EPORT NO. UMTA-MA-06-0049-81-4

EVALUATION OF SQUEAL NOISE FROM THE WMATA TRANSIT CAR DISC BRAKE SYSTEM: A PRELIMINARY INVESTIGATION

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MARCH 1981

FINAL REPORT

DOCUMENT IS AVAILABLE TO THE PUBLIC THROUGH THE NATIONAL TECHNICAL INFORMATION SERVICE, SPRINGFIELD, VIRGINIA 22161

Prepared for
U.S. DEPARTMENT OF TRANSPORTATION
RESEARCH AND SPECIAL PROGRAMS ADMINISTRATION
Transportation Systems Center
Cambridge MA 02142

NOTICE

NOTICE

Technical Report Documentation Page

1. Report No.	2. Government Accession No.	3. Recipient's Catalog No.
UMTA-MA-06-0049-81-4	PB-81-17736	

4. Title and Subtitle
EVALUATION OF SQUEAL NOISE FROM THE WMATA CAR DISC BRAKE SYSTEM: A PRELIMINARY INVESTIGATION

5. Report Date
March 1981

6. Performing Organization Code
DTS-331

7. Author's)
Larry A. Ronk, Michael A. Staiano

8. Performing Organization Report No.
DOT-TSC-UMTA-81-10

9. Performing Organization Name and Address
ORI, Inc.*
1400 Spring St.
Silver Spring MD 20910

10. Work Unit No. (TRAIS)
UM149/R1708

11. Contract or Grant No.
DTRS-57-80-P81005

12. Sponsoring Agency Name and Address
U.S. Department of Transportation
Urban Mass Transportation Administration
Office of Technology Development and Deployment
Office of Rail and Construction Technology
Washington DC 20590

13. Type of Report and Period Covered
Final Report
April 1980-October 1980

14. Sponsoring Agency Code
UTD-30

15. Supplementary Notes
*Under Contract to: U.S. Department of Transportation
Research and Special Programs Administration
Transportation Systems Center
Kendall Square, Cambridge MA 02142

16. Abstract

The WMATA rail transit car design adopted the use of disc brakes as the primary friction braking system. Unfortunately, while disc brakes are far more efficienct than the traditional tread brake designs, they are also prone to generate unpleasant squeal noise. The purpose of this study was to: (1) evaluate squeal noise generation at the WMATA transit property, (2) review the mechanisms for squeal noise generation and the options for its control, and (3) identity future research and development needs relative to disc brake systems for rail transit vehicles.

17. Key Words
Noise, rail transportation noise, urban rail transit noise, brake squeal noise, disc brakes

18. Distribution Statement
DOCUMENT IS AVAILABLE TO THE PUBLIC THROUGH THE NATIONAL TECHNICAL INFORMATION SERVICE, SPRINGFIELD, VIRGINIA 22161

19. Security Classif. (of this report)	20. Security Classif. (of this page)	21. No. of Pages	22. Price
Unclassified	Unclassified	49	

Form DOT F 1700.7 (8-72)   Reproduction of completed page authorized

PREFACE

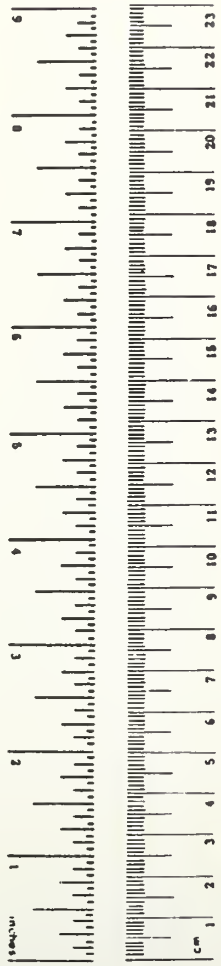
This report was prepared by ORI, Inc. for the Transportation Systems Center under order DTRS 57-80-P-81005. The study was conducted as part of the Urban Rail Noise Abatement Program, managed by the Transportation Systems Center, and sponsored by the Office of Rail and Construction Technology of the Urban Mass Transportation Administration.

As intended, the results of this study are preliminary in nature, and will provide direction to future efforts to reduce squeal noise from the WMATA transit car disc brake system.

METRIC CONVERSION FACTORS

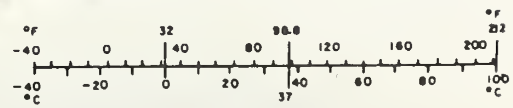
Approximate Conversions to Metric Measures

Symbol	When You Know	Multiply by	To Find	Symbol
		LENGTH		
in	inches	2.5	centimeters	cm
ft	feet	30	centimeters	cm
yd	yards	0.9	meters	m
mi	miles	1.6	kilometers	km
		AREA		
in^2	square inches	6.5	square centimeters	cm^2
ft^2	square feet	0.09	square meters	m^2
yd^2	square yards	0.8	square meters	m^2
mi^2	square miles	2.6	square kilometers	km^2
	acres	0.4	hectares	ha
		MASS (weight)		
oz	ounces	28	grams	g
lb	pounds	0.45	kilograms	kg
	short tons (2000 lb)	0.9	tonnes	t
		VOLUME		
tsp	teaspoons	5	milliliters	ml
Tbsp	tablespoons	15	milliliters	ml
fl oz	fluid ounces	30	milliliters	ml
c	cups	0.24	liters	l
pt	pints	0.47	liters	l
qt	quarts	0.95	liters	l
gal	gallons	3.8	liters	l
ft^3	cubic feet	0.03	cubic meters	m^3
yd^3	cubic yards	0.76	cubic meters	m^3
		TEMPERATURE (exact)		
°F	Fahrenheit temperature	5/9 (after subtracting 32)	Celsius temperature	°C



Approximate Conversions from Metric Measures

Symbol	When You Know	Multiply by	To Find	Symbol
		LENGTH		
mm	millimeters	0.04	inches	in
cm	centimeters	0.4	inches	in
m	meters	3.3	feet	ft
m	meters	1.1	yards	yd
km	kilometers	0.6	miles	mi
		AREA		
cm^2	square centimeters	0.16	square inches	in^2
m^2	square meters	1.2	square yards	yd^2
km^2	square kilometers	0.4	square miles	mi^2
ha	hectares (10,000 m^2)	2.5	acres	
		MASS (weight)		
g	grams	0.035	ounces	oz
kg	kilograms	2.2	pounds	lb
t	tonnes (1000 kg)	1.1	short tons	
		VOLUME		
ml	milliliters	0.03	fluid ounces	fl oz
l	liters	2.1	pints	pt
l	liters	1.06	quarts	qt
l	liters	0.26	gallons	gal
m^3	cubic meters	35	cubic feet	ft^3
m^3	cubic meters	1.3	cubic yards	yd^3
		TEMPERATURE (exact)		
°C	Celsius temperature	9/5 (then add 32)	Fahrenheit temperature	°F



Technical Report Documentation Page

1. Report No.
UMTA-MA-06-0049-81-4

2. Government Accession No.
PB 81-197360

3. Recipient's Catalog No.

4. Title and Subtitle
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5. Report Date
March 1981

6. Performing Organization Code
DTS-331

7. Author(s)
L.A. Ronk, and M.A. Staiano

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DTRS-57-80-P81005

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400 Seventh Street, S.W.
Washington, D. C. 20590

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16. Abstract
This study was conducted as part of the Urban Rail Noise Abatement Program sponsored by the Office of Rail and Construction Technology of the Urban Mass Transportation Administration.

The Washington Metropolitan Area Transit Authority (WMATA) rail transit car design adopted the use of disc brakes as the primary friction braking system. Unfortunately, while disc brakes are more efficient than the traditional tread brake designs, they are also prone to generate unpleasant squeal noise. The purpose of this study was to: 1) inspect the WMATA disc brake assembly; 2) assess disc brake squeal noise occurrence on WMATA vehicles and obtain representative A-weighted sound level measurements; 3) identify and evaluate alternative disc brake squeal; and 4) identify future research and development needs relative to disc brake systems appropriate for use on rail transit vehicles. This report states that disc brake squeal is a chronic and pervasive problem on the WMATA rail transit system. It results in a significant increase in operating sound levels during braking (4 dBA) in a large majority of braking operations (92 percent). Although a noise control treatment has been implemented, its effectiveness has not been clearly documented. Conclusions based on the findings of the investigation are presented in this report, and future research and development needs relative to disc brake systems appropriate for use on rail transit vehicles are identified.

17. Key Words
Rail Transit Vehicles
Urban Rail Noise Abatement Program
WMATA Railcar Noise
Subways
Disc Brake System
Rail Noise
Friction Braking System
Brake Squeal
Research & Development

18. Distribution Statement
Available to the Public through the National Technical Information Service, Springfield, Virginia 22161.

19. Security Classif. (of this report)	20. Security Classif. (of this page)	21. No. of Pages	22. Price
Unclassified	Unclassified	50	A03

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SUMMARY

The WMATA rail transit car design adopted the use of disc brakes as the primary friction braking system. Unfortunately, while disc brakes are far more efficient than the traditional tread brake designs, they are also prone to generate unpleasant squeal noise. The purpose of this study was to: (1) evaluate squeal noise generation at the WMATA transit property, (2) review the mechanisms for squeal noise generation and the options for its control, and (3) identify future research and development needs relative to disc brake systems for rail transit vehicles.

Ninety-two percent of the 143 train arrivals observed at the WMATA Metro Center Subway station during May, 1980 produced noticeable brake squeal. The average maximum sound levels measured before and during brake squeal were, respectively, 87.2 dBA and 91.1 dBA. Average maxima (both before and during squeal) for the side-platformed upper level of the Metro Center subway station were about 2 dBA higher than for the center-platformed lower level. The BART disc brake system, while very similar to WMATA's disc brake system, in terms of physical characteristics, does not exhibit squeal. None of the obvious operational and installation differences explain this disparity.

Contrary to the initial expectation of many investigators, disc brake squeal is not a stick-slip phenomenon. It is generally agreed that disc brake squeal arises "from a geometrically induced or kinematic constraint instability of the elastic system." What this means is that due to the geometry set-up between the piston, the brake pad, and the disc, and their

mechanical stiffnesses, a "digging-in" action of the pad and a slipping release of this action results in a vibratory excitation and, ultimately, airborne noise. Multiple-degree-of-freedom, lummped-parameter analyses have been performed which reasonably predict the gross characteristics of brake squeal. These analyses indicate that the squeal instability region is largely dependent upon the contact point position of the pad with the disc, disc stiffness normal to its plane of rotation, Young's modulus of the pad material, caliper system stiffness normal to the disc, and caliper mass. An analysis which allowed for asymmetries of the contact point positions of the pads on either side of the disc indicated that a strong relationship exists between squeal generation and the relative positions of the contact points.

Squeal noise controls used to date have been developed largely on a "cut-and-try" basis. The primary options which are available to quiet a squealing disc brake system are to increase rotor stiffness or internal damping. In the case of WMATA's brake assembly, the disc rotor is bolted directly to the wheel and some evidence exists that the wheel may be a significant radiating surface. Consequently, in this unique case benefits may be derived from the alteration of the disc/wheel assembly. A recent investigation has implied that the design of asymmetric features in the pads and caliper to either side of the disc may provide a means of reducing the squeal instability region.

A squeal noise control device has been applied to the WMATA Metro rail transit vehicles. This device consists of a thin strip of resilient material mounted on 1/8 inch thick steel plate inserted between the brake pad backplate and the caliper piston. This installation presumably reduces the stiffness of the caliper assembly and is reported to reduce squeal occurrence. However, it has not been effective since it is prone to rapid wear and not easily replaceable.

A number of investigations should be performed to fully define the WMATA squeal problem and develop solutions to prevent it:

(1) Squeal probabilities and spectra should be determined by measurements on at least a full day operation of a single transit car truck in the stock condition and with the current squeal control device.

(2) The resonant frequencies of the disc rotor, the wheel, and the disc/wheel assembly should be determined to help identify the squeal radiation mechanism.

(3) If the above studies indicate the current squeal noise control is fundamentally viable, more suitable applications of this approach should be explored.

(4) Alternative squeal noise controls should also be explored in the form of modifications to the system such as stiffening the disc rotor, decoupling the rotor wheel assembly, or otherwise altering the disc/wheel assembly resonances. A fundamental solution to the brake squeal should be explored by the design and testing of a prototype, high-damping disc rotor.

I. INTRODUCTION

BACKGROUND

The Washington Metropolitan Area Transit Authority (WMATA) Metro rail rapid transit system incorporates a number of noise and vibration design features which greatly reduce noise impacts compared to older transit system operations. In the early design stages of the WMATA Metro rail transit system it was recognized that noise impact would have significant influence on community and patron acceptance of the new transportation system. Because of the importance of controlling system noise and vibration, the WMATA Metro rail transit system has established stringent noise level criteria which are, in many cases, more restrictive than those established by other transportation systems or those mandated by most community noise standards and ordinances. However, in spite of the considerable efforts directed at controlling the acoustical environment, the WMATA Metro rail transit system has been confronted with an unexpected noise source produced by the operation of the Metro trains. This noise source is the intermittent disc brake squeal which occurs during vehicle braking, primarily at speeds of less than 15 mph. This high-pitched brake squeal noise is particularly noticeable in the reverberant subway station environments.

PURPOSE OF STUDY

The purpose of this effort was to: (1) inspect the WMATA disc brake assembly (2) assess disc brake squeal noise occurrence on WMATA vehicles and obtain representative A-weighted sound level measurements, (3) identify and, to the extent possible, evaluate alternative disc brake designs and/or maintenance procedures to control rail vehicle disc brake squeal, and (4) identify future research and development needs relative to disc brake systems appropriate for use on rail transit vehicles.

REPORT OVERVIEW

The findings of this investigation are presented in Sections II, III, IV, and V.

Section II

In this section, an assessment of the WMATA Metro rail transit vehicle disc brake squeal problem is presented. This assessment is based on data collected in the WMATA Metro Center subway station and is described in terms of the frequency of squeal occurrence and the distribution of maximum A-weighted sound levels measured before and during brake squeal. This section also presents a comparison of the spectral characteristics of WMATA Metro train operations with and without disc brake squeal and a comparison of the WMATA Metro and the Bay Area Rapid Transit District (BART) rail transit disc brake systems.

Section III

This section discusses some of the existing theoretical models used to explain and to predict disc brake squeal generation. The results of a number of laboratory and field investigations specifically addressing disc brake squeal generation and radiation are also discussed.

Section IV

Based on the results of these investigations, potential squeal noise reduction methods are identified. Other topics discussed in this section include the following: a description of the brake squeal control device currently used on WMATA Metro rail transit vehicles, and alternative designs and/or maintenance procedures to control brake squeal on WMATA Metro rail transit vehicles.

Section V

Conclusions based on the findings of this investigation are presented in this section and future research and development needs relative to disc brake systems appropriate for use on rail transit vehicles are identified.

II. ASSESSMENT OF WMATA METRO RAIL TRANSIT VEHICLE DISC BRAKE SQUEAL

In general, brake squeal is a probabilistic phenomenon. It occurs during one brake application and not during another and appears to be influenced by numerous parameters. In the following paragraphs the conditions under which squeal occurs will be discussed qualitatively, based largely upon the observations of Spurr (2). Following this discussion, quantitative observations of squeal occurrence on WMATA Metro transit cars, measured during a day of operations, are presented.

CONDITIONS UNDER WHICH BRAKE SQUEAL OCCURS

Squeal occurs with both drum and disc brakes. With both types of brakes, squeal is elusive; it may occur during one application but not on another, though conditions are apparently identical so far as speed, pressure, temperature, and deceleration are concerned. Squeal sometimes occurs over only a part of the revolution of the disc, giving rise to a series of squeaks instead of continual squeal.

The greater the coefficient of friction of the pad, the greater the likelihood of squeal, though a friction material of high coefficient of friction does not necessarily squeal. Squeal can occur if the lining shows "early morning sharpness," i.e., a temporarily high coefficient of friction caused by exposure to humid conditions, and disappears when the "sharpness" disappears. If the coefficient of friction of squealing pads is sufficiently reduced temporarily, squeal will disappear -- to return again at the same coefficient of friction as it disappeared. Squeal is most likely to occur when the brake is cooling down after severe use. It generally occurs at low speeds and, once started, repetitive braking under the same conditions tends to build it up.

Squeal generally occurs at very low pressures, corresponding to decelerations of 0.10 g and less. When the pads squeal in forward stops, they may not squeal in stops in reverse, but may do so if the pads are reversed in the calipers.

The relationships between brake system physical properties and squeal are complex. The greater the coefficient of friction, the greater the likelihood of squeal; but otherwise there is no obvious relationship. Also, squeal has been observed from pads with very high damping.

Squeal frequencies are typically around 1000 Hz for both drum and disc brakes, and besides the main frequency there are generally two or three much weaker harmonics. The frequency of squeal can sometimes change from one value to another. Observations have shown that squeal frequency occasionally changes in a continuous manner during the brake application.

BRAKE SQUEAL OCCURRENCE AND SOUND LEVEL DISTRIBUTION ON WMATA TRANSIT CARS

The Metro Center subway station was selected as the location to assess the frequency of disc brake squeal occurrence and to perform in-station sound level measurements. The Metro Center station is a two-level, three-line station with two tracks at each level. The "Red" line operates on the upper level and the "Blue" and "Orange" lines operate on the lower level.* The upper level of the Metro Center station is a side platform configuration; the lower level is a center platform configuration. The tracks on both the upper and lower levels of the Metro Center station are supported by floating slab trackbeds.

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*The "Red," "Blue," and "Orange" lines are the only lines currently operating in the WMATA Metro rail transit system. Thus, all trains pass through the Metro Center station during normal line operations.

Occurrence of Disc Brake Squeal

During the assessment period, May 1980, 143 train operations (i.e., trains braking to a stop) were subjectively evaluated to determine the probability of disc brake squeal occurrence. Because of the pure tone or screech character of disc brake squeal noise, trains with and without brake squeal were easily discernible.

Of the 143 train operations observed, approximately 92 percent produced noticeable brake squeal. However, since all of the trains observed were comprised of six or eight cars, it was not possible to determine which car or cars generated the squeal noise.

Sound Level Distribution

In addition to subjectively evaluating the noise from the train operations, sound level measurements were performed in the upper and lower levels of the Metro Center station using a B&K 2209 impulse precision sound level meter, with a 1-inch free field condenser microphone (B&K 4145), extension rod, and foam wind screen. The measurements were taken using the "Impulse Hold" sound level meter circuitry to obtain maximum RMS A-weighted sound levels during train arrivals. All sound level measurements were made at a distance of six feet from the platform edge and five feet above the platform surface. In the lower level of the station, sound level measurements were performed at both ends of the station at a distance of 100 feet from the station entrance portal. In the upper level, sound level measurements were also performed at both ends of the station; however, because of an obstruction at 100 feet, measurements were made at a distance of 50 feet from the station entrance portal.

A typical time history of the A-weighted sound level of a WMATA Metro train operation during entry into a subway station is shown in Figure 2.1. It can be seen from Figure 2.1 that the sound levels produced by brake squeal at a location approximately 100 feet from the station portal entrance are typically on the order of 4 dB above the sound levels produced by normal train operations.

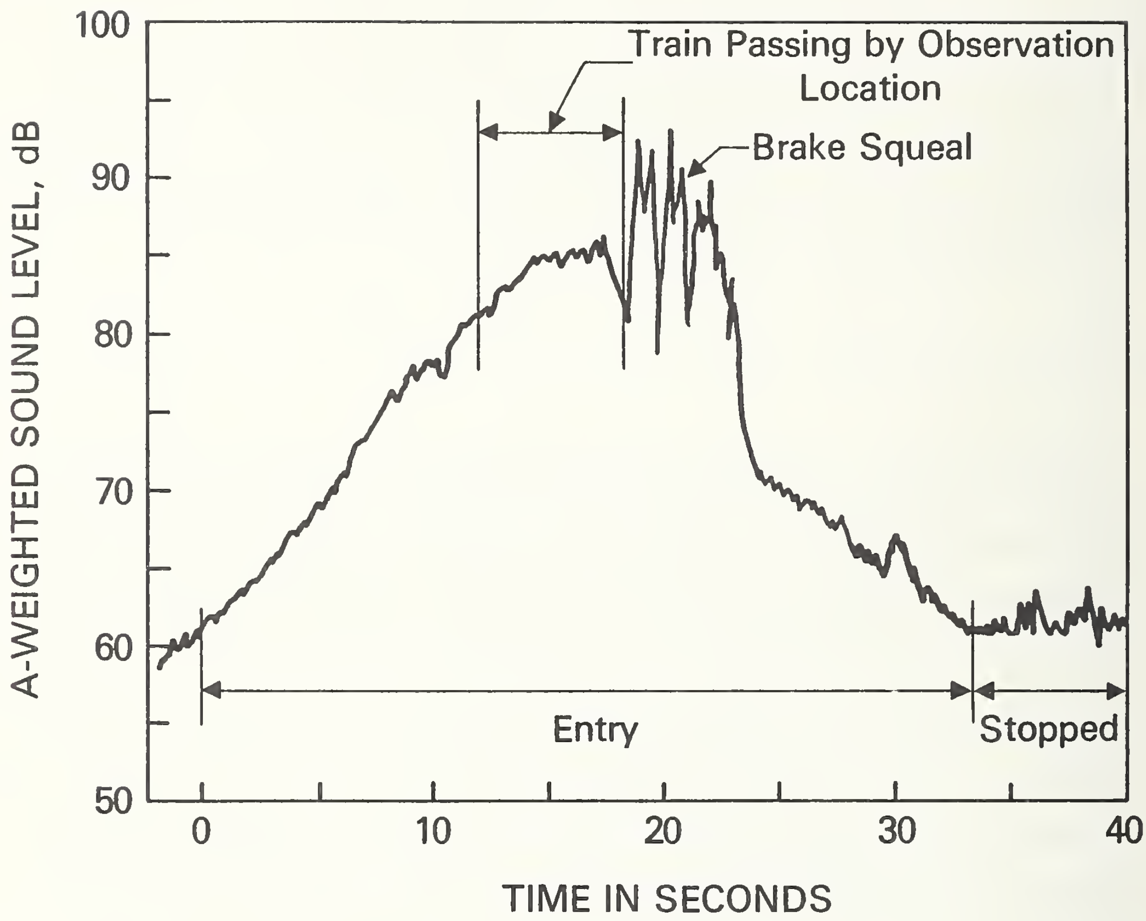


FIGURE 2.1. TYPICAL TIME HISTORY OF SOUND LEVEL OF A WMATA METRO TRAIN DURING ENTRY INTO A STATION FOR A LOCATION APPROXIMATELY 100 FT. FROM THE ENTRY END OF STATION

(From G.P. Wilson, Reference 1)

It should be noted that the occurrence of brake squeal noise did not always result in an increase in the sound level meter reading. That is, often brake squeal was perceived but the meter indication did not increase. Two possible explanations for this occurrence are:

- Due to other sources of noise, the brake squeal sound level was not greater than the maximum sound levels occurring at other times during the train's arrival.

- The brake squeal sound level did not dominate the train's A-weighted sound level although it was readily perceivable due to its characteristic pure tone content.

The purpose of performing the sound level measurements was to obtain typical levels for train operations with and without disc brake squeal. However, because of the high rate of brake squeal occurrence, a limited sample of sound levels without brake squeal was obtained. The brake squeal sound levels reported herein are only those for which the maximum sound level increased during squeal occurrence. These data were obtained by measuring the maximum sound level before and after the occurrence of brake squeal.

Table 2.1 presents a statistical summary of the measured A-weighted sound level data obtained in the lower and upper levels of the Metro Center subway station. From Table 2.1 it can be seen that, on the average, the sound levels measured in the upper level are approximately 2 dBA higher than those measured in the lower level. It should be noted that the 2 dBA difference between the upper level and the lower level average sound levels is observed for train operations before and during brake squeal. This suggests that the difference between the average maximum sound levels for train operations with brake squeal and those without brake squeal is unaffected by the station platform configuration. From the data shown on Table 2.1, it can be seen that the increase in station sound levels resulting from Metro train brake squeal is approximately 4 dBA. Wilson (1) reports that brake squeal sound levels produced by Metro trains during entry into the subway stations are typically

TABLE 2.1

ANALYSIS OF MAXIMUM A-WEIGHTED SOUND LEVELS OF WMATA METRO TRAINS DURING ENTRY INTO THE METRO CENTER SUBWAY STATION LOWER AND UPPER LEVEL LOCATIONS

		Train Operations During Brake Squeal			Train Operations Before Brake Squeal			
Measurement Location	Number of Sound Level Measurements	Range of Sound Levels dBA	Average* of Sound Levels dBA	Standard Deviation dBA	Range of Sound Levels dBA	Average* of Sound Levels dBA	Standard Deviation dBA	Difference Between Average Sound Levels (Squeal)-(No Squeal) dBA
Lower Level	17	87-94	90.3	1.8	84-90	86.2	1.6	4.1
Upper Level	12	88-95	92.3	2.3	85-90	88.5	1.8	3.8
Lower and Upper Levels	29	87-95	91.1	2.2	84-90	87.2	2.0	3.9

* - Arithmetic average

90 to 95 dBA and occasionally result in sound levels on the platforms as high as 100 dBA. Wilson also reports that in the absence of brake squeal noise, the typical sound levels for Metro train operations during station entry are 85 dBA or less, except for short time periods at locations near the ends of the station platforms. At the end of the station platform, the maximum levels are in the range of 84 to 90 dBA, depending on station configuration and train speed at the time of observation.

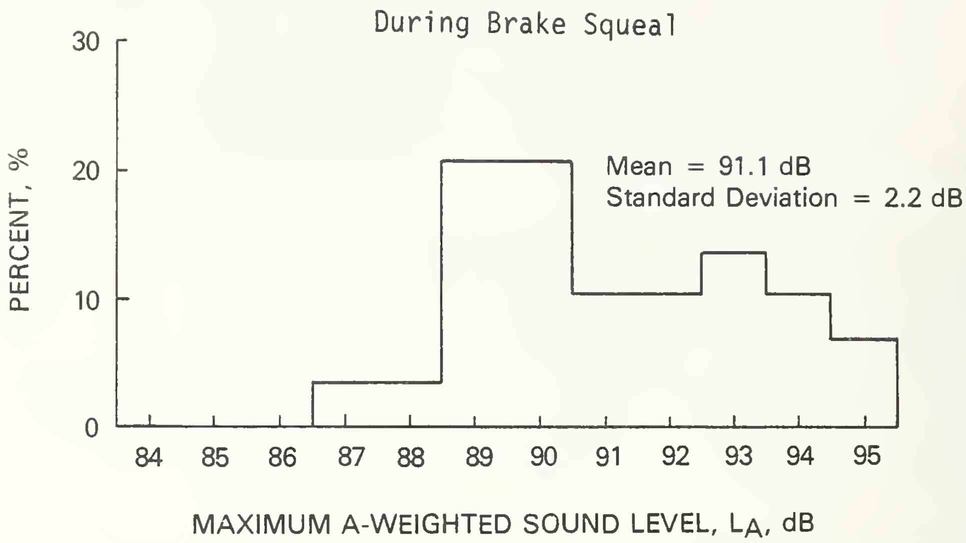
Figure 2.2 presents a distribution of the measured A-weighted sound levels obtained in the lower and upper levels of the Metro Center subway station. Lower and upper level sound level data are combined to show a representative distribution of levels before and during brake squeal for both center (lower level) and side (upper level) platform configurations.

Octave band sound pressure level spectra for Metro train operations with and without brake squeal are presented in Figure 2.3.(1) It should be noted that the sound level measurements were made at the center of the station platform and that the difference between the maximum A-weighted level with and without brake squeal is approximately 11 dB. In addition, Figure 2.3 shows that the Metro train brake squeal noise is concentrated in the octave band centered at 500 Hz.

COMPARISON OF WMATA AND BART BRAKE SYSTEMS

At this time, WMATA and the Bay Area Rapid Transit District (BART) are the only domestic rail rapid transit systems which use disc brakes as the primary friction braking system. The physical characteristics of these systems are tabulated in Table 2.2. However, the BART maintenance and engineering personnel have reported that the BART rail transit vehicles do not generate disc brake squeal noise.(16)

Both WMATA and BART rail transit vehicles use dynamic braking systems as the principal means of deceleration, and use the friction braking for normal and emergency stopping. Normal friction braking is performed at train speeds of approximately 15 mph or less. Unlike BART, the WMATA Metro trains



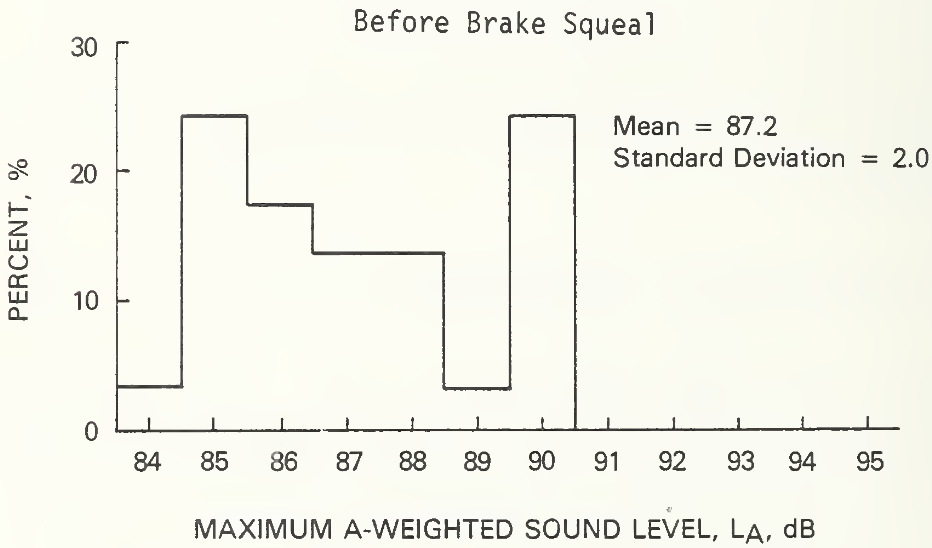


FIGURE 2.2. DISTRIBUTION OF A-WEIGHTED SOUND LEVELS OF WMATA METRO TRAINS DURING ENTRY INTO SUBWAY STATIONS

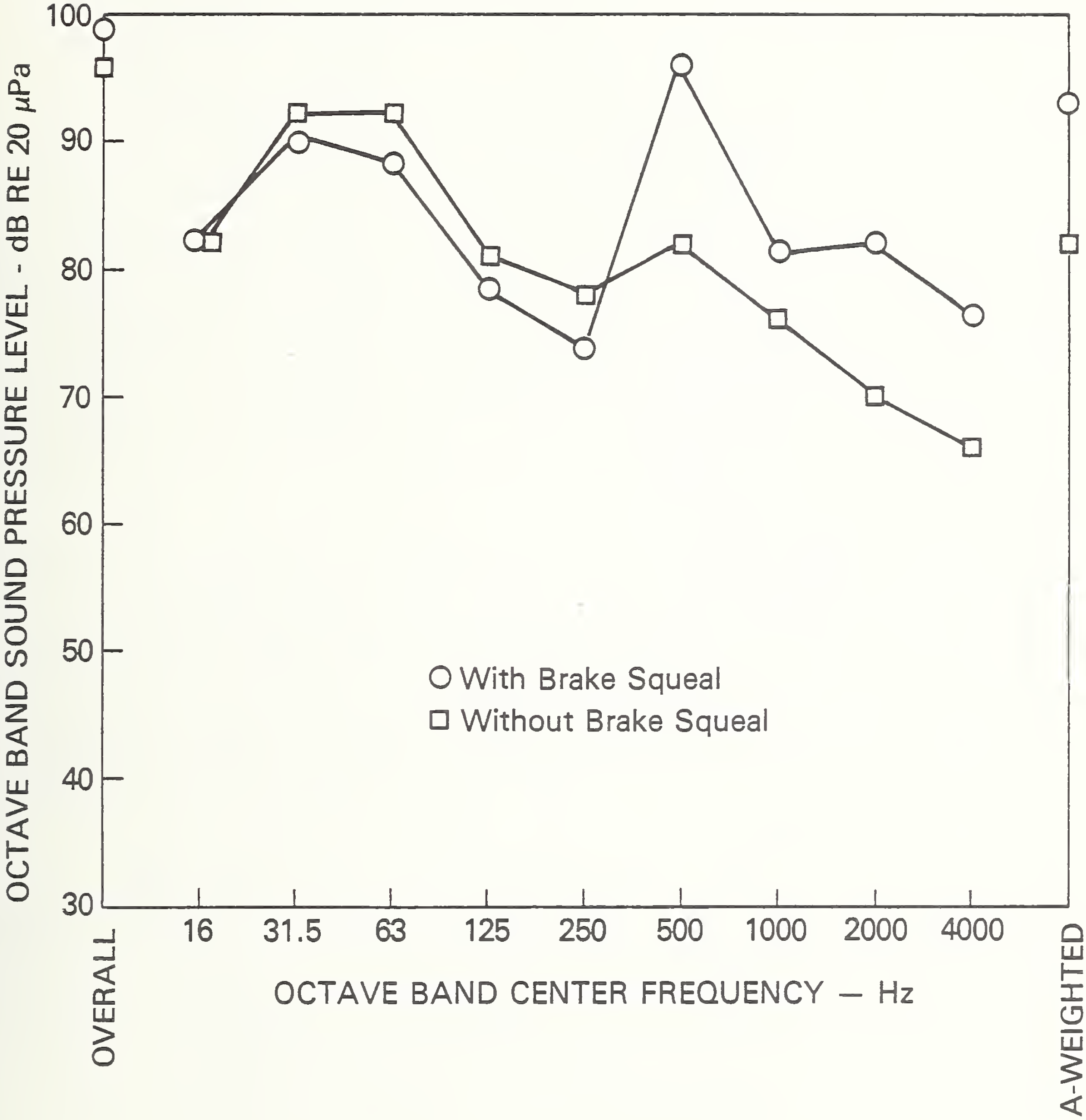


FIGURE 2.3. AVERAGE MAXIMUM SOUND LEVELS AT CENTER OF STATION LOCATIONS; 4-CAR METRO TRAINS

(From G.P. Wilson, Reference 1)

TABLE 2.2
DISC BRAKE SYSTEM CHARACTERISTICS[a]

Component	Characteristic	WMATA	BART
Friction Disc	Diameter (in.)	20.3	23.5
	Thickness (in.)		
	- overall	3.6	3.5
	- braking surface[b]	0.4	0.6
	Weight (lb.)	136.5	143.0
Friction Pad	Nominal Area ($in.^2$)	59.0	61.0
	Thickness -- unworn (in.)	0.75	0.6
	Coefficient of Friction	0.3[c]	0.36[c]
	Weight (lb.)	5.75	4.5[d]
Caliper	Weight (lb.)	75[e]	70[e]

a Data presented in this table were obtained from the following sources:

- Crist, R.C., BART, Personal communication with L. Ronk, May 26, 27, 1980.
- Straut, J. E., Abex Corporation, Personal communication with L. Ronk, May 29, 1980.
- Rusynko, S., Knorr Brake Corporation; Personal communication with L. Ronk, May 30, 1980.
- Bassily, F. P., WMATA, Personal communication with L. Ronk, June 13, 1980.

b Annular plate upon which the friction pad acts.

c Representative average over the range of operating temperatures.

d Lining weight only. Lining with anvil backing plate is 6.5 lbs., lining with piston backing plate is 14.5 lbs.

e "Dry" weight with brake pads.

also use the friction braking system for speed adjustments during normal running. The reduced braking requirements of the BART system may be a factor in its squeal-free performance.

The overall design of the disc brake systems used on the WMATA and the BART rail transit vehicles is different in many respects. A major difference between the two disc brake system designs is the location of the friction disc and caliper assemblies relative to the truck wheel. WMATA uses two brakes per axle. As seen from Figure 2.4, the WMATA friction disc and caliper assemblies are positioned outboard of and bolted to the truck wheel. On the other hand, BART uses a single brake assembly per axle. From Figure 2.5, it can be seen that the BART friction disc and caliper assemblies are positioned on the inside of the truck wheel.

The friction discs used with the WMATA disc brake assemblies are one-piece units which are attached directly to each truck wheel. The friction discs used with the BART disc brake assemblies are two-piece units (split disc configuration) which are attached to each truck axle.* Both the WMATA and the BART friction discs have a ribbed-core design for heat dissipation during braking operations. The specifications of the WMATA and the BART systems are comparable as can be seen from the Table 2.2.

The friction pads originally used with the WMATA disc brake assemblies were composed of an iron-lead-asbestos composite material.(17) However, the primary supplier of the iron-lead-asbestos pads (ABEX Corp.) has discontinued production of this type of friction pad. WMATA is currently exhausting its supply of the iron-lead-asbestos pads and is replacing them

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* The friction disc is divided into two parts along a diameter to allow attachment to the truck axle.

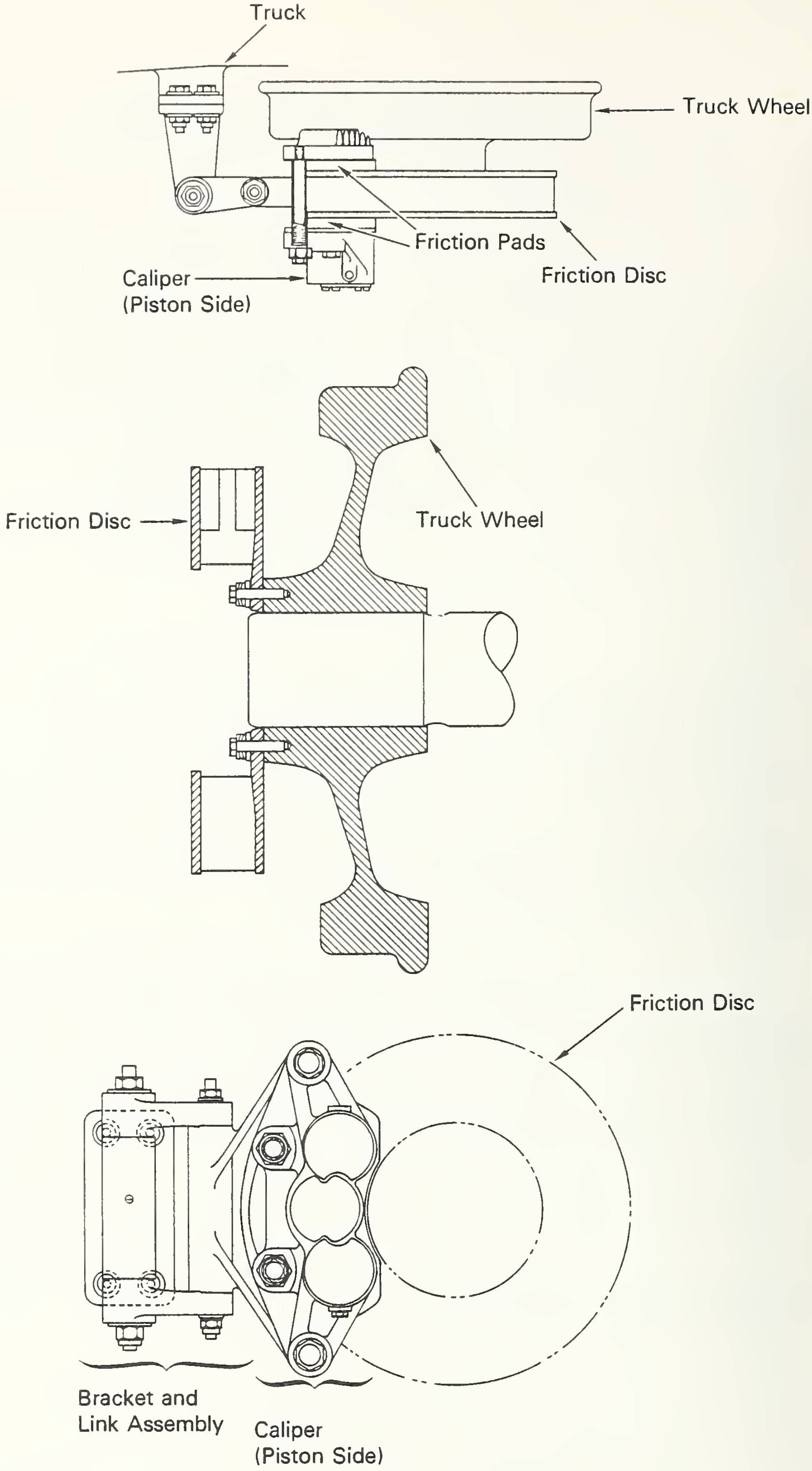

FIGURE 2.4. WMATA RAIL TRANSIT VEHICLE DISC BRAKE ASSEMBLY

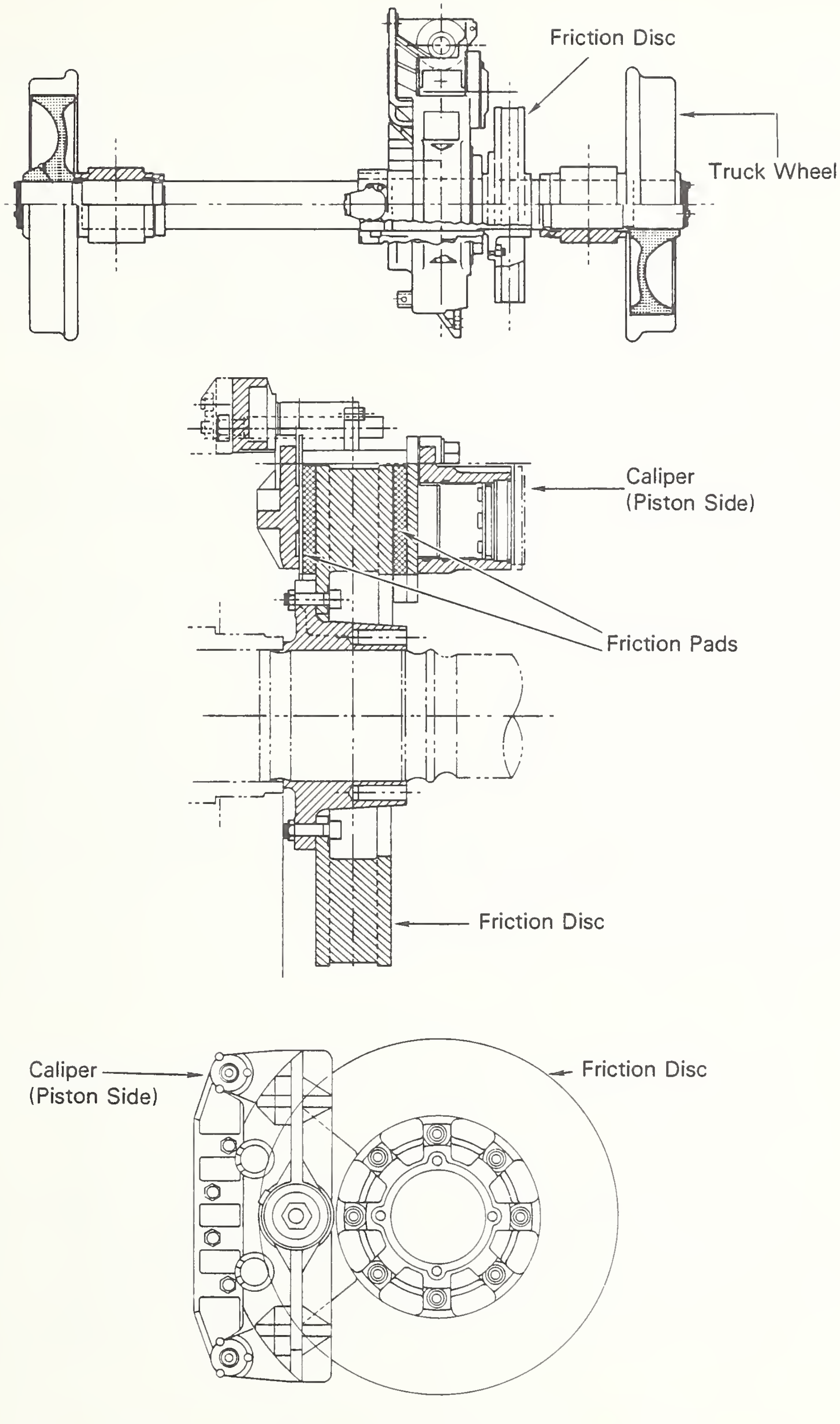

FIGURE 2.5. BART RAIL TRANSIT VEHICLE DISC BRAKE ASSEMBLY

with asbestos-free pads. Both types of pads are believed to be in service at the date of this report. Preliminary indications are that the new asbestos-free pads do not reduce brake squeal occurrence.

The friction pads used with the BART disc brake assemblies are composed of a semi-metallic material (SMD-2) manufactured by Molded Materials Company, a Division of Carlisle Corp.[16] A more detailed description of the components of the BART friction pads could not be identified.*

Comparison of the gross characteristics of the WMATA and BART systems does not readily explain why WMATA experiences squeal and BART does not. BART's in-board brake installation (vis-a-vis WMATA's installation outside the wheels) would tend to reduce squeal sound levels if they existed -- by imposing a more tortuous propagation path -- but would probably not make them imperceptible. The use of the friction brakes on WMATA at speeds greater than 15 mph for speed adjustments may increase the thermal distortion of the pad and, as a result, increase squeal generation. Finally, the BART split-disc configuration will significantly increase the internal damping of the bolted-up friction disc and may have the effect of shrinking the squeal instability range to outside BART's normal operational parameters. (However, an early generation, single-piece disc configuration did not exhibit squeal either.)

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*Semi-metallic friction materials are composed of "chopped" steel wires, a phenolic resin binder, plus a mix of proprietary ingredients.

III. DISC BRAKE SQUEAL GENERATION AND RADIATION

SQUEAL GENERATION MECHANISMS

Initially, investigators assumed disc brake squeal was explained by the effects of stick-slip or a variable coefficient of friction between the brake pad and the disc. However, these hypotheses failed to explain many of the observed characteristics of squeal. Spurr (2) first proposed the "sprag-slip" mechanism in which squeal is fundamentally related to the geometric configuration of the disc and caliper and their mechanical stiffnesses and is independent of variations in the coefficient of friction. With some modification the basic sprag-slip theory has been supported and extended (3,4). Consequently, disc brake squeal is generally agreed to arise "from a geometrically induced or kinematic constraint instability of the elastic system" (5,6).

The sprag-slip theory is unlike the stick-slip theory in that it does not depend on a variation in the coefficient of friction with sliding speed. The sprag-slip theory recognizes that due to the configuration and flexibility of the system, the normal force -- thus, the frictional force -- between the contacting surfaces can vary, thus resulting in variations in the relative sliding speed. Spragging is essentially a "digging-in" action of the pad element and a slipping release of this action. It is a self-induced periodic change which can produce a vibratory excitation of the system resulting in airborne noise.

The occurrence of spragging can be understood by a simple experiment. Hold a soft block eraser between the thumb and forefinger. Rub the eraser from right to left along the surface of a writing tablet such that it is free to rotate in a counterclockwise direction about the pivot formed by the thumb and forefinger. If the rotational axis of the eraser formed by the fingertips is directly above the point of contact of the eraser with the paper surface, the movement of the eraser should be smooth and free of vibration. This should also be true if the eraser rotational axis leads the point of contact (i.e., the top of the eraser is inclined in the direction of movement). Now, incline the eraser slightly such that the rotational axis follows the point of contact. Movement along the paper should now induce an oscillation of the eraser about the rotational axis -- spragging. (If oscillation does not occur, vary the pressure of the eraser on the paper, its inclination angle, or the surface on which it is rubbed.) Gradually increase the eraser's inclination such that the contact point leads the rotational axis more. An angle of inclination should be observed beyond which the spragging oscillation will cease. This angle, relative to the normal to the rubbed surface, can be shown to be equal to the arctan μ where μ is the coefficient of friction between the eraser and the surface.

Spragging can occur in real brake systems because the true contact area between the brake pad and the disc is only a very small proportion of the nominal pad area unless the pressure is very high. Circumferential locations (with respect to the disc) of the contact point between the pad surface and the disc and between the piston and the brake pad backplate can arise which form a squeal configuration, as illustrated in Figure 3.1. The effects of pad warpage due to thermal distortion will move the pad/disc contact point with virtually every brake application. This variation in pad geometry may account for the observed randomness of squeal generation.

Displacement of the pad/disc contact point along the radius of the disc can also be of significance in affecting the quality of the squeal noise. Earles and Soar (4) explored the effect of disc stiffness by varying

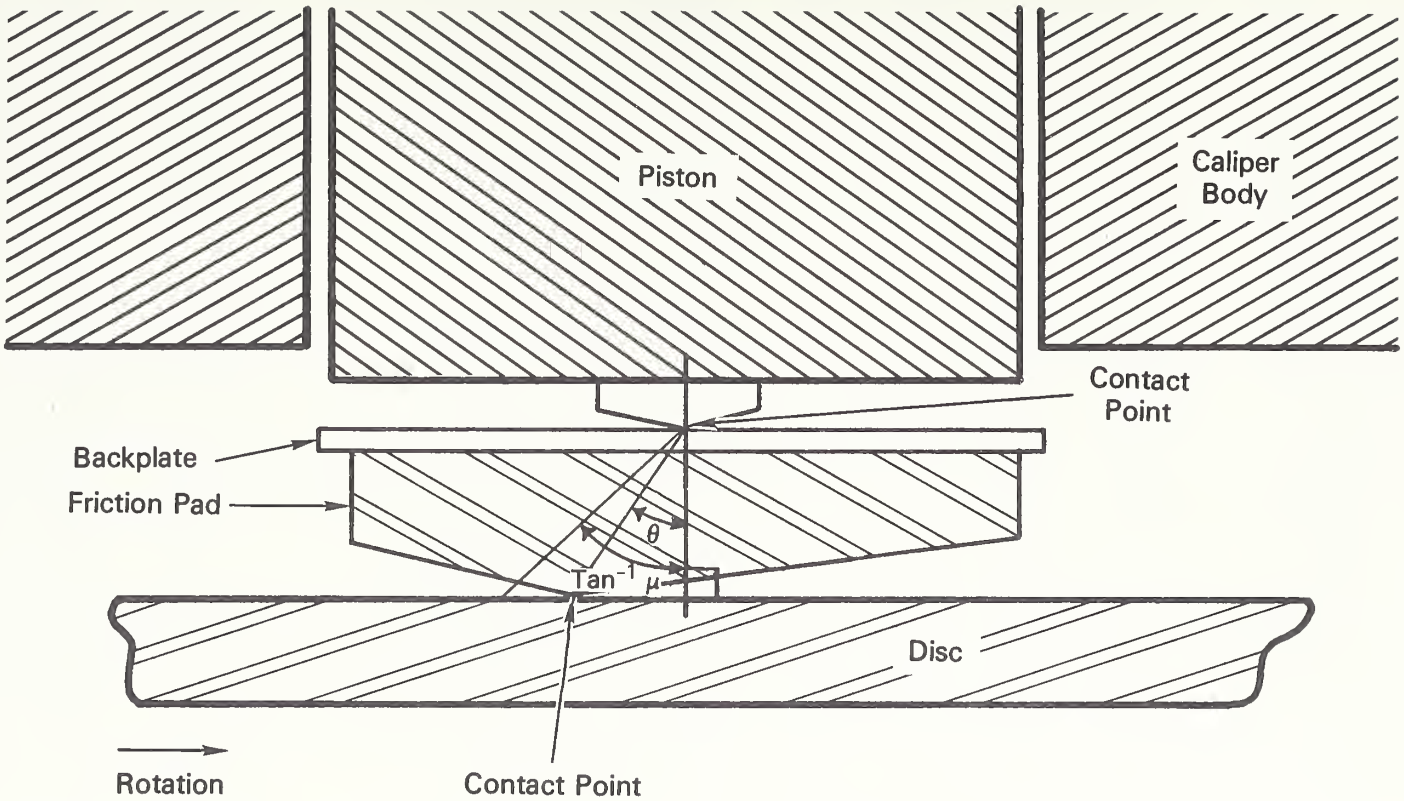

FIGURE 3.1. SCHEMATIC OF DISC/CALIPER SYSTEM (IN SQUEAL CONFIGURATION)

disc contact radius. They found the fundamental resonant frequency of the coupled pin/disc system* to be a function of contact radius.

Within the regime in which squeal can occurr ($\mu \geq .25$), squeal noise is essentially independent of the magnitude of μ. However, with increasing μ, the range of inclination angles for which squeal can occur, $0 < \theta < \tan^{-1} \mu$, increases. This accounts for the statement that "a squealing brake is an efficient unit." This also accounts for squeal caused by temporarily high μ due to exposure to humid conditions.

A number of observers and experimenters have found that, while squeal generally occurs at relatively low brake actuating pressures, <u>within the squeal regime</u> squeal is largely independent of load. Apparently, higher brake pressures compress the brake pad to reduce pad warpage, thus θ decreases. Also, since high brake pressures are likely to be used at high vehicle speeds (and disc speeds), both the dynamic μ and the squeal regime will be smaller.

In a recent ORE investigation (9), experimenters found that a grooved pad squealed while a smooth pad did not. Further, the addition of more grooves on an already squealing pad increased the duration of squeal over the braking process and broadened the squeal frequency range. While ORE offered no explanation of the phenomena, it may be explained by the sprag-slip theory. That is, the grooving of the brake pad creates a number of small "brake pads" each capable of establishing its own contact point. The increased number of "pads" increases the probability of one or more assuming a squealing configuration.

PREDICTION OF SQUEAL

A number of theoretical models using lumped-parameter systems have been proposed. Essentially all of these models assumed a single-sided disc

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* In Earles' (3,4,6,7,8) experimental work, he represented the brake pad using a steel pin ($\mu \cong 0.3$). His results were essentially replicated when pins made from brake pad material were used and were similar to experience with real disc brakes.

brake (i.e., they considered only one brake pad and assumed symmetry) with one significant exception -- to be discussed latter. These models ranged in complexity from:

- A damped single-degree-of-freedom system consisting of the disc and a pin (representing the pad and caliper) in which system vibrations in translational and torsional modes were considered individually (4), to

- An undamped six-degree-of-freedom system consisting of the disc, pad, and caliper (5), to

- A damped eight-degree-of-freedom system consisting of the disc, both pads, and caliper (10).*

The essential requirement for all these models was that they describe the torsional instability of the pad which arises from spragging.

A four-degree-of-freedom model is illustrative of the essential interactions which give rise to squeal (6,8). The disc subsystem is assumed to vibrate in a single translational mode normal to the plane of the disc surface. The pad and caliper are described as a pin having three degrees of freedom: translations parallel (in X-direction) and normal (in Y-direction) to the disc surface and rotations, θ, about a central Z-axis. Damping is ignored and the pin and the disc are assumed to remain in contact at all times. Assuming small oscillations, solutions are in the form

$$x = X e^{\lambda t}, \ y = Y e^{\lambda t}, \ \theta = \Theta e^{\lambda t},$$

and a frequency equation in the form

$$a\lambda^4 + b\lambda^2 + c = 0 \tag{1}$$

will be obtained. The coefficients a, b, and c are functions of the pin

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* Although both pads were considered an evaluation of the effects of contact configurations of the disc, pad, and piston and the effects of asymmetries of parameter values was not presented.

pivot/contact point geometry, the pin and disc subsystem mechanical stiffnesses and masses, and the moment of inertia of the pin subsystem about the Z-axis. Noting that x, y, or θ can grow without bound for $\lambda > 0$, it can be shown that for stability it is required that

$$a > 0, \ b > 0, \ c > 0, \text{ and } b^2 > 4ac.$$

Thus for certain magnitudes of a, b, and c the system is unstable and squeal generation is expected.

Investigators have assumed values for the parameters in equation (1) to explore regions of instability (5,6,8). They found that the occurrence of squeal instability is a strong function of:

- Contact point position of pad with disc (i.e., inclination angle, θ)
- Disc stiffness normal to its plane of rotation
- Young's modulus of pad material
- Caliper system stiffness normal to the disc
- Caliper mass.

An example of the results of this approach is provided in Figure 3.2a from Earles and Badi (7). In Figure 3.2a the shaded region denotes the region of squeal instability as a function of inclination angle, θ, and disc stiffness, k_d, for a "single pin," i.e., a one-sided system. This result agrees reasonably well with experimentally observed behavior (using a pin in lieu of a pad).

Earles and Badi applied their single-pin approach to the more realistic two-pin system with interesting results. For a single-pin-disc system a necessary (but not sufficient) condition for unstable motion to occur is that θ be negative (i.e., disc/pad contact point leads pad/piston pivot point), $0 < /\theta/ < \tan^{-1}\mu$. On the other hand, for unequal pin inclination angles, $\theta_1 \neq \theta_2$, the unstable region is either expanded or reduced depending upon the relative values of θ_1 and θ_2, as shown in Figures 3.2b, c, and d.

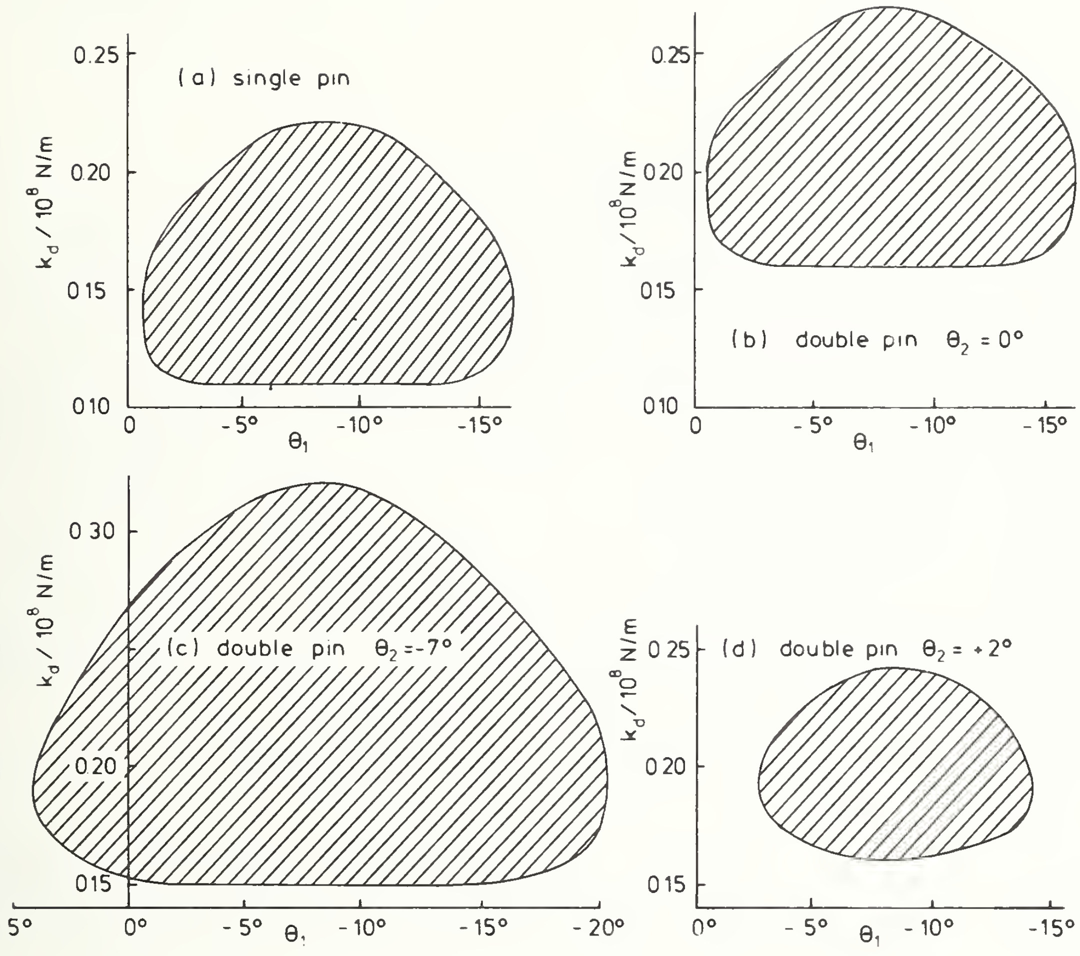

FIGURE 3.2. THEORETICAL OSCILLATORY UNSTABLE REGIONS

In spite of the greatly increased understanding of the squeal mechanism, analytical models are apparently rarely used in practice. For example, General Motors Corporation performs its initial evaluation of its automotive disc brake systems by measuring sound levels on prototype brake systems [19].

OBSERVED SOUND LEVELS AND SPECTRA

A number of investigators have measured disc brake squeal sound levels using a variety of test devices. These test devices were transit car and automobile brake assemblies and a purely experimental, single-pin-disc test rig. The results of these investigations are summarized in Table 3.1. Despite the differences in test procedures, the results from the various test devices appeared to be generally representative. For example, Earles' observed frequency range of 2 to 3 KHz using a pin-disc system compared very closely with the 1.6 to 3.15 KHz range observed by ORE. As discussed earlier, Wilson (1) measured a WMATA squeal spectrum with a maximum in the 0.5 KHz octave which is relatively low compared to these other findings. This result, however, correlated with investigations which indicated the natural frequencies of the brake disc and the wheel were both approximately 500 Hz[18].

Earles and Soar (3) found the fundamental resonant frequencies varied as a function of disc contact radius and disc thickness. The relative amplitudes of the harmonics varied in "some undefinable manner depending on the particular configuration being used."

SQUEAL RADIATION MECHANISMS

In general, squeal may be radiated by one of four possible surfaces:

- Brake disc
- Caliper assembly
- Structure to which the disc is attached
- Structure to which the caliper is attached.

TABLE 3.1
BRAKE SQUEAL SOUND LEVELS

Investigators	Test Device	Microphone Distance from Disc (m)	Overall Sound Level (dBA)	Squeal Frequency Range (KHz)	Observed Pure Tones: Amplitude of Fundamental (dB)	Observed Pure Tones: Frequencies (KHz)	Coefficient of Friction	Comments
ORE (1977)	Transit Car Brake Assembly on Test Rig	0.3	110-116*	1.6-3.15	—	— — —	—	* Mean L_{eq} (during braking) for all observations with or without squeal. Ferodo EP 2262 Grooved Pads
Earles & Soar (1971)	Pin-Disc Test Rig	0.3	—	2-5 (Typical)	90*	1.12, 2.24, 3.36, 4.48, 5.60, 6.72, 7.84, 8.96, 11.2	.29	* Disc Contact Radius = 82mm Disc: 203mm diameter, 1.8-5.1mm thick, steel Pin: 6mm long, 6mm diameter, steel
Earles (1977)	Pin-Disc Test Rig	—	—	2.3	—	— — —	.61	Squeal 20dB Higher than No Squeal Disc Contact Radius = 80mm Disc: 200mm diameter, 2.3mm thick, steel Pin: 5mm long, steel
Felske, et al., (1978)	Automobile Brake Assembly on Test Rig	—	—	—	120 120 110 105 100 100 110	2.5, 5.0, 7.4 3.1, 3.8, 6.4, 6.9, 10.0 3.2, 6.4, 9.7 2.0 5.0, 10.0 9.8 3.2, 6.4, 9.6*	—	* Pad with "Rain Groove"

Investigations by ORE (9) found that: "Both in the brake pad holders and in the brake discs very high vibration levels were measured. It was not possible to clarify which of the two parts initiates the vibration or which part sooner leads to airborne noise. On the basis of the results obtained from the damped brake pad holders, it can only be assumed that the brake pad holders cause the exitation of this noise."

On the other hand, Flaim's investigations[19] of automotive disc brake assemblies have indicated that the squeal frequency roughly correlates to the disc resonances with "a linear shift of a few hundred hertz" and that the disc is "the right size to be an efficient radiator." He expected the rest of the vehicle suspension to provide relatively little radiation since it is fairly well isolated from the brake assembly.

Investigators have focused upon the disc as the primary radiating surface. However, prediction of the squeal noise frequencies to be expected has proven difficult. Earles and Soar (4) found "The fundamental frequency generated by the coupled system was never found to coincide with the resonant frequencies of either subsystem pin or disc and it was determined ... that the harmonic generations were independent of any higher natural frequencies within the system." The same investigators later found (3) that in a single-pin-disc system, the coupled resonant frequencies were near the anti-resonances of the free disc. The translational (non-squealing) pin system mode appeared to excite a (2,0)* disc mode while the torsional (squealing) pin system produced a (3,0) disc mode. In their experimental set up, the single-pin-disc coupled resonant frequency exhibited approximately the same relationship to the free disc resonant frequency, as was observed by Flaim.

However, squeal excitation appears capable of exciting a variety of modes such as was found by Felske, et. al. (13). Furthermore, ORE (9) had no success in "detuning" a ventilated transit car disc (either by asymmetrical cooling ribs or by different thickness friction disc surfaces). Finally, the WMATA squeal frequencies observed by Wilson (1) correlated with the natural frequency observed for the assembly of the brake disc bolted to the wheel -- approximately 500 hertz (18).

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* (d,c) Is the modal form of the disc, where d is the number of the modal diameters and c is the number of modal circles.

IV. DISC BRAKE SQUEAL NOISE REDUCTION

SQUEAL NOISE CONTROL DESIGN

Experimental work by General Motors on automotive disc brake assemblies show that massive installations were relatively unlikely to squeal. When squealing was experienced, the primary alternatives were to stiffen the rotor or increase the rotor damping (19). Automotive weight reduction programs have resulted in lighter weight brake assemblies and have forced the use of high-damping, grey iron disc rotors. This work was an application of Miller (11), who found that the occurrence of brake squeal was reduced by the use of high-damping materials (those with greater than or equal to 0.20 percent critical damping). A disadvantage of high damping materials is that they are generally of lower strength.

Rotor stiffening can increase the frequency of the disc modes out of the audible range. Its disadvantages are increased weight of the rotor and increased heat propagation into the wheel bearings. Relatively little benefit was found in stiffening the caliper bridge because of the high compliance of the friction materials.

Squeal noise generation models indicate that alteration of the system stiffnesses and caliper mass offer the greatest potential for disc brake squeal noise reduction. Unfortunately these lumped-parameter models can only suggest trends for what is clearly a distributed system. Earles and Lee (8) questioned whether stiffnesses could be altered sufficiently to move out of the instability region. Miller's (11) high-damping disc and ORE's (9) damped brake pad holder suggest increasing system damping has potential for reducing the propensity for squeal. Unfortunately the existing models which address contact point geometry changes have not included damping. Consequently, little insight exists regarding this expected behavior, although North's (10) model indicated increasing damping would reduce the area of instability. Automotive experience has been that squeal generation is unpredictable and, when problems are encountered, experimental methods ("cut and try") have been used to find solutions (12).

The recent modeling work using two-pin-disc systems suggests some possibilities for noise reduction if the experimentally observed behavior is found in real systems and if the contact point inclination angles can be controlled. Figure 4.1 illustrates the instability area as a function of Θ_1, and Θ_2. If Θ_1 or Θ_2 can be kept greater than approximately 4°, the squeal region is avoided. This may be realizable for transit car disc brake systems, which see equal use in both directions, by asymmetrically eccentric loading of the brake pads. Also asymmetries of disc contact point radius and system stiffnesses may also provide similar benefits.

SQUEAL CONTROL CURRENTLY USED BY WMATA

A squeal noise control which apparently altered the brake system dynamics was implemented. The noise control treatment consisted of a 1/8 in. thick steel plate to which a thin (0.011 in.) strip of resilient material (Connecticut Hard Rubber Co. Fabric No. 1611 -- a silicone rubber-coated fiberglass fabric) is bonded using GE RTV-106 Silicone Rubber (14, 18, 20). This assembly is inserted between the friction pad backplate and caliper piston after removing a spacer pack from the brake assembly.

The squeal noise control was installed on transit cars and subjectively evaluated before and after installation and was reported by WMATA to "definitely work" (21). Furthermore, Abex indicated that subjective tests showed the insertion of the steel plate to be equally effective at normal braking temperatures with or without the bonded resilient material (18).

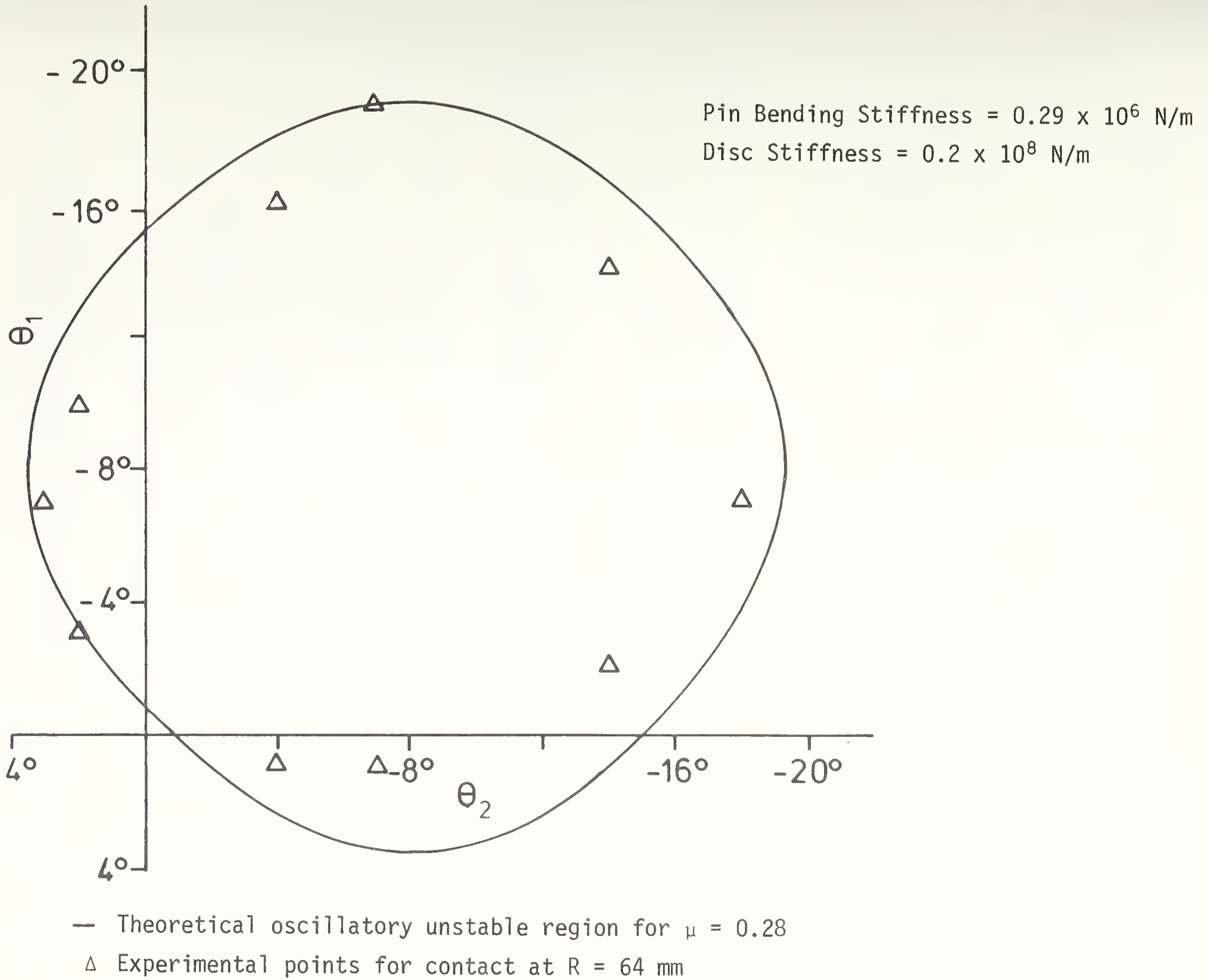

FIGURE 4.1. OSCILLATORY UNSTABLE REGIONS FOR TWO-PIN-DISC SYSTEM

Unfortunately, WMATA reported that the resilient material wears out quickly and must be replaced periodically (14). Furthermore, the installation of the resilient material assembly created several maintenance problems:

- First, insertion of the assembly reduces the clearance between the caliper piston and the friction disc. When the friction pad thickness is on the high side of its manufacturing tolerance, this installation becomes a difficult and time-consuming process.

- Second, insertion of this assembly requires removal of both captive screws,* a procedure which is also time-consuming and not necessary for friction pad replacement. (As can be seen from Figure 4.2, the front and the rear friction pad assemblies can be replaced by removing only one of the two captive screws.)

- Third, the resilient material wear rates have been such that the material should be replaced when new friction pads are installed. (Installation of new pads is typically performed at night in the maintenance facility yard.)

Thus, the resilient material is generally replaced only when the brakes are completely rebuilt and not during routine friction pad replacements (15).

Inspite of the preliminary test findings, the squeal noise control effectiveness appears to be nullified when the resilient material is destroyed. Consequently, with the rapid wear and infrequent replacement of the resilient material, the treatment is not a practical or effective squeal noise control device. (The bonding of the resilient material directly to the friction pad backplate was not initially considered because of cost and expectation of longer resilient material life. However, this approach may be a viable solution of the current wear problem.)

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* Captive screws are the bolt assemblies used to support the friction pad backplate in the caliper assembly.

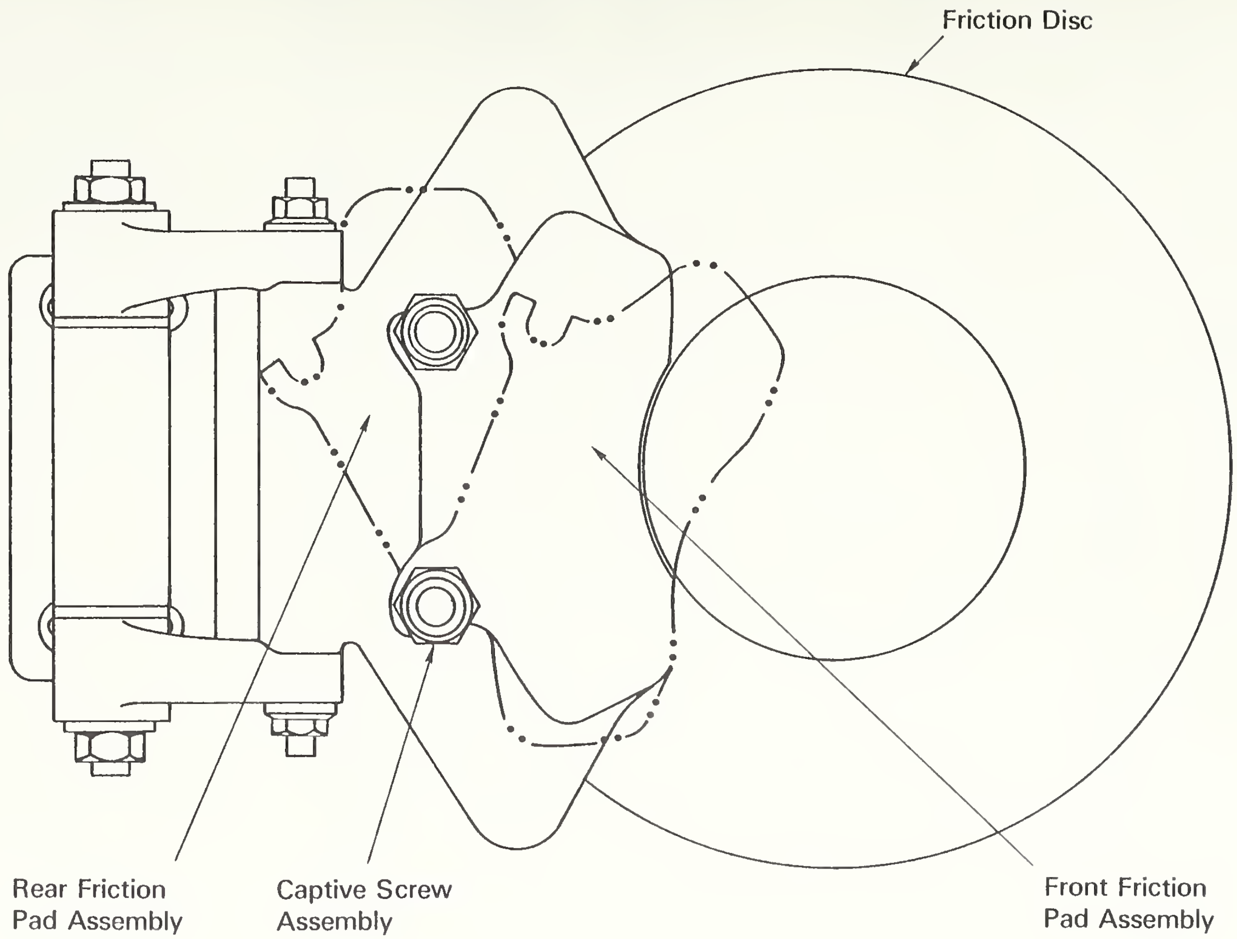

FIGURE 4.2. ILLUSTRATION OF FRICTION PAD REPLACEMENT FOR THE WMATA RAIL TRANSIT VEHICLE DISC BRAKE SYSTEM

SQUEAL CONTROL OPTIONS FOR WMATA

As discussed previously in this section, WMATA maintenance facility personnel reported a number of problems related to the installation of the resilient material assembly currently used on WMATA Metro rail transit vehicles. The primary problems are the rapid wear of the resilient material and the difficulty of its replacement. These problems could be alleviated by manufacturing the resilient material assembly and the friction pad assembly as an integral unit. A single unit configuration might be formed by bonding the resilient material between the friction pad and the pad backplate. This design would simplify the replacement of the resilient material and assure its renewal. Other alternatives include the use of a resilient material that is more resistant to the fretting wear experienced during normal operations or, to bond the resilient material and a wear plate to the caliper piston.

The theoretical and experimental work reported in the open literature indicates a number of other methods which may be used to reduce disc brake squeal generation. The most practical reduction methods include changing the friction pad assembly stiffness and changing the caliper assembly mass, stiffness, and damping capacity. Other squeal reduction methods which have been identified but may not be easily applied to an existing disc brake design include: a variation in the position of contact between the friction pad and caliper piston and an increase in the damping capacity of the friction disc. Use of these techniques, however, will require more extensive evaluation and testing efforts.

V. CONCLUSIONS AND RECOMMENDATIONS

CONCLUSIONS

Disc brake squeal is a chronic and pervasive problem on the WMATA rail transit system. It results in a significant increase in operating sound levels during braking (4 dBA) in a large majority of braking operations (92%). While a noise control treatment has been implemented, its effectiveness -- particularly in the sense of the reduced probability of squeal occurrence -- has not been clearly documented. Furthermore, this treatment has been prone to rapid wear and consequently has not provided an effective reduction or elimination of squeal problem.

A basic understanding of this disc brake squeal generation exists -- sprag-slip. However, while analytical models have been developed, these models are not a suitable supplement to experimental work.

The primary options which are available to quiet a squealing disc brake system are to increase rotor stiffness or internal damping. In the case of WMATA's brake assembly, the disc rotor is bolted directly to the wheel and some evidence exists that the wheel may be a significant radiating surface. Consequently in this unique case benefits may be derived from the alteration of the disc/wheel assembly.

RECOMMENDATIONS

A number of investigations should be performed to fully define the WMATA squeal problem and develop solutions to prevent it.

The probability of squeal should be determined by measurements on at least a full day operation of a single transit car truck (four disc installations) in the stock condition, with the insertion of the 1/8-inch steel plate, and with the insertion of the plate plus bonded resilient material. These measurements should include the determination of squeal frequencies.

The resonant frequencies of the disc rotor, the wheel, and the disc/wheel assembly should be determined to help identify the squeal radiation mechanism.

If the above studies indicate the current squeal noise control is fundamentally viable, more suitable applications of this approach should be explored such as by the direct bonding the resilient material to the friction material backing plate.

Alternative squeal noise controls should also be explored in the form of modifications to the system such as stiffening the disc rotor, decoupling the rotor wheel assembly, or otherwise altering the disc/wheel assembly resonances. A fundamental solution to the brake squeal should be explored by the design and testing of a prototype, high-damping disc rotor. (This approach requires a thorough evaluation of the structural, heat rejection, and wear aspects of the revised rotor material.)

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APPENDIX A
REPORT OF NEW TECHNOLOGY

During this investigation, acoustic measurements were made and hardware systems evaluated in an attempt to understand the mechanisms causing brake squeal on WMATA vehicles. This evaluation has documented the nature of the brake squeal problem, and has reviewed the relevant theories of squeal generation and brake squeal control. These preliminary results will provide direction to future efforts to reduce noise from the WMATA transit car disc brake system.

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