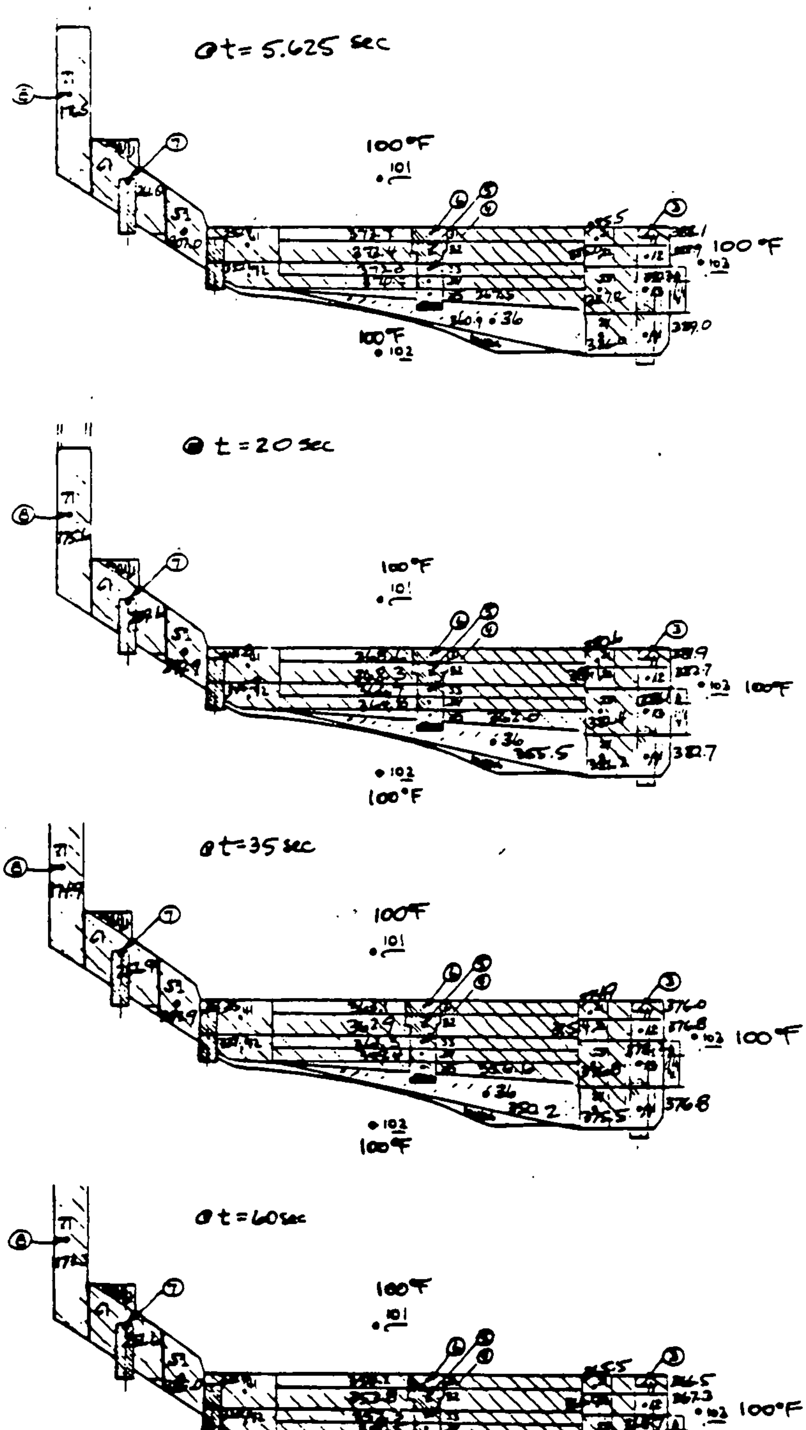


Figure 15. Temperature profile during cooling in front brake drum, GM RTS-II model.

gree of life improvement through temperature reduction were made on the basis of a temperature–wear model. Next, a correlation between convective flow conditions and predicted steady-state temperature reduction was made. Finally, practical techniques for reducing steady-state brake drum temperatures were investigated.

Determination of Necessary Overall Heat Transfer Coefficient

Using the thermal characteristics and reported brake lining mileage of the GM RTS-II, an estimate of the necessary temperature reduction and the amount of drum cooling needed to reach that temperature was calculated. The technical feasibility of the various retrofit schemes was then evaluated by comparing the amount of cooling that each method was capable of delivering with the amount of cooling needed.

The equation by Fazekas (*2*) described the steady-state brake drum temperature as a function of bus mass, velocity, overall heat transfer coefficient, and the time between stops:

$$T_{ss} - T_o = \frac{\frac{1}{2}mV^2}{hAt_o} \tag{3}$$

Above the threshold temperature, the lining life was found to follow this trend:

$$L_\ell = K \exp\left(\frac{E}{RT}\right) \tag{6}$$

The temperature difference needed to induce longer lining life was calculated using these two equations, and the thermal cooling conditions needed to induce the temperature change were also estimated. An interface temperature rise of 50 F above the steady-state temperature was assumed.

It was assumed that anytime the brake linings were exposed to transient temperatures above 400 F (350 F steady-state), the lining materials wore in an exponential manner with respect to temperature. The extension of brake lining life predicted to occur by reducing the drum steady-state temperature from 390 F to 350 F was calculated as follows. An average initial lining life of 22,000 miles, as suggested by data in Table 6, was assumed.

$$\frac{x \text{ miles}}{22{,}000 \text{ miles}} = \frac{\exp\left(\frac{E}{R}\frac{1}{T_1}\right)}{\exp\left(\frac{E}{R}\frac{1}{T_2}\right)}$$

where T_1 = reduced temperature, 390 F (472 K); T_2 = initial temperature, 440 F (500 K); and x = 39,000 miles.

By reducing the steady-state drum temperature 40 F to levels below 350 F, an improvement in lining life to levels observed in the GM 5307 "New Look" bus is predicted. Conservatively, a 50 degree temperature reduction was targeted. From the model and in-service data, the overall heat transfer rate from the GM RTS-II front drum was found to be 277 Btu/min at an average temperature difference between the drum and the ambient air of 270 F. The overall heat transfer coefficient for this situation is:

$$Q = hA\,\Delta T$$

$$hA = U = \frac{Q}{\Delta T} = \frac{(277 \text{ Btu/min})\,(60)}{270 \text{ F}} = 61.6 \text{ Btu/hr F}$$

The overall heat transfer coefficient needed to induce a 50 F drum temperature reduction was calculated, assuming identical stopping conditions exist for the same bus, with ambient air at 85 F:

$$\frac{390 - 85}{340 - 85} = \frac{\dfrac{\frac{1}{2}mV^2}{(61.6 \text{ Btu/hr F})t_o}}{\dfrac{\frac{1}{2}mV^2}{(U)t_o}}$$

$$U = 74 \text{ Btu/hr F}$$

To decrease drum temperature by 50 F, an overall heat transfer coefficient for the drum of 74 Btu/hr F should be targeted.

The overall heat transfer from the drum is a result of convective losses from the inside of the drum and the outside of the drum as well as conduction losses to the wheel:

$$Q = \left[\underset{\text{convection}}{(h_oA_o)_{\text{outside}}} + \underset{\text{convection}}{(h_iA_i)_{\text{inside}}} + \left(\frac{KA}{\Delta x}\right)_{\text{conduction}}\right]\Delta T$$

for improved brake performance,

$$\frac{Q}{\Delta T} = 74\,\frac{\text{Btu}}{\text{hr F}} = (h_oA_o) + (h_iA_i) + \frac{KA}{\Delta x}$$

This equation was used to determine the technical feasibility of the proposed schemes. The contribution of improvements in h_o, h_i and conduction area, A, were assessed individually and the cummulative contribution was determined by adding the effects together.

If the internal convection and conduction coefficients remain the same and the external area, A, is estimated to be 3.4 ft^2, the necessary exterior coefficient needed to induce the desired temperature reduction was (12 Btu/hr)/(ft^2 F).

Description of Convective Cooling Conditions on Drum Exterior. Convective cooling of exterior surfaces is a function of the air flow velocity across the surface and the surface geometry. This section of the report examines the conditions affecting the convective cooling of the exterior drum surface and establishes the air flow conditions necessary to affect convective cooling.

The convective coefficient, h, is expressed as a dimensionless quantity in the form of the Nusselt number:

$$Nu = (h)/(kf/D)$$

where $h \equiv$ convective coefficient; $kf \equiv$ thermal conductivity of media; $D \equiv$ length affecting convection in air flow.

This Nusselt number is usually expressed as a function of the Reynolds number, which is determined by the flow geometry and flow velocity:

$$Re = \frac{VD}{\nu}$$

where $V \equiv$ velocity; $D \equiv$length affecting convection and air flow; and ν = viscosity of media.

The distance between the brake drum and the wheel rim is typically around 2 in. for a 22-in. diameter wheel rim mounted on the GM RTS-II brake drum. The annulus region between the drum and the wheel rim becomes more constricted in the vicinity of the hub nuts. Air flow across the exterior of the drum flows through this annulus region, and the conditions in the annulus region determine the Reynolds number, the Nusselt number, and the resulting heat transfer.

Drum Surface Flow Condition: Fully Developed vs. Entrance Effects-Dominated. Under flow conditions, the heat transfer coefficient on a heated surface is related to the flow distance of the media across the surface. At the entrance to a pipe or annulus, the heat transfer coefficient is high and gradually decreases as flow proceeds along the surface. At a certain distance, the flow becomes fully developed and the heat transfer coefficient at the surface becomes constant.

An expression relating the Nusselt number and the Reynolds number for air flowing across a plate under nondeveloped conditions was derived by Colburn (*16*). The expression uses a Reynolds number based on a flow distance rather than on a flow diameter as in the fully developed flow condition:

$$\overline{Nu} = 0.036\ Pr^{1/3}\ Re^{0.8}$$

The Prandlt number, *Pr,* is 0.71 for air.

The velocity of air flow needed to cool the brake drum can be calculated using this expression. Previous thermal analysis determined that an exterior convective heat transfer coefficient of 12 Btu/hr-ft^2-F is needed to ensure adequate cooling of the drums. Under these conditions, the air velocity needed is calculated

$$V = (73.6)\left(\frac{hL}{kf}\right)^{1.25}\left(\frac{\nu}{L}\right)$$

Substituting fluid properties and letting *L,* the flow length, equal 10 in: $V = 52.1$ ft/sec. This velocity corresponds to an Re_x = 241,000. The boundary layer thickness can be calculated by the following equation:

$$\frac{\delta}{x} = 3.76\ Re_x^{-1/5}$$

$$\delta = 0.31 \quad x = 0.31 \text{ in.}$$

Because the boundary layer is so thin, the original assumption that the flow is not fully developed is reinforced.

Assuming an air inlet area of 0.97 ft^2, and air volume flow rate of about 50 ft^3/sec, or a little more than 3,000 cfm will be needed to cool the drums about 50 F. Retrofit schemes designed to cool the drums by increasing exterior convection were evaluated with respect to their ability to deliver 3,000 ft^3/min.

Description of Schemes for Temperature Reduction

Numerous techniques are available for reducing brake drum temperatures. Schemes requiring modification of the brake actuation mechanism or redesign of components were not considered in this analysis. The retrofit techniques for reducing brake drum temperatures have been restricted to those that involve both active and passive methods for increasing convective air flow in the vicinity of the drum, and those that require only bolt-on installation to the drum and wheel assembly.

Newcomb and Spurr (*17*) have outlined a number of methods used in the past to improve drum cooling. For example, drums having fins cast in the axial direction, as in the drums of the RTS-II, have been found to be more effective in cooling than fins cast in the circumferential direction, as in the GM 5307. This observation was supported by the experimental data and subsequent thermal model. In addition, commercial vehicles usually demonstrate lower drum cooling rates than do automobiles because the drum and wheel assemblies of commercial vehicles typically have a smaller annulus region between the brake drum and the wheels. This prevents cooling air from circulating over the drum. Increasing the clearance between the wheel rim and the outside diameter of the drum is particularly important in increasing drum cooling.

Previous methods used to induce greater cooling include drum or wheel-mounted vanes designed to direct air flow over the drums. These methods have been used with limited success. Interior cooling of the drum is not very feasible because of the difficulty of directing air flow over the rubbing path and because of the possibility of introducing water, dirt, or debris into the friction zone.

Since success has been achieved by extending the brake drum flange into the air stream, thus exposing more drum area to convection, this technique could be implemented in buses by bolting a flange to the outer rim of the drum. Thermal conductivity between the flange and the drum could be improved through the use of heat conducting paint. Solution schemes based on these and other ideas have been investigated. The technical feasibility of the solutions was determined by first establishing a target heat transfer coefficient needed to reduce drum temperatures and then establishing air flow rates or heat conduction rates needed to achieve that temperature. The following methods for reducing brake lining temperature were considered:

- Increase in convective cooling of drums by forced convection induced with blowers.
- Increase in convective cooling of drums through fixed air scoops.
- Reduction of wedge brake drag.
- Increase in drum-wheel annulus area to permit greater air circulation.
- Increase in convective cooling of drums through wheel-mounted fans or scoops.
- Increase drum convective surface area by flange.
- Increasing conduction cooling.
- Alteration of driving style.
- Use of alternative friction materials.

Discussion of the technical feasibility of these schemes is presented in the following sections.

Increase Convection Using Electrical Blower Units. Electrical blower units that direct air across the surface of the brake drums were evaluated. For adequate cooling of the drums, a 3,000 cfm blower capacity was needed.

Standard blower units using squirrel cage fans and shrouded construction were investigated. Because of the size restrictions, relatively small blower units having 8-in. or 9-in. diameter wheels were chosen. Under these restrictions, no commercially available units were capable of delivering the necessary flow rate. The most appropriate unit was capable of delivering 535 cfm at a power expenditure of 243 watts. For four blower units, this power usage would be about 972 watts or 1.3 hp per bus.

In a transit bus, the output of the blower was directed across the drum by means of a flexible duct. The blower units were designed to be fixed near the front and rear bulk heads of the bus frame. In actual installation, service of the blowers could possibly be done by providing an access port in the floor of the bus. D-C powered blower motors were selected because connection to the buses' present electrical system was possible.

Figure 16 shows the units selected for this program. A single motor has been fixed to two squirrel-cage fans, with ports made available for connecting to 4-in diameter flexible conduit. Switch connections allowing for three speeds were available.

To determine the effectiveness in reducing brake drum temperature, two units were installed on a transit bus in Los Angeles for cooling the front and rear street-side brake drums. The results of those experiments are described in a subsequent section of the report.

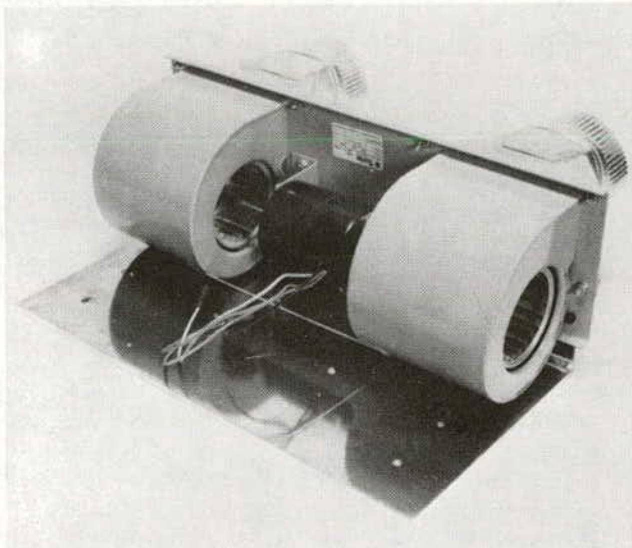


Figure 16. Blower units designed to direct cooling air across brake drums.

Increase Convection Using Air Scoops. Air scoops that direct air flow across the exterior drum surfaces were considered. Assuming a bus traveling at an average speed of 15 mph, the intake area of the scoop would have to have a minimum surface area of 2.3 ft^2 in order to direct the 3,000 ft^3/min necessary to induce cooling. This inlet region could have the dimensions of 1 ft $\times$ 2.5 ft, with the flexible conduit directing the air flow to the annulus region of the brake drums. Each wheel would have to be fitted with an air scoop.

The use of an air scoop presents possible problems of undercarriage clearance and maintenance. Ideally, the scoops would be slung under the bus to direct air flow. In addition to directing air, the scoops will also tend to direct debris and dirt toward the brake drums. Accumulated dirt on the outside of the drums would reduce heat transfer even more, while abrasive debris on the inside surfaces of the drum would reduce drum and lining life. Another disadvantage of the scoops is the increased bus drag and the resulting decrease in fuel economy that may result from the scoop placement. Although the installation of an air scoop may not be practical, the success of such a scheme would suggest possible changes in body configurations in the vicinity of the wheels to permit better air circulation.

Figure 17 shows one of the air scoops used for this program. The inlet of the scoop was designed for a thin profile to minimize possible problems in ground clearance. A width of 36 in. and a height of 9 in. were used.

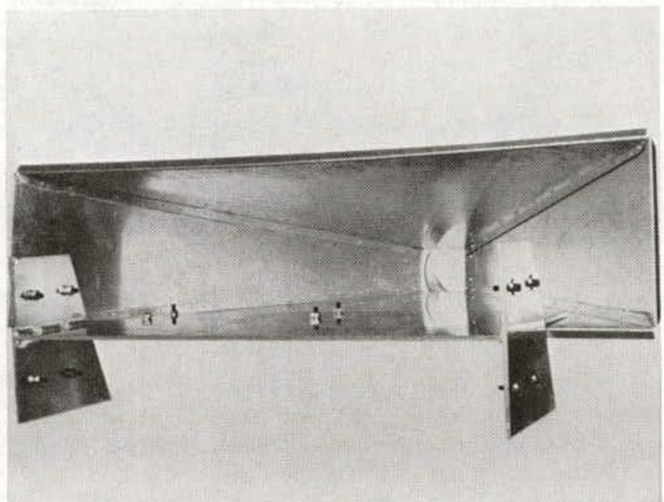


Figure 17. Air scoop designed to direct air flow across the front drum of an RTS-II ADB.

During Phase II of the program, the technical feasibility of using air scoops to reduce brake temperature was determined by mounting air scoops near a front brake drum and measuring the resulting change in the steady-state drum temperature in service.

Reduction of Wedge Brake Drag. The brake drums of the GM 5307 bus currently exhibit internal convective heat transfer coefficients of 22 Btu/hr-ft^2-F, while the drums of the RTS-II exhibit an internal heat transfer coefficient of 14 Btu/hr-ft^2-F. The difference is felt to be due to the S-cam mechanism of the 5307, which causes the linings to retract away from the drum surface, allowing for air circulation between the shoes and the rotating drum. This would result in a relatively higher convective heat transfer coefficient. On the other hand, the GM RTS-II wedge brake system has a floating wedge actuation device in which both of the shoes are self-actuating, i.e., the friction between the drum and the shoe acts to further draw the shoes into the interface. The wedge brake mechanism is self-activating and does not fully retract away from the brake drum surface after release of the brakes. This causes a reduction in the amount of air flowing between the lining and the drum and results in a lower internal heat transfer coefficient.

In order to fully retract the shoes from the drum after braking, an offset spring running diagonally between the two halves of the shoe can be installed. Figure 18 shows this technique. During braking, the self-activating character of the brake lining will stretch the spring, but after braking, the spring will act to retract the shoes away from the drum.

Retraction of the shoes from the lining will probably increase the internal convective heat transfer coefficient from 14 Btu/hr-ft^2-F to about 20 Btu/hr-ft^2-F, raising the overall heat transfer coefficient slighlty, but not enough to lower the temperature by 40 or 50 F. Even though the shoe retraction cannot induce all the necessary increase in cooling needed, a partial increase in cooling may be achieved through this technique. In addition to forcing the linings to retract, offset cutting of the shoes is also recommended because of reported improvements achieved by this technique.

A brake retractor was installed on the rear street-side brake shoes of a GM RTS-II in Los Angeles. Results of in-service experiments are presented in later sections of this report.

Improve Convection Conditions by Increasing Wheel-Drum Clearance. The wheel-drum clearance has been reported to be an influence on the brake drum temperature (*21*). This influence is probably not caused by any changes in forced convection because the flow profile around the drum is not fully developed and the heat transfer is influenced more by turbulent boundary layer effects instead of bulk fluid flow. The reported reduction in brake drum temperature with an increase in clearance is most likely due to an improvement in the free convection heat transfer conditions, rather than the forced convection conditions.

The convective heat transfer coefficient for a heated cylinder under free convection condition is expressed by the following:

$$h_c = 0.27 \left(\frac{\Delta T}{D}\right)^{1/4}$$

where T refers to the difference in drum surface temperature and air media temperature, and D refers to the diameter of the heated drum.

Increasing the clearance most likely improves cooling by allowing more expedient air exchanges, which increases T and improves h_c. A comparison of two cases, one in which the surrounding air is hot (~200 F) and one in which the surrounding air is cool (~80 F), for a brake drum at an average temperature of 400 F, shows the following:

$$\frac{(h_c)_1}{(h_c)_2} = \left[\frac{(400\text{-}200)}{(400\text{-}80)}\right]^{0.25}$$

$$(h_c)_2 = 1.12\ (h_c)_1$$

A 12 percent improvement in the exterior convective heat transfer coefficient is realized. This improvement alone is not sufficient to reduce brake drum temperatures entirely, but it may be successful when used in conjunction with other schemes.

A slight increase in the brake drum-wheel clearance can be realized by changing the wheel rim configuration (*18*). The majority of transit agencies use a tubeless tire rim, which has a 15 percent drop center configuration. This configuration restricts the wheel-drum clearance and may prevent effective bulk air exchange. Tube-type tire rims, however, have a 5 deg sloped configuration, which is less restrictive to bulk air exchange. A changeover from tubeless to tube tires may effect a reduction in brake drum temperature because of more effective bulk air exchange.

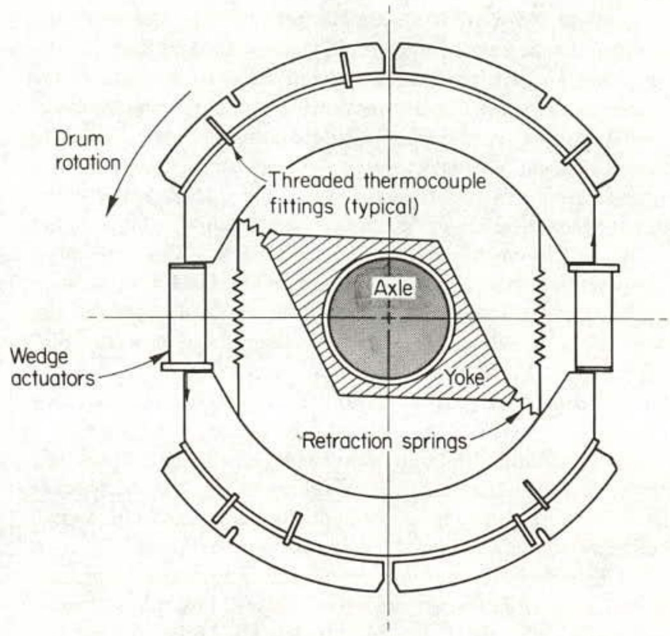


Figure 18. Schematic of retractor mechanism.

In order to verify or refute this situation, the researchers attempted to procure tube-type tires and rims to outfit a transit bus in Los Angeles. However, current trends have reduced the availability of tube tires, and the tires needed for this study could not be obtained locally. Therefore, this retrofit technique, although it offers promise, was not investigated.

Increase Convection Through Wheel-Mounted Fans. A scheme in which small fans are affixed to the rotating wheel for the purpose of directing air flow across the drum was considered as a possible retrofit scheme, but this method was determined not to be technically feasible. In the present design, the wheel rims of the buses have five air inlets positioned to allow convective cooling of the exterior brake drum surface to occur. A convective heat transfer coefficient of 12 Btu/hr-ft^2-F is needed on the exterior drum surface and an air flow rate of around 3,000 cfm is required to achieve this heat transfer coefficient. For a bus wheel rotating at 180 rpm, which corresponds to a forward speed of about 15 mph, each air inlet of the wheel would have to be filled with a scoop or fan having a surface area of 0.88 ft^2, or about 11 in. × 11 in. in size. This estimation was made assuming that only two of the fans are effective at any one time in directing air, since the relative velocity of the fan near the ground is relatively low.

This size is too large to be considered for a transit bus application, considering possible abuse during use and the possible safety hazard proposed by the use of these fans. A safe fan size would probably be 5 in. × 5 in., but this size would not induce the necessary flow rate across the brake drums. For this reason, this retrofit method was not considered to be technically feasible.

Increase Convective Surface Area of Brake Drums. This scheme involves changing the surface area of the drum in order to effect more convective cooling across the drum surface. The initial design calls for bolting on a flange to the outer rim of the drum facing in toward the bus, so that a portion of the drum extends out past the lining contact zone and into the undercarriage airstream of the bus.

The amount of area required for this scheme can be estimated assuming that the conduction to the wheel rim and bearing nut remain the same and the internal heat transfer coefficient and the internal area affected by this coefficient remain the same:

$$74 \frac{\text{Btu}}{\text{hr-F-ft}^2} = h_o A_o + A_i h_i + (KA/\Delta x)$$

where $h_i \equiv$ internal convective coefficient; $A_i \equiv$ internal area; $h_o \equiv$ external convective coefficient; and $A_o \equiv$ external area.

$$A_i h_i = 19.6$$

$$\frac{KA}{\Delta x} = 12.3$$

$$h_o A_o = 42 \frac{\text{Btu}}{\text{hr-F}}$$

$$\text{Assuming } h_o = 7.7 \frac{\text{Btu}}{\text{hr-F-ft}^2}$$

$$A_o = 5.5 \text{ ft}^2 \text{ Total}$$

Since the exterior area of the drum is already 3.4 ft^2, the outside area must be increased about 2 ft^2. A circular flange 6 in. deep would increase the area by the necessary amount.

Although this technique conceptually may be a viable technique for improving convective losses and reducing drum temperatures, current brake system designs do not permit any extensions of the brake drum. This technique was judged to be impractical and was not evaluated.

Increase Conduction Losses From the Brake Drum. Increasing conduction heat losses from the drum to the wheel rim and bearing hub was not considered to be practical for several reasons. First, the main driving force behind conduction heat transfer is the temperature gradient between the bulk drum section and the wheel rim itself. Since the temperature measured at the brake drum-wheel rim interface was not particularly high, no real increase in temperature gradient and conduction heat transfer can be realized by decreasing this temperature even further. Second, the conduction heat transfer could be increased by increasing the contact area between the wheel rim and the drum without increasing the temperature gradient. Increasing the contact area, however, could be accomplished only through major redesign of the brake drum and wheel components, and an obstruction in convective air flow across the drum would occur. Third, increased heating of the wheel rim and bearing hub could have a deleterious effect on bearing lubrication and tire life. For these reasons the reduction of brake drum temperatures by increasing conduction losses to the wheel rim and bearing hub was not considered further.

Alteration of Driving Style. Excessive braking speed is thought to be the main reason for excessive brake temperatures in the GM RTS-II ADB. Only slight increases in the braking speed can account for substantial increases in the amount of energy dissipated into the brake drums, and this increased energy is reflected in an increase in the drum temperature.

The braking speed could possibly be decreased through driver instruction programs in which drivers were instructed not to exceed a certain braking speed. The experiments in Los Angeles found that the average high energy braking speed for the GM 5307 was 24 mph. Since this braking speed resulted in acceptable temperature levels, this speed is a good target braking speed for the GM RTS-II ADB.

Use of Alternative Braking Materials. Alternative braking materials are now emerging as substitutes for asbestos-containing materials as the use of asbestos becomes more controlled by law. New asbestos-free materials based on fiberglass or Kevlar are coming into use in several transit authorities (*19, 20*) as the use of conventional phenolic-asbestos composites becomes less attractive.

Since the use of these materials is relatively new, no temperature–wear performance data are readily available. Therefore, no evaluation of this method was conducted during this program. However, materials having a higher threshold temperature than conventional asbestos-phenolic materials should show better temperature–wear properties.

TECHNICAL EVALUATION OF RETROFIT SCHEMES

The purpose of this task was to evaluate the capability of the selected retrofit schemes to reduce the steady-state brake drum temperatures in service. The following four configurations were evaluated during the Phase II experiments: (1) no retrofit schemes (baseline), (2) air scoop on front street-side brake drum, (3) blowers on both front and rear brake drums, and (4) brake shoe retractors on rear street-side shoe. For each configuration, tests were conducted for two days on urban transit routes in Los Angeles. Tests on one day were conducted on a bus designated for limited service, or urban service with selected stops, while tests on the other day were conducted on a bus designated for curb-to-curb local service. The limited service was characterized by fewer but faster stops, while the local service was characterized by many slow stops.

Table 8 describes the two routes used. The 80/20 route was continuous limited service along Wilshire Blvd. between downtown Los Angeles and Santa Monica beach. The 84/20 route was continuous local service between downtown Los Anglees and U.C.L.A. All experiments, except for the first baseline run, were conducted on weekdays to keep ridership consistent from day to day.

Table 8. Description of routes used in Phase II experiments.

Route #	Type of Service	Description of Routes
80/20	Limited service	Continuous limited service along Wilshire Blvd. between downtown L.A. and Santa Monica
84/20	Curb-to-curb service	Continuous curb-to-curb service along Wilshire Blvd. between downtown L.A. and U.C.L.A.

Table 9 describes the routes, retrofit schemes, and drivers used for the experiments. A total of six different drivers were used over eight test days, designated Day 1 to Day 8. Since all the service routes were longer than 8 hours, two drivers were used on each run. All of the experiments on the weekday 80/20 route used the same two drivers, while the afternoon run of the 84/20 route used the same driver. Because driving style and stopping speed are important factors determining brake system temperatures, consistency in driving style was desired in order to compare the effects of the retrofit techniques.

The instrumentation and related equipment used for these experiments were the same as those used for the Phase I experiments. Because of policy changes at SCRTD, however, the research team were not able to use the same type of asbestos-phenolic shoe material that was used in Phase I experiments. The thermal behavior of the system was not expected to be greatly altered by the selection of friction materials. Appendix B discusses this topic in more detail.

The following sections of the report present the results of the Phase II experiments.

Table 9. Routes and drivers used in Phase II experiements.

Test Day	Route Number	Retrofit Scheme	Driver (a.m.)	Driver (p.m.)	Day
1	80/20	Baseline	E	G	Sat.
2	80/20	Baseline	A	B	Mon.
3	84/20	Air scoop (front only)	C	D	Tues.
4	80/20	Air scoop (front only)	A	B	Wed.
5	84/20	Blowers (front and rear)	F	D	Thurs.
6	80/20	Blowers (front and rear)	A	B	Friday
7	80/20	Retractor (rear only)	A	B	Tues.
8	84/20	Retractor (rear only)	C	D	Wed.

Fluctuations in Brake Drum Baseline Temperatures (Rear)

Brake drum temperatures for two test days in the absence of retrofit schemes were compared to determine the variation in temperature levels that can be expected to occur because of daily fluctuation in traffic conditions and driver skill.

Figure 19 shows rear brake drum temperature under baseline conditions on the 80/20 route. Since the air scoop configuration directed air flow to the front drum only, data collected for the rear drums during days on which the air scoop was fitted to the bus were considered to be baseline data for the rear drums. These data are characterized by heating and cooling cycles that occur because of scheduled runs during the day. Blanks in the data are a result of failure of the data tape used to store the data.

These data show the influence of operator skill and driving style on brake drum temperatures. For the duration of the experiments, bus operation during the morning segment of route 80/20 was handled by a single driver, as shown in Table 9. Temperatures in the morning portion of the route for these two days were similar, with temperatures of the second day of the baseline experiments slightly lower. Bus operation during the afternoon session was handled by two drivers, one of which was an instructor and one of which was an apprentice. During the afternoon session of the baseline session on Day 2, the experienced instructor operated the bus. Drum temperatures ranged between 300 F and 400 F. During the afternoon session of Day 4, however, the apprentice operated the bus. His driving style was very cautious and marked by long, slow stops. Drum temperatures at that time ranged between 220 F and 300 F. Under identical routes with similar traffic conditions, driving style appeared to contribute to large fluctuation in brake drum temperatures.

Typical temperature levels for the rear brake drums are also seen in results obtained from two days of experiments on the 84/20 route with experienced drivers. Figure 20 shows these results. Temperature levels for both days are similar; brake drum temperatures ranged between 300 F and 400 F during the route. These results suggest that temperatures measured during the afternoon segment of route 80/20 on Day 4 are abnormally low and are not representative of rear baseline experiments.

Fluctuations in Brake Drum Baseline Temperatures (Front)

Front brake drum baseline temperatures were more variable than rear brake drum temperatures. The baseline conditions for the front drum were observed by comparing data from Day 2 and Day 7. Since Day 7 involved an examination of the effectiveness of a rear shoe retractor with no change to the front brake shoes, front brake drum data for that day were also considered to be baseline.

Figure 21 shows the temperature history for the front brake drum on two days on the 80/20 route. Temperatures between days appear to fluctuate by as much as 90 F in some cases, although typical fluctuations appear to be around 20 F to 40 F. Baseline data for only one day are available for the front brakes on the 84/20 route. Temperatures on this route ranged between 300 F and 430 F.

Figure 21 illustrates the extent to which temperatures can vary from day to day under controlled operating conditions. Specifically, temperatures on Day 7 between 9 a.m. and 11 a.m. ranged from 320 F to almost 500 F, while on Day 2, the temperature ranged between 320 F and 380 F. These two experiments were conducted on Monday and Tuesday, respectively, so similar traffic patterns were encountered. No explanation for this difference in temperature is given.

Of the three retrofit techniques evaluated, the use of blower units to force air across the brake drum surface was the most effective technique in reducing brake drum temperatures. The next section of the report describes the results of the use of blower units in-service to reduce brake drum temperatures.

Evaluation of the Effectiveness of Blowers in Reducing Drum Temperatures

Blower units depicted in Figure 16 were mounted under the bus near the front and rear street-side wheels. The output of

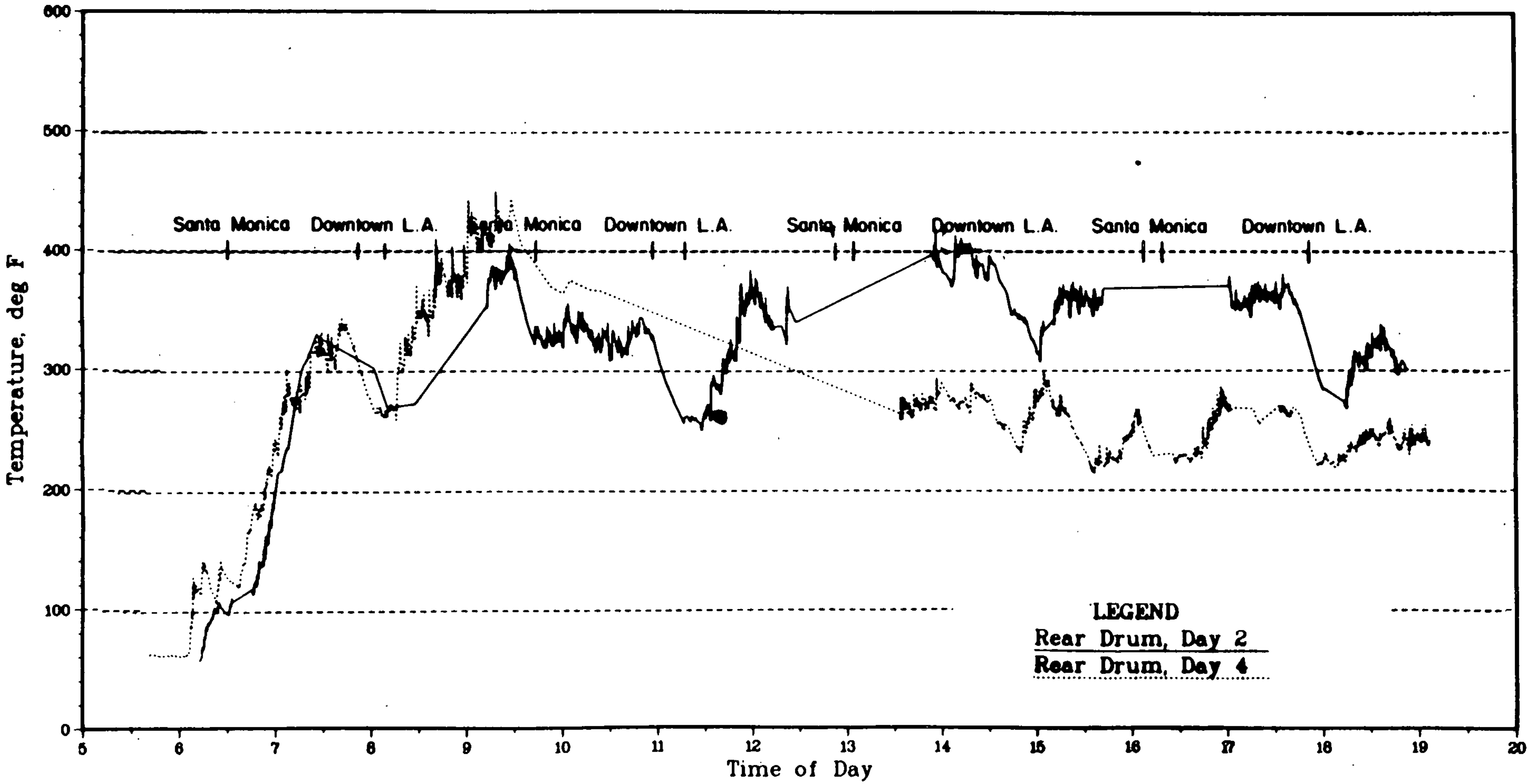


Figure 19. Near-surface brake drum temperature, GM RTS, 80/20 route (baseline temperatures for two separate days).

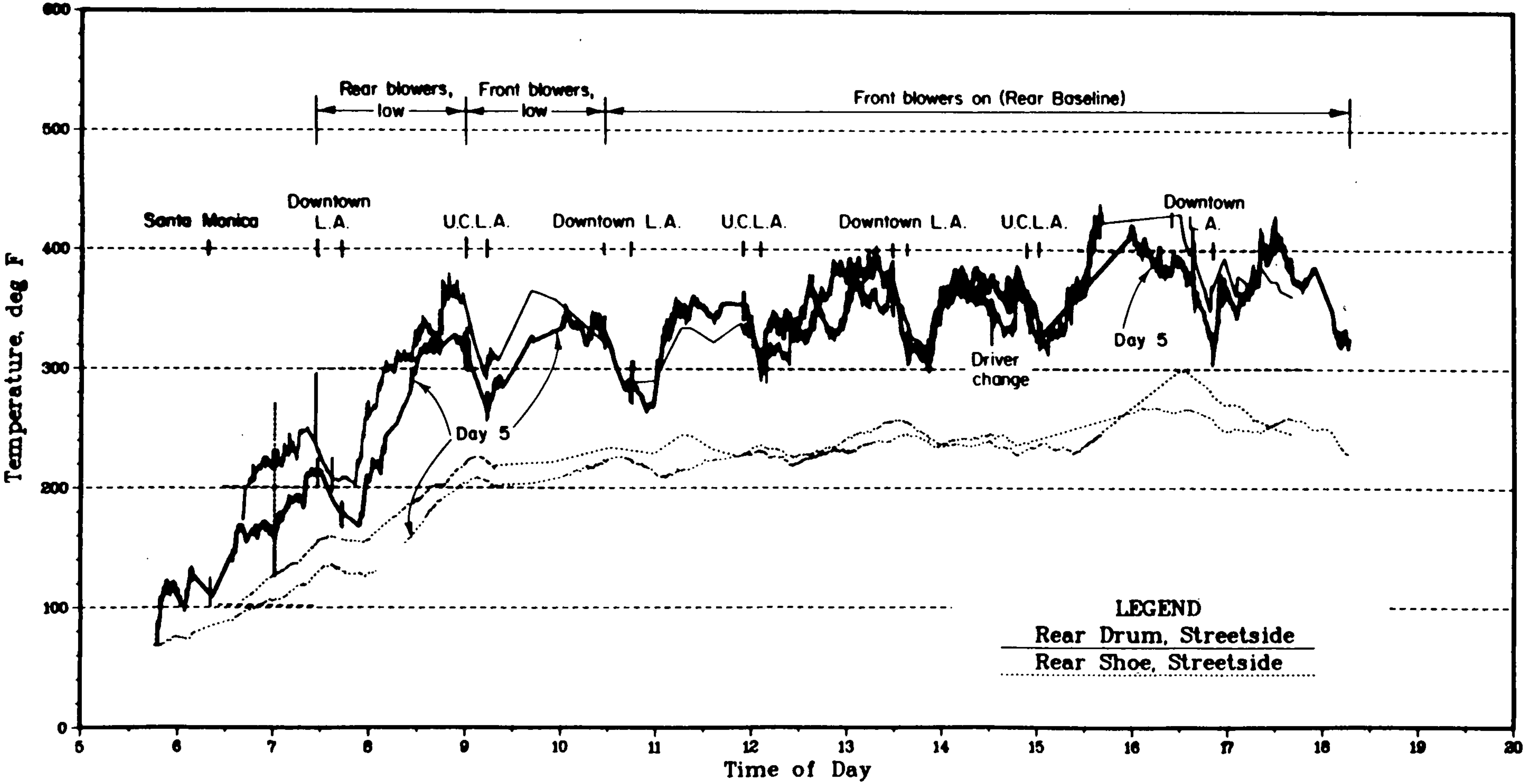


Figure 20. Near-surface brake drum temperature, GM-RTS, 84/20 route (baseline temperatures for two separate days).

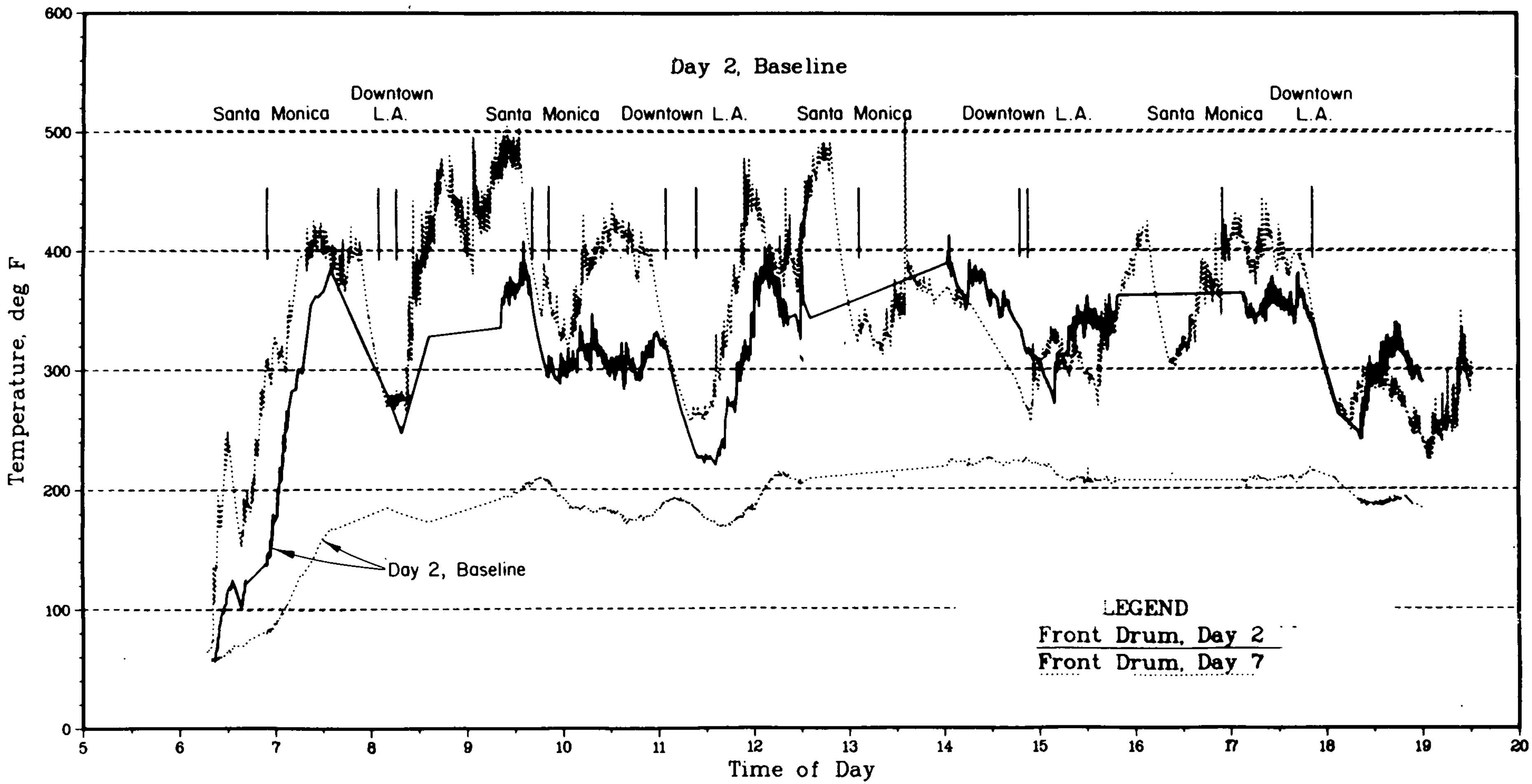


Figure 21. Front brake drum baseline temperature, GM-RTS, 80/20 route.

the blowers was directed by means of 4-in. flexible tubing to the annulus region between the brake drum and the wheel rim. Since each unit had two outputs, two flexible tubes were used for each brake drum.

The blowers were configured to operate at two motor speed settings and two output levels. Appendix C lists the design and operating specifications for the units. Under the selected low speed setting, these units delivered an air flow rate of 300 cfm at a power consumption of 108 watts. At the high speed setting, the units delivered 535 cfm with a power consumption of 243 watts. A switch was positioned under the bus to change the output of the blowers from outside the bus while the bus was stationary.

In-service experiments were conducted by running the brakes with the blowers off to allow brake drum heating, and then switching the blowers on to cool the drums. Front and rear blowers were alternately switched on and off during the service day.

Figure 22 shows the temperature history for the rear brake drum of the RTS-II on the 80/20 line. Blower conditions have been indicated on the diagram. In addition, the baseline data of Figure 20 have also been presented on the same graph to show comparative temperature levels.

The effect of the rear blowers is exhibited in the third route of the day, which originated at Santa Monica and terminated in downtown Los Angeles. The cooling rate in the initial part of the route, as represented by the slope of the time–temperature curve, is much greater than the cooling rates of the baseline. Cooling rates appear to have been doubled. At an output of 535 cfm, the blowers appeared to cool the drum to temperatures that were between 40 F to 80 F lower than the previous baseline temperatures.

The fifth route of the day, which originates in Santa Monica at a time of 13.2, also shows signs of an increased cooling rate as evidenced by the steep cool-down temperature-time slope at the beginning of this route. During this segment of the route, the blower unit was effective in keeping drum temperatures in the rear drums between 30 F and 60 F lower than the baseline.

The afternoon portions of the run show that under driving conditions that result in relatively low operating temperatures for the brakes, no further reduction in brake drum temperature was realized by the use of the blower units.

Figure 23 shows the results for the front brake temperature measurements of the 80/20 route. Compared with the baseline data collected on Day 7, a temperature reduction of between 160 F and 130 F was realized with the blowers set on high. With the blowers set on low, a 50 F to 70 F temperature reduction was realized.

The long-term effect of blowers on steady-state drum temperatures is also seen in Figure 24, which shows front drum temperatures over a 9-hour period with the blowers operating on a low setting. Temperature peaks above 400 F have been eliminated, although one short segment of the data shows temperatures as high as 370 F under blower conditions. At temperatures around 320 F, no effect on brake drum temperature is seen with the blower. Overall temperature reduction appears to range between 20 F and 80 F, depending on the time of day.

Summary of Blower Unit Evaluation Experiments. Based on the evaluation experiments, the following conclusions regarding the use of blowers in reducing brake drum temperatures can be made:

1. The use of blowers set at a flow rate of 535 cfm was observed to approximately double the cooling rate of the brake drums, as evidenced by the increased slope of the temperature–time curve during stand still.
2. Rear brake drum steady-state temperatures were observed to be reduced between 40 F and 80 F with the blowers at an output of 535 cfm. Front brake drum temperature reductions were more than twice that of the rear.
3. At a lower blower output (300 cfm), temperature reductions in the brake drums were likewise reduced. Rear brake drum temperatures were reduced between 30 F and 60 F, while higher temperature reductions (~50 F to 70 F) were observed for the front brake drums.

The use of blowers to reduce brake drum temperatures was shown to be technically feasible. Target temperature reductions of 50 F and more were achieved through various blower output settings.

The use of air scoops to direct cooling air across the brake drum was also investigated. The results of these evaluation experiments are presented in the following section of the report.

Evaluation of the Effectiveness of Air Scoops in Reducing Drum Temperatures

An air scoop was mounted under the front of the bus just below the front bumper. Because of ground clearance problems, an air scoop was not fitted on the rear of the bus. A 4-in. diameter flexible duct was used to direct air flow from the duct toward the heated front brake drum. Inasmuch as the rear brake drums were not fitted with an air scoop, the data collected for the rear drums were considered to be baseline data for the rear brake drums.

Data were collected for two days of service along the 80/20 and the 84/20 routes. The test day for the 84/20 route was Day 3, while the test day for the 80/20 route was Day 4.

Figure 25 shows data collected for the front drums with the air scoops in place, compared with data collected under similar baseline conditions. Data for the air scoops show an approximate 3-hour block of time in which data were lost.

No reduction in steady-state drum temperature is apparent. Temperature signatures for both days during the morning portion of the run are almost identical; the air scoop was not instrumental in keeping brake drum temperatures below 350 F during severe service. Temperature differences in the afternoon portion of the run were most likely due to difference in traffic or driving style. Presumably, the apprentice driver who operated the bus during the afternoon portion became aggressive from the fourth day of testing to the seventh day of testing.

Figure 26 shows data taken on the 84/20 route for two days using identical drivers both days. Steady-state temperature levels measured for the front drums with the air scoops in place were as high or higher than the temperature measured under baseline conditions.

The use of air scoops to reduce steady-state temperatures did not prove technically feasible. The most probable reason is that the relatively slow speed of the bus did not generate the static pressure necessary to overcome restrictions posed by the ductwork and scoop itself. As a result, no real directed air flow occurred, and temperatures near to baseline temperatures were recorded.

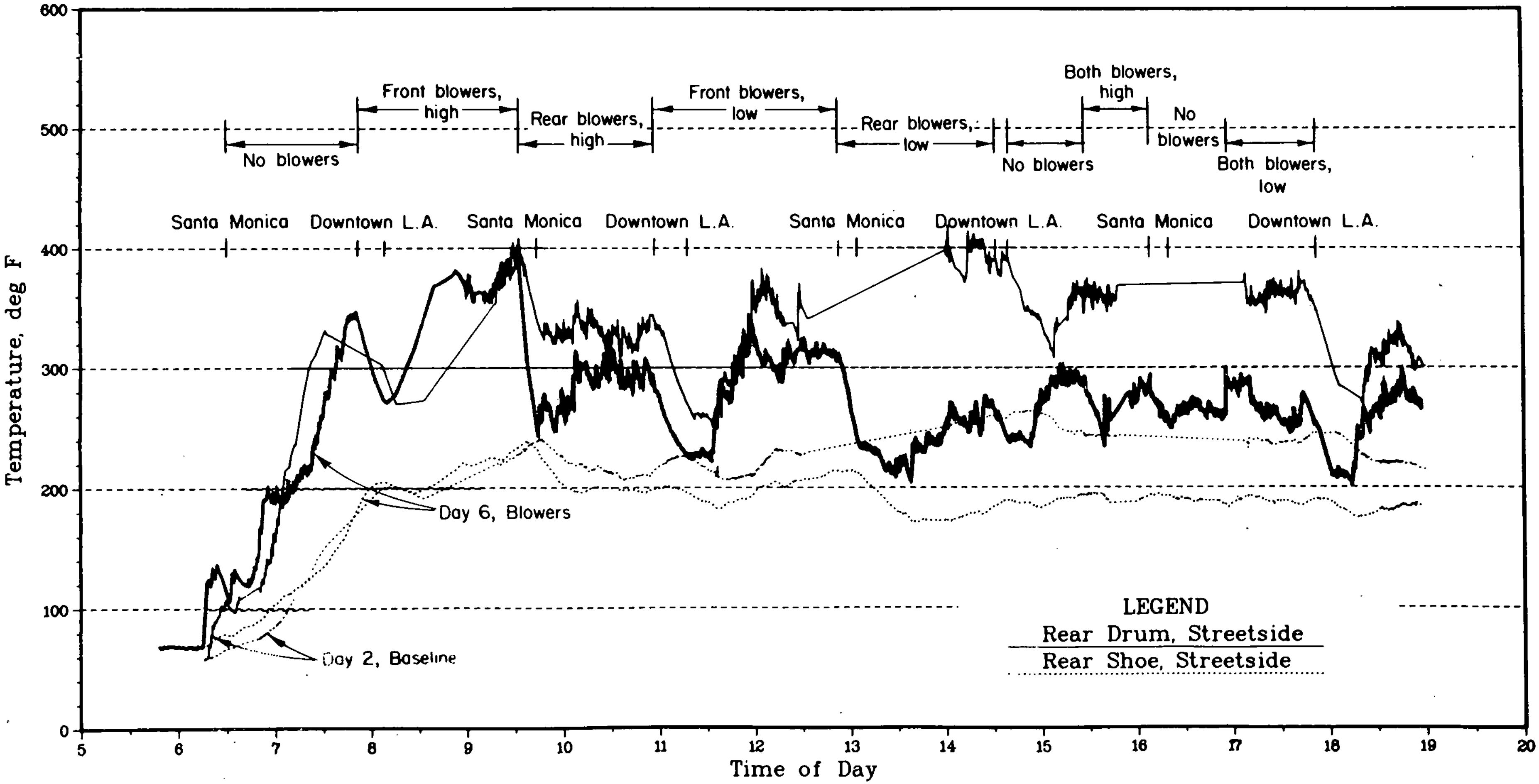


Figure 22. Rear brake drum temperatures, with and without electric blower units, 80/20 route.

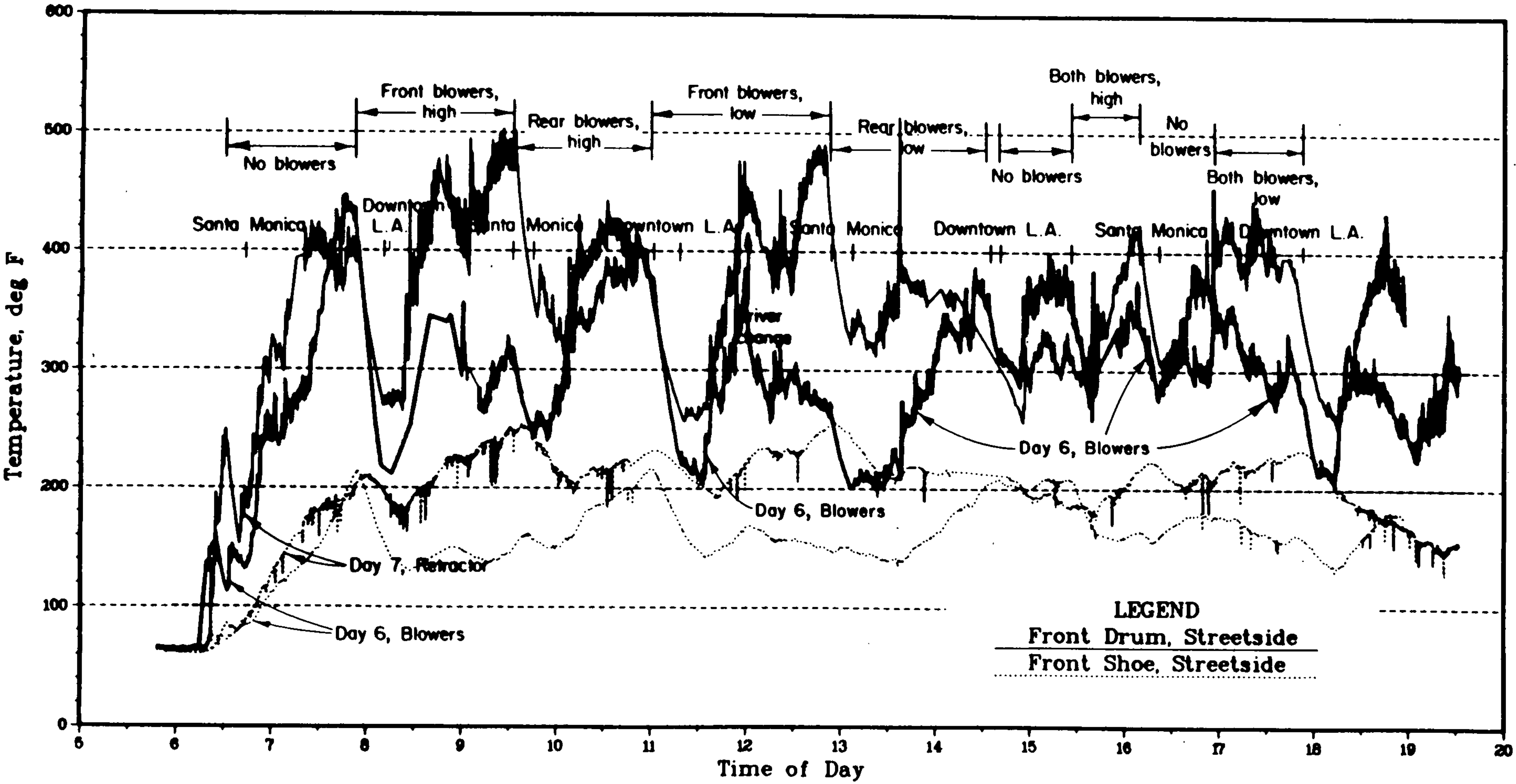


Figure 23. Front brake drum temperature, with and without blower units, 80/20 route.

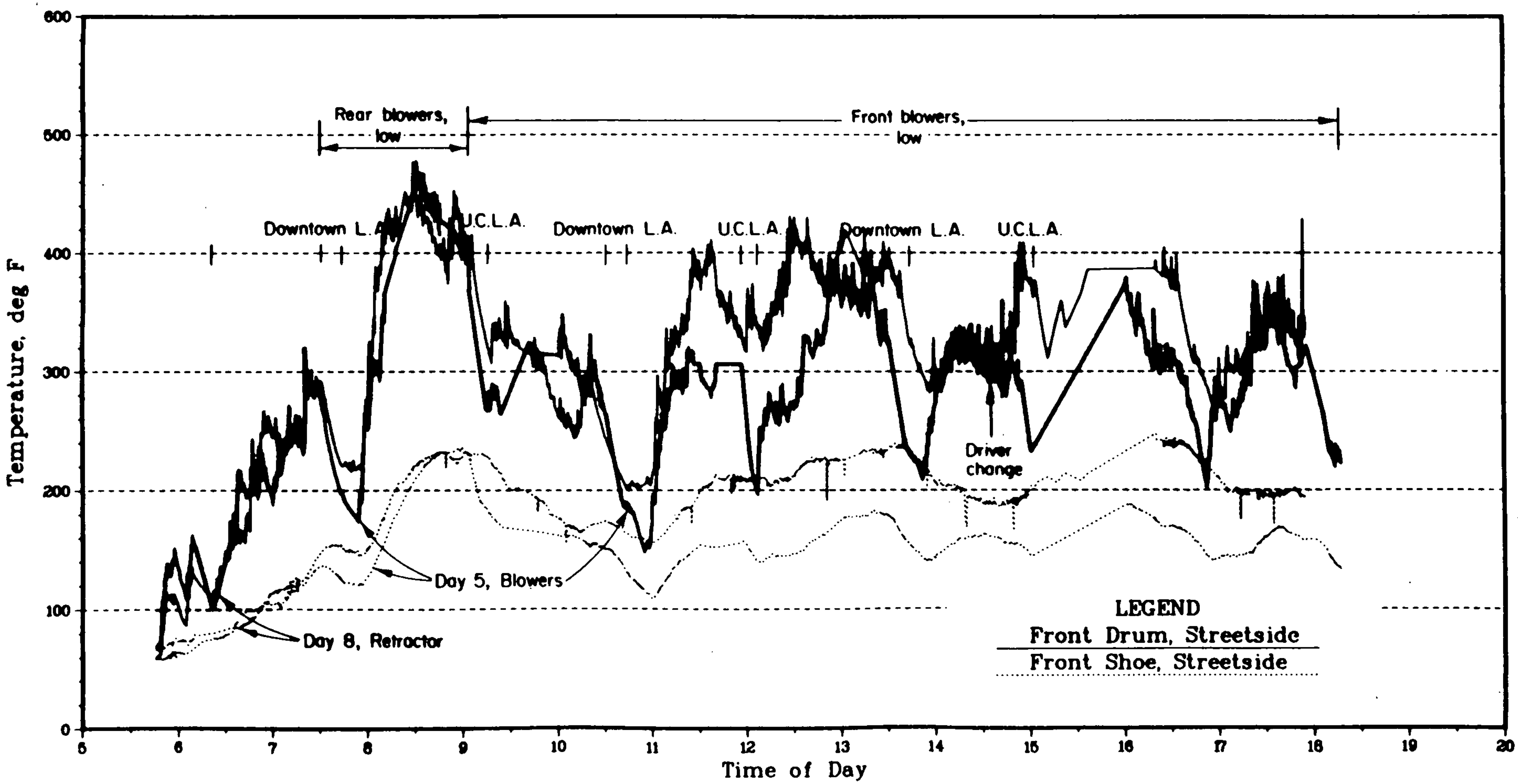


Figure 24. Front brake drum temperature, with and without blower units, 84/20 route.

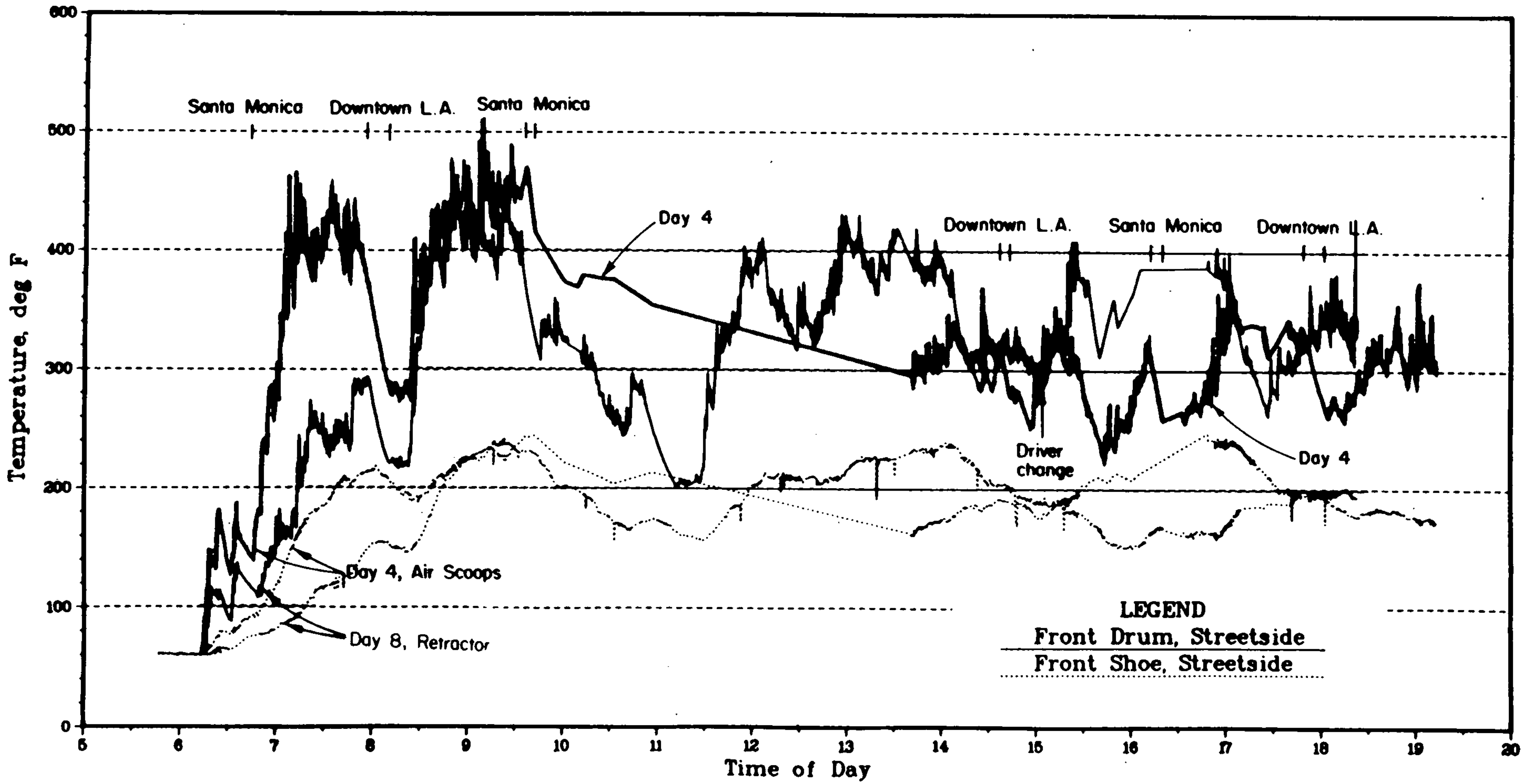


Figure 25. Front brake drum temperatures, with and without air scoop, 80/20 route.

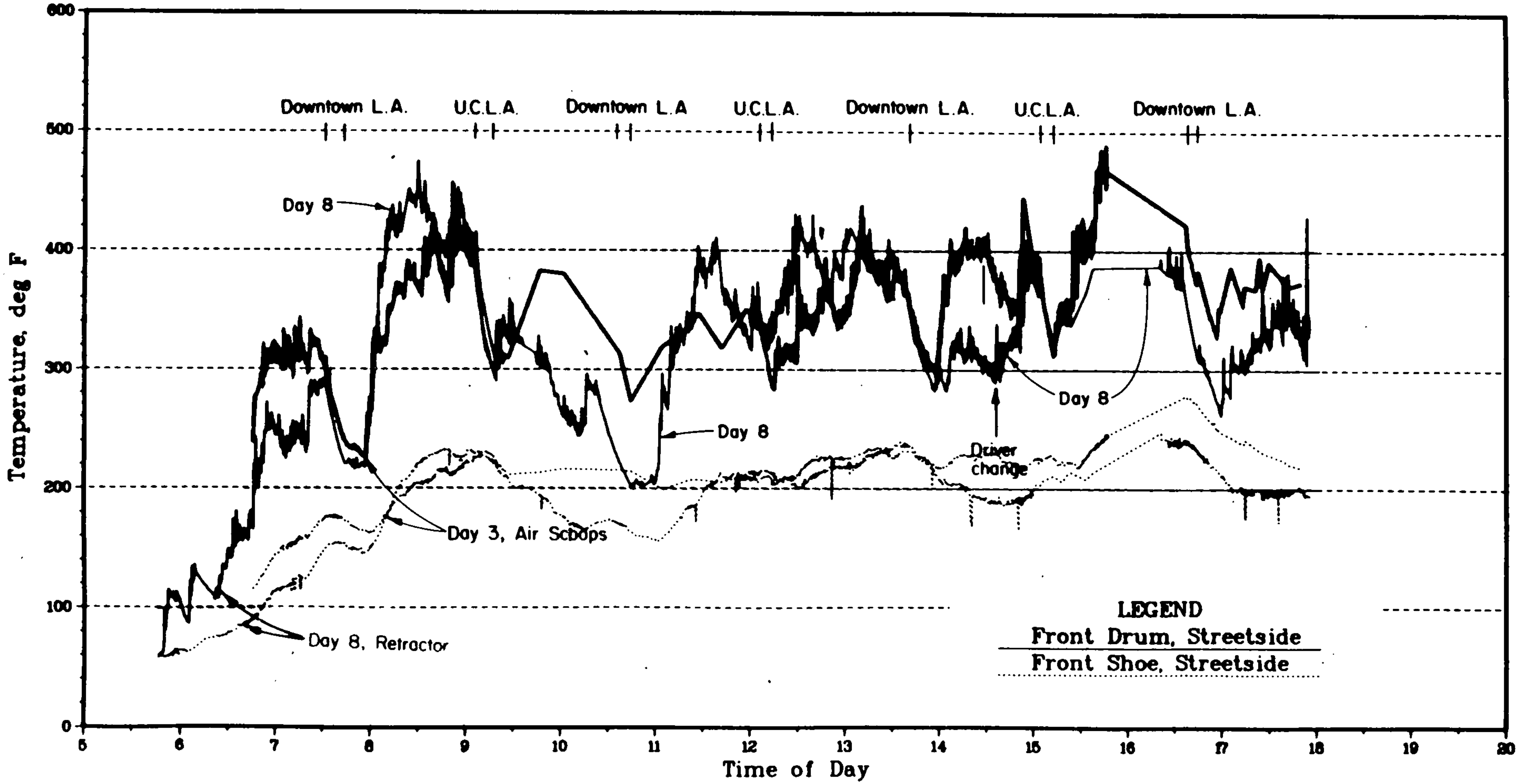


Figure 26. Front brake drum temperatures, with and without air scoops, 84/20 route.

Evaluation of the Effectiveness of Using Brake Shoe Retractors to Reduce Brake Drum Temperatures

A device designed to retract wedge brake shoes from the braking surface of the rear street-side brake drum was evaluated in-service on two test days. This retractor device is shown in Figure 18 and consists of two springs mounted diagonally between the brake shoes and a center ring positioned around the rear axle shield. Presumably, this device would prevent parasitic drag of the shoes against the drum and improve air circulation around the interior braking surface after braking. A retractor was not mounted on the front brake assembly; front brake drum data for these two test days were considered to represent baseline data for the front drums.

Figure 27 shows the rear brake drum temperature history on the 80/20 route for the brake system fitted with a brake retractor. Plotted simultaneously are the temperature histories for baseline data. Temperature histories for the morning portion of the route for the baseline data and the retractor data are almost identical, with the retractor-equipped system exhibiting temperatures sometimes 50 F higher than the baseline data. Temperature data in the afternoon segment show no real reduction in brake drum temperature as a result of the retractor mechanism.

The results of the experiments with the retractor-equipped brake assembly indicate that no reduction in brake drum steady-state temperature was achieved using brake shoe retractors.

Technical Summary of Effectiveness of Retrofit Schemes

The evaluation experiments conducted in-service in Los Angeles demonstrated that of the three techniques examined, the use of electrical blowers to increase convective flow around the heated brake drums was the only technique capable of reducing the steady-state brake drum temperature by the target amount of 50 F. Figure 26 shows that at a setting of 300 cfm the blower units used in these experiments consistently kept the steady-state brake drum temperature below the target temperature of 350 F. This target temperature was determined by examining literature sources of temperature–wear properties for phenolic-asbestos materials, in addition to comparing temperature histories for buses giving acceptable brake lining life.

The effect of temperature reduction via blower units on brake lining life was estimated in two ways. First, the general reduction in brake drum temperature on the GM RTS ADB to levels at or below 350 F would most likely result in brake lining lives similar to those exhibited by the GM 5307 "New Look" bus. This would result in an average lining life from 22,000 miles to around 35,000 miles, for a 60 percent increase in lining life. These data were taken from trends exhibited in Table 6.

The second technique involved the use of the temperature–wear equation of Lui and Rhee. The blowers were effective in reducing temperature by an average of 50 F. An estimate of the increase in lining life from this temperature reduction could be determined by using Eq. 6, which relates the wear of asbestos-phenolic friction materials to the operating temperature. However, a wide fluctuation in operating temperature, as evidenced by Figures 20 and 24, makes the assignment of a single operating temperature difficult throughout a typical service day. Steady-state near-surface temperatures fluctuated between 300 F and 450 F, with interface temperatures estimated to range between 350 F and 500 F. A reasonable average steady-state drum temperature was assumed to be 400 F, with an interface temperature of 450 F. The improvement in lining life as a result of a reduction in steady-state drum temperature from 400 F to 350 F was estimated with the following expression. A 50 F interface rise has been assumed, and all temperatures are represented in degrees kelvin. Average brake lining life at 450 F was estimated to be 20,000 miles.

$$\frac{1}{\text{Life}} = K_e^{\frac{-E}{RT}}; \quad 450^\circ\text{ F} = 505^\circ\text{ K}, \quad 400^\circ\text{ F} = 477^\circ\text{ K}$$

$$\frac{\frac{1}{20{,}000}}{\frac{1}{x}} = e^{\frac{-E}{R}\left(\frac{1}{505} - \frac{1}{477}\right)}$$

$$x = 35{,}000 \text{ miles}$$

Both estimates suggest an extension of brake lining life from approximately 20,000 miles to about 35,000 miles using the blower units.

The economic feasibility of using the blower units is evaluated in the next section of the report.

ECONOMIC EVALUATION OF COOLING BLOWER RETROFIT SCHEME

Of the three retrofit schemes technically evaluated during the Phase II experiments, only the use of electric-motor-powered blower units for forced convective cooling proved to be capable of significantly reducing brake temperatures and, thus, increasing brake lining life. The economic feasibility of that retrofit scheme is evaluated here. The other two retrofit schemes were found to be technically unfeasible and, therefore, are not subjected to economic evaluation.

Baseline and Retrofit Scheme Costs and Benefit/Cost Ratio

During Phase I, formulae for benefit-cost prioritization of selected retrofit schemes were defined. Those formulae and their definition were previously presented in the Phase I Interim Report (June 2, 1983). They are also presented in Appendix D of this report for reference purposes.

Applying those formulae, the costs and benefit/cost ratio for the baseline brake and cooling blower retrofit scheme were calculated to determine the economic feasibility of the cooling blower retrofit scheme. Table 10 presents a summary of the costs and benefit/cost ratio estimated for that retrofit scheme. Tables 11 through 15 show a more detailed breakout for each of the costs listed in Table 10.

As shown by the results presented in Table 10, only a marginal total cost reduction and relatively low benefit/cost ratio, i.e., 1.04, is estimated for the cooling blower retrofit scheme, assuming an initial average lining life of 22,000 miles. The principal reasons for this are:

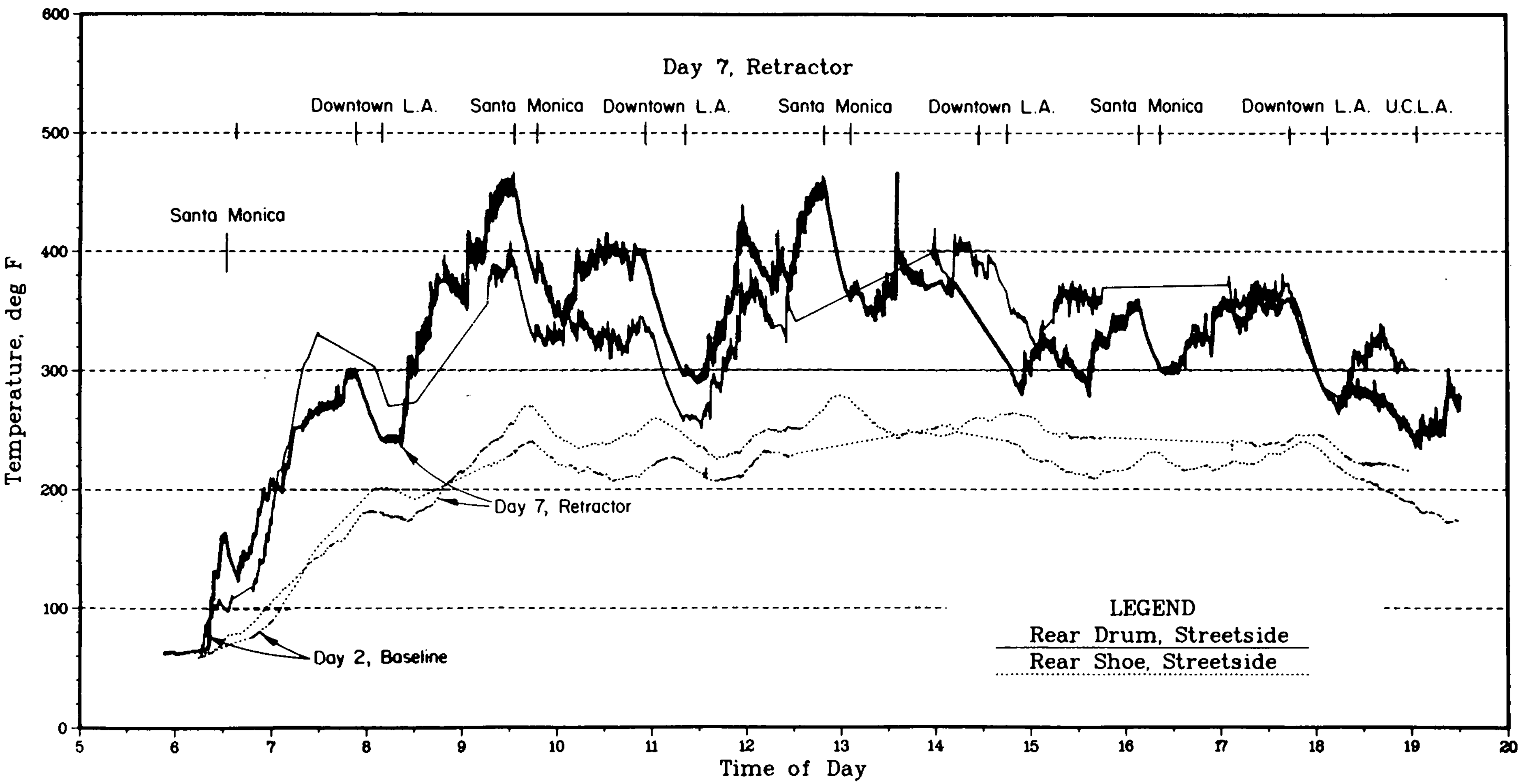


Figure 27. Rear brake drum temperature, with and without brake retractors, 80/20 route.

Table 10. Summary of costs and benefit/cost ratio for baseline brake and cooling blower retrofit scheme.

Cost Element	Estimated Costs, 1983 $/Vehicle		
	Baseline Brake GM RTS Bus	Cooling Blower Retrofit Scheme	Cost* Difference
Retrofit Capital Cost ($C)	$ 0	$ 2,000	+$2,000
Operations/Maintenance Cost Per Vehicle Life ($O/M)			
Reline Cost	16,275	9,975	-6,300
Drum Replacement Cost	2,613	1,446	-1,167
Cooling Blower Maintenance Cost	0	3,000	+3,000
Cooling Blower Increase in Fuel Cost	0	1,690	+1,690
Subtotal	$ 18,888	$ 16,111	-$2,777
Total Cost ($C + $O/M)	$ 18,888	$ 18,111	-$ 777
Benefit/Cost Ratio**	1.00	1.04	

*Cost Difference = Retrofit Scheme Cost - Baseline Brake Cost

$$\text{**Benefit/Cost Ratio} = \frac{\text{\$O/M Baseline Brake}}{\text{(\$C + \$O/M) Retrofit Scheme}}$$

Table 11. Cooling blower retrofit scheme, retrofit capital cost.

Cost Element	Front Axle		Rear Axle
Parts & Materials Cost, 1983$/Axle	$ 400		$ 400
Labor & Indirect Costs			
Labor, labor hours/axle	24		24
Labor Rate, 1983$/labor hour	25		25
Labor & Indirect Costs, 1983$/Axle	600		600
Total Cost,			
1983$/axle	$1,000		$1,000
1983$/vehicle		$2000	

Table 12. Relining cost for baseline brake and cooling blower retrofit scheme.

Cost Element	Baseline Brake GM RTS Bus		Cooling Blower Retrofit Scheme	
	Front Axle	Rear Axle	Front Axle	Rear Axle
Bus Life (BL), vehicle miles	500,000		500,000	
Lining Life (LL), vehicle miles	22,000	22,000	35,000	35,000
Number Relines/Vehicle Life*	21.7	21.7	13.3	13.3
Reline Cost/Reline				
Parts & Materials Cost, 1983$/reline	125	175	(Same as Baseline)	
Labor, labor hours/reline	9	9		
Labor Rate, 1983$/labor hour	25	25		
Labor & Indirect Costs, 1983$/reline	225	225		
Subtotal, 1983$/reline	$350	$400	$350	$400
Reline Cost/Vehicle Life,				
1983$/axle	$7,595	$8,680	$4,655	$5,320
1983$/vehicle	$16,275		$9,975	

*Number relines/vehicle life = $\frac{BL}{LL} - 1$.

Table 13. Drum replacement cost for baseline brake and cooling blower retrofit scheme.

Cost Element	Baseline Brake GM RTS Bus		Cooling Blower Retrofit Scheme	
	Front Axle	Rear Axle	Front Axle	Rear Axle
Bus Life (BL) vehicle miles	500,000		500,000	
Lining Life (LL), vehicle miles	22,000	22,000	35,000	35,000
Drum Life to Lining Life Ratio (DLR), lining sets/drum life	4	4	4	4
Number Drum Replacements/Vehicle Life*	4.7	4.7	2.6	2.6
Drum Replacement Cost/Drum Replacement				
Parts & Material Cost, 1983$/drum replacement	210	320	(Same as Baseline)	
Labor, labor hours/drum replacement	0.5	0.5		
Labor Rate, 1983$/labor hour	25	25		
Labor & Indirect Costs, 1983$/drum replacement	13	13		
Subtotal, 1983$/drum replacement	$223	$333	$223	$333
Drum Replacement Cost/Vehicle Life				
1983$/axle	$1,048	$1,565	$580	$866
1983$/vehicle	$2,613		$1,446	

*Number Drum Replacements/Vehicle Life = $\frac{BL}{LL \cdot DLR} - 1$.

Table 14. Cooling blower retrofit scheme, motor replacement and maintenance cost.

Cost Estimate	Front Axle	Rear Axle
Bus Life (BL)		
vehicle-miles	500,000	
vehicle-hours	30,000	
Motor Life (ML), vehicle-hours	7,500	7,500
Number Motor Replacements/Vehicle-Life*	3	3
Motor Replacement Cost/Motor Replacement		
Parts & Materials Cost, 1983$/Motor Replacement	200	200
Labor, labor hours/motor replacement	12	12
Labor Rate, 1983$/labor-hour	25	25
Labor and Indirect Costs, 1983$/Motor Replacement	300	300
Subtotal, 1983$/Motor Replacement	$ 500	$ 500
Motor Replacement Cost/Vehicle-Life,		
1983$/axle	$1,500	$1,500
1983$/vehicle	$3,000	

Number Motor Replacements/Vehicle Life = $\frac{BL}{ML} - 1$.

Table 15. Cooling blower retrofit scheme, estimated increase in fuel cost.

Cost Factor	Estimated Value	
Electric Power, Watts		
Front Axle	192	
Rear Axle	192	
Total Vehicle		384 (P)
Bus Life,		
Vehicle-Miles	500,000	
Average Speed, mph	17.8	
Vehicle-Hours		≈30,000 (BL)
Fuel-To-Electric Energy Conversion Efficiency, %		
Engine	30	
Gear Drive	95	
Alternator	60	
Overall		17 (%E)
Conversion Factors,		
BTU/watt-hour		3.413 (K_1)
Gallon diesel fuel/BTU		7.305 x 10^{-6} (K_2)
Fuel Cost, 1983$/gallon fuel		1.00 (FC)
Increase in Fuel Cost/Vehicle-Life*, 1983$/vehicle		$1,690

*Increase in Fuel Cost = $\frac{P \cdot BL \cdot K_1 \cdot K_2 \cdot FC}{\%E/100}$

- Brake lining life is estimated to increase only from 22,000 to 35,000 vehicle-miles, i.e., a 60 percent increase. This results in only an approximately 40 percent reduction in brake reline and drum replacement costs.
- Most of the reduction in brake reline and drum replacement cost is offset by the estimated retrofit capital, blower and motor maintenance, and increased fuel costs for the cooling blower retrofit scheme.
- NOTE: One other possible increase in cost for the cooling blower retrofit scheme was considered and discarded, i.e., a possible increase in the electrical power supply retrofit capital and maintenance costs. Those possible increases in cost were discarded because the additional alternator output required to power the cooling blower motors is quite small relative to the output capacity of the current bus alternator. ADB buses typically have an alternator rated at 24 volts and a maximum of 270 amps, or 6.5 kilowatts, output. The electric power required to drive the blower motors is approximately 0.4 kilowatts per bus or six percent of the maximum alternator output. It is estimated that this additional electric load could be picked up by the existing alternator and alternator drive without retrofit or additional maintenance costs.

Current (Baseline) Operations/Maintenance Cost Factors

Calculation of a benefit/cost ratio for any retrofitting scheme requires the determination of operations/maintenance costs for the current (baseline) system over the life of the bus. Table 16 presents a summary of operations/maintenance cost factors estimated for the baseline, i.e., GM RTC, bus brakes. The values listed for those factors are representative of data obtained from six transit agencies plus review of other past bus brake cost analyses.

Baseline operations/maintenance costs were calculated on a per axle basis using the baseline cost factors presented in Table 16. All costs are estimated in terms of 1983 dollars.

The baseline cost factor estimates listed in Table 16 are representative of data obtained from six of the approximately 1,000 transit agencies in the United States. A specific transit agency may calculate brake-related operations/maintenance costs by substituting operations/maintenance cost data unique to that agency for the data presented in Table 16. For example, the costs outlined in Table 10 assume an average lining life of 22,000 miles for the baseline (GMC RTS bus). The baseline brake costs increase if lining life is less than this average and the benefit/cost ratio of the blower retrofit technique increases. An increase in lining life from 15,000 miles to 35,000 miles, instead of from 22,000 miles to 35,000 miles, results in a benefit/cost ratio of 1.5, instead of 1.04 as before.

Table 16. Summary of current (baseline) operations/maintenance cost factors.

Factor	Representative Value
Bus Life, vehicle-miles	500,000
vehicle-hours	30,000
Brake Lining Life for GMC RTS Bus, vehicle-miles	
Front axle	22,000
Rear axle	22,000
Drum Life to Lining Life Ratio, lining sets/drum life	4
Reline Parts and Materials Cost for GMC RTS Bus, 1983$/axle-reline	
Front axle	$125
Rear axle	$175
Drum Replacement Parts and Materials Cost for GMC RTS Bus, 1983$/axle-drum replacement	
Front axle	$210
Rear axle	$320
Reline Labor for GMC RTS Bus, labor-hours/axle-reline	
Front axle	9
Rear axle	9
Drum Replacement Labor for GMC RTS Bus, labor hours/axle-drum replacement	
Front axle	0.5
Rear axle	0.5
Labor Rate Including Fringe Benefits and Overhead, 1983$/labor-hour	$ 25

Effect of Discounting Future Costs

UMTA recommends that discounting not be used in life-cycle cost evaluation procedures for bus procurements. Thus, present worth factors of 1.0 were applied in calculating all of the baseline and retrofit scheme costs and benefit/cost ratio listed in Table 10. As a check on what effect discounting would have on the estimated benefit/cost ratio for the cooling blower retrofit scheme, the costs are recalculated here assuming a discount rate of 10 percent per year. (See Appendix D for definition of the appropriate equations and present worth factors.)

Referring to Table 10,

- There would be no change in estimated Retrofit Capital Costs ($C).
- All Operations/Maintenance Costs Per Vehicle Life ($0/M) would be multiplied by a Present Worth Factor ($PWF_{o/m}$) of 0.64. Thus, $0/M costs for the baseline brake and cooling blower retrofit scheme would be reduced to $12,088 and $10,311, respectively. This reduces the estimated $0/M cost difference between the two from −$2,777 to −$1,777.
- Spare-bus costs must now be included in the calculation. Using Eqs. 10 and 11 in Appendix D and assuming a new bus price of $160,000/bus, 0.75 vehicle out-of-service days per axle lining replacement or 1.5 days per bus lining replacement, 180 vehicle miles per vehicle operating day, and a Present Worth Factor (PWF_{sb}) of 0.39 results in estimated discounted spare-bus costs of $1,148 and $700 per bus for the baseline brake and cooling blower retrofit scheme, respectively.
- Summation of the above costs results in total discounted costs of $13,236 and $13,011 per bus for the baseline brake and cooling blower retrofit scheme, respectively. This results in a benefit/cost ratio of 1.02 for the cooling blower retrofit scheme.

In conclusion, regardless of whether a nondiscounted or discounted economic evaluation approach is used, a relatively low benefit/cost ratio, i.e., 1.04 or 1.02, is estimated for the cooling blower retrofit scheme, assuming an average lining life of 22,000 miles prior to retrofit.

CHAPTER THREE

INTERPRETATION, APPRAISAL, APPLICATION

The main objective of this program was to aid transit system operators in improving brake lining life in Advanced Design Buses. The effort was prompted by numerous reports of reduced lining life in these newly designed buses, compared with lining life exhibited by older buses widely used in the transit industry.

The major premise of this study was that reduced brake lining life in the ADBs was prompted by higher system temperatures. Program efforts were directed toward confirming this premise and then devising techniques of economically reducing system temperatures and prolonging lining life.

The previous chapter presented experimental results which indicated that the use of electrical blowers on transit buses may reduce steady-state brake drum temperatures and extend the life of asbestos-phenolic friction materials. Because this retrofit technique requires the installation of a device that requires periodic maintenance and consumes electric power, this retrofit technique is not particularly satisfactory. For this reason, other alternative techniques are also discussed in this chapter. Individual transit companies are urged to collect brake lining data and assess lining wear data prior to implementing "bolt-on" retrofit techniques.

The purpose of this chapter is to discuss the pragmatic implications of the findings presented in Chapter Two and to recommend possible courses of action for improving bus brake life. Because much of the data collected during this program has been qualitative, specific lining life improvements will not be forecast; recommended maintenance actions or equipment changes will be presented. The following section summarizes the main findings of this program and discusses the influence of equipment and maintenance practices on lining life. Suggested courses of action to improve lining maintenance procedures and lining life are also discussed.

MAJOR FINDINGS

The major findings of this program are as follows:

1. Based on data provided by six transit companies, the brake linings of the two Advanced Design Buses produced by General Motors and Flxible were reported to have lining lives of approximately 22,000 miles, compared with 40,000 miles in the brakes of the older "New Look" bus produced by General Motors. According to these estimates, the brake lining life of the ADBs is about 45 percent less than the life of the older bus designs.

Costs for lining and drum maintenance for the older GM "New Look" 5307 and the GM RTS-II ADB are estimated to be $9,200 and $18,900, respectively, over the life of the bus. Those estimates were made by collecting data regarding replacement frequency, labor costs, and material costs.

2. For conventional asbestos-phenolic brake linings, temperature has no effect on wear up to a "threshold" temperature. Above the threshold temperature, lining wear increases exponentially with further increases in temperature. For bus brakes, a threshold temperature of 400 F was selected, based on temperature–wear data provided in the published literature. Because the lining wear is sensitive to temperature changes at levels above the threshold temperature, only slight increases in brake drum temperature can cause large increases in lining wear. For example, an increase in the temperature seen by the lining from 400 F to 450 F was estimated to result in a 42 percent decrease in lining life, assuming a lining life of between 35,000 to 38,000 miles at 400 F.

3. In-service measurements made in Los Angeles showed that the linings of the older "New Look" buses built by GM were exposed to temperatures below the lining threshold temperature of 400 F. On the other hand, identical linings on a GM RTS-II Advanced Design Bus were periodically exposed to temperatures estimated to be greater than the threshold temperature of the friction materials. Although the temperatures were not excessively higher than temperatures measured for the older buses, the temperature increase (estimated to be between 30 F and 60 F on the average) would be sufficient to reduce lining life.

4. Three techniques for reducing steady-state brake drum temperatures were evaluated in the field: (1) air sloops to direct ambient air to the drums, (2) electrical blowers to force convective cooling air to the drums, and (3) retractors for preventing possible residual drag in the wedge-brake shoes of the GM RTS-II ADB. Retrofit schemes were selected in part on the basis of their relative ease of retrofit by transit maintenance workers. No techniques requiring major component redesign were considered.

Only the electrical blowers were successful in reducing brake drum temperatures. However, the cost of retrofitting, maintaining, and operating a bus with these blowers was estimated to be $18,100, which was about equal to the projected costs savings due to improved lining life. The use of these blowers was not considered to be a cost-effective technique to improve lining life.

Although this program was not successful in designing retrofit techniques that could be easily implemented by transit companies, there are other factors that may influence brake lining life. These factors are described next.

FACTORS INFLUENCING LINING LIFE

Several factors were identified in the introducion as having influence on the amount of energy dissipated during braking and the subsequent life of the brake linings. The factors included vehicle and passenger weight, as well as alterations in the drive train efficiency of the bus. Although some of the factors identified as influencing lining life cannot be controlled by transit maintenance personnel, the program activities identified several controllable factors that may or may not influence brake lining life.

Influence of Brake Lining Materials

As discussed in the previous chapter and in Appendix B, the selection of brake lining material will most likely not affect the steady-state brake drum temperature because of the inherent low thermal conductivity demonstrated by such composite friction materials. Since most composite friction materials exhibit a threshold temperature above which wear is expected to increase exponentially, improved brake lining life will be realized by selecting a lining material exhibiting a higher threshold temperature.

For this program, phenolic-asbestos friction materials were assumed to have a threshold temperature of 400 F, as seen in Figure 5. Other friction materials employing a semimetallic composition or sintered metallic composition were seen to have threshold temperatures of 600 F and higher and are therefore expected to have better lining life than the asbestos-phenolic materials.

Because of the possible health hazards posed by the use of asbestos, there is now a general shift away from asbestos-phenolics to asbestos-free composite friction materials. Although specific temperature-wear properties of these materials are not publicly available, laboratory investigations and in-service experiments suggest that the Kevlar-reinforced, asbestos-free linings exhibit improved life over the asbestos-phenolic material (*13, 19, 20*). Investigations by Loken (*13*) with Kevlar-reinforced materials suggest that the threshold temperature for this material is approximately 600 F. If these early data are representative of current materials, substitution of asbestos-free materials would probably result in an improvement in the lining life of the ADBs up to levels currently exhibited by the GM 5307 "New Look" buses, since no temperature-induced failure is anticipated.

Although this program was not directed at determining the wear characteristics of various lining materials, experimental results suggest that a lining material exhibiting a threshold temperature greater than 400 F will most likely demonstrate improved wear performance. Asbestos-free lining materials appear to give improved performance by virtue of a higher threshold temperature. Although the use of these materials has been encouraged because of environmental concerns, longer brake lining life in the Advanced Design Buses may prove to be an additional benefit.

Influence of Driving Style

Phase II experiments illustrated how driving style can influence steady-state brake drum temperatures to a large degree. Figure 19 shows how, under identical route conditions, temperatures in the drums in the bus being driven by an aggressive experienced driver were more than 100 F greater than temperatures on the same bus under the direction of a student driver.

The influence of stopping speed on steady-state brake drum temperatures is seen in Eq. 3 derived earlier in this report. Since the steady-state drum temperature is proportional to the square of the stopping velocity, a slight increase in stopping velocity could be accompanied by a large temperature rise in the brake drums.

$$T_{ss} = T_o + \frac{m V^2}{2hAt_o} \qquad (3)$$

The increase in stopping velocity needed to cause an increase in steady-state temperature from 350 F to 400 F under 75 F ambient conditions is found by assuming identical cooling conditions and stopping frequency:

$$\frac{400-75}{350-75} = \left(\frac{V_2}{V_1}\right)^2$$

Assume $V_1 = 24$ mph and $V_2 = 26$ mph. A slight (2 mph) increase in stopping velocity can cause a 50-deg rise in the steady-state drum temperature, assuming similar cooling conditions.

These results strongly suggest that driver training may be one technique for reducing brake system temperatures and improving brake lining life. Although retraining of experienced drivers may not be feasible, new apprentice drivers could be instructed regarding the influence of driving habits on equipment life.

Influence of Wheel Rim Design

The influence of the amount of clearance between the wheel rim and the heated brake drum was discussed in the previous chapters. One particular transit system has reportedly achieved relatively long lining lives for GM RTS buses by employing 22-in., tube-type tires and rims (*21*). Tube-type tire rims have more rim-tire clearance than do tubless rims and so offer more possibility for improved free convection around the drum. A 12 percent improvement in convection cooling was estimated to be possible.

As tire manufacturers for heavy equipment continue to phase out tube tires, this possible retrofit scheme becomes more and more difficult to implement. As a possible alternative, drum manufacturers have introduced a 15-in. diameter drum for the GM RTS-II which has no external fins. Overall clearance between the tire rim and the drum is improved in this design, although the influence of extended fins is eliminated.

Because the influence of wheel-drum clearance on drum temperature was not investigated experimentally in this program, specific recommendations regarding the use of either tube-type tires or finless brake drums cannot be made.

Qualitatively, any scheme which promotes free or forced convection around the brake drums is encouraged.

Recent data from Sweden suggest that the composition of the wheel rim may influence brake drum temperatures. Aluminum-alloy wheels have been used to presumably aid in conducting heat from the drums and convecting this heat away from the surface of the wheel rim. This technique may also result in lower tire bead temperatures, allowing for the use of more efficient radial tires.

Influence of Brake Actuation Mechanism

The brake systems of the three bus designs employ various lining actuation mechanisms, as presented in Table 1. The brake system of the GM 5307 "New Look" bus employs a manually adjusted S-cam actuator, which fully retracts the shoes from the inside surface of the brake drums after braking. The Flxible 870 ADB uses a similar actuation device, except that an automatic slack adjustor is used. The GM RTS-II has a floating wedge actuator in which both halves of the shoes are self-actuating, i.e., the friction between the drum and shoe acts to further draw the shoe into the interface, increasing the braking force. Slack adjustment on these wedge actuators is automatic.

Because of the floating action of the wedge brake system, full retraction of the shoes from the drum after braking may not occur, leading to parasitic drag and reduced brake drum cooling. The retractor mechanism evaluated during this program was designed to reduce possible drag and improve brake drum cooling.

In-service experiments with a retractor showed no improvement or reduction in steady-state brake drum temperatures. This may be because: (1) the retractor was not able to retract the shoes fully from the drum after braking; (2) the amount of heat generated parasitic drag is small compared to heat input during braking; and (3) no parasitic drag occurs with the wedge-brake design and the retractors actually do nothing.

The in-service experiments were not able to determine why the retractor mechanism failed to aid in reducing brake drum temperatures. For this reason, modification of the actuation system has not been proven to be necessary, nor is it recommended.

Efforts by Flxible engineers have shown that early treadle actuators used on early Flxible 870 models were somewhat prone to maladjustment, causing the front-rear braking balance to be altered. This resulted in high rear brake system temperatures and subsequent high rear brake wear rates. Flxible engineers have successfully demonstrated that a change in the treadle actuator will alleviate this problem and improve brake lining life.

In summary, the two factors having the most influence on brake lining life, exclusive of any bolt-on retrofit techniques, are the brake lining material and the individual driving style of the operators. Lining materials having a threshold temperature of around 500 F or greater are recommended.

RECOMMENDATIONS

The results of this program indicate that the brakes of the Advanced Design Buses operate at higher temperatures than do the older buses, and that these higher temperatures are sufficient to cause an increase in the wear of asbestos-phenolic linings. Four observations about this issue can be made:

1. The brake systems of these buses will probably not be redesigned in the near future to fully dissipate the extra energy transferred during braking, i.e., unmodified brake systems on these buses can be expected to continue to operate at higher temperatures.
2. Trends to asbestos-phenolic brake materials with asbestos-free materials will probably continue.
3. The Kevlar-reinforced material shows evidence of having better temperature-wear properties than the asbestos-phenolics, although little field test data exist to confirm this.
4. The only "bolt-on" retrofit technique found to improve brake drum cooling was electrical blower units.

These observations suggest that transit maintenance officials first compare in-service the performance of different types of lining materials, using careful data collection and management techniques. The collection of brake lining wear data will require that more stringent attention be paid to quantities such as thickness changes, bus service conditions, and miles per lining set. To aid in this, Appendix E presents a possible format to be used to sort out the data. The tabular form of the data lends itself to being easily processed by a small personal computer aided by a spread-sheet software package.

Lining wear should be measured in terms of loss of pad thickness as measured by a caliper. The caliper should record the initial thickness of both the pad and the shoe prior to installation. During reline, the thickness of the worn pad would again be measured with the caliper, probably in four places along the arc of the shoe towards the center of the pad itself.

The installation of blower units to improve lining life was determined to be only marginally economical, assuming an initial average lining life of 22,000 miles. The benefit/cost ratio of this retrofit scheme increases with a reduction in initial lining life. For example, if a transit authority regularly changes ADB lining every 18,000 miles instead of 22,000 miles, the benefit/cost ratio increases from 1.04 to 1.3, for a 30 percent reduction in lining costs. However, improved lining life through replacement of lining material may make the use of bolt-on blowers unnecessary.

CHAPTER FOUR

CONCLUSIONS AND SUGGESTED RESEARCH

CONCLUSIONS

Braking systems convert kinetic energy into thermal energy in the process of stopping transit buses. Since most urban centers in the United States have relatively low grades on service routes, friction braking has been used exclusively on buses designed and produced in the United States.

Brake systems are designed and sized with the size of the vehicle and engine and the average stopping velocity and duty cycle in mind. For large buses, brake drums are purposely designed to have a greater mass in order to lengthen the time needed to attain the final equilibrium temperature.

In the initial Phase I experiments, temperature measurements were made on three buses of different designs in order to establish both cooling characteristics and temperature histories for the different buses. The results of these experiments were highly qualitative in nature because of the very small statistical sample selected, i.e., only one bus per design was evaluated. However, for the buses examined, the Advanced Design Buses exhibited slightly higher brake drum temperatures in-service from the older GM 5307 "New Look" buses that the ADB's were designed to replace. This slight temperature rise was estimated during the program to be responsible for the general reduction in lining life.

The most probable explanation for the increase in brake temperatures is an increase in the stopping velocity. The ADBs designed by General Motors and Flxible have been outfitted with more powerful drive trains which allow for faster acceleration during service. Since the amount of energy dissipated during braking is proportional to the square of the stopping velocity, only a slight increase ($\sim$ 2 to 3 mph) in stopping velocity is capable of causing the brake drum temperature to rise. No severe differences in brake drum cooling characteristics between the bus designs were found.

"Bolt-on" techniques for reducing the drum temperature in ADBs were designed using convective cooling data collected by in-service experiments. The only technique found to be technically capable of reducing the brake drum temperatures down to permissible levels was the use of cage blowers that operated on bus voltage and directed air via flexible ducts onto the heated drums. The cost of implementing this retrofit technique in terms of increased fuel use, capital equipment costs, and installation and maintenance costs was about equal to the savings realized by the improvement in lining life. The installation of blower units to improve lining life does not appear to offer any particular advantage.

Current trends in brake maintenance include a gradual change from asbestos-containing brake pads to asbestos-free pads. This trend may in itself improve brake lining life if the temperature–wear properties of these materials show improvement over the asbestos-phenolic pads.

FUTURE RESEARCH

There are several technical areas that merit future work, which are discussed in the following.

Brake Maintenance Data Collection

In light of the large amount of brake lining wear data produced by transit authorities, one possible program would involve generating a computer program to be used by transit companies to record and process brake lining wear data. This research/consulting project would first involve devising pragmatic techniques consistent with brake lining maintenance procedures for measuring changes in lining thickness. A second aspect of this project would then involve modifying existing computer spreadsheet programs to specifically analyze lining wear data and then statistically process the data to yield quantities such as median life, standard deviation, and so on. The computer program could also graphically display the normal distribution of lining life or wear rate for various lining materials under various duty cycles. The program could also be outfitted to calculate comparative costs of using various brake lining materials.

Examination of Temperature–Wear Behavior of Transit Coach Brake Materials

For this program the temperature–wear properties of the friction materials now used in transit coaches would be determined under laboratory conditions. These results would be used in conjunction with data generated during this program to forecast possible savings generated through the use of alternative friction materials.

Examination of Wedge Brake Retractor Mechanism

The wedge brakes used on the GM RTS appear to be very effective in stopping the bus. However, the self-actuating nature of the mechanism may promote parasitic drag, resulting in higher brake lining temperatures.

This program would design and experimentally evaluate the use of a retractor mechanism designed to retract the linings away from the drum after the bus has stopped. This would improve the convective heat transfer from the drum and improve brake lining life through lower lining temperatures.

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APPENDIX A

DESCRIPTION OF BUS SERVICE ROUTES USED IN PHASE I EXPERIMENTS

Table A-1. Summary of bus run-in and test mileage conditions for Los Angeles experiments, Phase I experiments.

Coach Design	Operating Conditions	Date	Route	Route Description	Miles
GM 5307 New Look	Run-in	18 June (Fri)	62/20	Wilshire line (urban local)	116
		19 June (Sat)	66/20	Wilshire line (urban local)	141
RTD #3197	Test	22 June (Tues)	83/20	Wilshire line (urban local and limited)	176
		23 June (Weds)	82/20	Wilshire line (urban local and limited)	118
		24 June (Thurs)	83/20	Wilshire line (urban local and limited)	176
GM RTS-02 ADB	Run-in	21 June (Mon)	62/20	Wilshire line (urban local)	116
		22 June (Tues)	66/20	Wilshire line (urban local)	141
RTD #8700	Test	28 June (Mon)	82/20	Wilshire line (urban local and limited)	118
		29 June (Tues)	83/20	Wilshire line (urban local and limited)	176
Flxible 870 ADB	Run-in	29 June (Tues)		Urban routes originating from Division 3	200
RTD #7616	Test	1 July (Thurs)	83/20	Wilshire line (urban local and limited)	75*
		2 July (Fri)	83/20	Wilshire line (urban local and limited)	154*

* Reduced mileage due to service problems with bus.

Table A-2. Description of individual routes found on the 83/20 line, Phase I experiments.

Route Number	Origination	Destination	Number of Stops	Average Stopping Speed	Route Description
308	6:39 - Santa Monica	8:03 - Los Angeles	24	24	Urban limited route during early morning rush hour
22	8:03 - Los Angeles	9:00 - Century City	38	23	Urban route connecting downtown Los Angeles with Century City
22	9:13 - Century City	10:10 - Los Angeles	36	24	Urban route after morning rush hour connecting Century City with downtown Los Angeles
22B	10:28 - Los Angeles	12:03 - Santa Monica	93	26	Long urban route with curb-to-curb stopping; routed through Brentwood section
20	12:15 - Santa Monica	1:45 - Los Angeles	58	24	Long urban route with curb-to-curb stopping
20	2:00 - Los Angeles	3:32 - Santa Monica	57	24	Long urban route with curb-to-curb stopping
308	3:55 - Santa Monica	5:21 - Los Angeles	51	26	Urban limited route during beginning of evening rush hour
308	5:35 - Los Angeles	6:51 - Santa Monica	53	25	Urban limited route during evening rush hour
22B	7:05 - Santa Monica	8:20 - Los Angeles	16	25	Urban route, post evening rush hour (low ridership)
21	8:38 - Los Angeles	9:38 - UCLA	insufficient data		Short urban route connecting downtown Los Angeles with UCLA

Note: Total mileage = 176 miles

Table A-3. Description of individual routes found on the 82/20 line, Phase I experiments.

Route Number	Origination	Destination	Route Description
308	6:35 - Los Angeles	8:29 - Santa Monica	Urban limited during early morning rush hour
20	8:44 - Santa Monica	10:06 - Los Angeles	Morning urban route
21	10:20 - Los Angeles	11:35 - UCLA	Mid-morning route connecting downtown Los Angeles with UCLA
21	11:50 - UCLA	1:05 - Los Angeles	Urban route with curb-to-curb stopping
21	1:20 - Los Angeles	2:36 - UCLA	Curb-to-curb stopping; route links downtown Los Angeles with UCLA
21	2:48 - UCLA	4:05 - Los Angeles	Mid-afternoon route between UCLA and Los Angeles
21	4:17 - Los Angeles	5:37 - UCLA	Urban route; early evening rush hour

Note: Total mileage = 118 miles

APPENDIX B

INFLUENCE OF BRAKE LINING MATERIAL ON BRAKE DRUM TEMPERATURES

Attempts were made to duplicate in the Phase II experiments the braking conditions and brake systems of Phase I to ensure consistent heating and cooling patterns. The Phase I experiments, which were used to determine cooling trends in various coach brake designs, were conducted using asbestos-phenolic brake shoe materials. Due to policy changes at SCRTD in Los Angeles where these in-service experiments were carried out, similar brake shoes were unavailable for the Phase II experiments. Suspected health hazards posed by the use of asbestos-containing friction materials have led to costly litigation and the subsequent banning of all asbestos-containing materials in SCRTD maintenance shops. As a result, an asbestos-free, Kevlar fiber based phenolic material manufactured by Worldbestos was substituted for the Phase II experiments.

No alteration in brake drum cooling or heating characteristics was expected due to the change in brake shoe material. Current estimates suggest that during braking, 94 percent of the heat is transferred to the drum, with 6 percent transferred into the linings (17). The proportion of heat generated which enters the drum or disc can be calculated by the following equation:

$$\sigma_1 = \frac{A_1 \ (K_1 \rho_1 c_1)^{\frac{1}{2}}}{A_1 \ (K_1 \rho_1 c_1)^{\frac{1}{2}} + A_2 \ (K_2 \rho_2 c_2)^{\frac{1}{2}}} \qquad \text{(B-1)}$$

where the quantities K, ρ, and c refer to thermal conductivity, density, and specific heat respectively; suffixes 1 and 2 refer to drum and lining respectively. Changes in material properties of the linings could influence this ratio and increase the amount of heat transferred into the drums, making the linings run cooler and thus prolong lining life. However, the amount of change necessary to affect brake drum temperatures during service is prohibitively large. Assuming the thermal conductivity, density, and specific heat of the asbestos-free linings are all 20 percent less than their asbestos counterparts, Equation (B-1) predicts that the ratio would become 96 percent instead of 94 percent previously. The resulting 2 percent decrease in heat to the brake drums would cause the drums to run slightly hotter than the drums which contact asbestos-phenolic linings. Assuming a steady-state drum temperature of 350 F, a 2 percent increase in the amount of heat rejected to the drum would result in a 7 F temperature rise. For this reason, the switch to asbestos-free brake shoes was not expected to greatly influence the heating and cooling characteristics of the brake systems.

A comparison of the temperature histories for the Phase I and Phase II experiments shows similar temperature levels for the brake drums of the GM RTS II. Figure B-1 shows temperature levels on a curb-to-curb urban route for the GM RTS-II fitted with asbestos-phenolic shoes, while Figure B-2 shows temperature levels on a similar route for a bus fitted with asbestos-free brake shoes. Temeprature levels in both cases range between 300 F and 400 F during the duration of the run.

Reported increases in brake lining life while using asbestos-free brake shoes in transit coach applications are most likely due to improved temperature-wear properties and not due to lower temperature in the brake system itself. Laboratory experiments conducted on lining materials made with Kevlar fibers showed the non-asbestos, Kevlar material to have improved temperature-wear properties compared to asbestos-phenolic materials (13). Threshold temepratures for early Kevlar-reinforced materials were around 600 F, compared to 400 F - 450 F for asbestos-phenolic materials.

For these reasons, the conditions of the Phase II experiments were considered to be thermally similar to earlier Phase I experiments conducted using an asbestos-phenolic configuration.

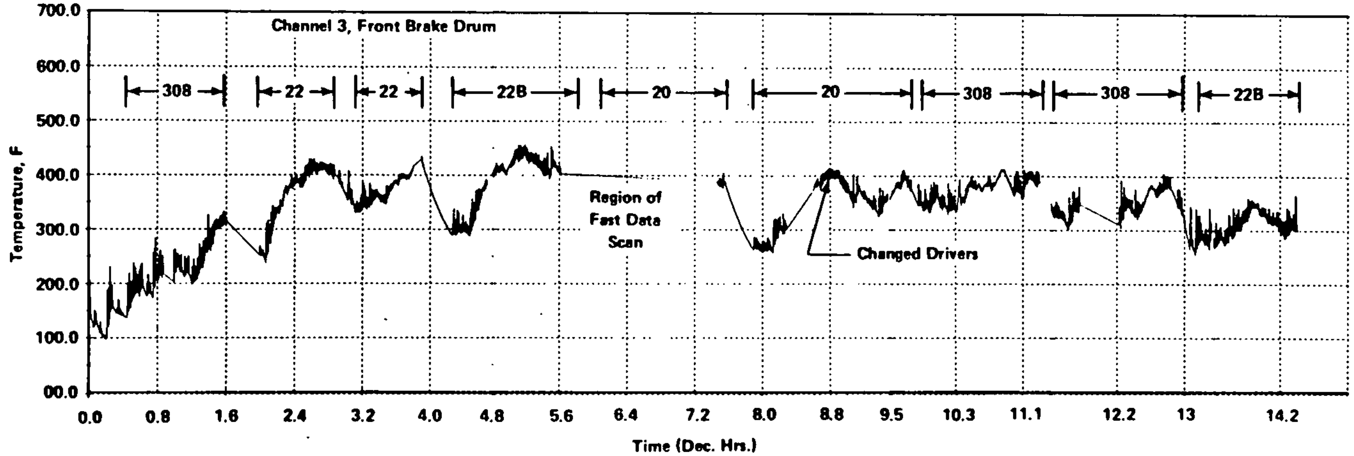


Figure B-1. Time–temperature history for GM RTS-II ADB, 83/20 line, front drum and shoe, asbestos-phenolic linings.

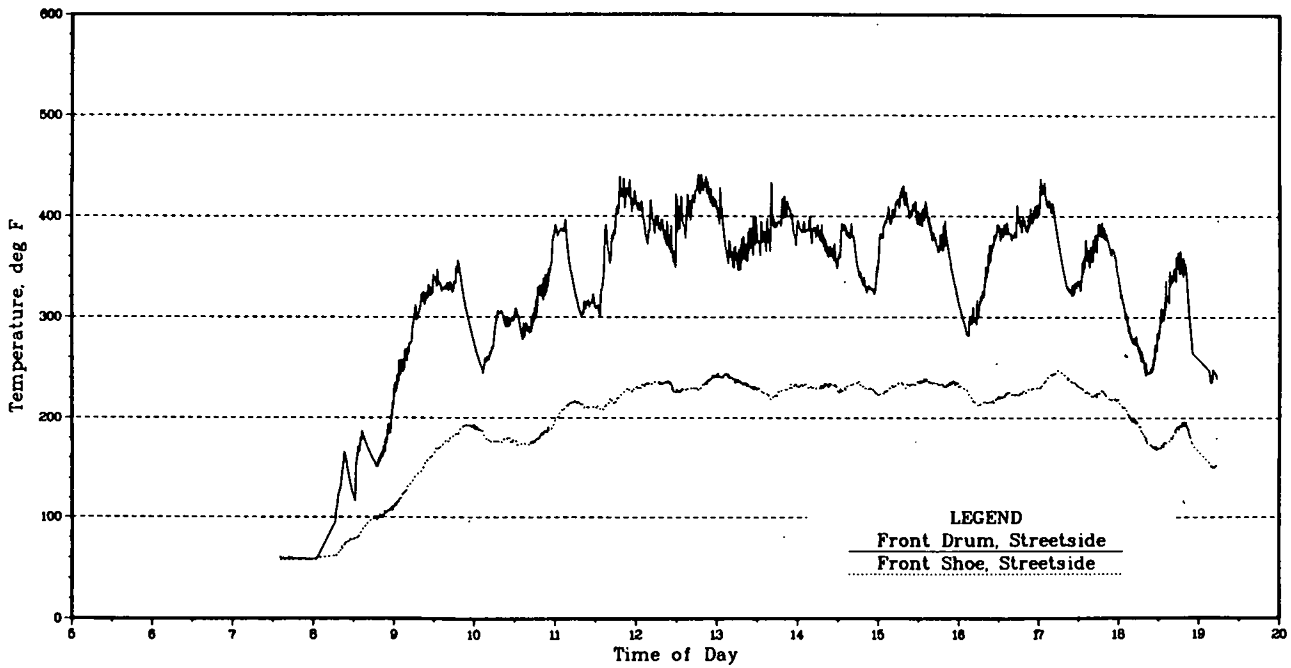
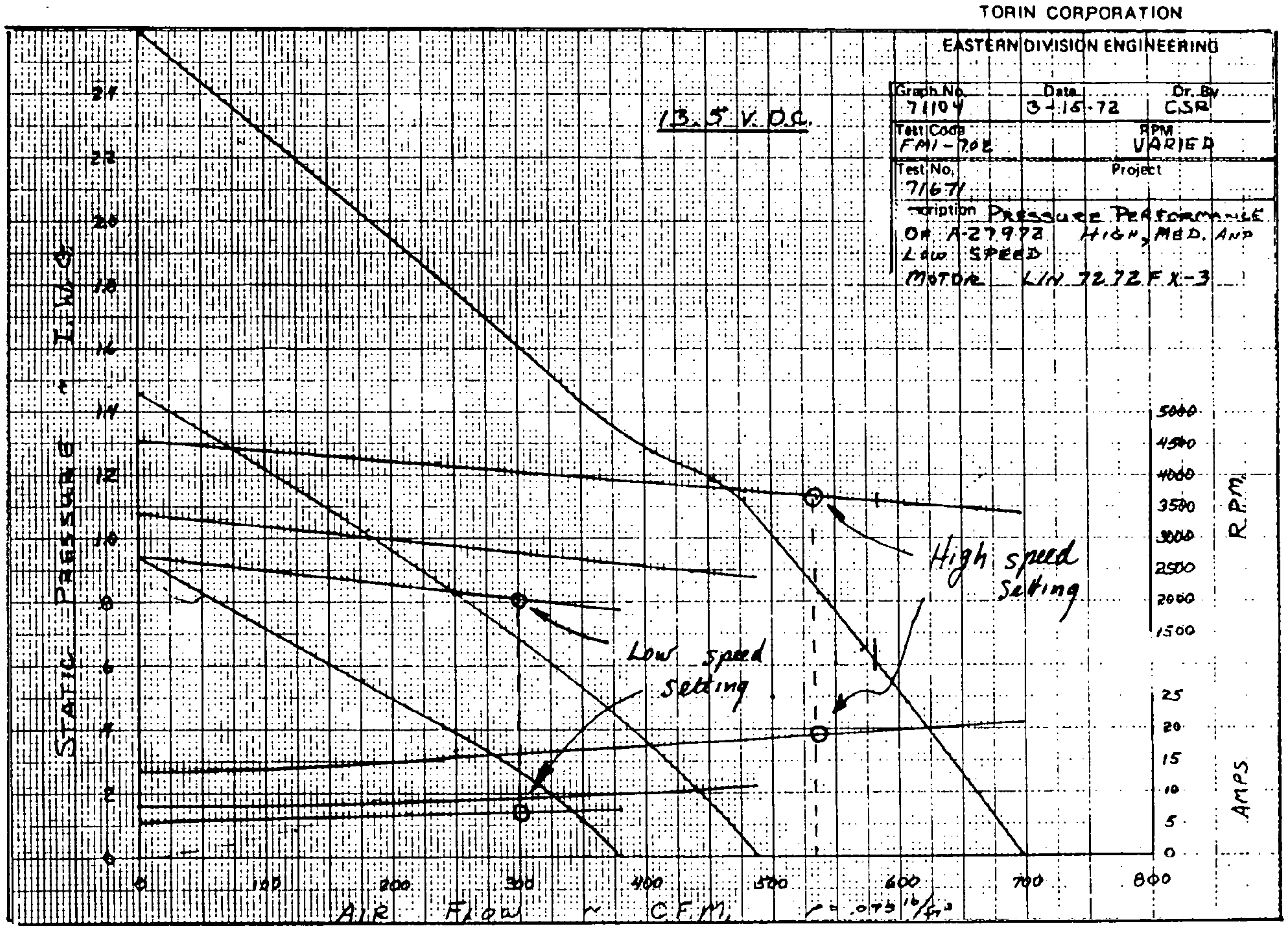


Figure B-2. Time–temperature history for GM RTS-II ADB, 80/20 line, front drum and shoe, asbestos-free linings.

APPENDIX C

OPERATING SPECIFICATIONS FOR ELECTRICAL BLOWER UNITS



APPENDIX D

DEFINITION OF FORMULAE FOR BENEFIT-COST PRIORITIZATION OF SELECTED RETROFIT SCHEMES

The formulae necessary to calculate benefit/cost ratios are presented in this appendix. These formulae are general in nature and can be applied to any retrofitting scheme and/or to any transit property.

Benefit/Cost Ratio

With respect to the benefit/cost ratio, the numerator, or benefit portion of the ratio, is defined as those future costs associated with the current (baseline) system which will be eliminated over the life of the bus. Those costs include operations/maintenance costs and spare-bus costs. The latter costs are related to bus out-of-service time associated with brake maintenance. The denominator, or cost portion of the ratio, is defined as the total life-cycle costs associated with a specific retrofitting scheme. Those costs include capital costs required to modify or retrofit the current system, operations/maintenance costs and spare-bus costs. Specifically,

$$\text{Benefit/Cost Ratio} = \frac{\$O/M\ \text{current} + \$SB\ \text{current}^*}{\$C\ \text{alternative} + \$O/M\ \text{alternative} + \$SB\ \text{alternative}^*} \quad (1)$$

where:

$O/M current = Operations/Maintenance costs for the current (baseline) system over the life of the bus.

$SB current = Spare-Bus costs for the current (baseline) system.

$C alternative = Capital costs required to modify or retrofit the current system for a specific retrofitting scheme.

$O/M alternative = Operations/Maintenance costs for a specific retrofitting scheme over the life of the bus.

$SB alternative = Spare-Bus costs for a specific retrofitting scheme.

Capital costs associated with the current (baseline) system are not included in the above formula as they are sunk costs and are a part of both the current system and all alternative retrofitting schemes. Thus, the capital costs listed for specific retrofitting schemes include only the capital costs required to modify or retrofit the current system.

Capital Costs

For purposes of this analysis, capital costs for a specific retrofitting scheme are defined as the cost of the labor, overhead (burden), and parts and materials required to modify or retrofit the current system. That is,

$$\$C\ \text{alternative} = (RL \cdot \$LR) + \$RP \quad (2)$$

where:

$C alternative = Capital costs required to modify or retrofit the current system for a specific retrofitting scheme ($).

RL = Retrofitting or modification labor required for a specific retrofitting scheme (labor hours).

* Note: Spare-bus costs go to zero when future cash value is not discounted, i.e., when present worth factors of 1.0 are assumed. This is explained under Spare-Bus Costs.

$LR = Labor Rate including fringe bemefits and burden ($/labor hour).

$RP = Retrofitting or modification Parts and materials cost ($).

Since the capital costs for retrofitting occur at the beginning of a project, they are not discounted to determine present worth.

Operations/Maintenance Costs

Operations/maintenance costs are defined as the maintenance costs typically associated with: (1) brake relining; (2) drum replacement; and (3) maintenance of any major new component introduced as part of a specific retrofitting scheme. All three of those costs are calculated in a similar manner. That is, first the cost per single maintenance action is calculated. That cost is then multiplied by the number of times the particular maintenance action is expected to occur over the life of the bus.

Cost per single maintenance action is defined as the cost of labor, burden, and parts and materials required to complete that particular maintenance action. The number of times a particular maintenance action is expected to occur over the life of the bus is determined as follows:

- The number of brake lining sets required over the life of the bus is determined by dividing the expected life of the bus by the expected life of a lining set (both lives expressed in vehicle miles).
- The number of times the brakes must be relined is equal to the number of lining sets required minus one, i.e., the original set which comes installed on the bus.
- The number of brake drums required over the life of the bus is determined by dividing the number of lining sets required by the expected number of lining set lives per single drum life.
- The number of times the drums must be replaced is equal to the number of drums required minus one, i.e., the original drum which comes installed on the bus.
- The number of times a major new component must be replaced or maintained is determined in a similar manner to that for brake drums except that a different factor for expected number of lining set lives per component life may be used.

Nondiscounted Costs

Based on the above discussion, nondiscounted operations/ maintenance costs over the life of the bus are calculated as follows:

$$\$o/m = \left(\frac{BL}{LL} - 1\right)\left[\left(ML_{\ell} \cdot \$LR\right) + \$MP_{\ell}\right] + \left(\frac{BL}{LL \cdot DLR} - 1\right)\left[\left(ML_{d} \cdot \$LR\right) + \$MP_{d}\right] + \left(\frac{BL}{LL \cdot XLR} - 1\right)\left[\left(ML_{x} \cdot \$LR\right) + \$MP_{x}\right] \tag{3}$$

where:

$o/m = Nondiscounted operations/maintenance costs for the current (baseline) system or any specific retrofitting scheme over the life of the bus ($).

BL = Bus Life (vehicle miles/vehicle life).

LL = Lining Life (vehicle miles/lining set).

DLR = Drum Life to lining life Ratio (lining sets/drum).

XLR = Life or maintenance interval of any major new component introduced as part of a specific retrofitting scheme to lining life Ratio (lining sets/component).

ML_{ℓ} = Maintenance Labor to reline brakes (labor hours).

ML_d = Maintenance Labor to replace drum (labor hours).

ML_x = Maintenance Labor to replace or maintain component X (labor hours).

$LR = Labor Rate including fringe benefits and burden ($/labor hour).

$\$MP_{\ell}$ = Maintenance Parts and materials cost to reline brakes ($).

$\$MP_d$ = Maintenance Parts and materials cost to replace drum ($).

$\$MP_x$ = Maintenance Parts and materials cost to replace or maintain component X ($).

All dollar ($) cost estimates are made in terms of present year dollars.

Present Worth

Since operations/maintenance costs are spread out over the life of the bus, they may be discounted to determine present worth. For this specific case, where the costs occur at regular intervals, a very close approximation of present worth may be calculated by multiplying the total nondiscounted operations/maintenance costs over the life of the bus by a single present worth factor, or

$$\$O/M = \$o/m \cdot PWF_{o/m} \tag{4}$$

where:

$O/M = The present worth of the total operations/maintenance costs occuring over the life of the bus ($).

$o/m = The total nondiscounted operations/maintenance costs occurring over the life of the bus as defined by Equation (3) ($).

$PWF_{o/m}$ = Present Worth Factor, operations/maintenance costs.

Calculation of the present worth factor is based on the equation used to calculate present worth factors for sinking funds. That equation, which is well documented in the literature, is

$$pwf_{sf} = \frac{(1+i)^n - 1}{n \cdot i\ (1+i)^n} \qquad (5)$$

where:

pwf_{sf} = Present worth factor for sinking funds, or the present value of a series throughout n periods of $1/n deposits made at the end of each period, with interest compounded at the end of each period.

n = Number of interest periods.

i = Interest or discount rate for one period (%/100).

As an approximation, a bus life of 10 years with equal levels of operations/maintenance costs occurring at the end of the first through ninth years is assumed. Combining Equation (5) and this assumption, gives

$$PWF_{o/m} = \frac{(1+i)^9 - 1}{9 \cdot i\ (1+i)^9} \qquad (6)$$

where: $PWF_{o/m}$ and i are as previously defined.

Examples of operations/maintenance cost present worth factors calculated for a range of discount rates are listed below.

Discount Rate % per year	$PWF_{o/m}$
0	1.00
5	0.79
10	0.64

Spare Bus Costs

In addition to capital and operations/maintenance costs, the out-of-service time associated with brake maintenance for the current (baseline) system and each alternative retrofitting scheme is also estimated. Based on that estimate and related bus operating time, an initial investment for spare buses and spare-bus costs are calculated for the current (baseline) system and each alternative.

Out-of-Service-Time

Out-of-service time (days) associated with brake maintenance over the life of the bus is equal to the estimated bus out-of-service time per lining replacement multiplied by the number of times the linings are expected to be replaced over the life of the bus, or

$$OST_t = OST_i \left(\frac{BL}{LL} - 1\right) \qquad (7)$$

where:

OST_t = Bus Out-of-Service Time over the life of the bus (vehicle out-of-service days/vehicle life).

OST_i = Bus Out-of-Service Time per lining replacement (vehicle out-of-service days/lining replacement).

BL = Bus Life (vehicle miles/vehicle life).

LL = Lining Life (vehicle miles/lining set).

Drum and other component maintenance is typically done at the same time that linings are replaced. Thus, no additional days of out-of-service time are added for those specific maintenance actions.

Operating Time

Operating time (days) over the life of the bus is equal to the bus life expressed in vehicle miles divided by estimated bus operating miles per operating day, or

$$OT_t = \frac{BL}{BD} \tag{8}$$

where:

OT_t = Bus Operating Time over the life of the bus (vehicle operating days/vehicle life).

BL = Bus Life (vehicle miles/vehicle life).

BD = Bus operating Day (vehicle miles/vehicle operating day).

Initial Spare-Bus Investment

Initial spare-bus investment per bus associated with brake maintenance is equal to the ratio of bus out-of-service time to bus operating time multiplied by initial bus price, or

$$\$sb = \left(\frac{OST_t}{OT_t}\right) \text{Initial Bus Price (\$/bus)} \tag{9}$$

where:

OST_t and OT_t are as previously defined, and

\$sb = Initial spare-bus investment per bus (\$/bus).

Combining Equations (7), (8) and (9) gives,

$$\$sb = \frac{OST_i\left(\frac{BL}{LL} - 1\right)}{BL/BD} \text{ Initial Bus Price (\$/bus)}$$

$$= OST_i \cdot BD\left(\frac{1}{LL} - \frac{1}{BL}\right) \text{Initial Bus Price (\$/bus)} \tag{10}$$

where:

\$sb, OST_i, BD, LL, and BL are previously defined.

Spare-Bus Costs

The calculated initial spare bus investment indicates the required increase in fleet size and initial investment resulting from bus out-of-service time associated with brake maintenance. However, that additional investment will eventually be recovered in terms of either longer calendar life for the fleet or increased disposal value at the end of a specified number of years.

The nondiscounted incremental increase in either fleet life or fleet disposal value is equal to the initial fleet investment for spare buses. Thus, when cash value is not discounted over time, calculated net spare-bus costs go to zero. However, when cash value is discounted over time, the present worth of the incremented increase in either fleet life or fleet disposal value is less than the initial investment for spare buses. That difference between initial investment and present worth of the incremental increase in either fleet life or fleet disposal value is defined as spare-bus costs, or

$$\$SB = \$sb\ (1-PWF_{sb}) \tag{11}$$

where:

$SB = Spare-Bus costs, i.e., difference in initial spare-bus investment per bus and present worth of bus at end of service period ($/bus).

$sb = Initial spare-bus investment per bus ($/bus).

PWF_{sb} = Present Worth Factor, spare-bus cost.

Calculation of the present worth factor is based on the equation used to calculate present worth or discount factors for a sum of money at some future date. That equation, which is well documented in the literature, is

$$pwf_{\$} = \frac{1}{(1+i)^n} \tag{12}$$

where:

$pwf_{\$}$ = Present worth of $1 due n periods hence, when interest is compounded at the end of each period.

n = Number of interest periods.

i = Interest or discount rate for one period ($/100).

As an approximation, a bus life of 10 years is assumed. Combining Equation (12) and this assumption, gives

$$PWF_{sb} = \frac{1}{(1+i)^{10}} \tag{13}$$

where:

PWF_{sb} and i are as previously defined.

Examples of spare-bus present worth factors calculated for a range of discount rates are listed below.

Discount Rate, % per year	PWF_{sb}
0	1.00
5	0.61
10	0.39

APPENDIX E

SAMPLE TABULAR DATA SHEET FOR COLLECTING LINING WEAR DATA

Table E-1. Forces required to pull two cable constructions in two wire sizes through 50 ft 4 bend, 2 plane loops—conduit insertion tests.

	Pull Force in Pounds						Failures
	1st Bend	2nd Bend	3rd Bend	4th Bend	Follow Thru	Max.	in 4 Cables
500 MCM THHN/THWN E23919 in 3" Conduit	1006	981	938	1013	1150	1150 1st Run	5
	1000	1150	1325	1500	1725	1725 2nd Run	6
	800	900	1062	1225	1375	1375 3rd Run	20
	935	1010	1108	1246	1417		
500 MCM XHHW ALCAN in 3" Conduit	775	1025	1300	1600	1600	1600 1st Run	1
	975	875	1000	950	850	1000 2nd Run	0
	825	938	838	938	1063	1063 3rd Run	4
	858	946	1046	1163	1171		
#1/0 AWG THHN/THWN E23919 in 1-1/4" Conduit	1000	1700	1175	1000	950	1700 1st Run	10
	325	375	600	500	625	625 2nd Run	20
	238	413	438	550	425	550 3rd Run	1
	521	829	738	683	667		
#1/0 AWG XHHW ALCAN in 1-1/4" Conduit	1075	1750	1700	1975	2900	2900 1st Run	1 Twist
	975	1250	2075	3100	Failed	3100 2nd Run	1
	975	950	1175	1525	2700	2700 3rd Run	1 Twist
	1008	1317	1650	2200	2150		

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