PB81-214702

REPORT NO. FRA/ORD-78/20, II

BRAKE SYSTEM DESIGN OPTIMIZATION
Volume II: Supplemental Data

L.L. Eshelman
C.C. Shelleman
J.P. Henderson

A.T. Kearney, Inc.
100 South Wacker Drive
Chicago IL 60606

APRIL 1981

FINAL REPORT

Prepared for

U.S. DEPARTMENT OF TRANSPORTATION
FEDERAL RAILROAD ADMINISTRATION
Office of Research and Development
Washington DC 20590

Technical Report Documentation Page

1. Report No. FRA/ORD-78/20,II	2. Government Accession No.	3. Recipient's Catalog No. PB81 214702
4. Title and Subtitle BRAKE SYSTEM DESIGN OPTIMIZATION Volume II: Supplemental Data		5. Report Date April 1981
		6. Performing Organization Code
7. Author(s) L.L. Eshelman, C.C. Shelleman, J.P. Henderson		8. Performing Organization Report No. DOT-TSC-FRA-78-1,II
9. Performing Organization Name and Address A.T. Kearney, Inc.* 100 South Wacker Drive Chicago IL 60606		10. Work Unit No. (TRAIS) RR128/R1309
		11. Contract or Grant No. DOT-TSC-1040 -2
12. Sponsoring Agency Name and Address U.S. Department of Transportation Federal Railroad Administration Office of Research and Development Washington DC 20490		13. Type of Report and Period Covered Final Report April 1977 to March 1978
		14. Sponsoring Agency Code

15. Supplementary Notes

*Under contract to: U.S. Department of Transportation
Research and Special Programs Administration
Transportation Systems Center
Cambridge MA 02142

16. Abstract

This report presents supplemental data which support Volume I, A Survey and Assessment, DOT-TSC-FRA-78-1,I,FRA/ORD-78/20,I, dated June 1978, 140 pages.

A detailed description of the functional performance of each freight car air brake system component is provided. This is followed by a discussion of the operation, performance, and design of air brake systems.

The results of a survey made to estimate the population of brake system components in service, and present operating problems with these components is presented. In addition, the problems which require immediate solutions from the perspective of the railroad industry are tabulated.

A summary of recent research work performed to investigate freight car wheel thermal capacity is presented.

17. Key Words Rail freight braking systems Rail freight equipment technology Thermal capacity		18. Distribution Statement DOCUMENT IS AVAILABLE TO THE PUBLIC THROUGH THE NATIONAL TECHNICAL INFORMATION SERVICE, SPRINGFIELD, VIRGINIA 22161	
19. Security Classif. (of this report) Unclassified	20. Security Classif. (of this page) Unclassified	21. No. of Pages 156	22. Price

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

PREFACE

The work described in this report was carried out under the direction of the Transportation Systems Center of the U.S. Department of Transportation in the context of an overall project of the Federal Railroad Administration (FRA) to provide a technical basis for the improvement of rail transportation service, efficiency, productivity and safety. The work was sponsored by the FRA Office of Research and Development, Office of Freight Systems.

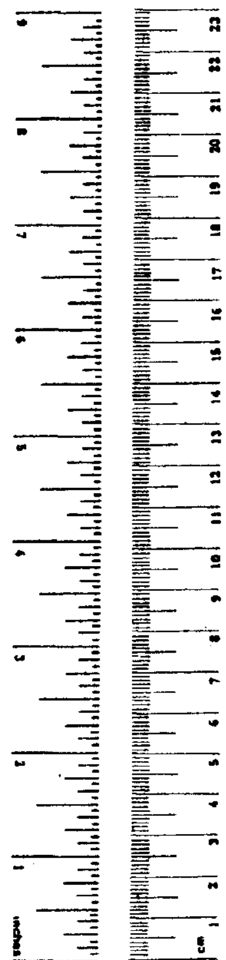
This final report is organized into two volumes. Volume I describes the work performed and presents the results of the study and is a complete report. Volume II contains supporting materials developed in the course of the study which add substantially to the information base but are not essential to the object of the study.

Kearney gratefully acknowledges the cooperation of the many railroads, rapid transit companies, manufacturers, research and development organizations, AAR Brake Equipment Committee members and staff, and other interested parties who provided the information required to conduct a professional, objective study. We would also like to thank DOT personnel in the Transportation Systems Center and the Federal Railroad Administration for their cooperation and assistance throughout the course of the study.

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

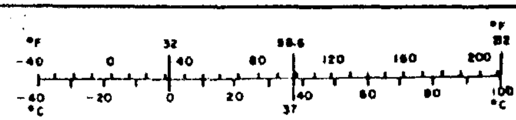


TABLE OF CONTENTS

TABLE OF CONTENTS (Cont'd.)

LIST OF FIGURES

LIST OF FIGURES (Cont'd.)

LIST OF FIGURES (Cont'd.)

LIST OF FIGURES (Cont'd.)

LIST OF TABLES

LIST OF TABLES (Cont'd.)

EXECUTIVE SUMMARY

This report represents supplemental data which support Volume I - A Survey and Assessment.

First a detailed description of the functional performance of each freight car air brake system component is provided. This is followed by a discussion of the operation, performance and design of air brake systems.

The results of a survey made to estimate the population of brake system components in service, and present operating problems with these components is presented. In addition, the problems which require immediate solutions from the perspective of the railroad industry are tabulated.

Finally, a summary of recent research work performed to investigate freight car wheel thermal capacity is presented.

1. INTRODUCTION

This volume represents supplementary data and information which support the material presented in Volume I - A Survey and Assessment.

This volume includes four sections followed by two appendixes. The contents of the sections which follow are:

Section 2 - Functional Performance of Brake System Components and Subsystems. This section presents a detailed description of the functional performance of freight car brake system components.

Section 3 - Operations, Performance, and Design of Freight Car Air Brake Systems. This section describes how the various brake system components interact as a system, and delineates the operating problems presently associated with each.

Section 4 - Railroad and Private Car Owner Survey. This section presents the survey results from railroads and private car owners. These results include rail system operating conditions, air brake system operating problems, air brake component populations, equipment being purchased, and special brake system equipment in service.

Section 5 - Thermal Capacity of Freight Car Wheels. This section summarizes recent research that investigated freight car wheel thermal capacity limitations.

Two appendixes are also included:

Appendix A - Glossary of Terms. This appendix presents and defines the terminology typically used by the air brake and railroad industry.

Appendix B - References and Bibliography. This appendix presents a documentation of the materials used in the study.

2. FUNCTIONAL PERFORMANCE OF BRAKE SYSTEM COMPONENTS AND SUBSYSTEMS

The functional performance characteristics of selected brake system components and subsystems presently in use are presented in this section in the following order:

1. Brake Air Hoses and Couplings
2. Angle Cocks
3. Brake Pipes and Fittings
4. Brake Control Valve
5. Air Reservoir
6. Brake Cylinder
7. Brake Cylinder Release Valve
8. Retaining Valve
9. Brake Rigging
10. Hand Brake
11. Slack Adjuster
12. Brake Shoes.

Each of the above brake system components and subsystems is discussed in terms of the need for their development, their performance, and any problems associated with their use.

Figure 2-1 is presented on the following page as an aid to illustrate the location of the components on a typical car and to indicate the functions and interactions of the components with respect to a freight car brake system.

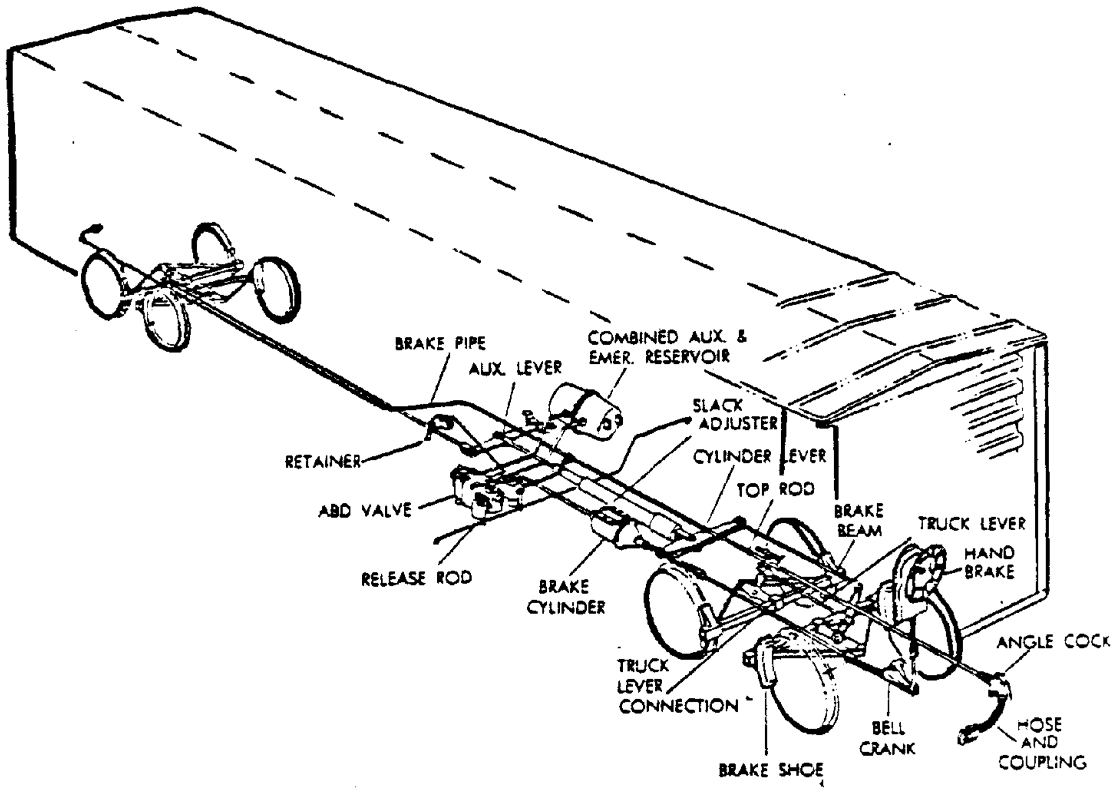


Figure 2-1

ARRANGEMENT OF BRAKE SYSTEM COMPONENTS
ON A TYPICAL FREIGHT CAR

Source: The Air Brake Association (1972). Used by permission of copyright owner.

2.1 BRAKE AIR HOSE AND COUPLINGS

The air hose and couplings ("glad hands") provide a flexible brake pipe connection between cars in a train.

As shown on the following page in Figure 2-2, the air hose is part to a subassembly that includes the hose coupling, two clamps, and a pipe nipple that attaches to the brake pipe of the car.

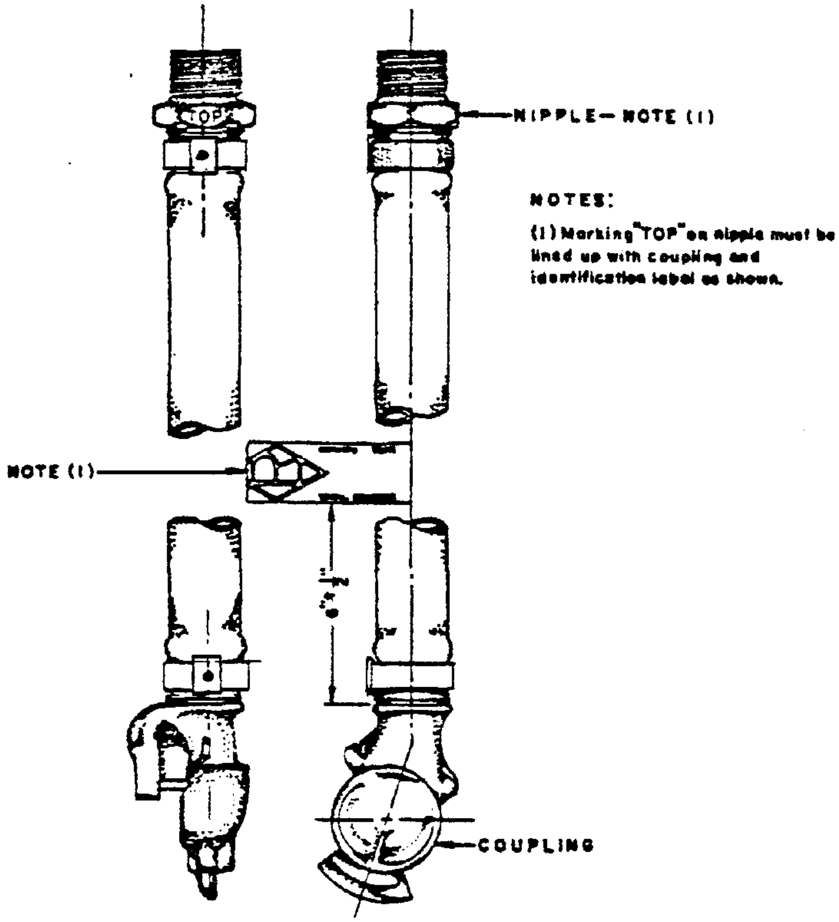


Figure 2-2

AIR HOSE AND AIR HOSE COUPLING

Source: Association of American Railroads (1978b, page E-51-1978). Used by permission of copyright owner.

The air hose couplings (or "glad hands"), which incorporate rubber gaskets to form a seal, are connected by manually raising the two air hose ends together at a proper angle, and lowering them in such a manner as to allow the couplings to rotate relative to each other, causing the flanges of the couplings to engage.

Air hose uncoupling is performed "automatically" when the two cars are being pulled apart during the mechanical uncoupling process. The tension force produced in the air hoses as the cars separate causes the couplings to move vertically upward and rotate so that their flanges become disengaged, permitting them to separate.

There are presently two AAR specifications for air hoses, namely, Specification M-601-71 (AAR standard), and Specification M-601-53 (AAR alternate standard).

Other types of air hoses have been tested in recent years with the primary emphasis on improving air hose life. These have included air hoses with braided steel wire reinforcement and the use of materials such as neoprene. The primary disadvantage of using alternate materials has been the extreme hose rigidity encountered during cold weather operation which makes it difficult, if not impossible, for one man to manually bend the hoses into their coupling position.

According to AAR Interchange Rule 5, the wear limits, gaging, and care for renewal of an air hose are as follows (Association of American Railroads, 1975, Rule 5):

a. Burst.

b. Leakage discernible without soap suds test.

c. Abrasions, cracks, soft spots, etc.

d. Loose or defective fittings, either end or hose.

e. End of hose 3/8 inch or more from shoulder on either nipple or coupling.

f. Porous, as determined by soap suds test.

g. Spliced.

h. Over 8 years old (determined by the date on the hose), date obliterated, at time of COT&S or IDT&S of air brake.

i. Missing.

To assure the proper functioning of the air hose and glad hands, the hose assembly must be properly mounted to and aligned with the car. Proper air hose mounting and alignment become difficult to assure under all operating conditions, especially with long freight cars that have long, overhanging couplers and are supported by shock absorbing cushioning devices that have long travel distances.

Although the specifications for air hose assembly installation arrangements are specified by the AAR (1975b), air hose separation problems are one of the most frequently cited deficiencies of the present air brake system. In the railroad data survey performed for the current project, 12 out of 17 railroads included the air hose coupling and the end of car arrangement as one of the areas that should be given the highest priority (Table 4-15) in improving the freight train brake system.

2.2 ANGLE COCK

As shown in Figure 2-1, an angle cock is normally attached to the end of the brake pipe at each end of the car. The purpose of the angle cock is to provide a manual positive airflow cutoff of the brake pipe. The angle cock is typically used when a portion of a train is to be moved separately during the "picking up" or "setting out" of a "block" of cars from a road train. For example, if a block of cars is to be set out of the head end of the train, the trainman would manually close the angle cock on the rear of the last car of the head end block to be set out. Upon uncoupling the train at this point, the air hoses would separate, throwing the rear unattended portion of the train into "emergency" to prevent it from moving. (This would occur because the angle cock on the front end of the front car would be left in its open position.) The front portion of the train could then be moved with the air brakes charged and properly functioning.

Angle cock location specifications are provided in AAR Specification No. 5218 and in Section E of the Manual of Standards and Recommended Practices.

Figure 2-3 shows the specifications for angle cock and air hose location on cars with an overhang of greater than 5'6" from the truck center to the end of the car.

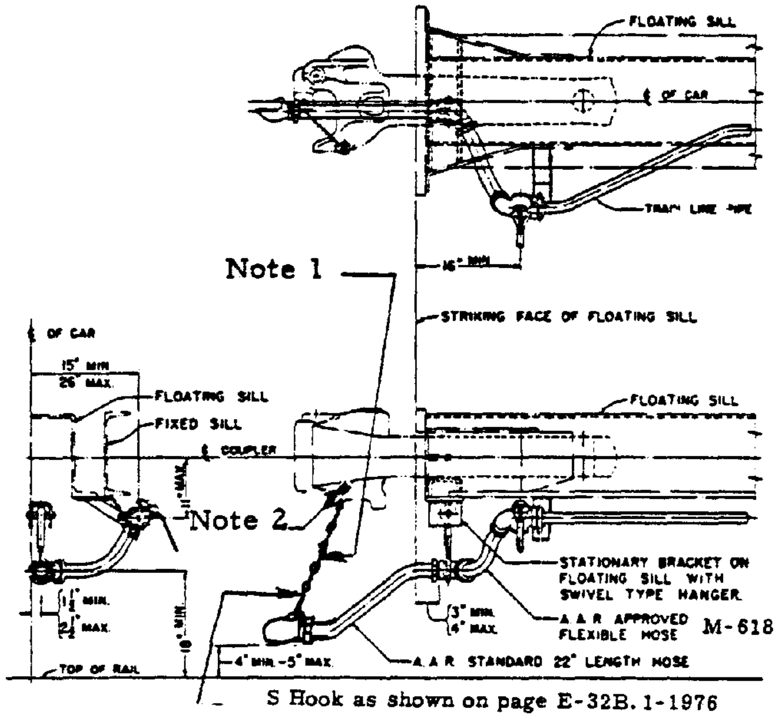


Figure 2-3

TYPICAL ANGLE COCK AND AIR HOSE LOCATION FOR CARS WITH MORE THAN 5' 6" OVERHANG

Source: Association of American Railroads (1978b, p. E-32A-1976). Used by permission of copyright owner.

A suitable angle cock location may be difficult to specify for long cars with certain coupler cushioning devices. In addition to the air hose separation problem discussed previously, both the angle cock and the air hose can be damaged by bypassed-coupler impact during switching and classification operations.

2.3 BRAKE PIPES AND FITTINGS

The brake pipe, like the one shown in Figure 2-1, is constructed of 1-1/4 inch extra heavy pipe. The 1-1/4 inch diameter was selected as an optimum diameter to satisfy both the air flow and volume requirements of the brake pipe in a long train.

As shown in Figure 2-4, a typical freight car includes other pipes in addition to the 1-1/4 inch brake pipe. The sizes and minimum bending radii of all brake system component piping are specified in the AAR Manual of Standards and Recommended Practices.

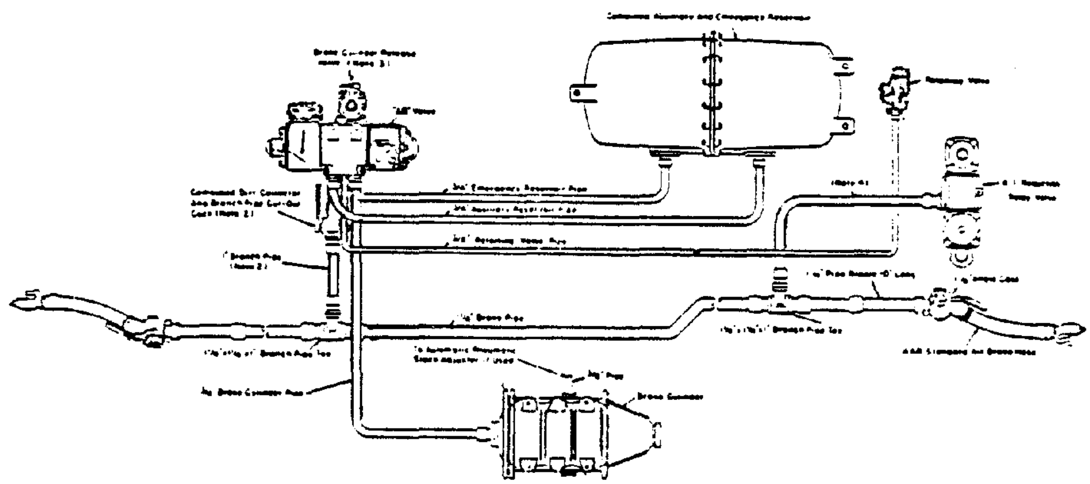


Figure 2-4

PIPING ARRANGEMENT ON A TYPICAL FREIGHT CAR

Source: The Air Brake Association (1972). Used by permission of copyright owner.

Pipe installation and securement specifications are published in the AAR Specification No. 2518.

One of the primary functional performance aspects of the brake pipe is the amount of air brake system leakage present between the connecting fittings on the various brake pipes.

The topic of air brake system leakage was addressed in a paper by Palmer (1975). This paper dealt with various air brake system leakage improvements made over the past 50 years, and defined the following two sources of leakages:

a. Brake Pipe Leakages - leakage in the brake pipe, hose assembly, angle cock and branch pipe to the charging ports of the AB type control valve.

b. System Leakage - with valves in their release position and fully charged, all leakage less the brake pipe leakage, such as reservoir and control valve leakage.

Although the leakage tests performed over the past 50 years varied considerably in their test methods and criteria, the results showed the total brake system leakage has decreased over this period of time. Table 2-1 compares the leakage test results from the tests made over the past 50 years.

TABLE 2-1

COMPARISON OF BRAKE SYSTEM LEAKAGE TEST RESULTS OVER A PERIOD OF 50 YEARS

	Percent of Cars Tested That Had Leaks					
	Based on Total Leaks			Based on Total Cars		
Source of Leakage	1925	1950	1975*	1925	1950	1975*
Hose and Gaskets	17.0%	19.8%	31.0%	71.0%	16.7%	11.1%
Angle Cocks	21.6	4.0	18.3	92.3	3.4	6.3
Brake Pipe	20.8	12.8	3.2	87.4	9.8	1.1
Branch Pipe	20.4	40.1	14.8	85.4	33.3	5.1
Brake Valve	17.1	1.2	19.4	84.8	1.0	3.9
Reservoir	3.1	22.1	12.5	18.0	18.6	7.4
Base (Number) or Percent	(10634)	(169)	(249)	439.0%	83.0%	34.0%

Note: * Based on audible leaks.

Source: Palmer (1975, Table 1).

The results of Table 2-1, while based on different test procedures and criteria, illustrate that air brake system leakage had decreased over the period. For example, in 1925, 87.4 percent of the cars checked had brake pipe leakage; in 1950, only 9.8 percent of the cars checked had brake pipe leakage. Furthermore, in 1975 only 1.1 percent of the cars checked had (audible) brake pipe leakage.

More recent test results were presented by Palmer that dealt with the effects that cold weather has on brake system leakage. Although these tests involved a small sample of observations, the following results were indicated:

a. "Cold weather adversely affects leakage."

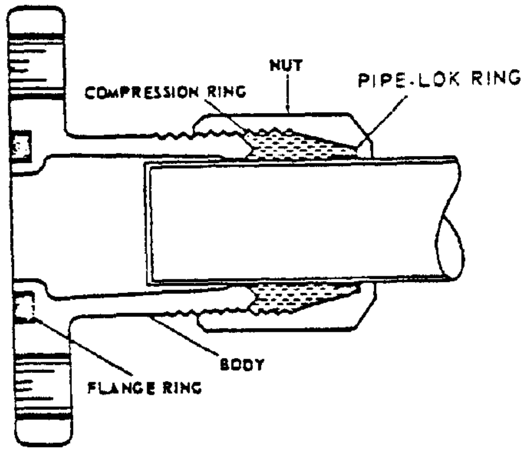
b. "Temperature has less effect on brake pipe leakage than on system leakage."

c. "The average car evaluated consumed 0.42 cubic feet of free air each minute."

d. "A poll of railroad repair facilities indicated that 40% of all cars tested have too much leakage to pass the single car test when originally tested. This tends to say that in the past 25 years, leakage levels have not changed appreciably."

The preceding tests were performed to sample air brake system leakage rates on a number of typical cars in acceptable operating condition. The results of these tests can be interpreted by considering the historical development of brake pipe fittings, and present requirements. Prior to the 1950's, brake system and leakage commonly resulted from fractures in the threaded area of the brake pipe fittings. In the early 1950's, fittings were introduced employing rubber compression grips. However, leakage problems persisted during cold weather due to insufficient tightening of the gland nuts of the compression fittings. This led to the introduction of welded fittings. Present AAR rules require the replacement of compression fittings with welded fittings when a car is shopped for 150 or more manhours of work. The compression fitting and the welded fitting are illustrated in Figure 2-5.

Compression Fitting



Welded Fitting

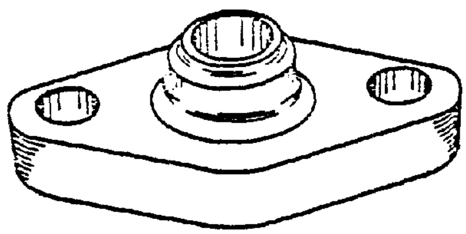


Figure 2-5

BRAKE PIPE FITTINGS

Also related to the functional performance of brake pipes and fittings is the ease with which these fittings can be repaired in the event of failure. This problem was discussed by Blaine and Hengel (1971) with respect to the special problems associated with repairing leaks on cars with cushion underframes, as follows:

> "Installation of brake pipe on cushion underframe cars between the sliding and stationary center sills has made locating and repairing leaks or damaged pipe very difficult without disassembling the underframe. More design attention is needed in this area."

The authors continued:

> "Excessive brake leakage is one of the major

> contributors to poor train handling. This defect can be due to leaks in piping caused by improperly anchored train line or fittings which allow the pipe to shift during coupling or slack action, defective air hose and gaskets, or angle cocks. Sometimes, this type of defect is not easily detected during terminal tests. During train movement or "stretching out" the train, leaks are more apt to occur if the train line is not adequately anchored. A car with compression fittings would seem to be more vulnerable, and this can become a real problem when the brake pipe is attached to the sliding sill of a cushioned car. Unfortunately, in many instances, this has led to a practice of torch cutting access holes in the stationary center sill in order to tighten hidden fittings."

In addition to affecting overall train braking performance, leakage has a direct impact on freight train operations. Because the rules governing the operation of freight trains require that air brake leakage be less than 5 psi per minute, and that brake pipe pressure gradient be less than 15 psi, it is a common practice to limit train length in cold weather operations in order to meet the legal minimum leakage requirements. As shown in Figure 2-6, on the following page the minimum brake pipe gradient of 15 psi is difficult to achieve for a 100-car train operated in ambient temperatures of below zero degrees Fahrenheit.

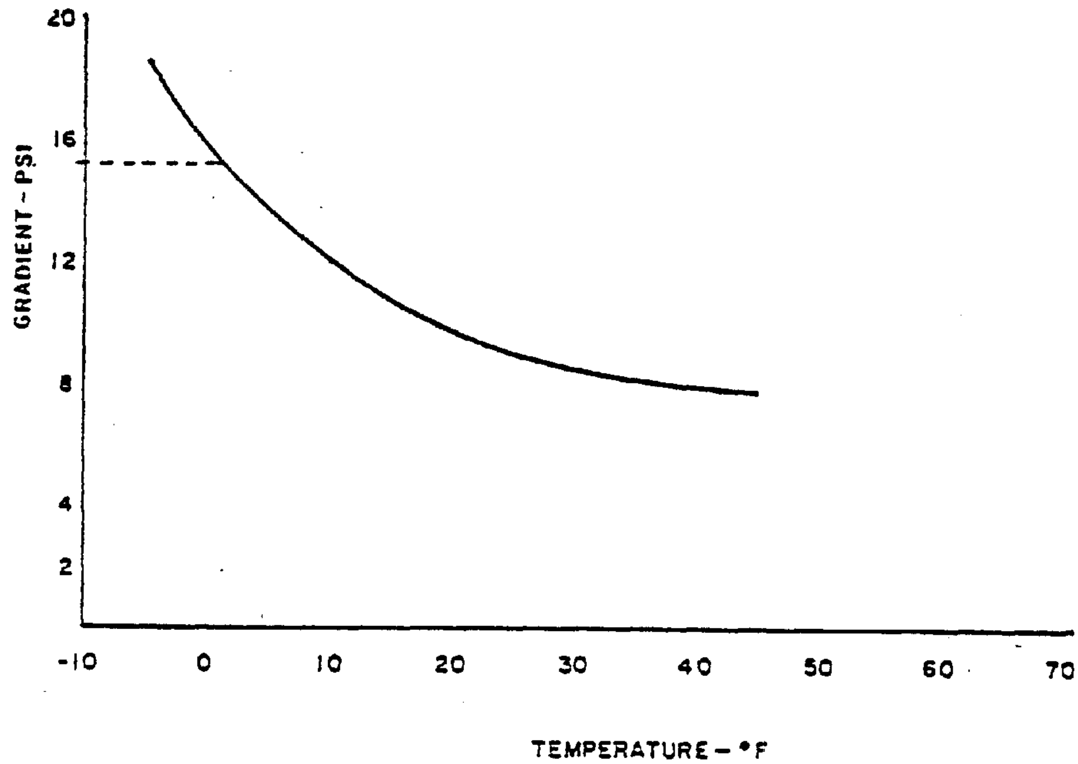


Figure 2-6

APPROXIMATE BRAKE PIPE GRADIENT VERSUS TEMPERATURE FOR "TYPICAL" 100-CAR TRAIN

Source: Palmer (1975, Curve 1). Used by permission of copyright owner.

A comparison curve showing brake pipe leakage versus ambient temperature is shown in Figure 2-7 on the following page.

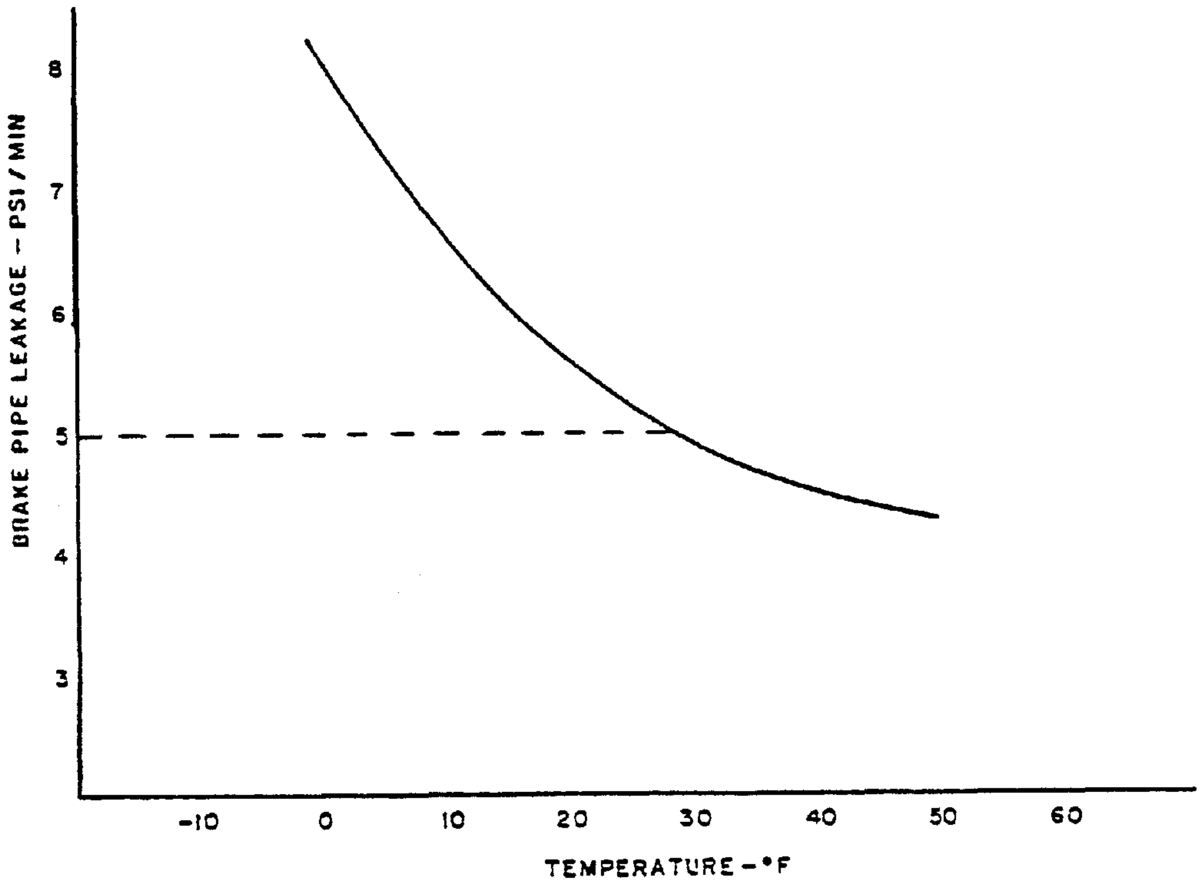


Figure 2-7

APPROXIMATE BRAKE PIPE LEAKAGE VERSUS
TEMPERATURE FOR "TYPICAL" 100-CAR TRAIN

Source: Palmer (1975, Curve 5). Used by permission of copyright owner.

As shown in Figure 2-7, the maximum leakage requirement of 5 psi per minute becomes difficult to achieve for a 100-car train when ambient temperature drops below the freezing point.

While both curves shown in Figures 2-6 and 2-7 are approximations, they do tend to illustrate that the leakage limit is more restrictive than the brake pipe gradient limit. This result has been partially responsible for requests that the validity of the 5 psi per minute leakage rate limitation be re-examined.

The use of gradient instead of brake pipe leakage as the only criterion for assuring operable train brakes was discussed in the paper by Blaine and Hengel (1971). In this paper the practical limit for satisfactory brake control on trains was assessed as an airflow demand of under 60 CFM, and preferably 50 CFM or less.

By establishing a relationship between brake pipe gradient and leakage, the authors concluded that

> to keep the airflow rate of 50 to 60 CFM or less..., the gradient must be held to 5 psi for 50 cars, 10 psi for 100 cars, and 15 psi for 150 cars. In effect, if all the leakage is in the brake pipe, this airflow would correspond to brake pipe leakage of 20 psi/min for 50 cars, 12 psi/min for 100 cars, and 8+ psi/min for 150 cars. However, at 8- to 10-psi/min brake pipe leakage stopping distance becomes adversely affected.

The effects of brake system leakage on train braking performance are discussed in part C of this section.

2.4 BRAKE VALVES

The functional performance specifications of brake valves are contained in the AAR Manual of Standards and Recommended Practices. These specifications pertain to the operational response of the brake valve with respect to its relationship to other valves and components of the air brake system of a freight train.

The functional performance characteristics of brake valves in use today can be described in terms of the AB and ABD types of brake valves. In addition, more advanced brake valves presently under evaluation include the ABDW and Z1A. The primary functional performance aspects of each of these valves are discussed in the following paragraphs.

2.4.1 AB Brake Control Valve - The AB valve is the oldest, most common control valve in use today. As such, the functional performance of the AB valve can be considered as a standard against which other brake valves can be compared.

The AB valve, which was introduced in 1933, replaced the "K" valve. (The last "K" valve was removed from interchange service in 1953.) Shown in Figure 2-8 the primary performance feature of the AB valve included a better quick-service function, retarded recharge, release-ensuring, and more sensitive application than the "K." In addition, the AB had a separate emergency portion allowing a much faster emergency transmission speed (over 900 feet-per-second). To control slack action during brake application, the AB valve provided an initial fast, limited brake cylinder pressure to bunch slack, followed by a more gradual pressure buildup giving full pressure in approximately 12 seconds.

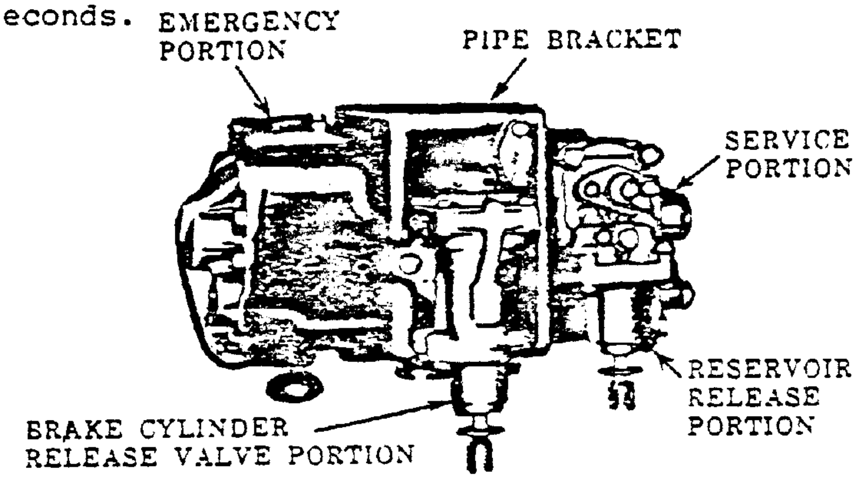


Figure 2-8

AB BRAKE CONTROL VALVE WITH QRB BRAKE CYLINDER RELEASE VALVE EXTERIOR VIEW

Source: The Air Brake Association (1972). Used by permission of copyright owner.

With the AB valve, a full emergency application on a 150-car train required approximately 17 seconds, whereas a full service application would require two to three minutes to develop (Blain, 1975b).

The maintenance requirements (of COT&S) of the AB valve are presently based on a period of 48 months.

2.4.2 ABD Brake Control Valve - By the 1950's, with the operation of long freight trains, the need for greater flexibility in train braking operations became apparent. The desirable direct-release feature of brakes was difficult to achieve in long trains, especially if leakage was concentrated in the rear. Unless brake pipe pressure could be increased at a high rate at the head end of the train, leakage at the rear end could cancel the "release" signal of increased brake pipe pressure, and the brakes at the rear of the train would not release. To prevent trains from being pulled apart during the release of a moderate or heavy brake application, trains would be allowed to stop completely before releasing the brakes.

Recognizing the need for a direct-release capability while trains were in motion, brake control valve designers developed an "accelerated service release" feature of the brake control valve. This feature allows a small amount of air from the emergency compartment of the air reservoir to enter the brake pipe locally to overcome the leakage effects. Thus the accelerated release propagates through the train at speeds of 450 to 600 feet-per-second.

The accelerated release feature, which was first demonstrated on the AC valve, was later incorporated into the ABD valve shown in Figure 2-9 below.

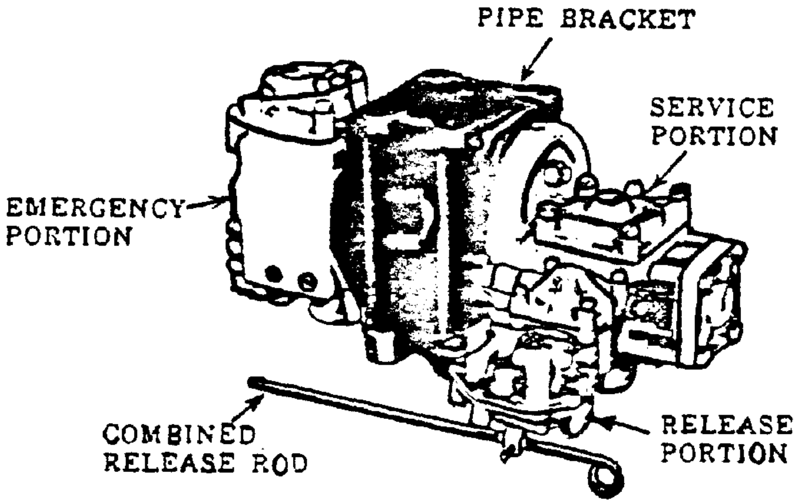


Figure 2-9

AIR BRAKE CONTROL VALVE EXTERIOR VIEW

Source: The Air Brake Association. Used by permission of copyright owner.

It should be noted that to facilitate conversion of the AB valves to ABD valves, the service and emergency portion of the ABD valve can be mounted to the original brake pipe bracket used for the AB valve (see Figures 2-8 and 2-9).

By changing the design and materials of the ABD valve, the COT&S maintenance period of the ABD valve was recently extended from 8 to 10 years, compared to 4 years for the AB valve (Association of American Railroads, 1975c).

2.4.3 ABDW Brake Valve - While the ABD valve was designed to provide a "running release" capability by producing a faster, more effective brake pipe pressure increase, there was also a need to decrease the time to apply the brakes of a train in order to reduce stop distances.

To assist in making brake applications on long cars having large brake pipe air volumes, a device known as the A-1 Reduction Relay Valve was designed and applied. The A-1 Reduction Relay Valve, which is illustrated in Figure 2-10, assists the AB or ABD valve in making a brake application at both the service and emergency rates.

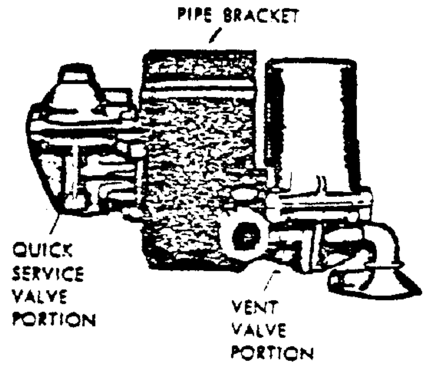


Figure 2-10

A-1 REDUCTION RELAY VALVE

Source: The Air Brake Association. Used by permission of copyright owner.

For brake pipe reduction at a service rate, the quick service valve portion of the A-1 Reduction Relay Valve locally vents brake pipe air through a choke at a service rate, thus assisting and propagating the service application through the train.

For a brake pipe reduction at an emergency rate, the Vent Valve Portion of the A-1 Reduction Relay Valve locally vents brake pipe air at an emergency rate.

The requirement for mandatory installation of the A-1 Reduction Relay Valve on all cars having more than 75 feet of brake pipe was intended to prevent a deterioration of braking performance caused by the increasing percentage of long cars. However, this feature is also desirable for use on all cars regardless of their length. In response to this need, the ABDW brake valve was designed with an "Integrated Continuous Quick Service" feature to upgrade service braking on all cars at lower cost and with improved functional performance over the A-1 Reduction Relay Valve (Wright, 1975). The ABDW valve is shown in Figure 2-11 and a schematic of the internal passages of the ABDW valve is presented in Section 3 Figure 3-3.

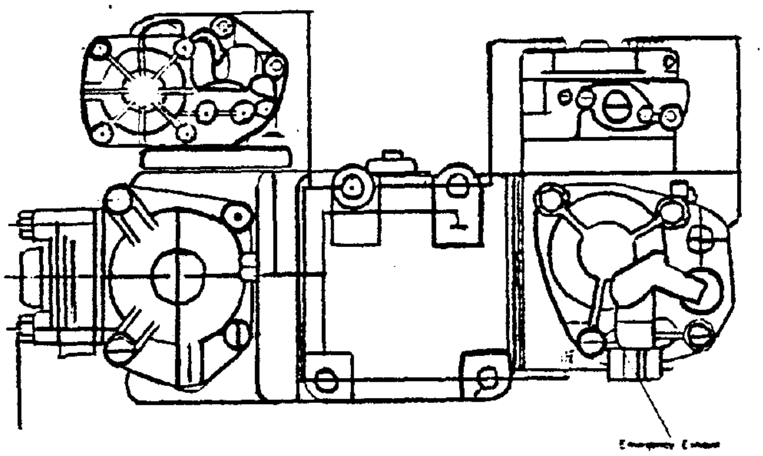


Figure 2-11

ABDW BRAKE CONTROL VALVE

Figure 2-12 shows the manufacturer's performance comparison between the ABD and ABDW control valves for full service brake applications.

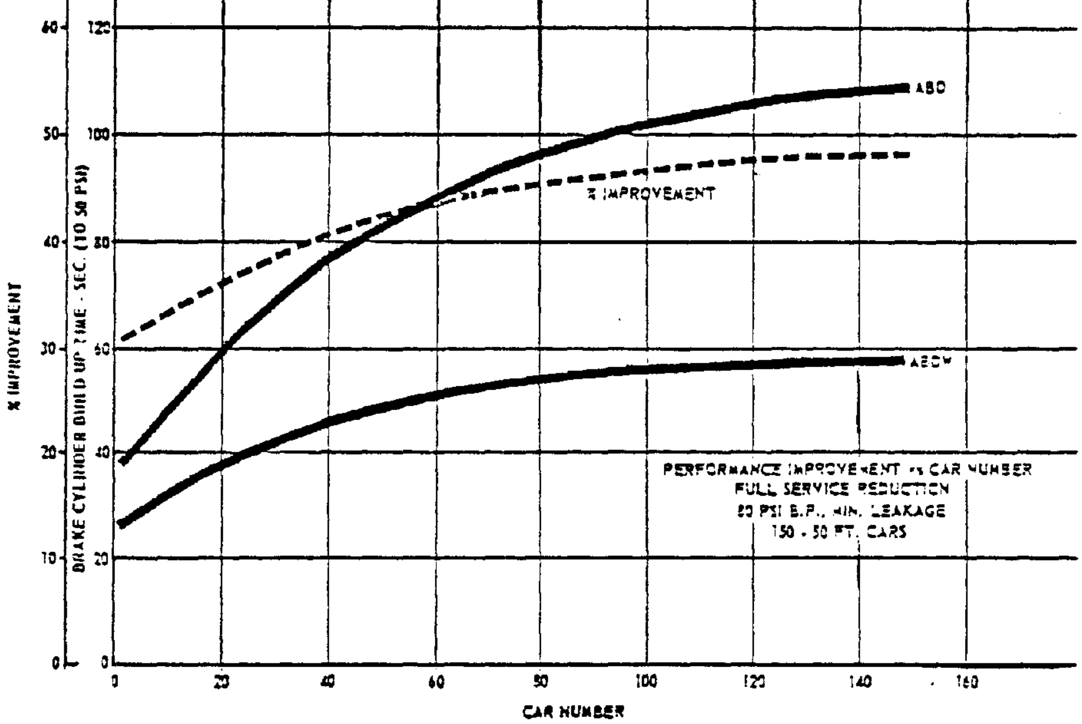


Figure 2-12

PERFORMANCE COMPARISON BETWEEN ABD
AND ABDW BRAKE VALVES

Source: Wright (1975, Fig. 5). Used by permission of copyright owner.

As shown in Figure 2-12, for a 150-car train completely equipped with either the ABD or the ABDW brake valve, the manufacturer predicts almost a 50 percent improvement in brake cylinder pressure build-up time.

It should be noted that the ABDW control valve has been demonstrated to the Brake Equipment Committee of the AAR and is presently under consideration for acceptance in interchange service by the member roads.

2.4.4 The Z1A Brake Control Valve - The Z1A brake control valve is another advanced brake control valve presently under consideration by the AAR for approval in interchange service. According to the manufacturers of the Z1A, the valve has been designed to provide both reduced initial costs and maintenance costs, and to provide an improvement in performance by reducing service propagation times and valve leakage. It was also designed to be fully compatible and interchangeable with the AB and ABD types of brake valves presently in use. An outline view of the Z1A brake control valve is shown in Figure 2-13.

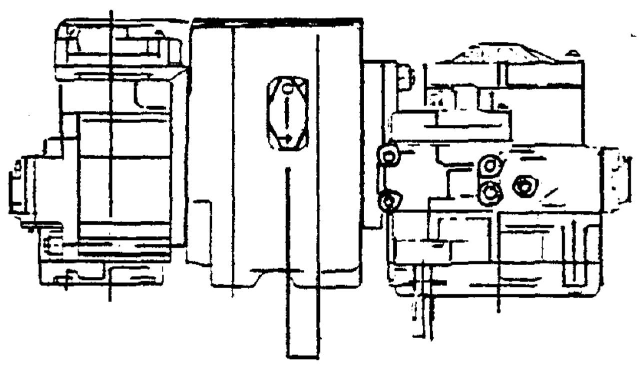


Figure 2-13

Z1A BRAKE CONTROL VALVE

According to the manufacturers of the Z1A, it would have longer potential life and lower maintenance costs than present brake valves because of the use of poppet valves instead of spool type valves. They attribute two main causal factors to the restricted life of the spool valves.

1. "Rubber seals wear more rapidly in applications such as spool valves which involve the movement of the seal over ports in the bore".

2. "The high velocity air moving around the spool removes the lubrication which is critical for long seal life."

The functional performance during full service application and release of the Z1A valve is shown in Figure 2-14.

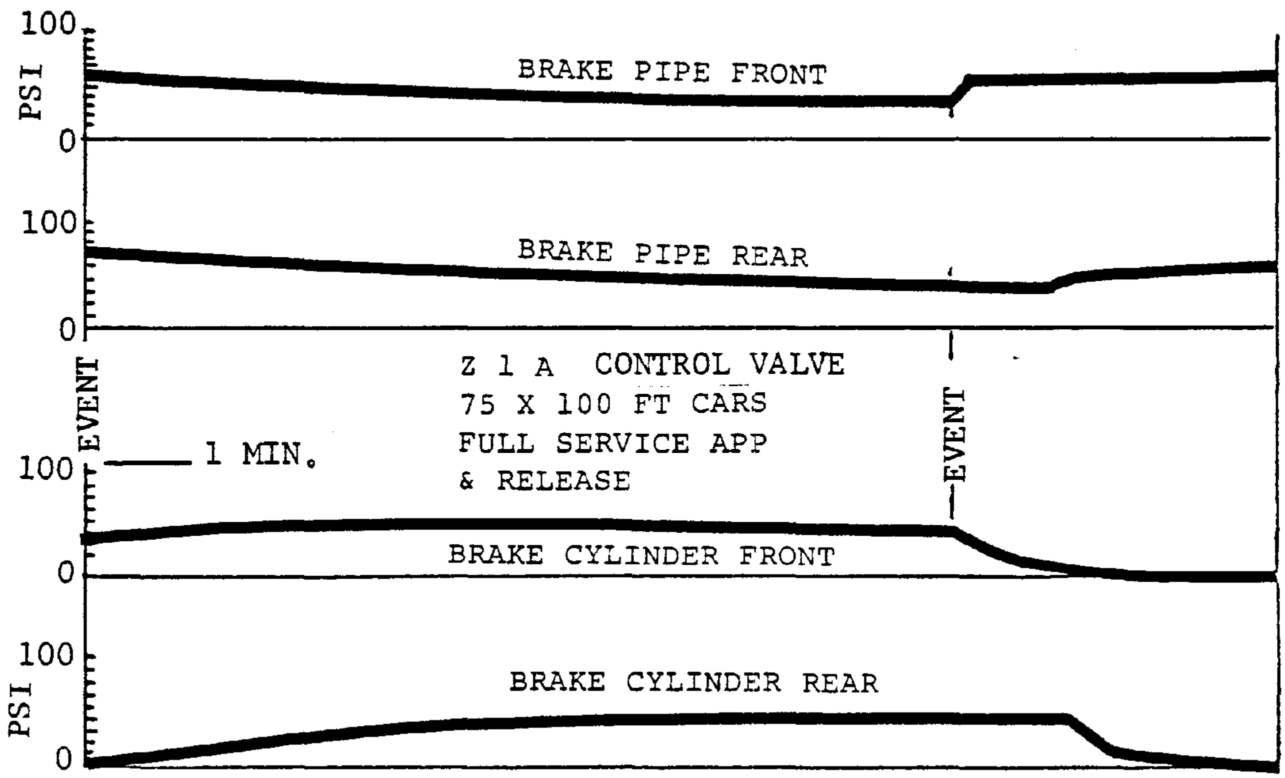


Figure 2-14

FUNCTIONAL PERFORMANCE OF Z1A BRAKE CONTROL VALVE DURING FULL SERVICE APPLICATION AND RELEASE

Source: Westinghouse Brake and Signal (1975).

2.5 AIR RESERVOIR

The purpose of the air reservoir is to store compressed air as braking energy during brake release to be used when needed by being admitted to the brake cylinders(s) by the brake control valve during brake application.

The air reservoir, shown in Figure 2-15, has two separate compartments - the auxiliary compartment which stores air for service brake applications, and the emergency compartment which stores air for emergency brake applications.

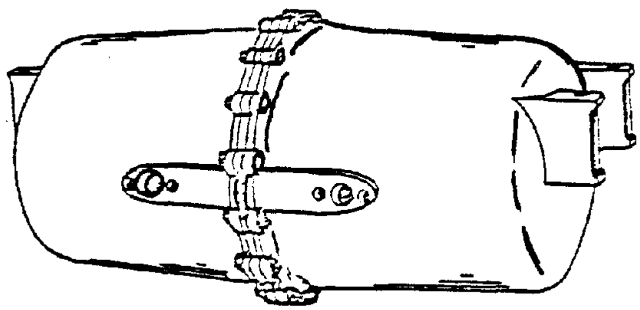


Figure 2-15

AIR RESERVOIR

Source: Track Train Dynamics (1973). Used by permission of copyright owner.

The volume of the auxiliary air reservoir compartment system charge of 70 psi will produce a brake cylinder pressure of approximately 50 psi following a full service application.

The emergency compartment of the air reservoir has a volume (typically 3,500 cubic inches) such that with the equipment initially charged to 70 psi, both reservoir volumes and brake cylinder pressures will equalize at approximately 60 psi following an emergency application (with a standard 10-inch cylinder with a 8-inch piston travel) (Wabco, 1967, p.9).

Table 2-2 shows the full service and emergency brake cylinder equalization pressure requirements for different values of initial brake pipe charge pressure.

TABLE 2-2

FULL SERVICE AND EMERGENCY EQUALIZATION BRAKE CYLINDER PRESSURE VERSUS INITIAL CHARGE BRAKE PIPE PRESSURE*

Initial Charge Brake Pipe Pressure psi	Equalization Brake Cylinder Pressure Full Service psi	Equalization Brake Cylinder Pressure Emergency psi
70	50.0	60.0
75	53.5	64.0
80	57.0	69.0
85	60.5	72.0
90	64.0	77.0
100	71.0	85.0
110	78.0	93.0

Note: * Based upon 2,500 cubic inch and 3,500 cubic inch auxiliary and emergency reservoirs, respectively, with a 10-inch diameter cyclinder and 8-inch piston travel.

Source: Blaine and Hengel (1971).

As shown in Table 2-2, the emergency equalization brake cylinder pressure is approximately 20 percent greater than the full service brake cylinder pressure regardless of the value of the initial brake pipe pressure. This is a result of the pressure-volume relationship between the emergency and auxiliary air reservoir compartments and the brake cylinder.

2.6 BRAKE CYLINDER

The purpose of the brake cylinder (and piston) is to provide the braking force on the car by applying the brake shoes against the wheels during brake applications. Brake cylinders exist in two types - one type for conventional indirect-acting brake rigging (AB-1 or ABU), and a different type for direct-acting, truck-mounted brake rigging.

Figure 2-16 illustrates the AB-1 type of brake cylinder used with conventional brake rigging.

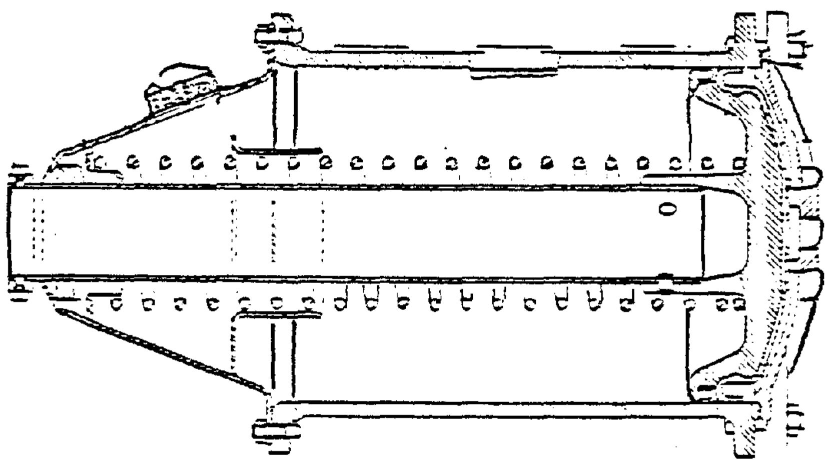


Figure 2-16

AB-1 TYPE OF BRAKE CYLINDER USED WITH CONVENTIONAL BRAKE RIGGING

Air enters the brake cylinder through the port at the left and displaces the piston, compressing the return springs and applying force to the brake rigging.

Total brake cylinder volume must be properly related to the air reservoir volume to produce the required brake cylinder pressure. To produce the proper braking force developed by the brake cylinder depending upon whether high friction or low friction brake shoes are used, a brake cylinder of proper diameter must be selected by the designer of the brake system. In addition, the brake system of a freight car can be converted from using low friction to high friction brake shoes by installing a "bushing kit" in the brake cylinder to reduce the effective piston diameter, thereby producing a lower braking force. A brake cylinder with a bushing kit installed is shown in Figure 2-17.

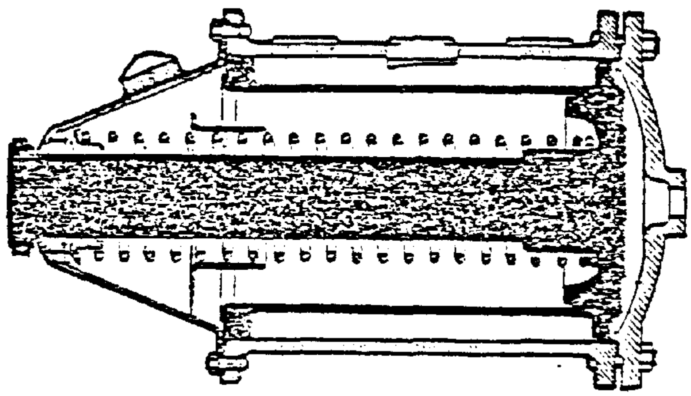


Figure 2-17

BRAKE CYLINDER BUSHING KIT

2.7 BRAKE CYLINDER RELEASE VALVE

The purpose of the brake cylinder release valve is to permit the manual "bleeding" of the air from the brake cylinder(s) of a car after brake pipe pressure has been depleted in preparation for switching and classification. Using the brake cylinder release valve eliminates the need to bleed the air from the air reservoir compartments and therefore reduces the time needed to recharge the reservoir during outbound train preparation. One of the manufacturers of the brake cylinder release valve estimates that 40 percent of the time can be saved in recharging a 150-car train if each car has the valve installed.

The brake cylinder release valve can be mounted directly to the pipe bracket for AB valves of recent manufacture, and can be installed with older AB brake valves by the use of an adapter kit. The ABD brake valve incorporates the brake cylinder release valve as a standard feature, as shown in Figure 2-18.

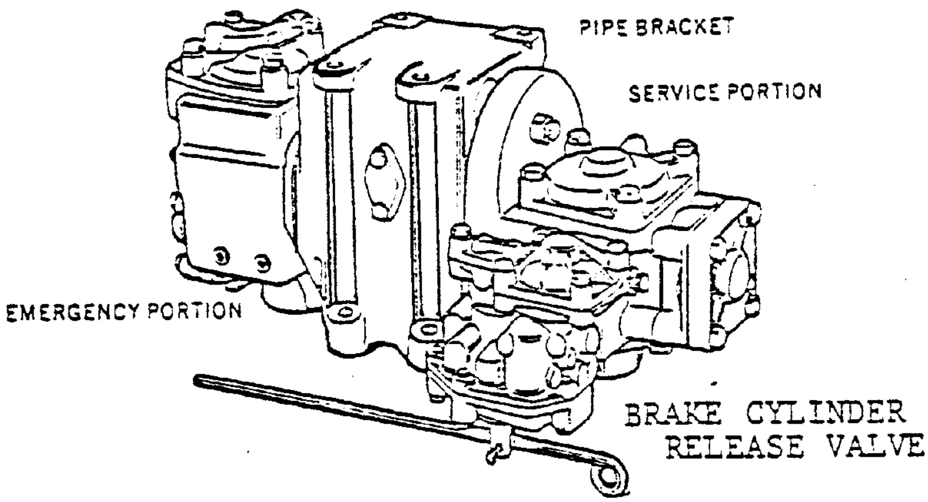


Figure 2-18

BRAKE CYLINDER RELEASE VALVE AS INTEGRAL PART OF ABD BRAKE VALVE

Source: Track Train Dynamics (1973). Used by permission of copyright owner.

The performance specifications for the brake cylinder release valve are provided in the Association of American Railroads (1875b pp. E-35 - E-42).

The manual actuation of the valve is such that a single momentary pull on the release rod would deplete the brake cylinder pressure. Although this feature is desirable, car inspectors in several yards routinely bleed both the brake cylinders and the air reservoir (by holding the release rod) to prevent air in the reservoir from leaking through defective brake cylinder release valves and reapplying the brakes. This problem was also described in the Railroad Survey Results - Part B (see Section 4, Table 4-3 through 4-11).

2.8 RETAINING VALVE

The purpose of the retaining valve is to maintain a minimum amount of air pressure in the brake cylinder after a brake "release" has been made and the car reservoirs are being recharged. This feature is used in severe descending grade territory so that train brakes are not inadvertently released. The retaining valves on each car must be manually set for both activation and release.

Prior to 1967 the retaining valve incorporated a four-position setting as follows:

1. "EX" (Direct Release). No air restriction.

2. "EX - HP". Retains 20 psi in the brake cylinder when train brakes are "released".

3. "EX - LP". Retains 10 psi in the brake cylinder when train brakes are "released".

4. "EX - SD" (Slow Direct Exhaust). Brake cylinder pressure is completely exhausted but at a slow rate.

In 1967 the three-position retaining valve was introduced (see Figure 2-19). The three-position retaining valve omits the "EX - LP" position, but includes the "EX," "HP," and "S" positions.

The functional performance specifications of both types of retaining valves are presented in the Association of American Railroads (1975b).

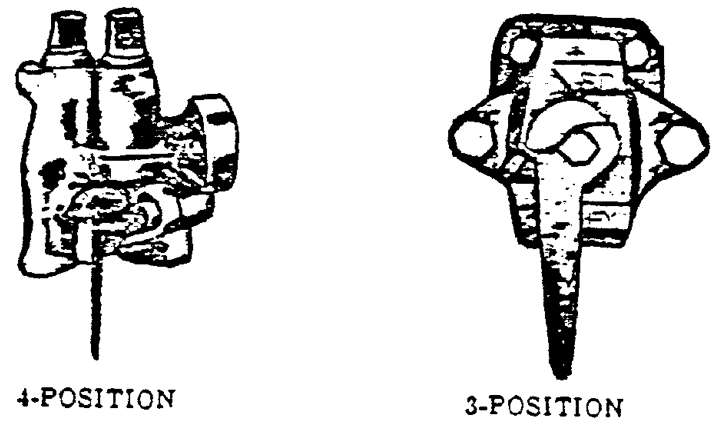


Figure 2-19

FOUR-POSITION AND THREE-POSITION RETAINING VALVES

Source: The Air Brake Association (1972). Used by permission of copyright owner.

Prior to the introduction of dynamic brakes on diesel-electric locomotives, retaining valves were used routinely for operating heavy freight trains on steep descending grades. Today, however, retaining valves are used only under certain circumstances--typically on severe grades when dynamic brakes are not available, or from coal mine tipples to yards and main lines, or on certain exceptionally steep (over 2 percent) main line descending grades (see Section 4, Table 4-2).

2.9 BRAKE RIGGING

The purpose of brake rigging is to multiply and distribute the braking forces developed by the brake cylinder(s) to the brake shoes of the car.

Because of the introduction of many different types, lengths, and capacities of freight cars, demands have been placed upon the designers of air brakes to design rigging that both produces an adequate braking level, and that can be physically attached to the freight car without interfering with other devices and appliances.

Brake rigging in use today includes two types: indirect-acting ("conventional") rigging, and direct-acting ("truck-mounted") rigging.

The functional performance of both types of rigging must be such that the AAR Net Braking Ratio (NBR) specifications (see Section 3) are adhered to.

2.9.1 <u>Conventional Rigging (Indirect Acting)</u> - Figure 2-20 illustrates four typical configurations of the conventional type of brake rigging. Any one of several different brake rigging arrangements can be used, depending upon the configuration of the car and other factors.

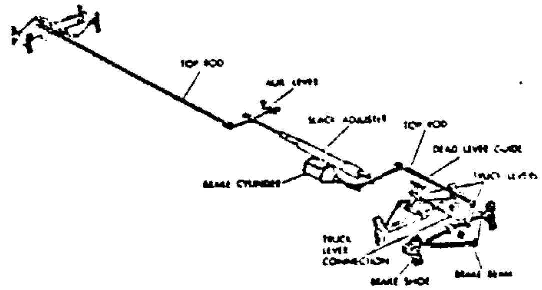
To achieve the required Net Braking Ratio, the designer of the conventional brake rigging can cause the forces produced by the brake cylinder to be multiplied by selecting the proper lever ratios. However, the total lever ratio must be limited to be no greater than 12.5:1 (if a double-acting automatic slack adjuster is used) so that the pressure-volume relationship between the air reservoir and the brake cylinder is not adversely affected by brake shoe wear. Also, the stresses produced on levers, rods, jaws, and pins must be limited to their inherent design values.

The AAR specifications require that a metal badge plate be fastened to the underframe of each freight car showing the design brake lever dimensions of the brake rigging. An example of a brake rigging badge plate is shown in Figure 2-21.

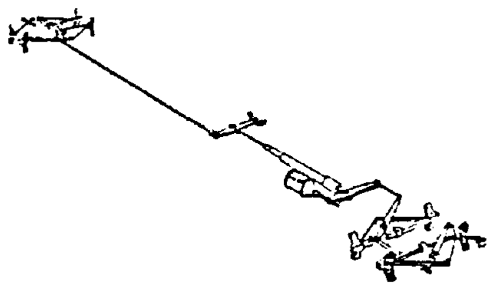
Because the Net Braking Ratio includes the losses of efficiency of various brake rigging components (see Section 3), estimates of efficiency are needed by the designer so that efficiency losses can be compensated for in the brake rigging force design calculations.

The efficiency losses of various brake system components have been estimated from tests. The results of these tests indicate that the efficiencies of various configurations of brake rigging differ considerably. However, approximate efficiency values have been determined as shown in Table 2-3 on page 32.

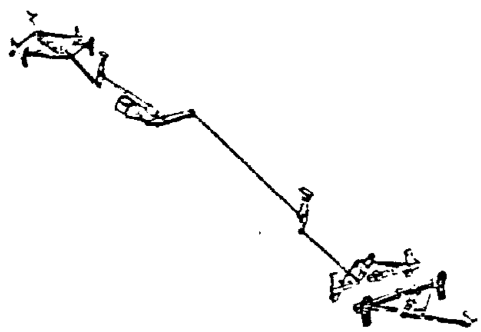
1. Bottom Rod Through Bolster



2. Bottom Rod Under



3. Top Rod Under Hook and Eye



4. Three-Lever Truck Arrangement

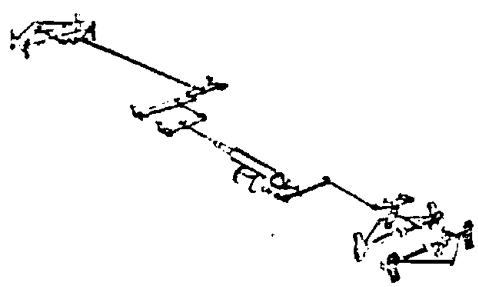
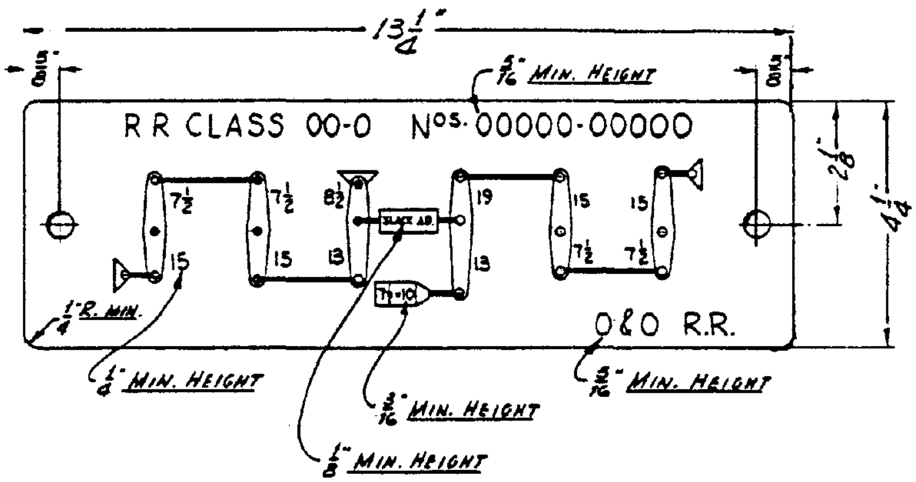


Figure 2-20

FOUR CONFIGURATIONS OF CONVENTIONAL BRAKE RIGGING

Source: The Air Brake Association (1972). Used by permission of copyright owner.



BRAKE BADGE PLATE SHOWING DIMENSIONS
FOR FREIGHT CARS.

NOTES:

1. MATERIAL - STAINLESS STEEL, ALUMINUM, PRESSED OR CAST METAL.
2. LETTERS, NUMBERS AND DIAGRAM ON STAINLESS STEEL OR ALUMINUM PLATES TO BE ETCHED BLACK.
3. HOLES DIA. 7/16
4. SHARP EDGES TO BE GROUND SMOOTH.
5. DIAGRAM, DIMENSIONS AND R.R. CLASS INFORMATION ON PLATE FOR ILLUSTRATIVE PURPOSES ONLY.

Figure 2-21

BRAKE RIGGING BADGE PLATE

Source: Association of American Railroads (1978b p. E-71-1976, Fig. 47). Used by permission of copyright owner.

TABLE 2-3

APPROXIMATE VALUES OF EFFICIENCY LOSSES OF BRAKE RIGGING COMPONENTS

Type of Component	Approximate Efficiency Loss
Brake Cylinder 7-1/2 inch	11%
Brake Cylinder 8 inch	10
Brake Cylinder 8-1/2 inch	9
Brake Cylinder 10 inch	8
Brake Cylinder 12 inch	7
Pins	1 per pin
Slack Adjuster (Pneumatic)	5
Horizontal Pads	1
Unit Beams	1 per beam

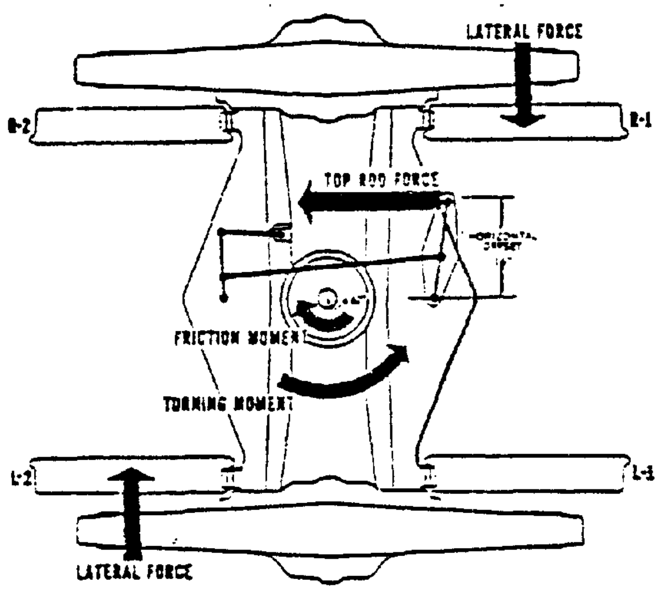
Source: St. Louis Air Brake Club (1972).

Field testing has also produced approximate efficiency ranges of the brake rigging on various types of cars. These results have shown that conventional brake rigging efficiency can range from a minimum of approximately 45 percent to a maximum of approximately 75 percent (St. Louis Air Brake Club, 1972).

An inherent disadvantage of certain arrangements of the conventional type brake rigging is the truck turning moment induced by the reaction forces of the brake rigging during brake application. According to the manufacturers of the truck-mounted type of brake rigging, this turning force can produce a lateral force exerted by the flange of the wheels against the rail*. The magnitude of this force can be on the order of 490 pounds, depending upon the brake rigging configuration. Figure 2-22 illustrates these forces for two configurations of conventional rigging.

*Tests have shown that the asymmetrical truck forces cause the truck to "parallelogram" (McLean, 1973).

a. TOP ROD OVER BOLSTER WITH DEAD LEVER ANCHORED TO TRUCK BOLSTER



b. BOTTOM ROD UNDER BOLSTER WITH DEAD LEVER ANCHORED TO CAR BODY

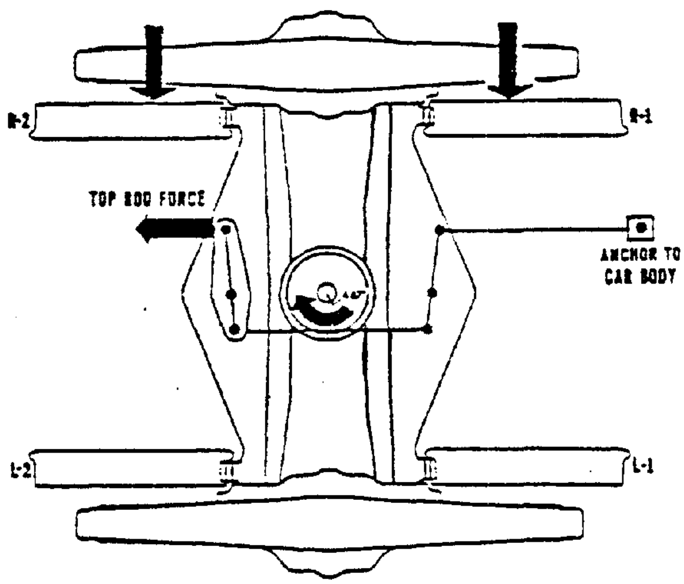


Figure 2-22

TRUCK TURNING MOMENT AND LATERAL FORCES
PRODUCED BY TWO CONFIGURATIONS OF
CONVENTIONAL BRAKE RIGGING

Source: WABCO (1975). Used by permission of copyright owner.

Interviews conducted with freight car wheel manufacturers and private car line leasing companies confirmed that for a large portion of wheels removed for worn flanges, the wear was pronounced on two wheels of the truck, one on each axle. This asymmetric wheel wear is said to be caused by the described truck turning moments produced by the conventional brake rigging.

A second problem area with the conventional type of brake rigging is the uneven brake shoe force distribution produced at various wheels of the car. By producing a higher than average brake shoe force on one wheel of the car compared to the others can tend to cause the wheels to slide if the car is empty, and severe thermal loadings can result on wheels if the car is loaded.

Conventional rigging can also "bottom out" if the rigging is not manually adjusted when multiple-wear wheels are replaced by one-wear wheels. Depending upon the type of rigging, the additional brake shoe travel required for one-wear wheels can cause the rigging to come into contact with the truck bolster or the center sill, making the brakes useless.

2.9.2 Truck-Mounted Rigging (Direct Acting) - The truck-mounted type of brake rigging combines the functions of the brake cylinder, brake rigging, and brake beams into a compact truck-mounted arrangement. Figure 2-23 illustrates the truck-mounted brake rigging designed by Wabco. (This rigging has the trade name of "WABCOPAC". Similar rigging produced by New York Air Brake Company has the trade name "NYCOPAC".)

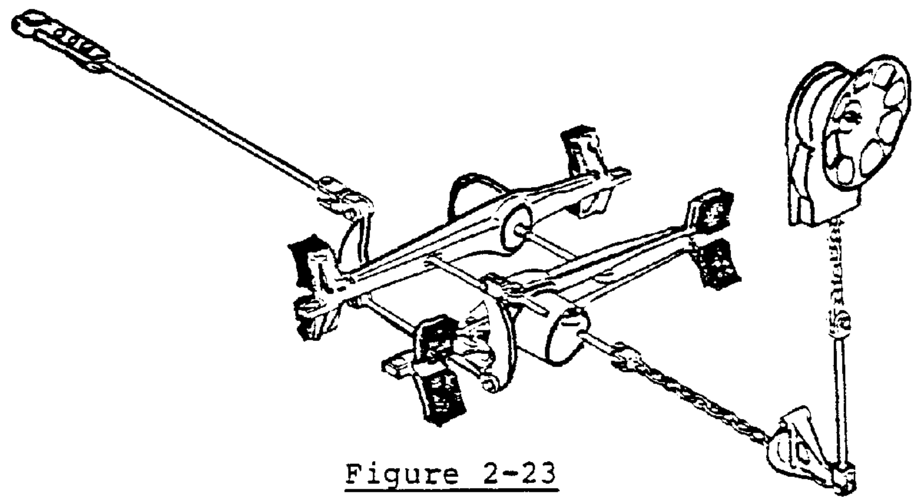


Figure 2-23

ILLUSTRATION OF TYPICAL TRUCK-MOUNTED BRAKE RIGGING (ONE TRUCK SHOWN)

Source: The Air Brake Association (1972). Used by permission of copyright owner.

As shown in Figure 2-23, the truck-mounted brake rigging requires two brake cylinders per truck (four cylinders per car).

Because the brake cylinders act directly upon the brake beams, the efficiency losses inherent in the conventional type of rigging are not present. Truck-mounted brake rigging efficiency typically ranges between 80 and 90 percent.

Other benefits associated with the truck-mounted type of brake rigging include the absence of truck turning moments resulting from brake application. As shown in Figure 2-24, the truck-mounted brake rigging produces symmetrical application of braking forces.

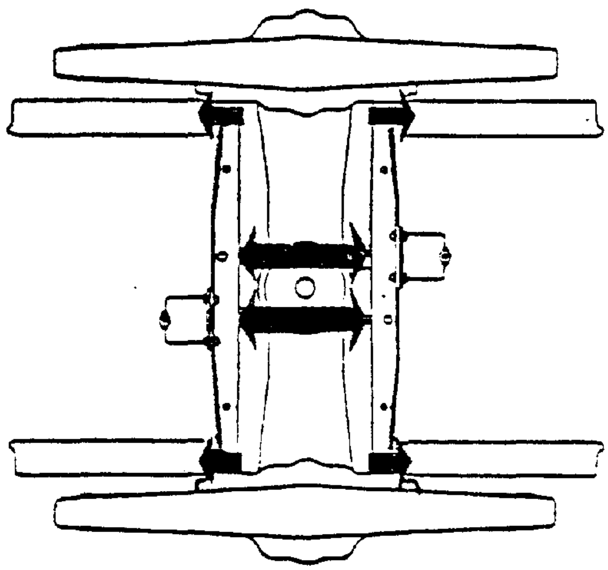


Figure 2-24

FORCE DISTRIBUTION OF TRUCK-MOUNTED (DIRECT-ACTING) BRAKE RIGGING

The more uniform force distribution produced by the truck-mounted brake rigging serves to reduce the tendency for wheel sliding on empty cars and high wheel thermal loadings on only one wheel of a car.

A potential brake system design problem with the truck-mounted brake rigging is that, even though it is more efficient than conventional rigging, the effective lever ratio is only 2:1 compared to lever ratios of up to 12.5:1 for conventional brake rigging.

In order to meet the Net Braking Ratio requirements for high capacity cars using truck-mounted rigging, high friction brake shoes are required for use in conjunction with large-diameter brake cylinders*.

Operating disadvantages associated with the truck-mounted type of brake rigging have included the more costly COT&S expenses because of the maintenance required on the three additional brake cylinders and the damage to brake cylinders on coal hopper cars caused by exposure to flames in thawing sheds. The cylinders are also frequently damaged when cars are derailed.

A more serious problem, which has recently been corrected by the manufacturer, has been the hand brake arrangement of the truck-mounted rigging. Until recently, the hand brake of that rigging applied the brakes on only one end (the "B" end) of the car. To provide adequate holding capability on one truck, the hand brake force had to be substantially greater than that produced on a truck of a car equipped with conventional rigging. Furthermore, inspection of the car by various ground crews would often result in the incorrect interpretation that the released brakes of the "A" end of the car meant that the hand brake of the car had been released. The resulting relatively high frequency of unreleased hand brakes, combined with the substantially higher hand brake forces, produced a high incidence of "B" end wheel removals due to slide, spalling and built-up tread. The correction has been to provide hand brake rigging that applies to both trucks of the car (Wabco, 1975).

2.10 HAND BRAKES

The main purpose of the hand brake is to hold the car stationary for long periods of time, although hand brakes are also used in a limited number of operations to control manually the speed of cars during switching and classification to reduce car impact forces.

Hand brakes on freight cars can be generally described in terms of their means of physical application. These include a vertical wheel assembly, or a horizontal wheel (usually a drop shaft) assembly, or a lever type assembly.

---

* Alternate versions of truck-mounted brakes use one cylinder per car and a lever arrangement to produce the brake shoe forces.

Figure 2-25 illustrates the typical vertical wheel hand brake assembly.

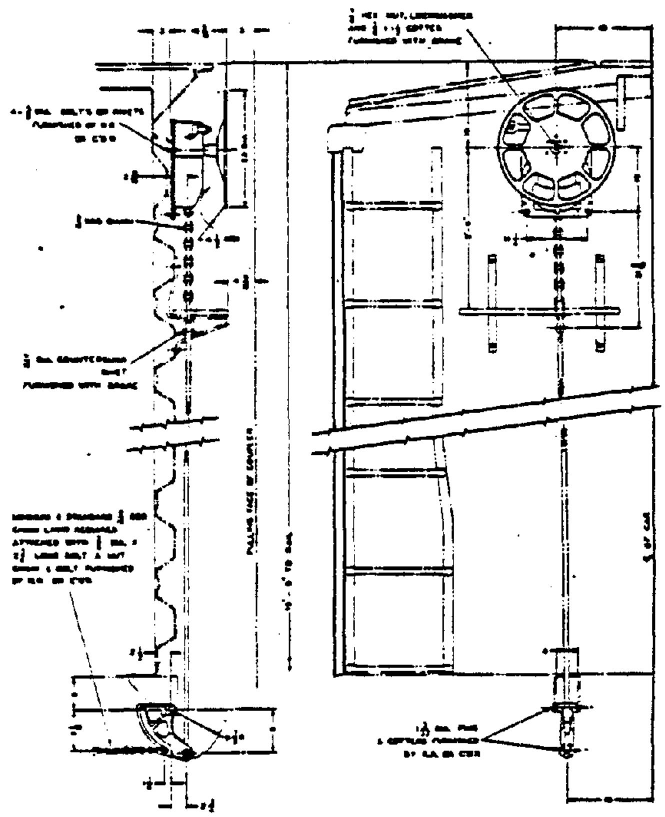


Figure 2-25

VERTICAL WHEEL HAND BRAKE ASSEMBLY

Source: Simmons-Boardman (1970). Used by permission of copyright owner.

The hand brake is manually operated by a crew member rotating the wheel which winds the chain around a sprocket. The sprocket is geared to the wheel and restrained by a ratchet or a reduction gear. The resulting force is transferred to the brake cylinder or brake rigging through a combination of sheave wheels and bell cranks, depending upon the design of the car and the type of brake rigging.

The hand brake must be released manually by activating a release lever, or by unwinding the wheel, depending upon the design. Unfortunately, the releasing of the hand brake is sometimes overlooked, and the car is moved in the train or within the yard with the hand brake applied. Depending upon whether the car is empty or loaded, the wheels might slide or "skip", causing flat spots or high tread temperatures, respectively.

The performance specifications for hand brakes are published in Association of American Railroads (1975b). These specifications vary according to the types of brake rigging and brake shoes installed on the car.

2.11 SLACK ADJUSTER

The purpose of the slack adjuster is to remove slack in conventional brake rigging due to brake shoe and other wear so that the proper brake cylinder piston travel is maintained. Proper piston travel must be maintained so that the pressure-volume relationship between the air reservoir and the brake cylinder is not out of balance. With brake rigging lever ratios of 8:1, a shoe wear of 1/8 inch can cause a 1-inch piston travel extension. This is the reason the FRA and the AAR require less than a 9-inch piston travel at the initial terminal, and that brake shoe condemning limits are specified.

It should be noted that, while proper piston travel is important, piston travel is not an indication of the total condition of the entire brake rigging. Provisions typically are provided on brake rigging levers and rods to correct lever angularity or interference by manually relocating pin placement to "square up" the rigging.

Figure 2-26 illustrates the operation of one type of automatic slack adjuster.

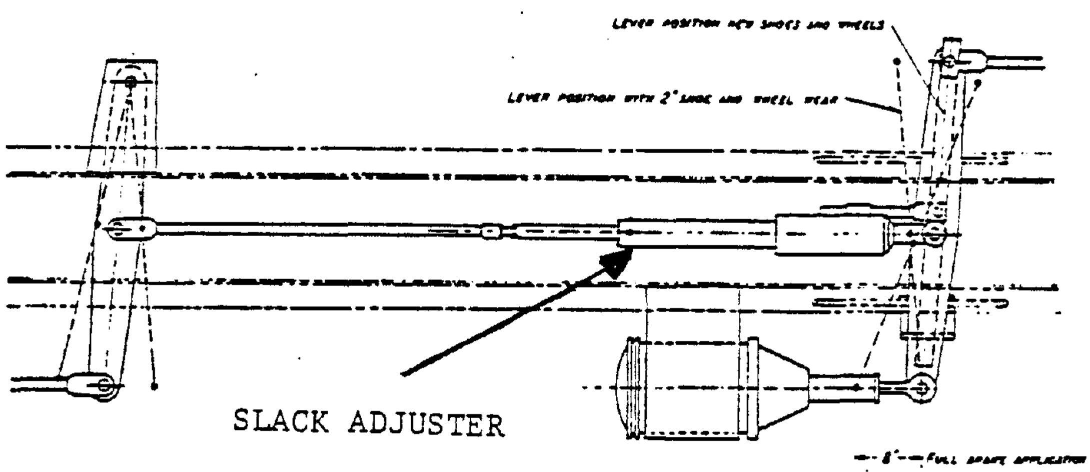


Figure 2-26

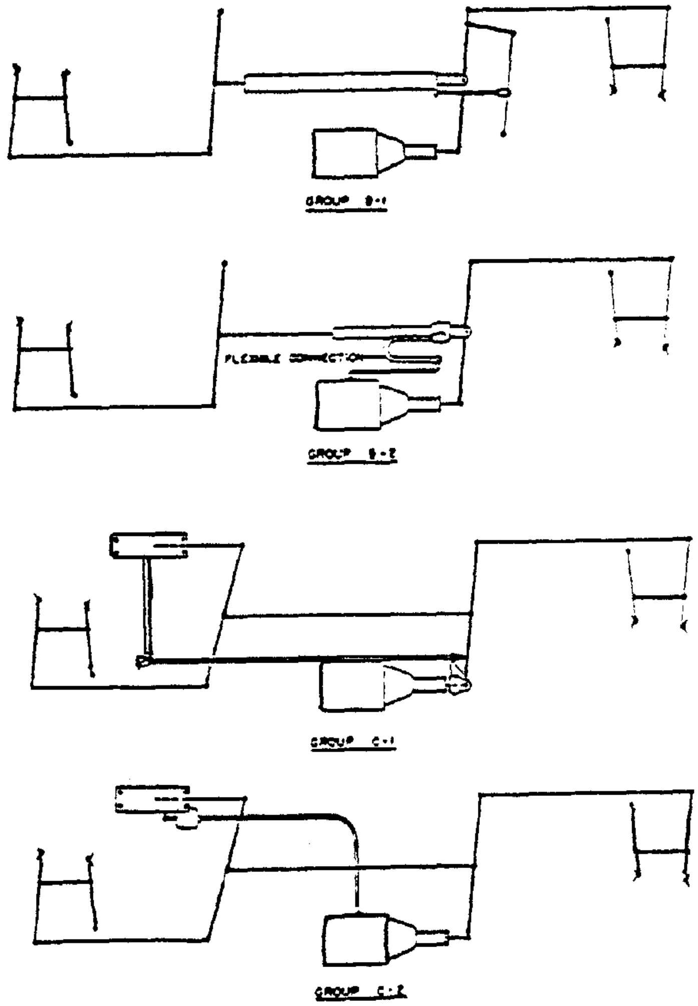
AUTOMATIC SLACK ADJUSTER
OPERATION

Source: Simmons-Boardman (1970). Used by permission of copyright owner.

The mechanical slack adjuster shown in Figure 2-26 consists of an enclosed long spiral worm which is rotated by an attached spring. The unit is designed to shorten automatically when the brakes are released to adjust for brake shoe wear. When worn brake shoes are replaced with new ones, the unit automatically lengthens on the first brake application to provide proper piston travel.

Slack adjusters are grouped according to their point of application to the brake rigging, and by their type of operation (mechanical or pneumatic) as shown in Figure 2-27.

The performance specifications and certification procedures for automatic slack adjusters are published in Association of American Railroads (1976, pp. 57-63).



A.A.R. typical applications of automatic slack adjusters. Group B includes adjusters as applied to center (cylinder) rod. Group B-1 is Mechanical Type; Group B-2 is Pneumatic Type. Group C includes adjusters mounted as floating lever fulcrum. Group C-1 is Mechanical Type; Group C-2 is Pneumatic Type.

Figure 2-27

TYPICAL AAR APPLICATIONS OF AUTOMATIC SLACK ADJUSTERS

Source: Simmons-Boardman (1970). Used by permission of copyright owner.

2.12 BRAKE SHOES

A number of different types of brake shoes are presently used in freight service. These shoes differ primarily in their materials of construction, which alter their frictional, wear, and sparking characteristics.

The standard cast iron brake shoe is often referred to as a "low friction" shoe. This shoe, which has been in service since 1935, is presently being phased out by the introduction of the high-phosphorus, ("Hi-Phos") cast metal shoe.

In 1965 the AAR adopted an alternate type of brake shoe known as the composition shoe. This shoe, which is also referred to as a "high friction" shoe, was developed partly to be used with the truck-mounted brake rigging, which because of its lower effective lever ratio, required a higher friction shoe to achieve the required level of braking. It should be noted, however, that all composition shoes are not high friction shoes; low friction composition shoes are also manufactured and are used primarily on locomotives, which typically have higher brake shoe forces than freight cars. A summary of brake shoe terminology is shown in Table 2-4.

TABLE 2-4

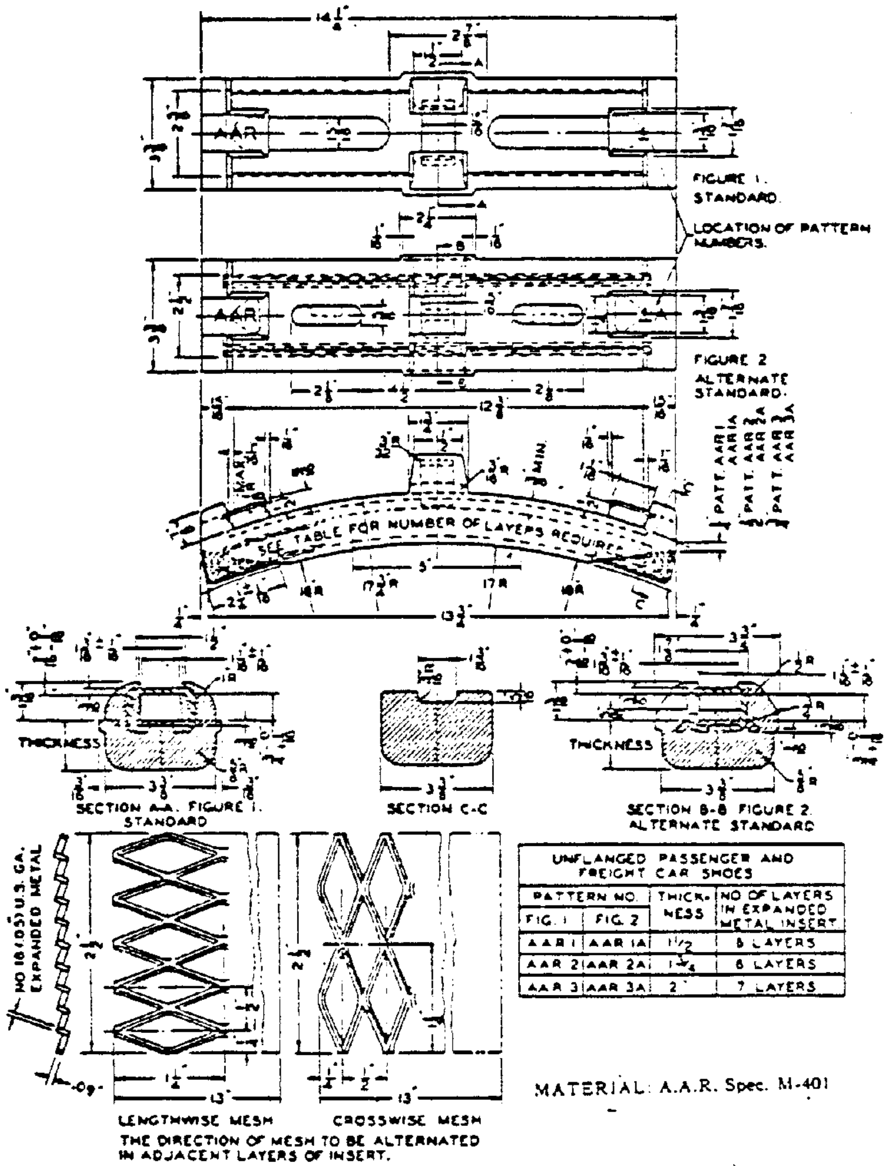
SUMMARY OF BRAKE SHOE TERMINOLOGY

Materials	"High Friction"	"Low Friction"
Metal		
Cast Iron	No	Yes
"Hi-Phos"	No	Yes
Composition	Yes	Yes

Table 2-4 illustrates the terminology problem with brake shoes, for the purposes of this section the following meanings are used.

"Cast Iron" : Low friction, metal
"Hi-Phos" : Low friction, metal
"Composition": High friction, composition (low friction is ignored unless otherwise specified).

2.12.1 Cast Iron Shoe - The cast iron brake shoe, designated the AAR-1, was adopted in 1935. The material specifications of the cast iron shoe are published in AAR Specification M-401-56. Figure 2-28 illustrates the dimensions and pattern numbers for different cast iron shoe thicknesses.



UNFLANGED PASSENGER AND FREIGHT CAR SHOES			
PATTERN NO.		THICKNESS	NO OF LAYERS IN EXPANDED METAL INSERT
FIG. 1	FIG. 2		
AAR 1	AAR 1A	1 1/2	5 LAYERS
AAR 2	AAR 2A	1 3/4	6 LAYERS
AAR 3	AAR 3A	2"	7 LAYERS

Figure 2-28

AAR REINFORCED METAL BRAKE SHOES

Source: Association of American Railroads (1978b, p. E-93-1972). Used by permission of copyright owner.

The functional performance of the cast iron brake shoe involves the friction characteristics it has relative to steel freight car wheels. Frictional values vary considerably depending upon operating conditions, and to some extent by brake shoe forces; in particular, they are directly affected by speed. The values of coefficient of friction shown in Table 2-5 can be considered an approximate range for cast iron brake shoes.

TABLE 2-5

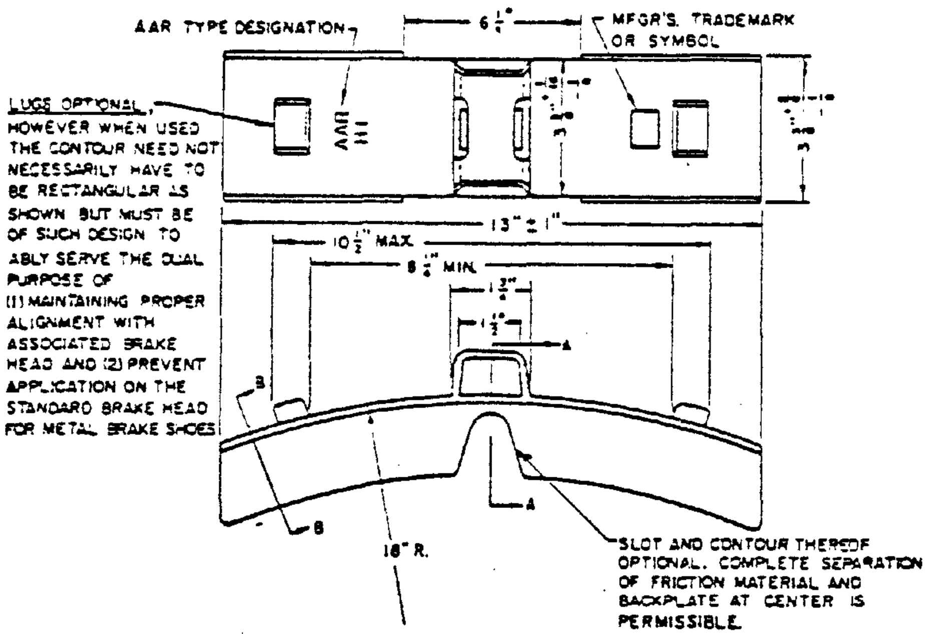
APPROXIMATE FRICTIONAL CHARACTERISTICS OF CAST IRON BRAKE SHOES AS A FUNCTION OF TRAIN SPEED

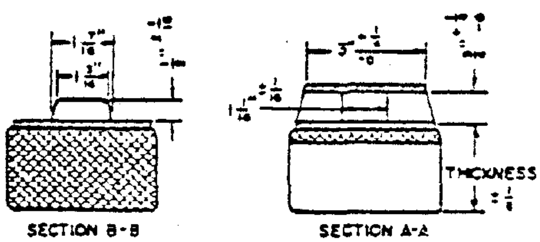
Train Speed (Mph)	Approximate Coefficient of Friction (Holloway, 1955)	Approximate Coefficient of Friction Calculated*
0	-	0.29
5	0.27	0.22
7.5	0.24	-
10	0.24	0.19
20	0.19	0.13
50	0.12	-

Note: * Calculated using breakaway test data. AAR specification M-401-56 stipulates a minimum coefficient of friction at 40 mph of 0.18.

2.12.2 Composition Shoe - The composition brake shoe was developed in the early 1960's to provide a higher frictional level required for the truck-mounted brake rigging, and to provide improved wheel wear and thermal capacity characteristics (see Section 5) as compared to the cast iron shoe. The specifitions for the high friction composition brake shoe are published in AAR Specification M-926-72.

Figure 2-29 on the next page illustrates the dimensions of the composition shoe and the AAR shoe type designations.





TYPE DESIGNATION WITH GUIDE LUGS	THICKNESS	TYPE DESIGNATION WITHOUT GUIDE LUGS
AAR H1	1 1/4"	AAR H1A
AAR H2	1 1/2"	AAR H2A
AAR H3	1 3/4"	AAR H3A
AAR H4	2"	AAR H4A
LETTER "H" INDICATES HIGH FRICTION. NUMERALS INDICATE THICKNESS		

Figure 2-29

HIGH FRICTION COMPOSITION TYPE BRAKE SHOE

Source: Association of American Railroads (1978b, p. E-94A-1971). Used by permission of copyright owner.

The functional performance features of composition brake shoes indicate that in addition to the overall higher frictional values, the coefficient of friction increases from zero train speed compared to the cast iron shoe, which has a coefficient of friction that decreases with train speed. Table 2-6 illustrates approximate values of the coefficient of friction of composition (high friction) brake shoes. These values were calculated using test data.

TABLE 2-6

APPROXIMATE VALUES OF THE COEFFICIENT OF FRICTION OF COMPOSITION (HIGH FRICTION) BRAKE SHOES

Train Speed (Mph)	Approximate Coefficient of Friction
0	
5	
10	0.30
15	0.30
20	0.30
25	0.30
30	0.30

Source: Calculated from breakaway test data.

Extensive dynamometer tests have demonstrated the increased wheel thermal capacities achievable using composition brake shoes as compared to cast iron brake shoes (See Section 5). Results of interviews with professionals in the industry generally support the conclusions of this research. Brake shoe and wheel wear are also said to be greatly reduced with the use of composition brake shoes.

2.12.3 Hi-Phos Shoe - In an effort to reduce the sparking characteristics of the cast iron shoe, the high-phosphorus content cast metal shoe has been introduced as an alternate to the standard cast iron shoe.

The frictional characteristics of the hi-phos shoe are approximately the same as the cast iron shoe. Table 2-7 on the following page illustrates approximate values of the coefficient of friction of the hi-phos shoe. These values were calculated using breakaway test data.

TABLE 2-7

APPROXIMATE VALUES OF THE COEFFICIENT OF FRICTION OF "HI-PHOS" BRAKE SHOES

Train Speed (Mph)	Approximate Coefficient of Friction*
0	
5	
10	0.25
15	0.17
20	0.15
25	0.15
30	0.14

Note: * Calculated using breakaway test data. The minimum static AAR required coefficient of friction is 0.45.

A graphical comparison of the friction characteristics of the three types of brake shoes discussed in the preceding is presented in Figure 2-30. The other functional performance characteristics are compared in Table 2-8.

TABLE 2-8

OTHER FUNCTIONAL PERFORMANCE CHARACTERISTICS OF "HI-PHOS" AND COMPOSITION BRAKE SHOES

	Hi-Phos	Composition
Wear Characteristics (compared to Cast Iron)	Improvement reported in brake shoe wear.	Significant improvement reported in both shoe and wheel wear.
Sparking Characteristics (compared to Cast Iron)	Greatly improved--almost no sparking.	Virtually sparkless.
Wheel Thermal Capacity (compared to Cast Iron)	Unknown.	Improved (Section 5).

Brake shoes are replaced in the field by removing a spring type brake shoe key which holds the brake shoe to the brake head of the brake beam. After the new brake shoe has been positioned, the key is vertically reinserted through slots provided in the brake head and the back of the brake shoe (see Figure 2-28). Two types of brake shoe keys are shown in Figure 2-30.

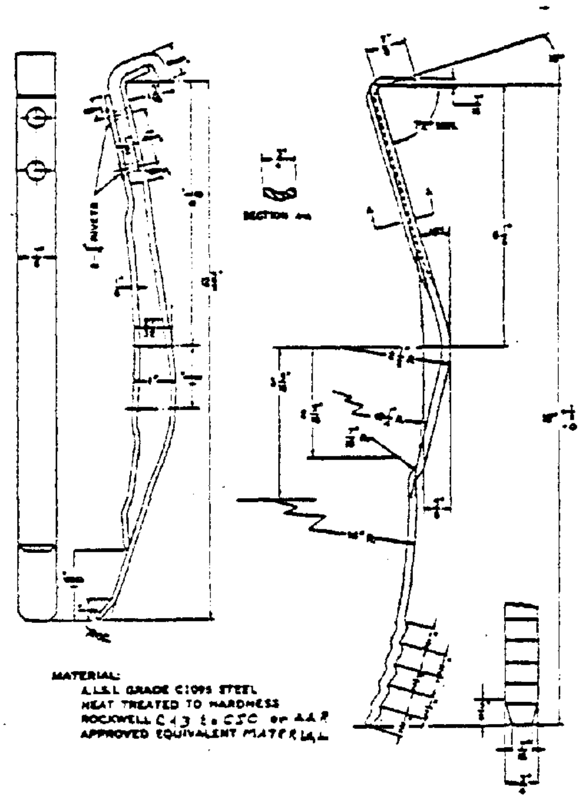


Figure 2-30

TWO TYPES OF BRAKE SHOE KEYS

Source: Association of American Railroads (1978b. p. E-94-1972). Used by permission of copyright owner.

Because of the substantially greater frictional level of the high friction composition brake shoes, it is important that the composition shoe is not misapplied to a car designed for low friction shoes. Although this can presently occur physically, procedures are in effect to prevent the misapplication of brake shoes. Furthermore, rules which will become effective in 1977 will make it physically impossible to misapply brake shoes. The procedures in effect presently, and those which will become effective in the future, are shown in Table 2-9 on the following page.

TABLE 2-9

SUMMARY OF RULES DESIGNED TO PREVENT THE MISAPPLICATION OF BRAKE SHOES (AS OF SEPTEMBER 16, 1975)

Freight Car Component	Description of Rule	Freight Cars Affected	Date in Effect	Source of Rule
Brake Beam - New Installation	Brake Beam Head E-84B rejects lug of high friction composition brake shoe	New and rebuilt	Presently	Office Manual - Interchange Rules - 88B
		Cars Modified for Composition Shoes	Presently	Office Manual - Interchange Rules - Rule 88D
Brake Beam - Renewal	Brake beams renewed with composition brake shoes must be renewed with brake heads that reject cast iron shoes	All cars equipped with composition shoes	Presently	Field Manual - Interchange Rules - Rule 6B
Brake Shoes - Renewal	All high friction composition brake shoes in use must have rejection lugs that prevent their installation on brake beams equipped with cast iron brake shoes	All cars	January 1, 1977	Office Manual - Interchange Rules - Rule 88D
Brake Shoes - Manufactured	All high friction brake shoes must be manufactured with rejection lugs that will reject them from brake beam equipped with cast iron brake shoes	Not Applicable	January 1, 1977	Manual of Standards and Recommended Practices Specification G-926-72

Source: Summarized from Cabble (1975).

3. OPERATIONS, PERFORMANCE, AND DESIGN OF FREIGHT CAR AIR BRAKE SYSTEMS

3.1 INTRODUCTION

This section presents a discussion of the operation, performance, and problems of freight car braking systems as they exist today. The material covered under each of these topics is summarized below:

- Operations: This section presents a detailed functional description of brake system components, brake rigging, brake system design, and train brake operations.

- Performance: This section discusses the regulatory requirements applicable to brake systems and describes their impact on train brake performance.

- Problems: This section delineates problems associated with air brake systems from the standpoint of yard processing and train handling. Also included are recommendations concerning the training of enginemen.

3.2 OPERATIONS

It is the purpose of this section to describe the operation of air brake systems in terms of freight train handling. First a functional description of air brake system components is presented, then the various kinds of brake rigging are discussed. This is followed by a brief discussion of air brake system design and train brake operations.

3.2.1 Brake System Components - The most common freight car brake system component arrangement used on freight cars today is shown in Figure 3-1.

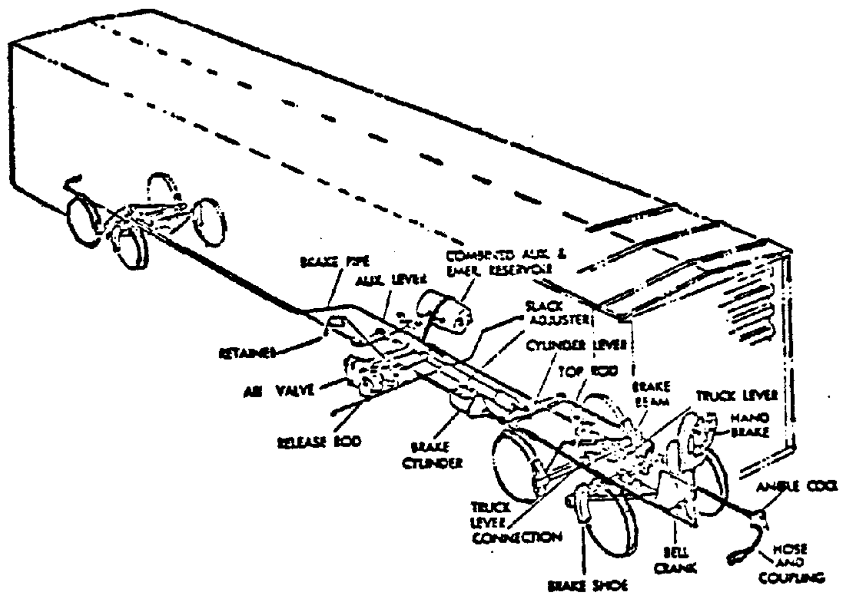


Figure 3-1

TYPICAL ARRANGEMENT OF FREIGHT CAR BRAKE SYSTEM COMPONENTS

Source: Air Brake Association (1972, p. 44).
Used by permission of copyright owner.

The freight car shown in Figure 3-1 above is a typical four-wheel truck boxcar equipped with an "AB" type brake arrangement. The pneumatic segment comprises the "AB" operating control valve, combined auxiliary and emergency reservoirs, a brake cylinder and a pressure retaining valve, all of which are permanently mounted to the car body.

The mechanical segment comprises a brake cylinder lever, a slack adjuster, brake rods, truck levers, brake beams, brake heads, and brake shoes. The hand brake is also permanently mounted to the car body and connected to the live cylinder lever by means of a chain, bell crank, and connecting rod.

The 1-1/4 inch diameter brake pipe extends from end to end of the car and has connecting hoses and angle cocks mounted on each end of the car. It is through this pipe that compressed air travels from the locomotive to each car and through the respective freight car control valves to charge the auxiliary and emergency reservoirs. Approximately 6,000 cubic inches of compressed air are stored in the two reservoirs on each car as braking energy.

Figure 3-2 illustrates a typical piping arrangement found on freight cars today.

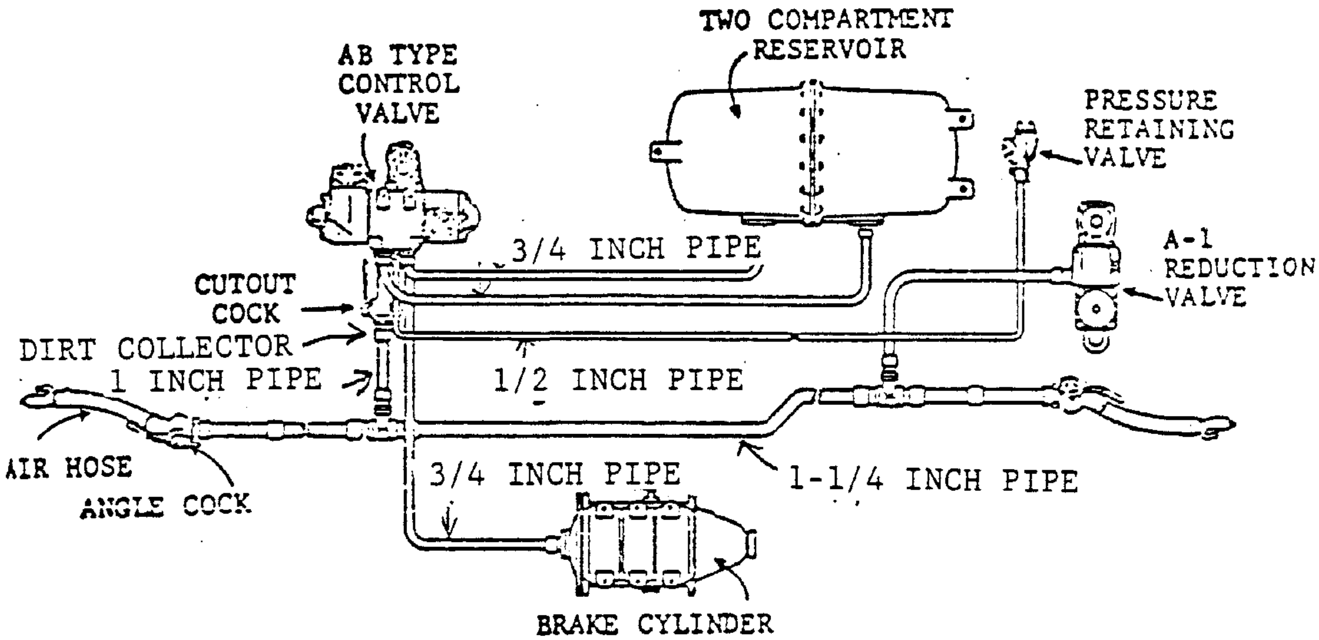


Figure 3-2

TYPICAL AIR BRAKE PIPING ARRANGEMENT

Source: Air Brake Association (1972). Used by permission of copyright owner.

.Figure 3-2 shows the 1-1/4 inch brake pipe extending from end-to-end on the car and from a tee connection. Brake pipe air from the tee connection passes through a 1-inch brake pipe branch pipe, through a bowl type dirt collector, through a cut-out cock, and into the pipe bracket to which the control valve is mounted. Air continues to flow through a micro type filter in the pipe bracket, and then passes through the service portion of the control valve in charging the two air reservoirs. The pipes extending from the pipe bracket to the reservoirs and brake cylinder are 3/4 inch in diameter, while the pipe to the retainer valve is 1/2 inch in diameter. All diameters, pipe lengths and bend radii must conform to installation requirements approved by the Association of American Railroads (AAR).

The A-1 reduction relay valve shown is currently required on freight cars having a total length of brake pipe of 65 feet or longer. The purpose of this valve is to assist the control valve in propagating service and/or emergency rates of brake pipe reductions.

The pressure retaining valve is used by railroads where descending grade conditions require a partial brake application while the car reservoirs are being recharged. The pressure retaining valve shown in Figure 3-2 is currently mandatory on all freight cars offered for interchange service. With the retainer valve handle in down position, air from the brake cylinder is directed to the atmosphere, following the release of the train brakes, without restriction. The modern retaining valve has three positions, namely, release, high pressure, and slow-direct position. When the valve is placed in the High Pressure ("HP") position, a pressure of 20 psi is retained in the brake cylinder when the brakes are "released." This provides a minimum constant brake application on a train while descending a steep grade. When the retainer valve is placed in the Slow-Direct position, brake cylinder pressure is completely exhausted when the brakes are released, but at a slow rate.

The retainer valves must be manually operated individually on each car and must be properly positioned before descending the grade and manually re-positioned to the "Release" position at the bottom of the grade.

The brake cylinder shown in Figure 3-2 has an inside diameter of 10 inches, and a length of 12 inches. The cylinder must be installed on the car so an 8-inch piston travel will position the live cylinder lever at a 90-degree angle. (The 90-degree angle is the most efficient position for transferring mechanical forces.) It is important to note that the 8-inch piston travel is necessary to retain the proper volume relationship between the air reservoirs and the brake cylinder during all degrees of brake applications. Pressure developments within the cylinder with respect to train brake pipe reductions are discussed below.

The plain AB brake control valve was first introduced to the railroad industry in 1921 and adopted as a standard in 1933. It is still acceptable in interchange service. It should also be noted that because of material and car shortages during the Second World War, the older "D" type triple valve was not excluded from interchange service until about 1950.

The ABD type valve was introduced in 1962 and was granted unlimited approval for interchange service in 1966. It was adopted as the standard valve for all new freight cars in 1974.

The principal operating changes introduced with the ABD valve were: (1) a brake cylinder release feature and (2) an accelerated release feature. Also, as a result of many coordinated tests and examinations conducted by the AAR, the ABD control valve apparently has operated with satisfactory reliability to receive a 10-year cleaning period (COT&S) versus the 4-year cleaning period now required with the AB valve.

With the endeavor to improve slack control and reduce stopping distance during service braking, a new feature known as "Continuous Quick Service" has been incorporated as an integral part of the new ABDW valve introduced to the rail freight industry in 1974. This valve is presently being evaluated by the AAR for performance reliability. To date it has not been granted AAR unlimited approval for interchange service. Figure 3-3 below shows a schematic view of the ABDW valve. The features of this control valve will be discussed later in this section.

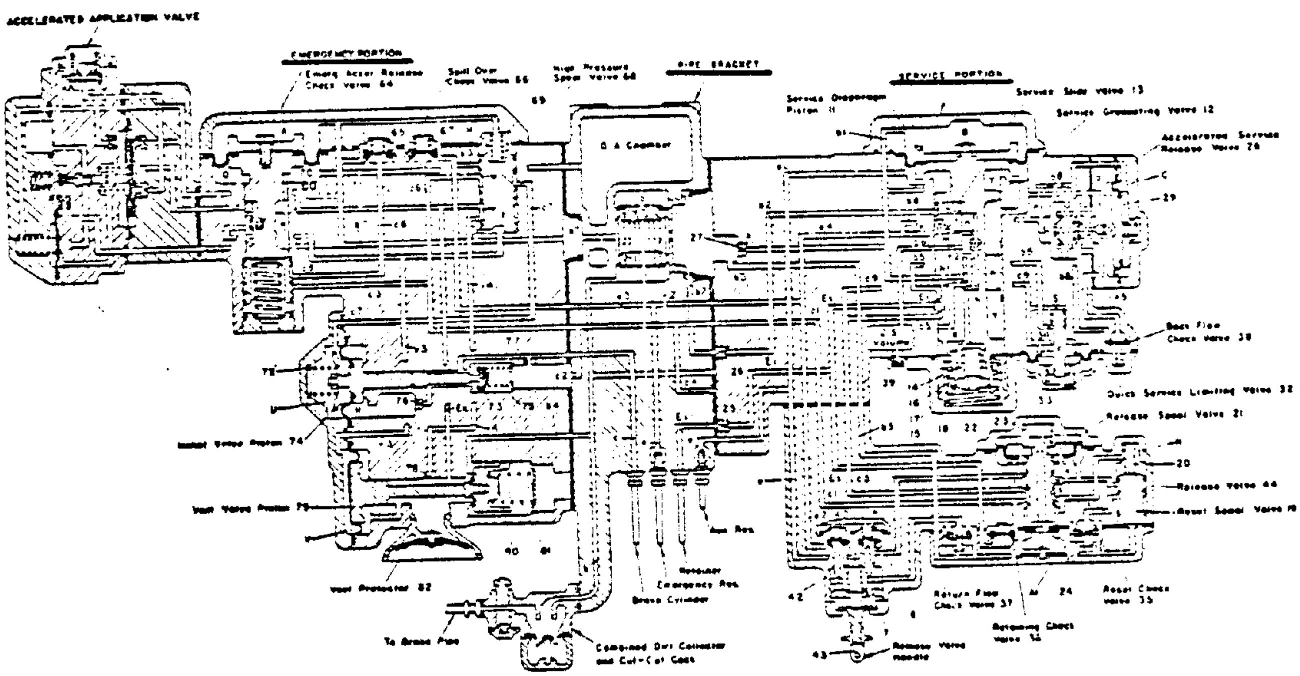


Figure 3-3

ABDW CONTROL VALVE WITH INTEGRATED CONTINUOUS QUICK SERVICE

Source: Westinghouse Air Brake Company Presentation to the Air Brake Association Annual Meeting, September 16, 1975. Used by permission of copyright owner.

3.2.2 Brake System Rigging - The brake rigging provides the mechanism to transfer and apply the pneumatic forces produced by the brake cylinder to the wheels of the freight car. Figure 3-4 below shows three different brake rigging arrangements.

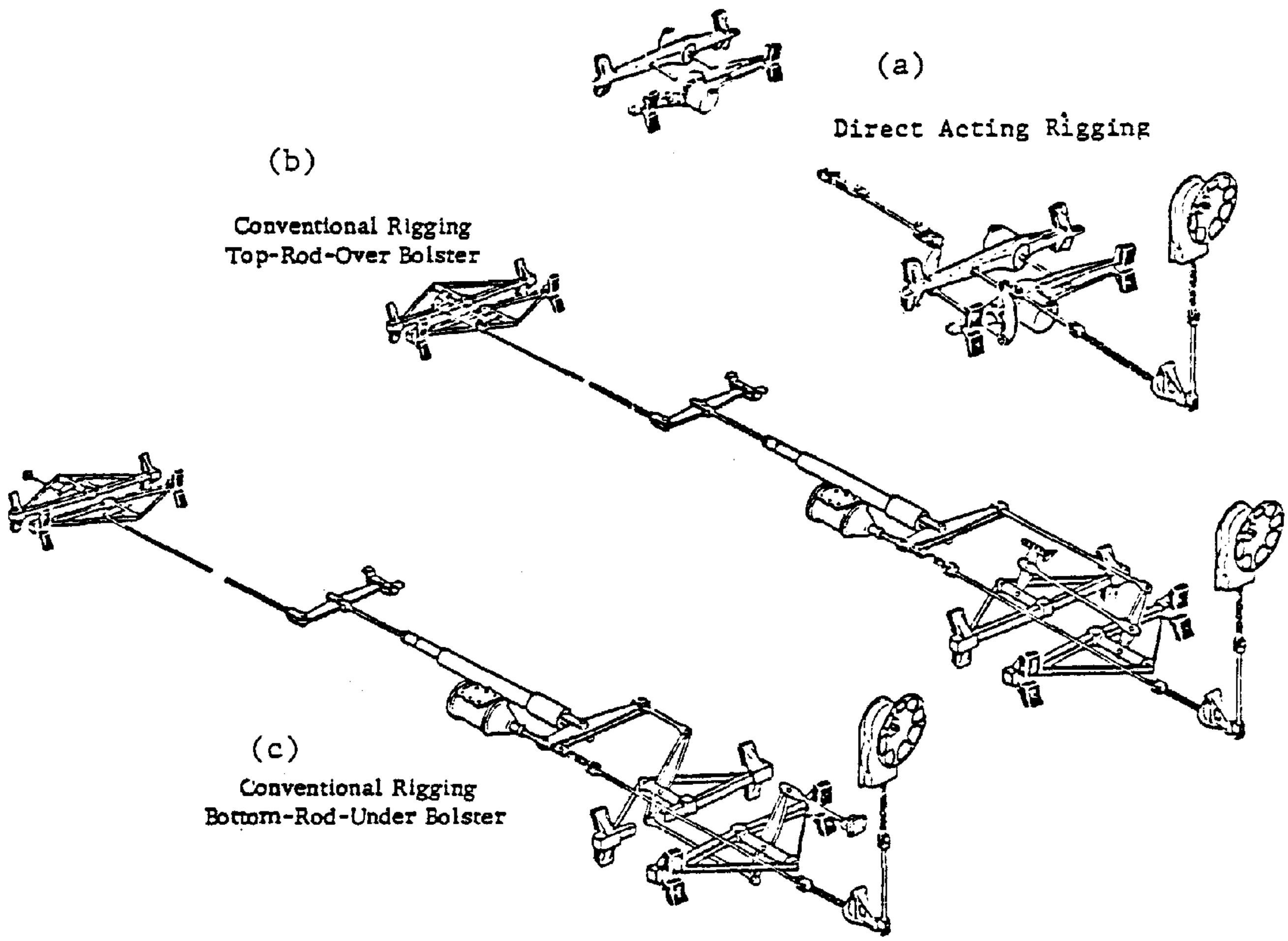


Figure 3-4

THREE TYPES OF BRAKE RIGGING ARRANGEMENTS

Source: WABCO (1974). Used by permission of copyright owner.

In Figure 3-4, (a) is a direct acting type of rigging, employing one cylinder for each brake beam, with the hand brake functional on only one truck*. Having no slack adjuster for controlling piston travel, it is necessary that the brake shoe thickness not exceed 1-1/4 inches when making initial installation. This is necessary to prevent excessive piston travel after the brake shoes have become worn and is part of the design to maintain the proper volume relationship between the air reservoirs and the brake cylinders.

View (b) of Figure 3-4 shows a conventional rigging, having one brake cylinder, connecting rods to the levers, and arranged so the top rod is over the truck bolster.

View (c) of Figure 3-4 is similar to (b) except it is designed so that the connecting rods are located under the truck bolster. Both (b) and (c) show conventional rigging with the hand brake functional on both trucks.

Figure 3-5 compares the force diagrams of the conventional and direct acting brake rigging arrangements.

---

* Hand brake rigging that applies to both trucks is presently available.

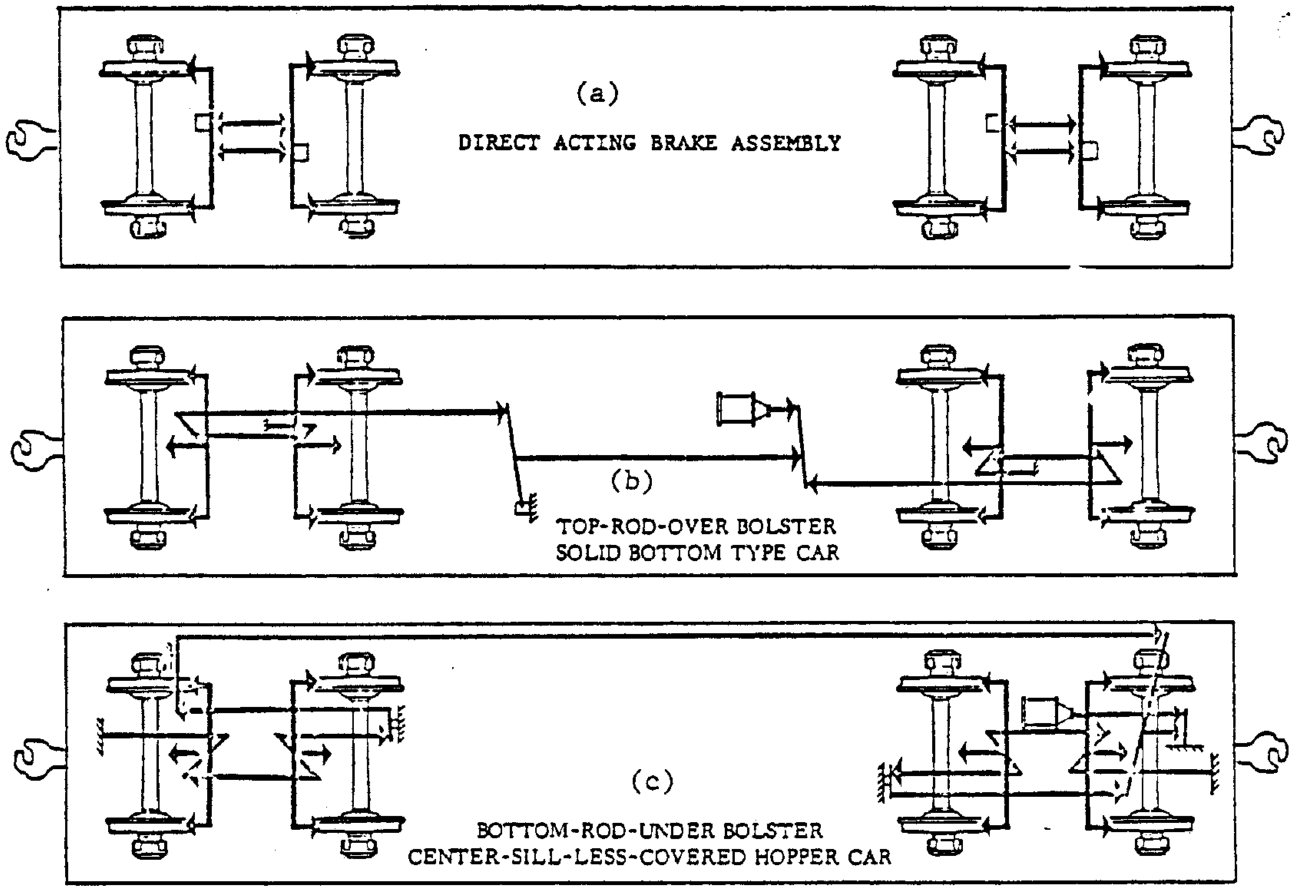


Figure 3-5

BRAKE RIGGING FORCE DIAGRAMS (BOTTOM VIEW)

Source: WABCO (1974). Used by permission of copyright owner.

A comparison can be made in Figure 3-5 between the force distributions of the direct-acting and the conventional riggings and between the top-rod-over and the bottom-rod-under types of conventional rigging.

Direct-acting and conventional rigging arrangements exhibit differences in efficiency losses and leverage ratios. Efficiency losses occur within the brake cylinders themselves and from each pivot point or pin connection of the rigging. As shown in Figure 3-5, the direct-acting rigging has four brake cylinders, as compared to one brake cylinder with the conventional rigging. This produces larger brake cylinder efficiency losses for the direct-acting type of rigging. However, the conventional rigging has considerably more rigging pivotal and pin connections than the direct-acting rigging. As a result the direct-acting rigging is generally considered to have a lower net efficiency loss.

The overall lever ratio that can be produced by the direct-acting rigging (without multiplying levers) is less than that which can be produced by the conventional rigging. A theoretical lever ratio of 12.5:1 is considered a maximum for conventional rigging. The direct-acting rigging using double-acting cylinders has a "lever ratio" equivalent of 2:1.

3.2.3 Brake System Design - The design of the brake system for a freight car must include many considerations. For example, depending upon the type of car, conventional rigging might not be a possible alternative because of lack of space for the associated rods and levers (e.g., tank cars without center sills). However, the primary consideration in brake system design is the determination of the proper level of brake forces that should be applied to the particular car when it is both loaded or empty.

In determining braking forces for various modes of braking, parmount concern must be given to wheel-to-rail adhesion limits and wheel thermal capacities. Realizing the importance of these two specific criteria, the AAR has adopted Performance Specifications which must be complied with when designing brake systems for new freight cars.

To determine the design level of brake shoe forces, the light weight and the gross rail load of the car must be known. Also, the preferred type of brake shoe (high friction "composition" or low friction metal) must be specified in advance. Using this information, the brake system is designed to produce the net brake shoe force specified by the AAR in terms of the empty and loaded car weights. This value of brake shoe force, when divided by the car weight, is known as the Net Braking Ratio.

Prior to the adoption by the AAR of the Net Braking Ratio as a design parameter, the theoretical braking ratio was used and referred to as simply the Braking Ratio. The theoretical Braking Ratio could be calculated using engineering equations as follows:

$$BR = \frac{\text{TOTAL THEORETICAL BRAKE SHOE FORCE}}{\text{CAR WEIGHT}}$$

BR CAN BE CALCULATED (BASED ON 50 PSI BRAKE CYLINDER PRESSURE).

$$BR = \frac{PANL}{W}$$

WHERE:
P = BRAKE CYLINDER PRESSURE (PSI)
A = BRAKE CYLINDER AREA (SQ. IN.)
N = NUMBER OF CYLINDERS
L = LEVER RATIO
W = CAR WEIGHT (LBS.)

Thus, the Braking Ratio is "theoretical" in the sense that efficiency losses of the brake cylinder and brake rigging are not considered in the calculation.

Because of the many variations of brake system design, a more exact calculation of the design level of brake shoe forces was needed. The design parameter known as Net Braking Ratio (NBR) is currently in use and is defined as follows:

$$NBR = \frac{PANEL}{W}$$

W = CAR WEIGHT
NBR = NET BRAKING RATIO
P = BRAKE CYLINDER PRESSURE
A = BRAKE CYLINDER AREA
N = NUMBER OF BRAKE CYLINDERS
E = RIGGING EFFICIENCY
L = LEVER RATIO (12.5:1 MAX)

CALCULATED BASED ON 50 PSI BRAKE CYLINDER PRESSURE

Thus, the brake rigging efficiency is included in the calculation of Net Braking Ratio.

As mentioned above, the AAR requires all new cars to meet a specified Net Braking Ratio, both loaded and empty, and depending upon the type of brake shoes selected. The AAR Net Braking Ratio specifications are as follows:

A. CAST IRON (LOW FRICTION) BRAKE SHOES
[15% COEFFICIENT OF FRICTION AT 60 MPH WITH 50 PSI BRAKE CYLINDER PRESSURE:]

NBR_{GRL} 13%

NBR_{LW} 53%

B. HIGH FRICTION "COMPOSITION" BRAKE SHOES
[33% COEFFICIENT OF FRICTION AT 60 MPH WITH 50 PSI BRAKE CYLINDER PRESSURE:]

NBR_{GRL} 6.5%

NBR_{LW} 30%

WHERE: NBR = Net Breaking Ratio
GR = Gross Rail Load
LW = Light Weight

The net braking ratios actually produced under static conditions are sampled on new cars according to FRA requirements.

3.2.4 Train Air Brake Operation - This section discusses how the train air brake system responds to the various control inputs performed by the locomotive engineman.

Beginning with the charging system in the locomotive, air flows through the connecting brake pipe hoses between cars to each car in the train.

On a particular car, air passes through the branch pipe cutout cock, through the bowl direct collector, through the micro filter, and onto the face of both the service and emergency pistons of the brake control valve. Both pistons are forced downward into release/charging position. The auxiliary and emergency reservoirs are charged at a controlled rate governed by the charging chokes in the service portion of the control valve. This controlled rate of charging is necessary to alleviate a heavy drop in brake pipe pressure at any point in the train and also to keep the pressure as high as possible for a uniform propagation of release throughout the train. Through the emergency portion, brake pipe air charges the Quick Action Chamber located in the pipe bracket of the control valve assembly.

3.2.4.1 Brake Service Application - To apply the brakes, the engineman measures the amount of brake pipe reduction by observing his air gauges. In normal train handling, a minimum reduction is usually 6 psi. Therefore, assuming brake pipe pressure was reduced on the locomotive from 70 to 64 psi, the exhaust of brake pipe air will commence on the locomotive and continue until the pressure reaches 64 psi at the head end of the train. When this brake pipe reduction is felt on the outer surface of the service piston in the control valve on a car, the pressure beneath the piston moves the piston upward, causing the graduating valve to move, followed by the slide valve. Movement of the graduating valve permits a measured amount of brake pipe air to flow into the quick service volume of the control valve. (The exhaust activity in the control valve serves as an aid in reducing brake pipe pressure throughout the train during an application to assist in propagating the application throughout the train.) In sequence, brake pipe and auxiliary reservoir air also flow through the limiting valve to the brake cylinder(s) to apply the brakes.

The limiting valve prevents brake pipe air from entering the brake cylinder after the brake cylinder pressure reaches its limiting valve setting, which is approximately 10 psi. This feature is called "Preliminary Quick Service."

Additional service rate reductions may be made by the engineman by further reducing brake pipe pressure on the locomotive. However, when the pressure in the auxiliary reservoir and in the brake cylinder become equalized, the maximum brake cylinder pressure is reached. The equalization of auxiliary reservoir and brake cylinder pressures is known as a "Full Service" application.

It should be noted that for cars equipped with either the AB or ABD valves, all reductions in the air from the brake pipe must vent through a controlled orifice in the locomotive automatic brake valve. This requires a substantial amount of time to make a complete brake application on a long freight train. This is the reason that cars with a brake pipe length of 65 feet or longer are now required to have an A-1 Reduction valve installed in the brake pipe to assist in propagating service applications. The purpose of this feature is to provide continuous quick service by locally venting brake pipe air during all stages of brake pipe reductions. The ABDW control valve incorporates this feature as an integral part of the control valve.

The emergency portion on the left of the pipe bracket (see Figure 3-3) serves no real purpose during service application; however, brake pipe pressure does change on top of the emergency piston. To prevent the piston from moving to emergency position it is necessary that quick action chamber air vents from under the piston during service reductions. This air is vented to the atmosphere at the bottom of the emergency portion of the control valve.

With a brake pipe reduction, air pressure builds up in the brake cylinder as follows. Assuming an initial brake pipe pressure of 70 psi, the most pressure that could build up in the brake cylinder would be approximately 50 psi. This is due to the volume relationship between the auxiliary reservoir and the 10-inch brake cylinder, based on a piston travel of 8 inches. The ratio is about 2-1/2 to 1. For example, if brake pipe pressure is reduced by 10 psi to 60 psi, the volume of air released from the auxiliary reservoir would produce approximately 25 psi in the brake cylinder. If brake pipe pressure were reduced to 50 psi (a 20 psi reduction), the auxiliary reservoir pressure would be 50 psi, and the pressure in the brake cylinder would be 2-1/2 times greater than the reduction, or 50 psi. This would be considered a full service application because the auxiliary reservoir and brake cylinder pressures have equalized. An additional brake pipe pressure reduction (at a service rate) to a pressure lower than 50 psi would not affect a change in the auxiliary or brake cylinder pressures.

As mentioned before, the brake pipe running from the locomotive to the rear of the train is the only communication from the locomotive brake valve to the brake valves on the cars. This single pipe is used to charge the system, make all reductions, and release the brakes on the train.

3.2.4.2 Brake Release - To release the train brakes, the engineman places the automatic brake valve of the locomotive in release position, allowing brake pipe pressure to be restored to its governing setting (called "feed valve" on present locomotives).

When brake pipe pressure is increased about 1-3/4 psi higher than auxiliary reservoir pressure, the service piston of the car's control valve moves to release position, thus allowing emergency reservoir air to flow to the brake pipe to assist in accelerating the release. (This feature is found in the ABD type valves and not in the plain AB valve.) Both the AB and ABD valves permit emergency reservoir air flow into the auxiliary reservoir to assist in charging. With the service piston in release position, the slide valve opens to permit brake cylinder air to exhaust by way of the retainer valve.

3.2.4.3 Brake Emergency Application - An emergency application is made by reducing the brake pipe pressure at a rate significantly faster than a service rate. Thus, the emergency piston moves to its extreme position and permits quick action chamber air to flow to the face of the vent valve piston, forcing the vent valve to the open position. This allows a local venting of brake pipe air at a fast rate, thus sustaining and propagating the brake pipe reduction at a rate faster than a service rate throughout the train.

During emergency applications, the emergency reservoir air combines with the auxiliary pressure to provide a higher pressure in the brake cylinder. During emergency, the buildup of pressure in the brake cylinder is ultimately 20 percent higher than from a full service application. Again, it should be mentioned that the A-1 valve on cars with brake pipe length of 65 feet or longer operates separately from the AB type valves in venting brake pipe air during emergency, and serves as an aid in propagating the emergency rate.

3.3 PERFORMANCE

3.3.1 Requirements - The AB type control valves have been designed to meet the Performance Specifications for Freight Brakes as adopted by the AAR in 1933, revised 1947. Some of the important requirements are as follows*:

1. General Requirements - "The design of the operating valve shall be such as will insure efficient and reliable operation, both in its application and release functions and

---

* The complete Power Brake Law of 1958 and implementation orders are contained in Association of American Railroads (1975b) and in Federal Railroad Administration, Publication 49 CFR 232.

when intermingled with other types of power brakes. It shall be so constructed that the rate of brake cylinder pressure development may be adjusted to meet such changes in train operating conditions as may develop in the future."

"The design of the service and emergency valves shall be such as to permit their removal for cleaning and repair without disturbing pipe joints."

"The portions of the car brake which control the brake application and release, and also the brake cylinder, shall be adequately protected against the entrance of foreign matter."

"The apparatus conforming to the requirements of these specifications shall be so constructed, installed and maintained as to be safe and suitable for service."

2. Service Requirements - "The apparatus shall be designed and constructed such that based on 70 pounds brake-pipe pressure and train length of 150 cars, with a service reduction of 5 pounds in the equalizing reservoir at the brake valve, all brakes will apply."

"An initial equalizing reservoir reduction at the brake valve will produce substantially 10 pounds pressure in the brake cylinder of the 1st and 150th brakes will be nominally 20 seconds or less."

"A brake pipe reduction of 10 pounds will result in a pressure in each brake cylinder of not less than 15 pounds nor more than 25 pounds."

"A total brake pipe reduction of 25 pounds will result in an equalization of brake cylinder pressure with pressure in the reservoir from which compressed air is supplied to the brake cylinder, and brake cylinder pressure of not less than 48 pounds nor more than 52 pounds will be obtained."

"Quick service activity of the train brakes will cease when the initial quick service action has been completed."

"The quick action feature of the brake will produce substantially uniform time of quick service transmission regardless of the unavoidable variations in friction resistance of the parts."

"The brake will so function as to prevent a degree of wave action in brake pipe pressure sufficient to cause undesired release of any brake while the brakes are being applied."

"The degree of stability will be sufficient to prevent undesired service application occurring as a result of unavoidable minor fluctuations of brake pipe pressure."

"The brake-cylinder pressure increase resulting from quick service operation will be less when the brake is applied with pressure retained in the cylinder than with application made when the brake cylinder pressure is zero."

"Undesired quick action will not result with any rate of change in brake pipe pressure which may occur during service application or release of the brake."

"In the normal release of train brakes, an individual car brake will not start recharging until the brake pipe pressure has increased sufficiently to have accomplished the release of adjacent valves."

"The recharge of auxiliary reservoirs in the forward portion of the train will be automatically retarded while full release position of the brake valve is being used to initiate the release of train brakes."

"After a 15-pound service reduction has been made and brake valve exhaust has closed, in a release operation in which the brake valve is moved to release position and after 15 seconds is moved to running position, all operating valves will move to release position within 40 seconds after brake valve is placed in release position."

"The rate of release of pressure from the brake cylinder will be nominally 23 seconds from 50 pounds to 5 pounds."

3. <u>Emergency Requirements</u> - "The apparatus shall be designed and constructed that, based on 70 pounds of brake pipe pressure and a train length of 150 cars, emergency application operation will always be available irrespective of the existing state or stage of brake application or release."

"Emergency application initiated during a release of a previous brake application will produce a material increase in brake cylinder pressure to that which would result from a full service application made under same conditions."

"When the operating valve acts in emergency it will so function as to develop nominally 15 pounds brake cylinder pressure in not more than 1-1/2 seconds and maximum pressure in nominally 10 seconds.

"With an emergency reduction of brake pipe pressure, all brakes including the 150th will start to apply within 8.2 seconds and develop not less than 15 percent nor more than 20 percent in excess of 50 pounds brake cylinder pressure within 18.2 seconds from the movement of the brake valve to emergency position."

"The operating valve will so function that, when an emergency application is made subsequent to a service application which has produced not less than 30 pounds brake cylinder pressure, the maximum brake cylinder pressure will be attained in nominally four seconds from the beginning of the emergency action of the valve." ("Emergency applications from a charged system will produce between a 15 and 20 percent increase in brake cylinder pressure over that which results from a full service application, and irrespective of any degree of prior service application.")

"With any group of three consecutive brakes cut out, an emergency reduction made with the brake valve will cause the remainder of the brakes to operate in emergency and produce normal emergency pressures in the same time as when all brakes are cut in."

"The brakes will so function as to accomplish the release of an emergency application with the same degree of certainty secured in the release of service application."

"When releasing brake following an emergency application, each brake will so function as to decrease the auxiliary reservoir pressure prior to the actual release pressure."

"Both service and emergency brake application will be released when the brake pipe pressure is increased to not more than 1-3/4 pounds above that of the auxiliary reservoir, and irrespective of the increased frictional resistance to release movement of the piston and slide valves after a period of operation in train service."

3.3.2 Discussion - The preceding requirements of air brake system component performance imply actual brake system performance as discussed in the following.

3.2.2.1 Service Requirements - As presented in the above, the AAR Service Requirements can be briefly summarized as:

WITH A BRAKE PIPE PRESSURE OF 70 PSI, AND FOR A 150-CAR TRAIN:

1. A 5 PSI REDUCTION AT A SERVICE RATE MUST PROPAGATE THROUGH THE TRAIN AND CAUSE ALL BRAKES TO APPLY.

2. THE 5 PSI REDUCTION MUST PRODUCE AT LEAST 10 PSI BRAKE CYLINDER PRESSURE ON EACH CAR THROUGHOUT THE TRAIN.

One of the primary implications of this requirement is shown graphically in Figure 3-6 below.

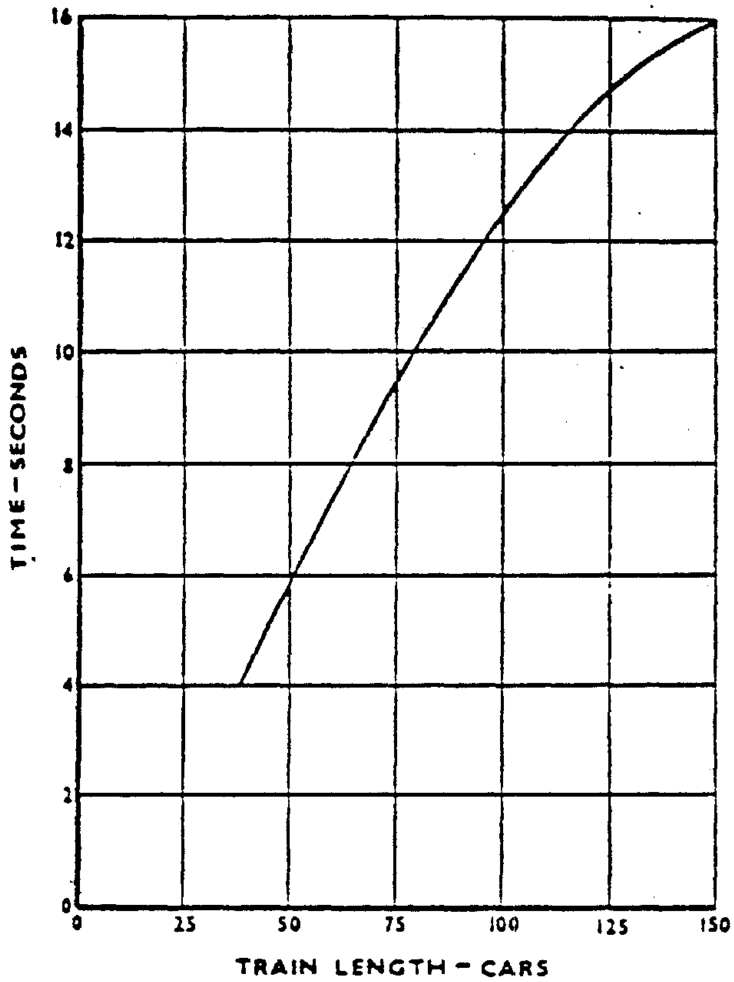


Figure 3-6

BRAKE PIPE PRESSURE PROPAGATION TIME VERSUS TRAIN LENGTH

Source: The Air Brake Association (1972). Used by permission of copyright owner.

As shown in Figure 3-6, the head car brakes typically start to function in less than 4 seconds, but the brakes on the 150th car do not start to function until 16 seconds later from the time the engineman makes the reduction to apply the brakes.

It should be noted that Figure 3-6 shows the brake pipe pressure signal propagation through the train. The actual brake cylinder pressure buildup on a given car begins at the time shown.

Figure 3-7 illustrates both the brake pipe pressure reduction and the brake cylinder pressure buildup propagation times for a typical 150-car train.

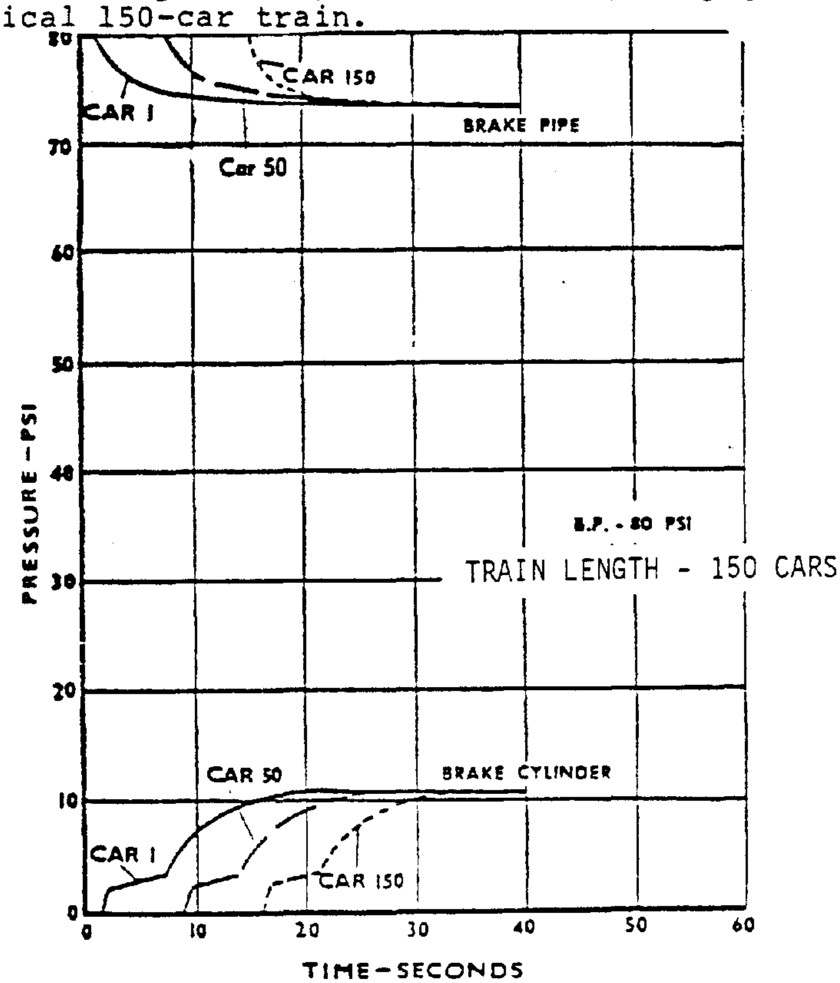


Figure 3-7

BRAKE PIPE REDUCTION AND BRAKE CYLINDER BUILDUP PROPAGATION TIME (6 PSI B.P. REDUCTION)

Source: The Air Brake Association (1972). Used by permission of copyright owner.

As shown in Figure 3-7, the brake cylinder on the 150th car does not reach 10 psi for 30 seconds from the time the engineer made the application. The distance a heavy tonnage freight train would travel before the speed would start to reduce following a 5- or 6-psi brake pipe reduction can be considerable. Therefore, unless time and distance are favorable, the engineman may start the second reduction before the brakes actually apply on the rear portion of train. This could move the slack toward the head end of the train causing violent slack action forces.

Figure 3-8 illustrates how the ABDW valve with the "Continuous Quick Service" (accelerated application) feature tends to apply the brakes on the rear portion of the train more rapidly than the current AB or ABD type valves. From this illustration, the ABDW valve appears to be an improvement towards alleviating slack action under certain conditions and towards reducing train stop distances. Again, it should be noted that the ABDW valve has only recently been granted a limited approval for testing in interchange service.

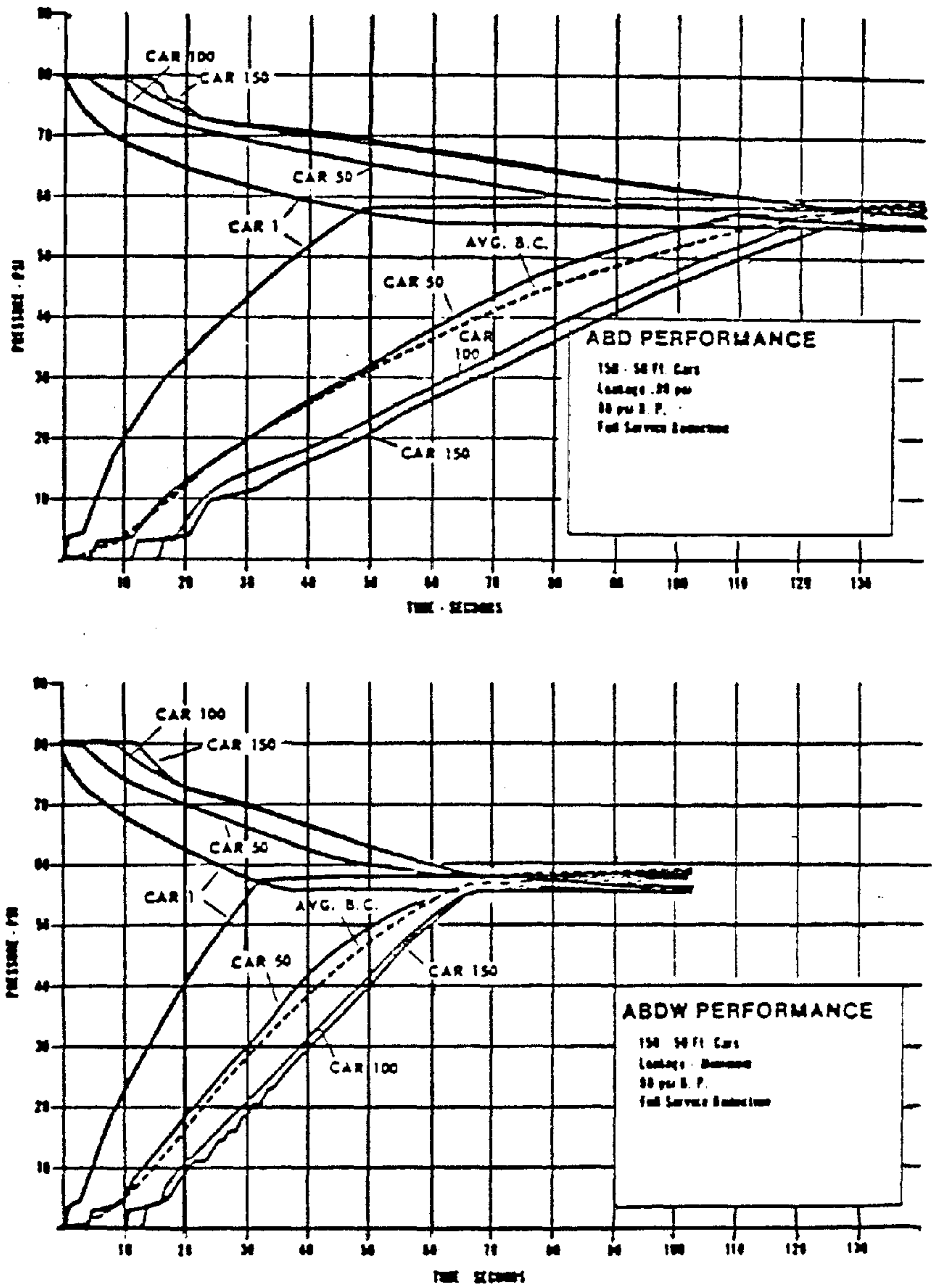


Figure 3-8

COMPARISON OF ABD AND ABDW FULL SERVICE REDUCTION APPLICATION TIMES

Source: Wright (1975, p. 8, Fig. 3). Used by permission of copyright owner.

As shown in Figure 3-8, the ABDW produces a substantial reduction in the time delay for the Full Service reduction compared to the ABD valve.

3.3.2.2 Release Performance - The AAR performance specification for the air brake release rate is partially summarized as follows:

> THE RATE OF RELEASE OF PRESSURE FROM
> THE BRAKE CYLINDER WILL BE NOMINALLY
> 23 SECONDS FROM 50 PSI TO 5 PSI.

The rate of release of pressure from the brake cylinder must be uniform to alleviate slack action. Assuming that the release rate for each car is matched fairly well, there is still a problem of communicating the release signal to each car at the same time. This is shown in Figure 3-9.

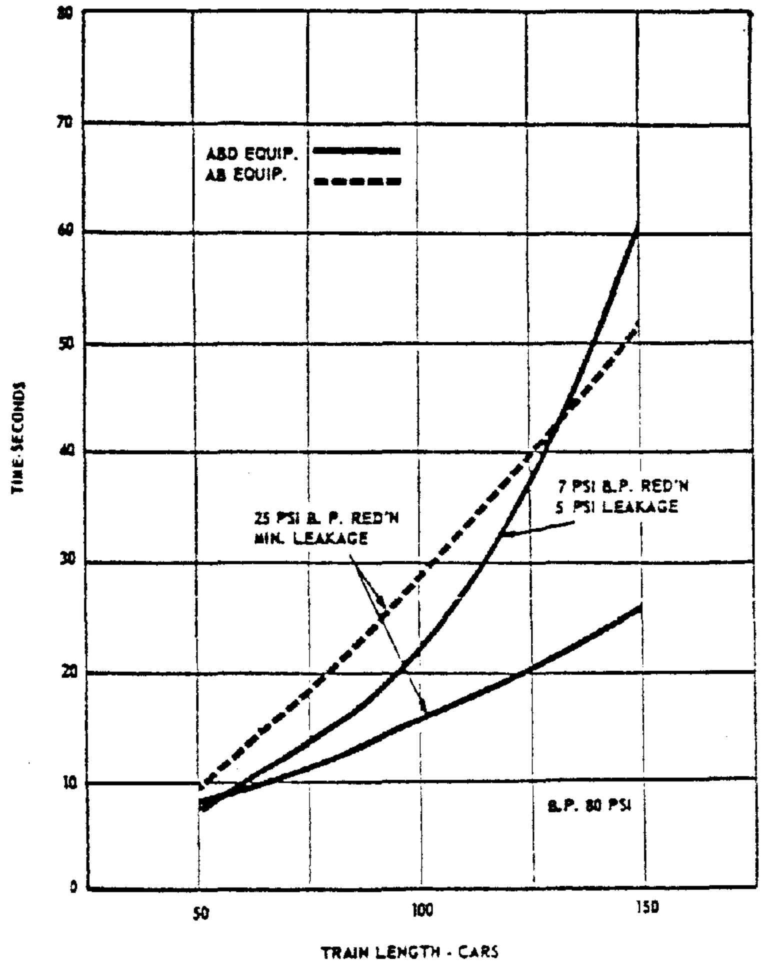


Figure 3-9

TIME TO START BRAKE RELEASE
FROM 7 OR 25 PSI REDUCTIONS

Source: The Air Brake Association (1972). Used by permission of copyright owner.

As shown in Figure 3-9, the 40th car starts to release within 10 seconds following a 25 psi brake pipe reduction, but the 150th car with an AB brake valve requires 25 seconds longer than the ABD (with the accelerated release feature) equipped car. This is a 100 percent improvement in communication rate from the head car to the 150th car.

3.4 PROBLEMS

The preceding section described the operations and performance requirements of air brake systems. This section will delineate and discuss the problems associated with the freight train air brake system in use today.

3.4.1 Yard Processing - In virtually all classification yards the brake system air must be bled manually from the reservoirs and cylinders to expedite the handling of cars. After classification, when the train is being assembled, all air hoses must be coupled manually and each angle cock must be checked to see that it is properly positioned for air to move through the brake pipe. Other routine items include the examination of each car to determine that each brake is "cut in," if the brake shoes are replaced as needed, and hand brakes and retainer valves are released.

After the above inspection is performed, the train air brake system must be charged with compressed air using either the locomotive consist or yard plant air.

As shown in Figure 3-10, the charging time typically ranges from between 35 and 45 minutes, and could require much longer time if there are brake pipe or system leakages throughout the train.

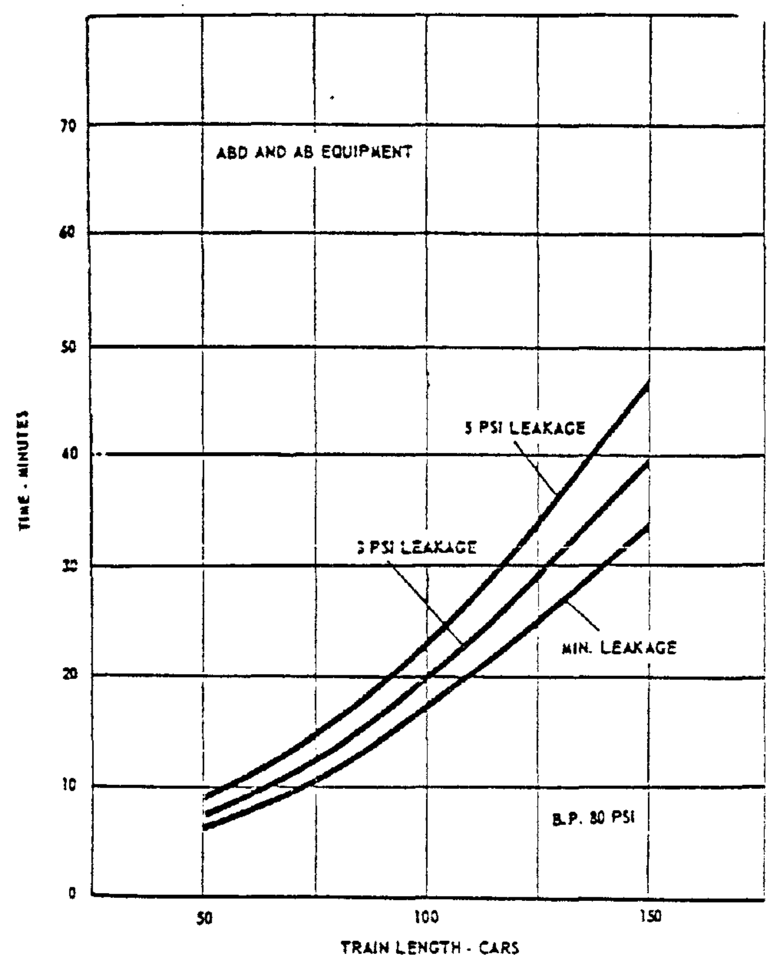


Figure 3-10

TRAIN BRAKE SYSTEM CHARGING TIME VERSUS TRAIN LENGTH

Source: The Air Brake Association (1972). Used by permission of copyright owner.

When the brake pipe pressure (at least 60 psi and/or no less than 15 psi difference from head end to rear end pressure) is established on the rear of the train, a standing brake test must be made to determine that brake pipe leakage is not in excess of 5 psi per minute, piston travel is not excessive, brake shoes contact wheels properly, rigging is not binding, and other conditions are satisfied.

After the brake test has been satisfied, the train is ready to depart from the initial terminal. No additional brake tests are required within the limit of 500 miles unless the train consist has been changed, or unless interchange rules or other special instructions that may be specified by an individual carrier must be enforced.

Air leakage can create problems in charging trains, cause undesired emergency applications and undesired releases of train brakes (if intermittent), and limit the amount of braking energy stored in the car reservoirs towards the rear of the train.

Figure 3-11 shows how air leakage concentrated in the rear portion of the train can affect the brake pipe gradient.

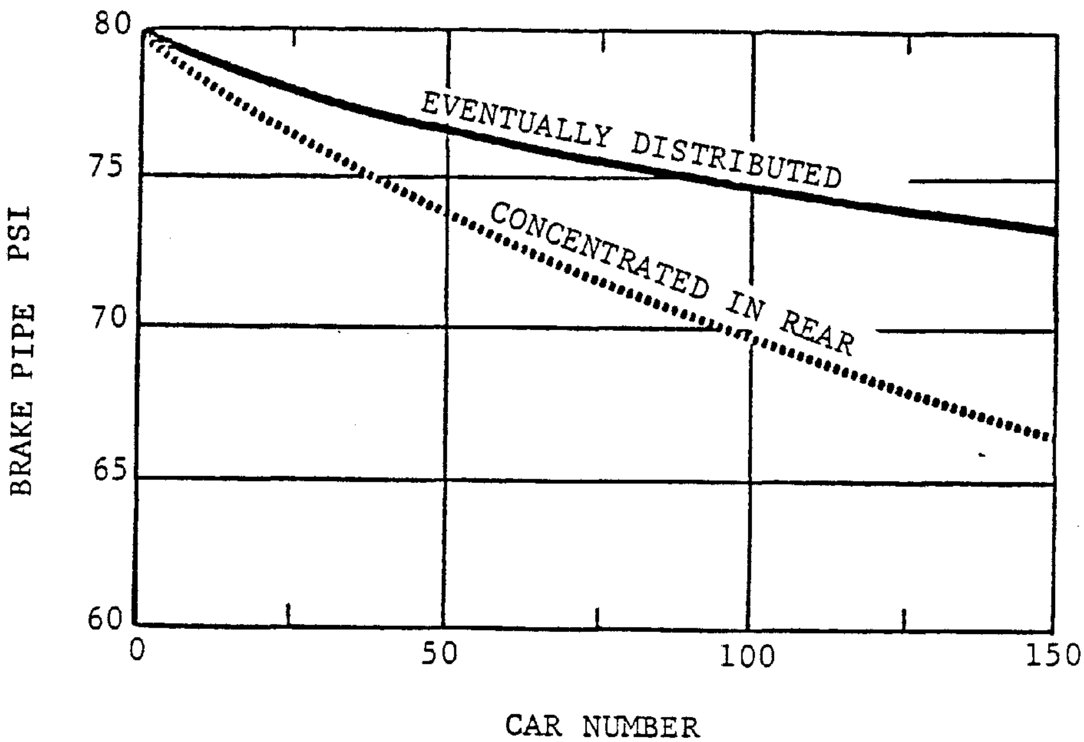


Figure 3-11

LEAKAGE LOCATION EFFECTS ON TRAIN BRAKE PIPE PRESSURE

Source: Derived from leakage data provided by WABCO.

To compensate for leakage, several roads use remote air charging systems in long trains when operating in low ambient temperatures. These systems typically consist of a repeater unit car with an air compressor which responds to the release signal in the train brake pipe. Also used to accomplish the same purpose are remotely controlled locomotive units placed in the rear portion of the train. The approximate effects of these systems are shown graphically in Figure 3-12.

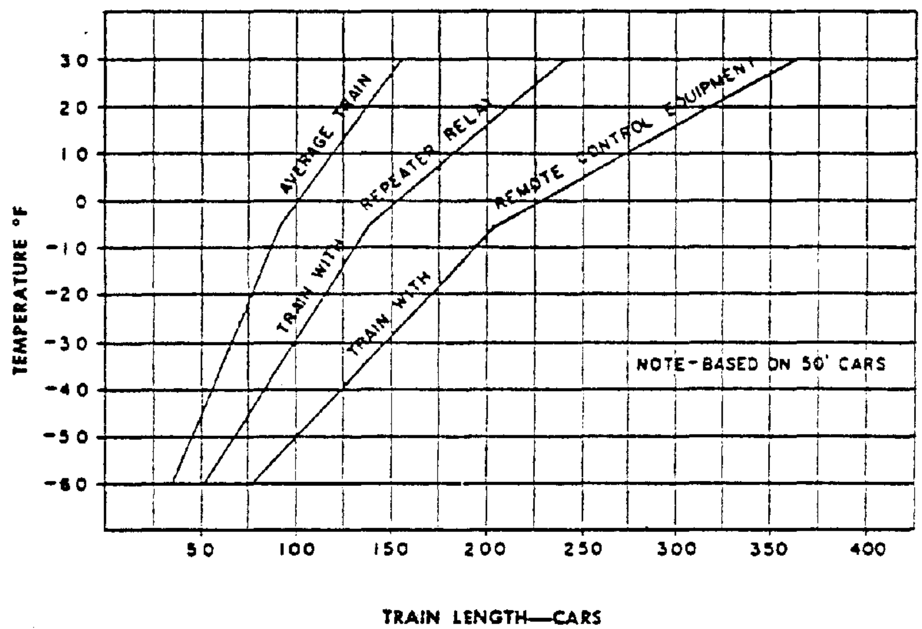


Figure 3-12

APPROXIMATE MAXIMUM TRAIN LENGTH (IN CARS) VERSUS AMBIENT TEMPERATURE

Source: The Air Brake Association (1972). Used by permission of copyright owner.

The top curve of Figure 3-12 shows a train with the locomotive supplying air from the head end; at a temperature of 0 degrees Fahrenheit the train length must be reduced to about 100 cars or less. The middle curve shows the use of a repeater relay unit to increase brake pipe pressure. This method shows that train length can be increased by approximately 50 percent with the repeater unit. Finally, the bottom curve shows that a radio-controlled locomotive unit can boost the brake pipe pressure enough to permit more than a 100 percent increase in train length compared to the average train. Although these methods have proven beneficial where train length is important to the carrier, there is a problem in supervising the use of these methods. The problems include knowing where to place the unit in the train, making the proper brake tests, and finally, switching the unit in and out of the train getting it to and from point of use.

3.4.2 Over-the-Road Problems - Over-the-road problems resulting from air brake systems include train delays, train derailments, damage to rolling equipment, and the associated cost of damage to contents. In addition to these costs, there is a sizable investment made in the training of enginemen in management of the train over all types of terrain. A few of the principal problems associated with moving fast manifests and/or slow tonnage trains are:

a. Light Reduction Desired: Heavy Reduction Required

b. Air Hose Failure

c. Partial Undesired Service Release

d. Undesired Emergency Application

e. Uneven Brake Forces Throughout the Train.

Thorough training of the engineman is essential. In particular, he must know the layout of the railroad with respect to grades, curves, signal locations, meeting points, speed restrictions, track conditions, and special moves associated with a specific portion of the railroad. Knowing the railroad in these terms will benefit the engineman during adverse weather conditions. It likewise enables him to exercise good judgment in the use of pulling and/or retarding power.

Knowing the characteristics of the route also enables the engineman to cope with some of the unexpected incidents or braking variables over which he has no control. Some of the unexpected variables that must be coped with during over-the-road operations are summarized above. The faster a train is operated, the more significant the problems become.

Uneven brake forces throughout the train represent one of the most difficult problems for an engineman to cope with, since train make-up and braking characteristics vary considerably from day-to-day.

The problem of uneven train brake forces can be discussed by referring to Figure 3-13.

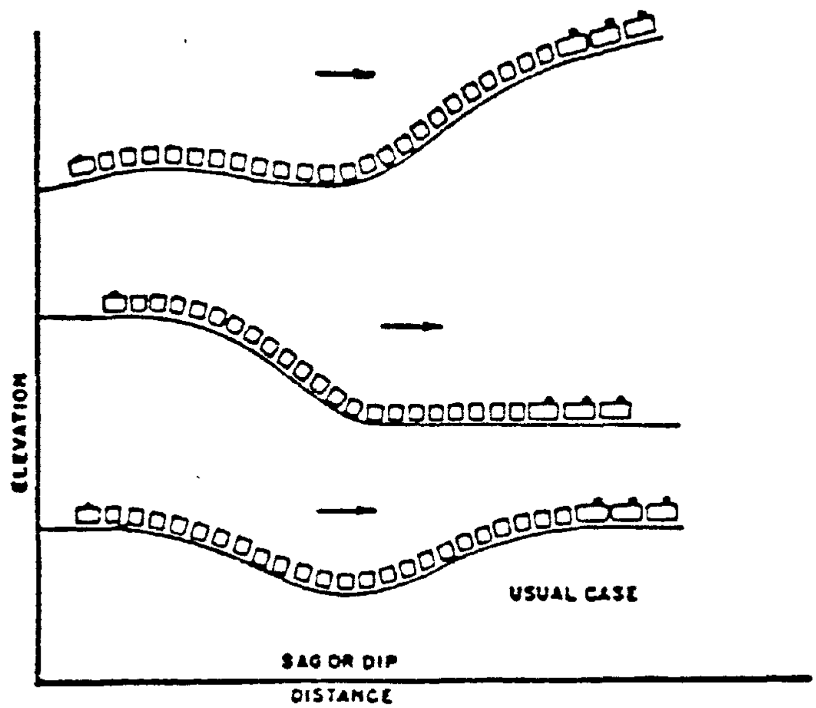


Figure 3-13

SYMBOLIC REPRESENTATION OF A TRAIN
BEING OPERATED OVER UNDULATING TERRAIN

Source: Track Train Dynamics (1972). Used by permission of copyright owner.

Figure 3-13 considers the operation of a mixed train consisting of empty cars weighing between 25 to 30 tons, and fully loaded cars with gross rail loads of 100 tons or more. The engineman must be aware of train length and the characteristics of various draft gear designs to exercise good judgment in adjusting pulling power and timing brake response.

If the train is moving over a terrain requiring slack to be held out as much as possible, a light brake pipe reduction of 6 psi may be most desirable. With reference to the previous discussion of brake application response time and pressure, it may be necessary to wait 30 seconds before attempting to make another reduction. Depending upon the speed and weight of train, a large distance may be traveled before the partial application of the brakes has affected the speed of train appreciably. Often, the second heavy reduction is made too soon, causing slack to roll in harshly and resulting in excessive in-train shock.

In the above example, perhaps a light reduction of 6 psi would have slowed the train down smoothly, particularly if it were a long train. But there may be a risk of having the brakes stick from such a light reduction. So, thinking of the possibility of brakes sticking, the engineman may have a heavier reduction than needed. Now, unless extremely good judgment in use of power and the timing of the release is made, both a heavy run-in and run-out of slack may be experienced.

There are unavoidable circumstances in which the engineman has no control, such as burst air hose or train separation. The problems could be acute if the train is lengthy and operating at a moderately fast speed, and worse if moving over undulating terrain. An emergency brake application resulting from the large opening in the brake pipe could cause the slack to be very uncontrollable. Therefore, it would be ideal in this case for every car to be controlled at an equal deceleration rate regardless of gross rail load. Any deviation from this rate can cause the slack to move in any direction. However, there are few trains of mixed consist with equal braking forces throughout the trains.

There also may be other circumstances where time will not permit the engineman properly to program the locomotive power reduction with selective brake pipe reductions in order to adjust train slack.

Emphasis must be placed on training enginemen to enable them to fully understand all ramifications involved in train handling.

Educational programs should cover simulations of all facets of train handling, including all models of locomotives to familiarize the trainee with the various locomotive control stand arrangements.

Firsthand information should be provided to describe the behavior of the various freight car brake arrangements. Brake response information concerning the various car types should include long coupler arrangements and cushion-type draft gear or moveable center sills. The information should describe causes for high lateral forces as well as draft and buff action.

Moreover, more thought should be exercised on improving or developing methods for the engineman to read: air consumption, braking power, throttle power, buff and draft forces, all of which can be used in planning the use of power and brakes.

4. RAILROAD AND PRIVATE CAR OWNER SURVEY

4.1 INTRODUCTION

To acquire data of sufficient detail concerning the population distribution of various kinds of freight car brake system components in present service, it was necessary to request the data directly from the railroads and the private car owners.

4.2 DATA REQUESTED IN THE RAILROAD SURVEY

Four types of data and information were requested in the railroad data surveys. These included the following:

a. Part A - General System Operating Conditions. This portion of the survey represented data in relation to the steepest descending grades, the most severe curves and train tonnages, lengths, and top speed for various classes of service.

b. Part B - Air Brake System Operations. This portion of the survey requested information concerning cold weather operating problems, the use of retainer valves, the success of brake cylinder release valves, slid-flat wheel problems and causes, and air brake system charging problems. In addition, opinions were requested concerning what should be given highest priority to improve freight car braking systems.

c. Part C - Standard Air Brake Equipment in Service. This portion of the survey requested air brake component population statistics for each of 10 car types. The data requested included:

1) Number of cars.

2) Average car age.

3) Brake valves by type.

4) Brake shoes by type.

5) Brake rigging type.

6) Number of slack adjusters.

7) Fittings by type.

8) Number of load proportional devices.

9) Number of empty/load devices.

This portion also requested information of Equipment Being Purchased by car type:

1) Type of brake shoes.

2) Type of rigging.

3) Use of load proportional or empty/load devices.

4) Type of wheels.

d. Part D - Special Brake System Equipment in Service. This portion of the survey requested information concerning the use of repeater relay systems, remotely-controlled locomotives, and other special brake systems.

4.3 DATA REQUESTED FROM PRIVATE CAR OWNERS

The private car owners were requested to supply the data of Part C, above. In addition, opinions were solicited concerning problems being experienced with the present braking system (or individual components), possible solutions, and general recommendations for improvements needed in freight train braking systems.

In addition to Part C, the private car owners were also requested to provide life cycle cost information with respect to the present freight brake system.

4.4 RESULTS OF THE SURVEY

The aggregated results of the railroad survey are presented in Tables 4-2 through 4-11. The results from the private car owners are presented in Table 4-12 and 4-13 (on brake component population) and Table 4-14 (life cycle cost information). Some of the data are disaggregated by type of freight car.

The general presentation order appears in Table 4-1.

TABLE 4-1

SURVEY PRESENTATION

I - RAILROADS

Part of Survey	Description of Data	Table
A	General Operating Characteristics	4-2
B	Air Brake Systems Operations	
	- Train Length Reduction in Cold Weather	4-3
	- Use of Retainer Valves	4-3
	- Brake Cylinder Release Valves	4-3
	- Slid-Flat Wheel Problems	4-3
	- Faster Brake System Charging	4-3
	- Faster Testing Benefits	4-3
	- Most Serious Problems	4-3
	- Highest Priority Improvements	4-3
C	Brake System Component Population Distributions	
	- Average Freight Car Age of Sample	4-4
	- Type of Brake Valve Population	4-5
	- Type of Brake Rigging Population	4-6
	- Type of Brake Shoe Population	4-7
	- Type of Brake Pipe Fittings Population	4-8

TABLE 4-1

SURVEY PRESENTATION (Cont'd.)

I - RAILROADS (Cont'd.)

Part of Survey	Description of Data	Table
C Cont'd	- Load Proportional, Load/Empty Devices	4-9
	- Air Brake Components Being Purchased	4-10
D	Special Brake System Devices in Use	4-11

II - PRIVATE CAR OWNERS

C	Summary	4-12
	- Brake System Component Population Distribution	4-13
	- Life Cycle Cost Information	4-14
B	Highest Priority Improvement	4-15

TABLE 4-2

SUMMARY OF RAILROAD SURVEY RESULTS - GENERAL SYSTEM OPERATING CONDITIONS

Territory	Reporting Road	Steepest Descending Grade (Percent)		Most Severe Curvature (Degrees)		Routine Helper Service?	Heavy Train Gross Tonnage	Fast Freight Train Characteristics			Slow Freight Train Characteristics	
		Main Line	Branch Line	Main Line	Branch Line			Maximum Cars	Length Feet	Top Speed MPH	Maximum Cars	Length Feet
Eastern	1	2.20%	4.00%	11.0	16.0	Yes	15,000	100	5,500	70	160	8,000
	2	2.20	4.00	10.0	16.0	Yes	22,000	100	-	50	160	-
	3	2.36	5.89	9.5	12.5	Yes	13,000	-	6,000	60	-	5,000
Western	4	4.00	5.00	10.0	16.0	No	12,000	-	5,000	79	-	7,000
	5	3.30	3.40	14.0	16.0	Yes	9,500	85	5,000	65	175	10,000
	6	2.20	3.75	16.0	17.0	Yes	14,000	90	6,000	50	130	6,500
	7	2.20	5.60	12.0	22.0	Yes	19,000	90	-	60	170	-
	8	2.20	4.00	16.0	16.0	-	-	-	-	-	-	-
	9	1.50	2.00	16.0	16.0	No	14,000	120	-	60	150	-
	10	2.22	2.90	15.5	33.0	No	16,000	75	4,900	60	150	8,250
	11	1.44	1.20	8.0	5.4	No	11,000	110	-	50	150	-
	12	3.00	4.50	15.0	24.0	Yes	8,000	50	-	60	90	-
Southern	13	1.50	1.50	10.0	12.0	No	18,000	175	10,000	60	175	10,000
	14	2.50	2.20	16.0	23.0	Yes	10,200	130	-	50	130	-
	15	2.34	3.48	16.5	20.5	Yes	28,000	100	7,500	65	180	9,000
	16	2.00	2.50	16.0	16.0	No	15,000	80	4,300	60	150	8,100
	17	4.70	3.75	17.0	22.0	No	13,000	125	-	55	150	-

TABLE 4-3

SUMMARY OF RAILROAD SURVEY RESULTS - AIR BRAKE SYSTEMS OPERATIONS

Territory	Reporting Road	(1) Train Length Reduced in Cold Weather?	(2) Retainer Valves Used?	Explanation of Use of Retainer Valves
Eastern	1	Yes	No	Only on severe grades in the event that dynamic braking is not available.
	2	Yes	Yes	Usually from mine tipple down to main line.
	3	Yes	No	
Western	4	Yes	No	Exception certain mountain areas when the dynamic brake is inoperative.
	5	Yes	Yes	
	6	Yes	No	Left up to the discretion of the engineer. Retainers not a significant benefit.
	7	Yes	Yes	Not used often, but provide for safe movements of a train when required.
	8	Yes	Yes	Have severe long descending grades in excess of 2.2%, and retaining valves are used to control trains for safe operations.
	9	Yes	No	
	10	Yes	No	
	11	No	No	
	12	Yes	Yes	
Southern	13	No	No	
	14	Yes	No	
	15	No	No	
	16	Yes	No	Pressure retaining valves not used on our railroad.
	17	No	Yes	Used in mainline operation and in certain coal mine operations.

TABLE 4-3

SUMMARY OF RAILROAD SURVEY RESULTS - AIR BRAKE SYSTEMS OPERATIONS (CONTINUED)

		(1)	(4)	(5)	(6)	
Territory	Reporting Road	Benefits Resulting from Brake Cylinder Release Valves	Slid Flats a Problem?	Caused by Hand Brakes?	Particular Type Car or Brake?	Explanation of Causes of Slid Flats
Eastern	1	Releasing brakes for altering piston travel; classification yards.	No	Yes	No	Unreleased hand brakes are a deliberate occurrence rather than accidental.
	2	In yards, bleeding of brake cylinder pressure only.	No	No	No	
	3	Decrease time for switching and humping. Decrease charging time after train is assembled.	No			
Western	4	In hump yard operation.	No			
	5	Bleeding trains at terminals.	Yes	Yes	Yes	Truck-mounted brakes.
	6	Saves time in preparing cars for switching.	Yes	Yes	Yes	Truck-mounted brake systems where have hand brake is on "B" end only.
	7	Save charging time when making up train and reduce time on inbound train.	Yes	Yes	Yes	Truck-mounted brakes not a problem in unit train "Hi-Phos" shoes also.
	8	In switching and hump yard operations.	Yes	No		
	9	Not used due to brakes leaking back on.	No	Yes	Yes	Truck-mounted brakes due to failure of crew checking both trucks.
	10	Expediting switching operation.	No			
	11	Preparing cars for classification and switching.	Yes	Yes	Yes	Truck-mounted brakes with hand brake pressure on one set of trucks.
	12	In releasing air brake for humping and switching.	Yes	Yes	No	
Southern	13	In hump operations; after switching can recharge in less time.	No	No	No	Early truck-mounted brakes experienced some slid flat wheels due to hand brake.
	14	Yard service; hump service.	No	Yes	No	
	15	Release brakes for switching. Economic justification questionable.	No			
	16	Permits brakes to be released quicker without depleting air in reservoirs.	No	No	No	
	17	Expediting cars in yards	No			The single truck hand brake on truck mounted brakes is a significant contributor to slid-flat wheels but flat wheels are not a significant problem in overall operations.

TABLE 4-3

SUMMARY OF RAILROAD SURVEY RESULTS -
AIR BRAKE SYSTEMS OPERATIONS (CONTINUED)

Territory	Reporting Road	(7) Faster Charging of Value?	Explanation of Faster Charging Benefits	(8) Faster Testing of Value?	Explanation of Faster Testing Benefits
Eastern	1	Yes		Yes	Eliminate brake pipe leakage test and institute a graduated scale of gradients versus train length.
	2	Yes	As long as the proper brake pipe gradient is established.	Yes	Decrease terminal time and would increase overall train speed.
	3	Yes	Any method that decreases cost, expedites train movements and otherwise not detrimental or complicated.	Yes	See explanation to left.
Western	4	Yes		Yes	
	5	Yes		Yes	
	6	Yes	Faster terminal charging of trains during winter would relieve congested train yards.	Yes	Time consuming to walk each car and adjust long or short piston travel.
	7	Yes	Save time in terminals and on road.	Yes	Save time in terminals.
	8	Yes		Yes	
	9	Yes	Could eliminate or reduce terminal delays.	Yes	Same as explanation at left.
	10	Yes	Reduce time for train inspection, testing and departure.	No	
	11	Yes	Anything that will move a train quicker after assembly is of value.	Yes	Same as explanation at left.
	12	Yes	Expedite train makeup. Help train handling when using cycle braking, or where frequent braking required.	Yes	Move trains out of yards sooner after makeup.
Southern	13	Yes	Less delay in preparing train for brake test; however, do not want as a trade off.	Yes	Would result in less terminal delays.
	14	Yes	Expedite testing.	Yes	Expedite train out of the terminals.
	15	No		No	
	16	Yes	Expedite movement of trains out of terminals and on road.	Yes	Expedite movement of trains out of terminals.
	17	No	Most trains are charged from yard supply. More time required to work train than to charge.	Yes	Eliminating time to walk train to inspect brakes.

TABLE 4-3

SUMMARY OF RAILROAD SURVEY RESULTS - AIR BRAKE SYSTEMS OPERATIONS (CONTINUED)

Territory	Reporting Road	(9) Most Serious Air Brake Problems: Terminals	(9) Most Serious Air Brake Problems: Line-Haul	(10) What Improvements Should Be Given Highest Priority
Eastern	1	Passing the five-psi leakage test in winter.	Same.	Improved hose coupling and gasket to reduce leakage.
	2			Use of equipment such as ABDW, more reliable automatic slack adjusters, all cars equipped with composition brake shoes.
	3	Leakage and obtaining gradient during cold weather. Damage to air hose arrangement.	Cold weather leakage. Undesired emergency often caused by hose damage or separation.	Decrease time to couple and test train brake equipment; simplify and standardize, especially hose arrangement.
Western	4	Angle cock location.	Undesired emergency application.	Reduce UDE's and locate angle cocks where air hose will not foul and uncouple.
	5	Brake pipe leakage.	Stuck brakes and undesired emergency applications, also hose separations.	Determine cause of undesired emergencies, prevention of hose separation while train is in motion.
	6	Adjustment of piston travel; leaking compression fittings; leaks between underframes.	Uncoupling of air hoses on long overhang cars; poor end hose design arrangements.	Prevent undesired uncoupling of air hoses; standardize design on cushion underframe cars at angle cock location.
	7	Leakage, long piston travel cars without slack adjusters.	Cold weather brake pipe gradient, train separation on long cars, undesired hose uncoupling.	Equip all cars with composition brake shoes, improve end of car arrangements on long cars, reduce leakage.
	8	Mishandling of train brakes; water in system; broken or leaking pipes and hoses.	Same.	Application of ABDW, accelerated application valves.
	9	Long piston travel; brakes fail to set or leak off during air test.	Stuck brakes.	Elimination of brake pipe leakage.
	10	Adjusting piston travel.	Sticking brakes, undesired emergencies.	Elimination of undesired and unauthorized brake applications and releases.
	11	Brake pipe leakage.	Brake pipe leakage and flexible rubber train line components.	Instantaneous application of all brakes in a train; instantaneous release of all brakes in a train.
	12		Failure of brake pipe fittings, connections and hoses.	Better brake pipe design; faster charging and recharging; faster, more uniform application of brakes through train.
Southern	13	Ruptured hoses due to separating under pressure; recharging after switching.	Air hose separation on long cars with sliding sills on end of car cushioning.	Faster, perhaps instantaneous application to reduce stop distance, faster release, quicker charging, eliminate air hose coupling entirely, elimination of variation of braking ratio between loaded and empty cars.
	14	Separation and leaks.	Separations.	Air brake hose couplings.
	15	Switch out cars for FRA testing and servicing that is not necessary.	No significant problems.	Elimination of regulations requiring excessive testing and servicing of the equipment.
	16	Inability of obtaining air pressure on long trains in cold weather.	Separation of brake hose glad hands.	Improved application and release times and variable or empty load brake equipment.
	17	Adjustment of air hoses.	Air hose separations, train line fittings and undesired emergencies.	Eliminate the air hose problem.

TABLE 4-4

SUMMARY OF RAILROAD SURVEY RESULTS -
SAMPLE AVERAGE FREIGHT CAR AGE (YEARS) BY CAR TYPE

Car Type	All Territories	
	Sample Population	Average Car Age
1. Box	297,847	13.6
2. Hi-Cube Box	7,346	7.9
3. Refrigerator	72,743	12.5
4. Flat (TOFC/COFC)	1,516	10.5
5. Flat (Non-TOFC/COFC)	43,315	16.0
6. Hopper	249,954	15.2
7. Covered Hopper	116,252	12.9
8. Gondola	112,052	15.4
9. Tank	565	24.2
10. All Others (Stock, Pulpwood, Caboose, Etc.)	29,675	26.5
11. Others Not Properly Formatted	77,000	23.2
Total	1,008,265	(15.2)

TABLE 4-5

SUMMARY OF RAILROAD SURVEY RESULTS - NUMBER OF CARS SAMPLED EQUIPPED WITH AB, AB PLUS(1), OR ABD TYPE AIR BRAKE VALVES

Car Type	Average Car Age of Sample (Years)	Total Number of Cars Equipped (All Cars Sampled)		
		AB	AB Plus	ABD
1. Box	13.6	151,484	68,715	77,367
2. Hi-Cube Box	7.9	36	298	7,012
3. Refrigerator	12.5	23,142	18,367	26,544
4. Flat (TOFC/COFC)	10.5	855	384	277
5. Flat (Non-TOFC/COFC)	16.0	21,519	11,848	9,948
6. Hopper	15.2	91,726	94,970	63,261
7. Covered Hopper	12.9	32,668	21,571	61,973
8. Gondola	15.4	58,867	20,654	32,531
9. Tank	24.2	457	102	6
10. All Others (Stock, Pulpwood, Caboose, Etc.)	26.5	20,628	5,892	3,142
11. Others Not Properly Formatted	23.2	59,389	7,816	9,795
Subtotal	(15.2)	460,771	250,617	291,856
Total			1,003,244	

Note: (1) AB plus represents AB brake valves equipped with brake cylinder release valves.

TABLE 4-6

SUMMARY OF RAILROAD SURVEY RESULTS -
NUMBER OF CARS SAMPLED EQUIPPED WITH INDIRECT — ACTING BODY-MOUNTED RIGGING, DIRECT — ACTING TRUCK-MOUNTED RIGGING, AND SLACK ADJUSTERS

	Car Type	Average Age of Cars Sampled (Years)	All Cars Sampled		
			Indirect (Body Mounted)	Slack Adjusters	Direct (Truck Mounted)
1.	Box	13.6	286,351	146,869	11,485
2.	Hi-Cube Box	7.9	1,544	1,743	5,802
3.	Refrigerator	12.5	66,939	52,106	5,804
4.	Flat (TOFC/COFC)	10.5	1,015	1,511	501
5.	Flat (Non-TOFC/COFC)	16.0	40,587	24,782	2,728
6.	Hopper	15.2	248,049	131,128	1,907
7.	Covered Hopper	12.9	96,273	64,832	19,973
8.	Gondola	15.4	109,261	54,581	3,175
9.	Tank	24.2	515	60	50
10.	All Others (Stock, Pulpwood, Caboose, Etc.)	26.5	29,649	5,944	24
11.	Others Not Properly Formatted	23.2	70,253	8,529	6,747
	Subtotal	(15.2)	950,436	492,085	58,196
	Total (Indirect and Direct)			1,008,632	

TABLE 4-7

SUMMARY OF RAILROAD SURVEY RESULTS -
NUMBER OF CARS SAMPLED EQUIPPED WITH METAL,
HI-PHOSPHORUS, OR COMPOSITION TYPE BRAKE SHOES

Car Type	Average Age of Cars Sampled (Years)	All Cars Sampled		
		Metal	Hi-Phosphorus	Composition
1. Box	13.6	162,482	37,340	98,014
2. Hi-Cube Box	7.9	248	0	9,761
3. Refrigerator	12.5	20,054	28,729	23,370
4. Flat (TOFC/COFC)	10.5	610	0	906
5. Flat (Non-TOFC/COFC)	16.0	20,443	9,461	12,339
6. Hopper	15.2	164,053	2,361	83,542
7. Covered Hopper	12.9	38,649	14,955	62,646
8. Gondola	15.4	70,445	6,190	35,416
9. Tank	24.2	514	0	51
10. All Others (Stock, Pulpwood, Caboose, Etc.)	26.5	14,630	10,924	4,121
11. Others Not Properly Formatted	23.2	62,015	0	14,985
Subtotal	(15.2)	554,143	109,960	345,151
Total			1,009,254	

TABLE 4-8

SUMMARY OF RAILROAD SURVEY RESULTS - NUMBER OF CARS SAMPLED EQUIPPED WITH WELDED AND OTHER TYPES OF AIR BRAKE PIPE FITTINGS

Car Type	Average Age of Cars Sampled (Years)	All Cars Sampled	
		Welded	Other
1. Box	13.6	97,094	174,031
2. Hi-Cube Box	7.9	3,849	2,553
3. Refrigerator	12.5	34,559	33,417
4. Flat (TOFC/COFC)	10.5	976	365
5. Flat (Non-TOFC/COFC)	16.0	11,797	28,336
6. Hopper	15.2	61,882	179,658
7. Covered Hopper	12.9	51,542	54,997
8. Gondola	15.4	42,532	62,152
9. Tank	24.2	19	546
10. All Others (Stock, Pulpwood, Caboose, Etc.)	26.5	4,372	25,093
11. Others Not Properly Formatted	23.2	4,370	72,630
Subtotal	(15.2)	312,992	633,778
Total		947,770	

TABLE 4-9

SUMMARY OF RAILROAD SURVEY RESULTS -
NUMBER OF CARS SAMPLED EQUIPPED WITH LOAD PROPORTIONAL AND LOAD/EMPTY DEVICES

Car Type	Average Age of Cars Sampled (Years)	All Cars Sampled	
		Load Proportional	Empty-Load
1. Box	13.6	0	0
2. Hi-Cube Box	7.9	0	0
3. Refrigerator	12.5	0	0
4. Flat (TOFC/COFC)	10.5	0	0
5. Flat (Non-TOFC/COFC)	16.0	0	0
6. Hopper	15.2	0	4,378
7. Covered Hopper	12.9	0	2,379
8. Gondola	15.4	0	1,952
9. Tank	24.2	0	0
10. All Others (Stock, Pulpwood, Caboose, Etc.)	26.5	0	672
11. Others Not Properly Formatted	23.2	0	0
Total	(15.2)	0	9,381

TABLE 4-10

SUMMARY OF RAILROAD SURVEY RESULTS -
AIR BRAKE SYSTEM COMPONENTS BEING PURCHASED BY RAILROADS SAMPLED IN SURVEY(1)
(EXPRESSED IN TERMS OF THE NUMBER OF ROADS THAT ARE PRESENTLY PURCHASING EACH COMPONENT)(2)

Car Type	Type of Brake Shoes			Type of Rigging		Special Equipment		Wheel Type		
	Metal	Hi-Phosphorus	Composition	Body	Truck	Load Proportional	Empty-Load	One-Wear	Two-Wear	Multiple-Wear
1. Box	1	0	11	12	0	0	0	8	6	0
2. Hi-Cube Box	0	1	9	8	3	0	0	6	6	0
3. Refrigerator	0	1	8	9	0	0	0	5	6	0
4. Flat (TOFC/COFC)	0	0	7	7	0	0	0	4	4	0
5. Flat (Non-TOFC/COFC)	0	1	8	7	1	0	0	6	5	0
6. Hopper	0	1	11	12	0	0	1	7	6	0
7. Covered Hopper	0	0	11	8	3	0	2	6	6	0
8. Gondola	0	1	12	12	0	0	1	8	6	0
9. Tank	0	0	2	2	0	0	0	0	2	0
10. All Others (Stock, Pulpwood, Aggregate, Caboose, Etc.)	0	1	8	9	0	0	1	5	5	0

Notes: (1) These data reflect both the kinds of cars being purchased and the air brake system components.
(2) Some railroads purchase more than one kind of component for a given car type.

TABLE 4-11

SUMMARY OF RAILROAD SURVEY RESULTS -
SPECIAL BRAKE SYSTEM EQUIPMENT IN SERVICE

Territory	Reporting Road	(1) Repeater Relay Systems Used?	(2) If So, How Many	(3) Remote Controlled Locomotives Used?	(4) If So, How Many	(5) Special Brake Systems Used?	(6) Describe Special Brake System
Eastern	1	No	-	No	-	No	-
	2	No	-	No	-	Yes	Hydraulic-pneumatic brake system.
	3	No	-	Yes	7 lead units 8 remote units	No	-
	4	No	-	Yes	60	No	-
Western	5	Yes	1	Yes	9 sets	No	-
	6	Yes	8	Yes	76 master 80 remote	No	-
	7	-	-	Yes	57	No	-
	8	No	0	Yes	1 set	No	-
	9	No	-	No	-	No	-
	10	No	N.A.	No	N:A.	No	N.A.
	11	No	-	Yes	-	No	-
	12	No	-	Yes	10 master 10 booster	No	-
Southern	13	No	-	Yes	62	No	-
	14	No	-	No	-	No	-
	15	No	-	No	-	No	-
	16	No	N.A.	No	N.A.	No	N.A.

TABLE 4-12

SUMMARY OF PRIVATE CAR OWNERS' RESULTS

FLAT CARS

Component	Population
1. Average Car Age of Sample (Years)	8-9
2. Type of Brake Control Valve	
AB	9,000
AB Plus*	36,000
ABD	28,000
3. Type of Rigging	
Body Mounted	72,620
Slack Adjusters	72,520
Truck-Mounted	380
4. Type of Brake Shoes	
Metal (Cast Iron)	68,000
Hi-Phos	0
Composition	5,000
5. Type of Fittings	
Welded	14,000
Other	59,000
6. Load Sensitive Devices	
Load Proportional	0
Load/Empty	0

Note: * AB Plus represents AB brake valves equipped with brake cylinder release valves.

TABLE 4-13

SUMMARY OF PRIVATE CAR OWNERS' RESULTS

AIR BRAKE SYSTEM COMPONENTS BEING PURCHASED BY PRIVATE CAR OWNERS

Component Being Purchased	
1. Brake Shoes	
Cast Iron	No
Hi-Phos	No
Composition	Yes
2. Brake Mounting	
Body	Yes
Truck	No
3. Load Sensitive Devices	
Load Proportional	No
Load/Empty	No
4. Wheel Type	
One-Wear	Yes
Two-Wear	No
Multiple-Wear	No

TABLE 4-14

PRIVATE CAR OWNER LIFE CYCLE COST DATA

Job Code(s)	Rule(s)	Description	Annual Cost Per Car	Percent of Cost
1000-1120	2	COT&S	$ 13.04	5.8%
1140-1144	3	IDT&S	7.31	3.3
1160-1612	4	Air Brakes and Parts	22.29	10.0
1628	5	Air Brake Hose	6.11	2.7
1640-1812	6-11	Brake Beams and Hangers	7.22	3.2
1828	12, 13	Brake Shoes - Cast Iron	32.03	14.3
1830	12, 13	Brake Shoes - Hi Phos	24.59	11.0
1832	12, 13	Brake Shoes - Composition (LF)	0.03	-
1836-1840	12, 13	Brake Shoes - Composition (HF)	7.13	3.2
1856-1940	14	Hand Brakes	3.15	1.4
3000-3123	41	Wheels (Application Why Made	53.27	23.8
315-3180	41	Codes Only)	47.56	21.3
		Totals	$223.73	100.0%

TABLE 4-15

OPINIONS OF PRIVATE CAR OWNERS CONCERNING

HIGHEST PRIORITY IMPROVEMENTS NEEDED

Priority	Improvement Needed in Brake System
1	Longer lasting brake shoe
2	Faster means of replacing brake shoes
3	Longer lasting wheels
4	Maintenance-free brake control valve (eliminate COT&S)
5	Center-of-car valve to replace angle cocks
6	Automatic air coupling included in coupler (eliminate end-of-car air hose arrangement problems)
7	Control valve designed for long cars to reduce cost of present valves plus auxiliary valves
8	Automatic retaining valve function

5. THERMAL CAPACITY OF FREIGHT CAR WHEELS

5.1 INTRODUCTION

The thermal capacity of freight car wheels represents one of the major design criteria that must be taken into account by the air brake system designer. Either the design braking level must be limited so that the thermal capacity is not exceeded under normal (and abnormal) operating conditions, or the braking effort must be achieved by augmenting tread braking with other forms of braking.

Through various forms of research, several attempts have been made, especially within the past 20 years, to determine the actual thermal capacity of freight car wheels typically found in service in North America. This research has included experimental as well as theoretical approaches. In particular, much use has been made of wheel dynamometers to perform various kinds of experiments to investigate the thermal capacity of freight car wheels.

The purpose of this section is to summarize and discuss the most significant portions of this research activity, and to present a collective assessment of the thermal capacity of freight car wheels.

5.2 PARAMETERS TO CONSIDER

Because of the many variations found in freight train brake equipment, train operations, and other factors, wheel thermal capacity must be evaluated with respect to the many parameters that can affect it. These parameters include the following:

a. Cyclic versus Drag Braking.

b. Wheel Geometry (Size and Shape).

c. Wheel Material and Heat Treatment.

d. Worn versus New Wheels.

e. Brake Shoe Material and Shoe Alignment.

f. Other Considerations.

Each of these topics is discussed in the following in terms of the research efforts performed within recent years.

5.2.1 Cyclic versus Drag Braking - The thermal capacity of a wheel depends upon the types of operations which the wheel experiences in service. In terms of freight service, cyclic braking is used by enginemen to control train slack action when operating long trains over undulating terrain. Although the braking levels used in this type of braking (also called "stretch" braking) are normally not large, the wheels are exposed to alternating cycles of heating and cooling.

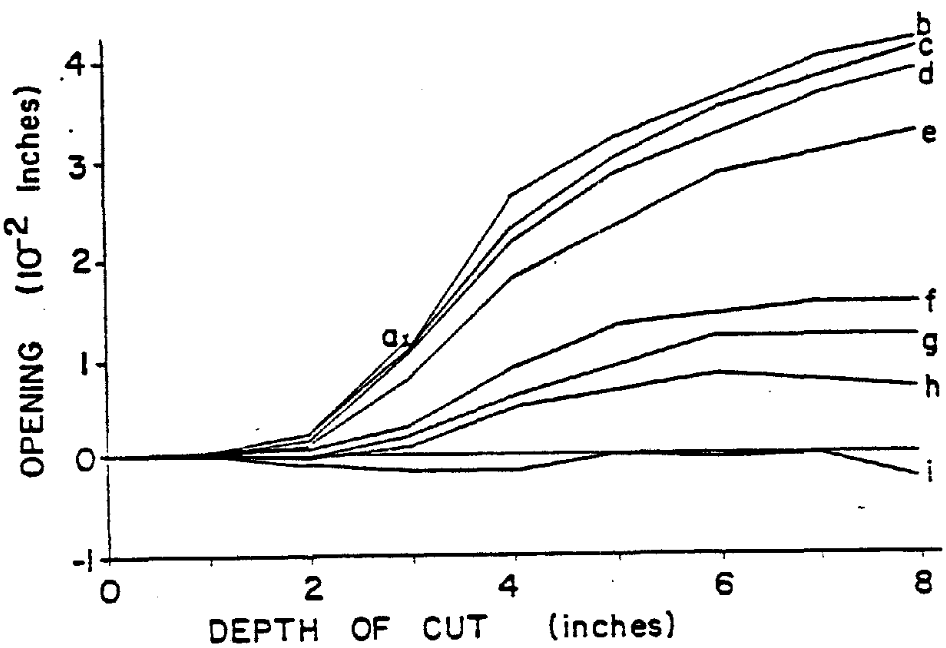
Another form of braking that can be considered as cyclic braking is a full service (or emergency) brake application used by the engineman to produce a large change in train speed (for example, from 50 mph to a stop). Again, the wheel tread temperature increases to a high value in a short period of time, and then drops back to the ambient level at a rate determined by the ambient conditions.

The other type of freight train brake operation, referred to as "drag braking," is normally associated with a steady brake application over a long period of time. This typically occurs when a train is being operated down a long, steep grade where train brakes are applied to "balance" the gravitational forces of the grade.

5.2.1.1 Safe Thermal Load - In a research project sponsored by the Griffin Wheel Company, a safe thermal load was defined as "a thermal load which does not produce inelastic deformation in the wheel" (Wetenkamp and Kipp, 1975, p. 2). Inelastic deformation produced in service because of braking is a manifestation of residual stresses in the wheel which can be caused by large variations in temperature between the wheel tread surface and other portions of the wheel. This variation in temperature, referred to as thermal gradient, is believed by some investigators to be more significant in causing wheel residual stresses than an equivalent (or even higher) tread temperature sustained for a long period of time. In other words, the thermal "shock effect" of cyclic braking is considered by some researchers to be more destructive to the wheel than a slowly applied, uniform, high temperature brake application, followed by a slow cooling process.

To place the thermal capacity of wheels in perspective with respect to cyclic and drag brake applications, the braking levels of both types of applications must be known. For example, a high level of drag braking could theoretically exist at which the wheel temperature would reach a destructive level (assuming the brake shoes would not disintegrate). The wheel would fail and the amount of cyclic braking might not be a factor.

5.2.1.2 Recent Test Methods - In recent tests conducted at the University of Illinois, several types of wheel braking levels and cycles were investigated using a wheel dynamometer. The residual stresses were measured after subjecting the wheels to various braking levels by measuring the opening of saw cuts made into the wheel rims after testing. The results of these tests are shown in Figure 5-1.



Saw cut opening versus depth of cut.

a-100 hp, 15 min (fractured)
b-40 hp, 45 min; 60 hp, 30 min; 80 hp, 15 min; 100 hp, 10 min
c-60 hp, 30 min
d-40 hp, 45 min; 60 hp, 30 min; 80 hp, 15 min
e-80 hp, 15 min
f-100 hp, 10 min
g-40 hp, 45 min
h-100 hp, 5 min
i-Class U wheel (as manufactured)

Figure 5-1

WHEEL STRESSES EXPRESSED IN TERMS OF SAW CUT OPENINGS AFTER EXPOSURE TO VARIOUS CAR BRAKE LEVELS

Source: Wetenkamp and Kipp (1975, Fig. 4). Used by permission of copyright owner.

The researchers made the following conclusions concerning the results shown in Figure 5-1 (Wentenkamp and Kipp, 1975, p. 15):

> "Starting with a given wheel design and heat treatment it is apparent that the more severe the brake application the greater the opening of the wheel upon performing the radial saw cut. The results would imply that the most severe load in this series was the 100 hp (74.6 kw) test for 15 minutes. This type of test will virtually consume a new composition shoe in one test and could possibly be experienced in service with a stuck brake. A substantially more severe brake cycle in freight service is difficult to imagine. The wheel subjected to this thermal load fractured when the radial saw cut progressed through the rim. None of the other wheels fractured upon sawing. Fortunately the limited data have a rather well defined dividing point and for the present it would seem that the thermal loads producing substantially less saw cut openings than the wheel that fractured upon sawing should be reasonably safe."

It should be noted that for the above mentioned case (100 hp, 15 minutes), the wheel did not fracture until the saw cut was made, the tread temperature reached 1,146°F (Wetenkemp and Kipp, 1975, Table I) and a new composition brake shoe was consumed during the test.

How this wheel would have withstood the combined physical loads and repeated thermal cycling of actual train operations can only be surmised.

5.2.2 <u>Wheel Geometry (Size and Shape)</u> - The wheel size (diameter), and the shape of its cross section are additional factors that affect wheel thermal capacity. Both of these factors are discussed below.

5.2.2.1 Wheel Size - In research performed by the Westinghouse Air Brake Company using its full scale, dual wheel dynamometer, an investigation was made into the tread temperatures that would be produced in various sizes of wheels at given levels of horsepower dissipation during braking (Cabble, 1973). The following diameters and types of Class B wrought steel wheels were tested in both the new tread and worn tread condition: 40-inch multi-wear, 36-inch multi-wear, 33-inch multi-wear, 36-inch one-wear, 28-inch multi-wear, and 28-inch one-wear.

The results of these tests are presented in Figure 5-2.

- New Wheels -

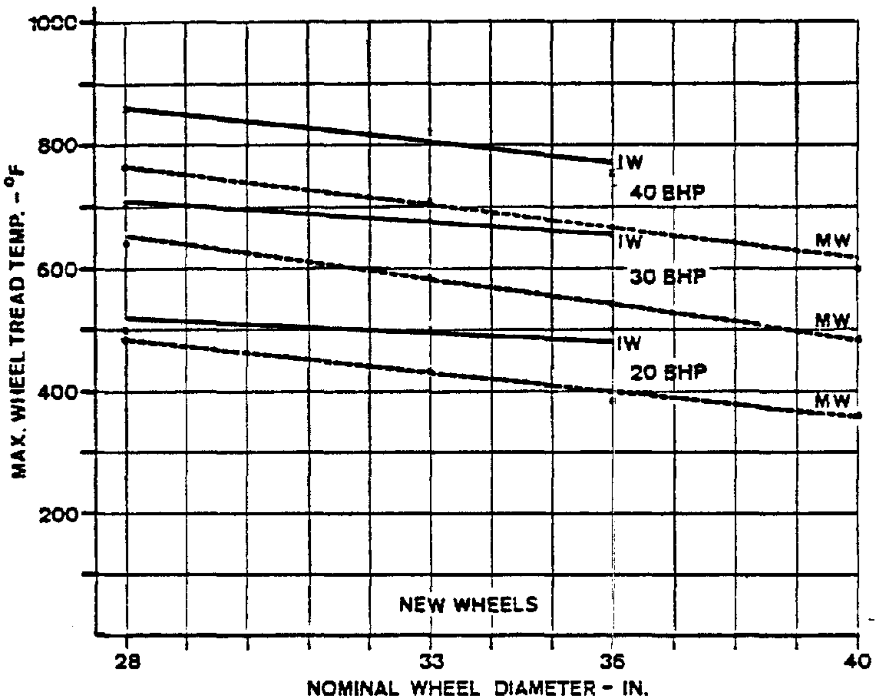


Figure 5-2

MAXIMUM WHEEL TREAD TEMPERATURE VERSUS WHEEL DIAMETER AFTER 60 MINUTES AT CONSTANT BRAKE HORSEPOWER

Source: Cabble (1972). Used by permission of copyright owner.

The conclusions made from the test results were as follows (Cabble, 1972):

a. "As the brake horsepower per wheel is increased, the temperature rise of a given wheel is increased the same length of time."

b. "As the nominal wheel diameter is decreased, the temperature rise of the wheel is increased."

c. "As the actual wheel diameter due to wear is decreased, the temperature rise of the wheel is increased."

d. "The approximate range of temperatures through which these changes occur versus the nominal wheel diameter approximates a straight line."

e. "The maximum wheel tread temperature that a wheel can stand without a problem determines the allowable brake horsepower per wheel and, therefore, the grade braking operation."

5.2.2.2 Wheel Shape - The effects that thermal stresses produced by tread braking, combined with physical stresses produced by vertical and lateral loading, have been investigated by performing analytical stress analysis (Novak et al, 1975). This research was performed in part to explain the difference in the wheel plate service failures of 28-inch wheels with straight and curved cross-sections.

Using two-dimensional finite difference and finite element mathematical methods of calculating octahedral shear stresses, the 28-inch straight plate (B-28) and curved plate (CB-28) wheels were evaluated under the service load application of vertical and lateral wheel loads at the flange throat in combination with an emergency brake application. The load applications are shown schematically in Figure 5-3.

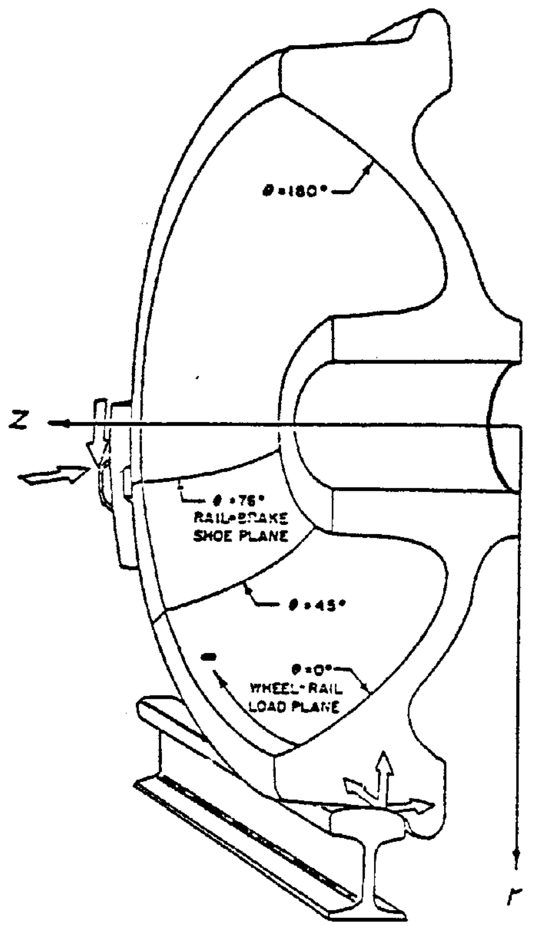


Figure 5-3

COORDINATE SYSTEM, LOAD APPLICATION, AND REFERENCE PLANES TO ANALYZE WHEEL STRESSES

Source: Novak, et al. Used by permission of copyright owner.

The service loads applied were as follows:

a. Car Wheel-Rail Contact Loading

Vertical: 46,100 pounds

Lateral : 20,000 pounds.

b. Tread Braking Load

Brake Shoe Force: 6,200 pounds

Tractive Force : 720 pounds.

5.2.2.3 Thermal Load - The particular thermal load applied to the wheel corresponded to the temperature distribution in the wheel after 55 seconds of brake application when tread surface temperatures were at a maximum. The temperature fields at this braking time interval are shown in Figures 5-4 and 5-5.

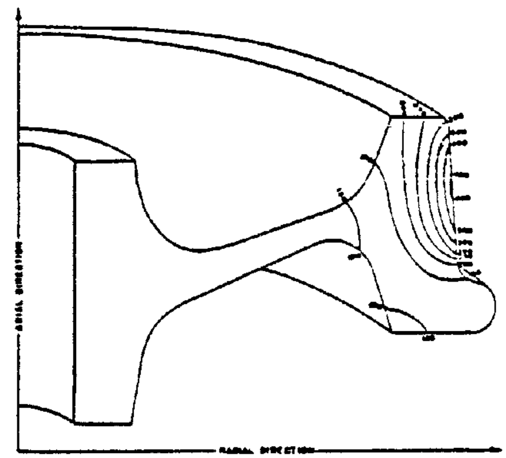


Figure 5-4

STRAIGHT PLATE WHEEL THERMAL LOADING

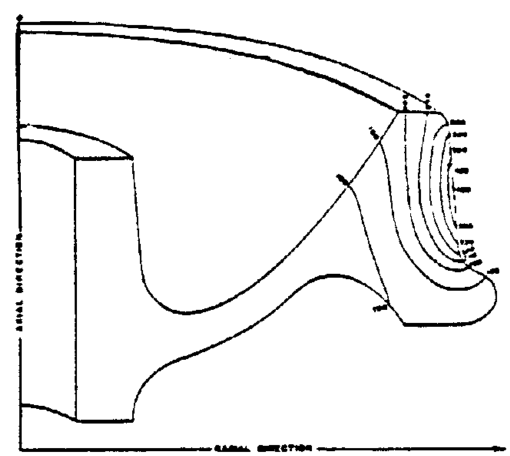


Figure 5-5

CURVED PLATE WHEEL THERMAL LOADING

Source: Novak et al. (1975). Used by permission of copyright owner.

The maximum octahedral shear stresses calculated were described by the authors as follows:

> "The maximum octahedral shear stress produced adjacent to the front hub fillet of the straight plate wheel by this load application is significantly higher than the stress generated by only the rolling loads (vertical or vertical and lateral wheel loads). This maximum stress occurs at approximately 45 degree of wheel rotation, and diminishes as the wheel rotates to the 180-degree position. The maximum octahedral stress intensity of about 13,500 psi (93.08 Mpa) near the back rim fillet also occurs after 45 degree of wheel rotation and diminishes as wheel travel approaches 180 degrees."

The authors noted that, as in previous work, it was found that mechanical wheel-rail contact loads comparable to those found in service may not be sufficient to induce plate failure (in either the straight plate (B-28) or the curved plate (CB-28) 28-inch wheel).

The authors continued:

> "However, introduction of the thermal and mechanical loads produced by emergency braking combined with the mechanical rolling loads produces considerably higher octahedral shear stress levels in the plate section of the B-28 design compared to the CB-28 design. This would indicate that the B-28-inch wheel could develop a greater number of plate failures than the CB-28 depending on the number of braking cycles induced during normal service operation. Since both wheel designs are subjected to essentially the same service conditions, and plate failures have occurred in only the B-28 design, it is most likely that the stresses in the CB-28 plate are below the threshold necessary to initiate a fatigue failure."

5.2.3 <u>Wheel Types</u> - There are several types of wheels used in the railroad industry today. These types vary according to material, method of manufacturing, and design differences.

Presently both wrought steel and cast steel wheels are approved for replacement and new car construction. Cast iron wheels were prohibited from Interchange Service after January 1, 1970.

5.2.3.1 Cast Steel Wheels - Cast steel wheels are made in one-wear, two-wear, and multiple-wear designs and are furnished in various heat treated classes and carbon levels according to AAR specifications (Association of American Railroads, 1973).

Cast steel wheels are permitted to cool in their molds to complete solidification. The wheels are removed from the mold and cooled under controlled conditions either before or after the center of the hub is removed to form a rough bore.

All cast steel wheels receive heat treatment to produce the desired metallurgical changes and to provide a favorable stress distribution in the wheel.

Except for one class of cast steel wheels (Class U), all cast steel wheels receive a quenching treatment on the tread surface and are produced to the hardness required by AAR specifications. Classes A, B, and C wheels vary in their carbon content and hardness.

5.2.3.2 Wrought Steel Wheels - Wrought steel wheels are made by successive hammering and squeezing operations on hot steel blocks. After final shaping the wheel is control-cooled. Wrought steel wheels are often heat-treated after initial cooling, either fully or rim-quenching only.

The purpose of the heat treatment is to achieve further modification in the wheel metallurgy to strengthen and harden the wheel.

5.2.3.3 AAR Specification Governing Wheel Manufacture - The AAR specifications define the chemical requirements for wheel carbon content and impurities, the physical requirements of hardness at various points on the wheel, and the wheel surface finish requirements. The wheel classifications incorporating these requirements are shown in Table 5-1.

TABLE 5-1

BASIC WHEEL CLASSIFICATION

Class	Treatment	Carbon Content		Recommended Use
		Minimum	Maximum	
U	Untreated	0.65	0.80	General service where untreated wheels are satisfactory.
U1	Untreated	0.95	1.20	
A	Treated	0.47	0.57	High speed service with severe braking conditions but moderate wheel loads.
B	Treated	0.57	0.67	High speed service with moderate braking conditions and heavier wheel loads.
C	Treated	0.67	0.77	Services with light braking conditions and high wheel loads, or service with heavier braking conditions where off tread brakes are employed.
L	Treated	----	0.47	High speed passenger and transit.

Source: Simmons-Boardman (1973).

Class A wheels (lowest carbon range) are the most resistant to thermal cracking. Class C wheels are more subject to thermal cracking, but have improved wear properties under heavy loads. Class B wheels represent a compromise between Class A and Class C wheels.

5.2.4 Worn Versus New Wheels - The amount of wheel wear also affects the thermal capacity of the wheel. In general, the more worn the wheel is, the higher its tread temperature, and the lower its thermal capacity at a given level of braking. This is especially true for small diameter wheels.

In tests performed by the Westinghouse Air Brake Company wheels were subjected to constant brake horsepower levels for long periods of time (up to 60 minutes). Figure 5-6 compares the temperature-versus-time curves for new wheels and worn wheels used in these tests.

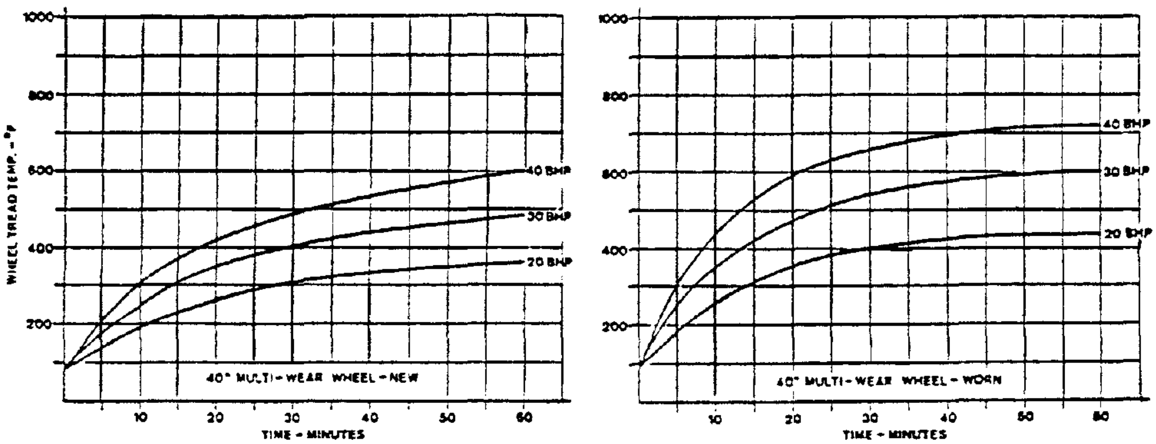


Figure 5-6

COMPARISON OF TREAD TEMPERATURE VERSUS TIME CURVES FOR NEW AND WORN 40-INCH MULTI-WEAR WHEELS

Source: Cabble (1973, Figs. 3 and 11). Used by permission of copyright owner.

The maximum tread temperature reached at 40 bhp was approximately 600 °F for the new wheel versus approximately 720 °F for the worn wheel, an increase of 120 °F for the worn wheel compared to the new wheel.

The higher temperature effects of worn wheels are even more apparent for the smaller diameter wheels. This is shown in Figure 5-7.

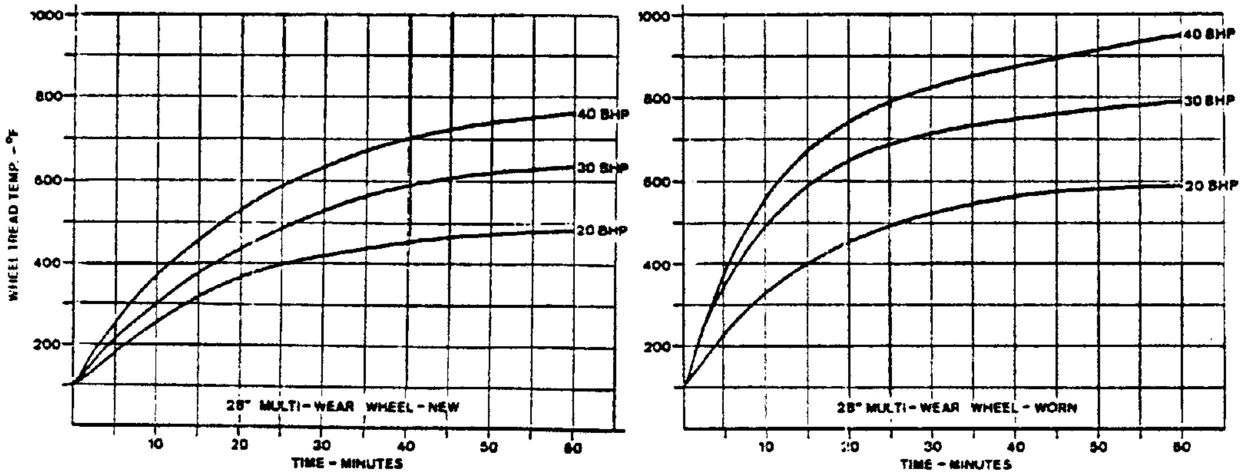


Figure 5-7

COMPARISON OF TREAD TEMPERATURE VERSUS TIME
FOR NEW AND WORN 28-INCH MULTI-WEAR WHEELS

Source: Cabble (1973, Figs. 6 and 14). Used by permission of copyright owner.

The maximum tread temperature after 60 minutes at 40 bhp for the new 28-inch wheel was approximately 760 °F versus approximately 950 °F for the worn wheel, an increase of 190 °F for the worn wheel compared to the new wheel.

Closely related to the amount of wheel wear and its effects upon maximum tread temperature is the wear category (multi-wear versus one-wear) of the wheel. This effect can be seen in Figure 5-8 which summarizes test results by showing the maximum temperatures reached for each wear category of the wheels tested.

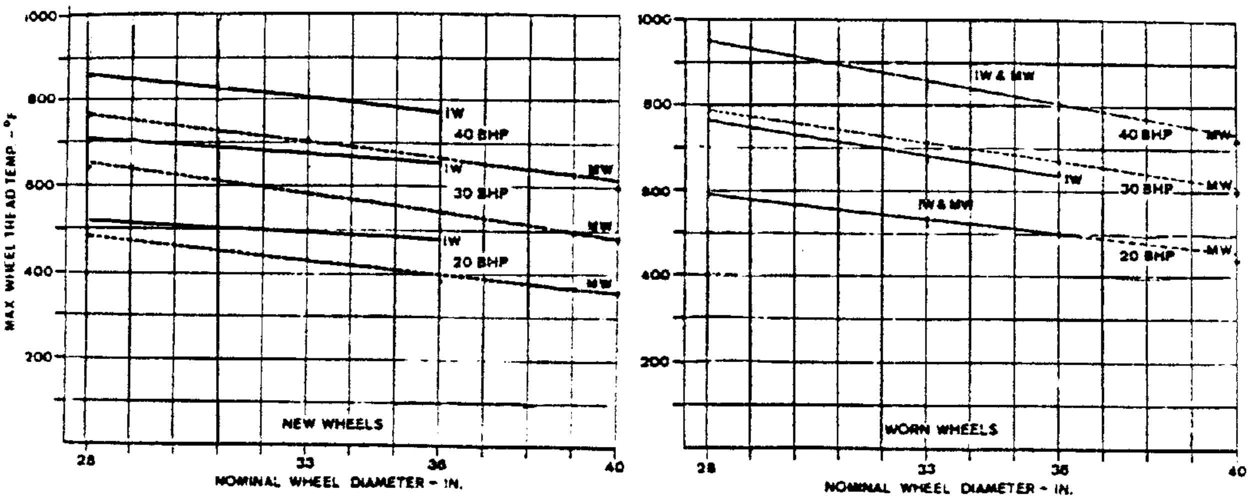


Figure 5-8

SUMMARY OF WHEEL TEMPERATURE TEST RESULTS
COMPARISON OF WEAR CATEGORIES

Source: Cabble (1973, Figs. 10 and 18). Used by permission of copyright owner.

Figure 5-8 illustrates the following phenomena:

1. In general, one-wear wheel temperatures are higher than multi-wear wheel temperatures when the wheels are new; for worn wheels, the differences between one-wear and multi-wear temperatures become almost negligible.

2. Wheel tread temperature increases inversely with wheel diameter, and directly with the amount of wheel wear.

3. When wheels are new, the temperature-size relationship is more significant for multi-wear wheels, however, when the wheels become worn the effect is negated.

5.2.5 Brake Shoe Material and Alignment

5.2.5.1 Brake Shoe Material - Up until about 1969, almost all wheel thermal capacity tests were performed using cast iron brake shoes held in a clasp arrangement (Weaver et al, 1969). In 1969, a report was published by the ASME that documented the testing of Class C wheel thermal capacity using composition

COBRA* brake shoes.

This report briefly documented earlier tests performed at the University of Illinois in 1950. These tests were "stop tests" that simulated a rapid deceleration rate from 115 mph. In these tests a simulated wheel load of 20,000 pounds was used in conjunction with a brake shoe force of 20,000 pounds on each of the two cast iron brake shoes held in clasp arrangement. The braking duty represented a total kinetic energy of 9,000,000 foot-pounds per stop. The results of these tests, as summarized by the authors, are:

> "Of the five Class CR wheels (Class C rimtreated) which were tested in this manner, three developed thermal cracks after only one stop. The remaining two wheels developed cracks after five stops and seven stops, respectively."

A following series of tests performed were documented by Wardisco and Dewey (1960). These tests simulated stops with a 31,000 pound wheel load instead of the 20,000 pound load of the above (13) tests. The results of these tests were as follows:

> "This study was extremely valuable in that it resulted in a much better understanding of the mechanism theories for the formation of defects that develop in wheels as a result of braking heat. <u>It is noteworthy to extract one particular result of these tests which supports the results obtained in 1950 at the University of Illinois: The average number of stops required to produce a thermal crack in CR wheels was one stop</u>."

In 1968, another test series was conducted with the purpose of obtaining a direct comparison of wheel thermal capacity of wheels braked with a single flanged COBRA composition brake shoes and wheels braked with cast iron shoes. A secondary purpose was to verify previous test results with cast metal shoes. For these tests, dynamometer stops were partially oriented towards "Metroliner" braking conditions. The test parameters are shown in Table 5-2.

---

* COBRA is a registered trademark of Railroad Friction Products Corporation.

TABLE 5-2

TEST PARAMETERS OF CAST IRON AND COBRA BRAKE SHOE COMPARISON TEST

Parameter	Wheel Number 8925 Cast Metal Brake Shoes	Wheel Number 10,793 COBRA Flanged Brake Shoes
Wheel location	Outside	Inside
Equivalent wheel load	23,128 lb	23,083 lb
Total simulated car weight	185,024 lb	184,644 lb
Brake arrangement	Clasp Shoes	Single Shoes
Net shoe force per wheel	19,150 lb	7,419 lb
Initial speed	110 mph	110 mph
Kinetic energy per stop foot-pounds	9,359,413	9,341,203
Average stop distance	6,845 ft	5,225 ft
Average stop time	68.02 sec	66.31 sec
Average work rate per stop foot-pounds-per-second	137,598	140,872
Average maximum wheel tread temperature	1,033 degrees F	664 degrees F
Average retardation rate	1.62 mphps	1.62 mphps
Total braking work done on wheels* foot-pounds	470,379,039	697,053,524

Note: * Includes wear-in stops.

Source:: Berg, N.A., Kucera, W. J., "A Review of Thermal Damage in Railroad Wheels", September 15, 1970, Note 13, Page 368, Table 1.

The test results were summarized by the authors as follows:

> "After approximately twenty emergency stops were completed on each wheel, wheel No. 8925 (metal brake shoes) began to develop hairline, skin-deep thermal cracks. The abundance of these thermal cracks increased with each successive stop. When thirty-seven stops were completed on wheel No. 10793 with the flanged COBRA brake shoe, and the dynamometer was being accelerated for the thirty-seventh stop on wheel No. 8925, a sharp "ping" was heard. Instead of proceeding with the emergency stop, the machine was stopped without brakes on the wheel in question. An examination revealed a severe thermal crack

extending almost across the complete width of the tread of wheel No. 8925. The tests were discontinued at this point.

This braking duty, dissipating over 9,000,000 foot-pounds of kinetic energy at rates of over 137,000 foot-pounds-per-second with cast iron brake shoes and over 140,000 foot-pounds-per-second with the COBRA brake shoe, is severe by any practical service standards, and exceeds the previously recommended limit of 125,000 foot-pounds-per-second."

In another paper presented at the Air Brake Association Annual Meeting (Berg and Kucera, 1970) further test results were documented. In particular, the following conclusions were made:

a. "The initiation of the thermal cracks in dynamometer testing is well documented...and offers strong proof that the wheel-rail load is not a necessary component in thermal crack formation."

b. "It has been virtually impossible for us to induce thermal cracks using composition brake shoes on full scale dynamometers. It is no problem at all to induce thermal cracks with cast iron shoes under these circumstances."

c. "Thermal cracks, once formed, can be propagated on a dynamometer using either type brake shoe."

The second conclusion stated above has been explained by examining the tread heating pattern characteristics of the cast iron and the composition brake shoes. These characteristics are compared on the following page in Figure 5-9.

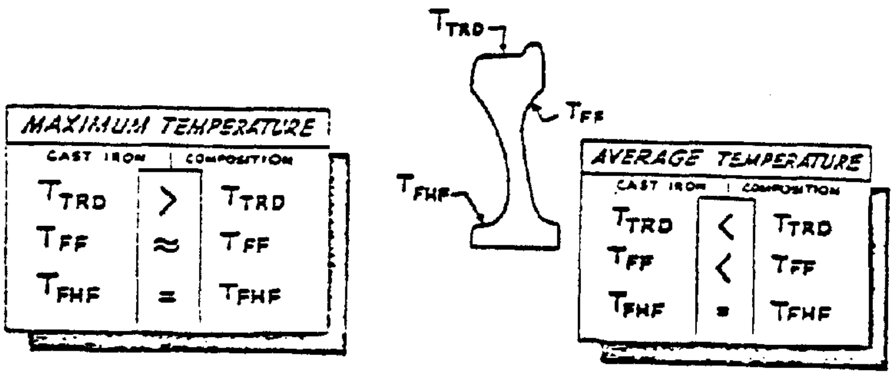


Figure 5-9

COMPARISON OF AVERAGE AND MAXIMUM TREAD TEMPERATURE OF CAST IRON VERSUS COMPOSITION BRAKE SHOES

Source: Berg and Kucera (1970, p. 17, Figs. 15 and 16). Used by permission of copyright owner.

Figure 5-9 characterizes the uneven heating of the wheel by the cast iron brake shoe.

The maximum tread temperature for a composition shoed wheel is more equal to its average tread temperature than an equivalent wheel braked by a cast iron brake shoe. At a 110 brake horsepower dissipation rate, the difference between maximum and average temperatures can be as much as 500 percent (Berg and Kucera, 1970, p. 18). This temperature differential is one of the primary causes of thermal damage.

5.2.5.2 Brake Shoe Alignment - Brake shoe misalignment is also known to affect the thermal capacity of wheels at a given level of brake shoe force. Berg and Kucera (1970) made the following statements concerning brake shoe alignment (p. 15):

> "Not enough emphasis can be placed on the importance of having properly aligned brake rigging. Theory has been confirmed by service experience that localized heating has a detrimental effect on the performance of the wheel.
>
> A British hypothesis put forward earlier this year, suggested that the rolling contact action between the wheel and rail produced beneficical surface residual compressive stresses. A thermal crack, they claim, cannot cause a brittle fracture until it penetrated this layer.

They furthermore say that the brake shoes must not be allowed to wander over to the front rim face lest they initiate a thermal crack in an area unprotected by preceding rolling contact. It is for this same reason the British railroads have abandoned the use of flanged shoes except for slow speed vehicles."

5.2.6 Other Considerations - Several other considerations must be investigated in estimating wheel thermal capacities. These are briefly discussed in the following.

5.2.6.1 Aerodynamic Cooling - Most of the research performed to investigate wheel thermal capacity did not include the beneficial cooling effect that wheels would receive in actual train operation. Although some of the researchers do position electric fans to circulate air over the wheel being tested, none of the research results quantified the effects of the cooling air flow.

Closely related to aerodynamic cooling is the ambient air temperature of the environment through which the train is being operated. Based upon the experimental and theoretical results found in research papers, it can be assumed that the most severe wheel temperature condition would be produced by a severe brake application cycle made on a wheel that is operating within a low ambient temperature.

5.2.6.2 Variations in Brake Forces - Although brake systems are designed to produce uniform braking levels on all wheels of a car, experience indicates that this is not always the case. Because of binding rigging, misapplied brake shoes, and other causes, actual brake shoe forces can vary substantially from one wheel to another on the same car. This can cause higher than normal wheel tread temperatures for one or more wheels within the train.

5.2.6.3 Slid-Flat Wheels - Slid-flat wheels are a common problem with present train brake systems. It is rare that a freight train can be observed passing without at least one pair of slid-flat wheels of one degree or another. Slid-flats can be caused by unreleased hand brakes, misapplied brake shoes, malfunctioning slack adjusters, exceptionally high brake pipe pressure settings, binding brake rigging, malfunctioning control valves, and probably many other causes. Regardless of the cause of a slid-flat wheel, the sliding process itself generates a high, localized wheel tread temperature. For an empty car this temperature would conservatively correspond to a brake horsepower dissipation rate of approximately 100 bhp/wheel*.

5.3 SUMMARY

Because the process of estimating wheel thermal capacity must take into account several variables as discussed previously, and because the individual research series performed to date typically included only a few of these variables in any given test series, it is difficult to provide a collective assessment of the thermal capacity of freight car wheels subjected to tread braking. However, a summary can be made concerning the nature of the braking technology and operating practices that can return the most benefits from braking, while simultaneously maintaining a safe thermal loading. This summary is presented below.

5.3.1 Summary of Research Results

5.3.1.1 Type of Brake Shoes - Research has shown that the composition brake shoe produces lower wheel temperature gradients at an equivalent horsepower dissipation rate than those produced by cast iron shoes. Two reasons have been put forth to explain this result:

a. The tread temperature distribution across the tread (in the axial direction) is more uniform for a composition brake shoe than for a cast iron shoe. A cast iron shoe tends to form "rings" or bands of extremely high temperatures on the tread which are local "hot spots."

b. The more uniform temperature distribution produced by the composition brake shoe causes a smaller temperature gradient (in the radial direction) and lower wheel stresses.

---

* Using an empty car weight of 30 tons, a low coefficient of friction between the wheel and the rail of 10%, and a speed of 50 mph produces the following bhp: $(30 \text{ tons} \times 2{,}000 \text{ pounds/ton} \times \frac{1}{8 \text{ wheels}} \times 0.10 \times 50 \text{ mph})$ 375 mi-lb/hp-hr = 100 bhp/wheel.

Furthermore, as stated in the results of one test series (Berg and Kucera, 1970) it has been virtually impossible to induce thermal cracks in a wheel using composition brake shoes on full scale dynamometers, although once cracks are formed by some other means, the cracks can be propagated using a composition shoe. It should be further noted that, in service tests using dynamometers, the composition shoes in most cases were consumed before wheel thermal cracks occurred.

5.3.1.2 Wheel Diameter and Cross-Sectional Shape - Test results indicated that large-diameter wheels exhibit lower tread temperatures at a given level of braking than small-diameter wheels. For new multi-wear wheels, tread temperatures produced on a 40-inch wheel were as much as 25 percent lower than the temperatures produced on a 28-inch wheel, although this effect was not as significant for worn wheels. Theoretical analyses of the 28-inch wheel have indicated that under combined physical and thermal loads, curved-plate 28-inch wheels exhibited substantially reduced wheel stresses compared to straight-plate 28-inch wheels. This result was substantiated by the fact that wheel plate failures in selected freight train operations occurred exclusively with straight-plate wheels.

5.3.1.3 Type of Brake Rigging - To avoid extreme variations in braking level on individual wheels of a freight car, the brake rigging must be capable of providing a uniform, balanced brake shoe force on each wheel. Also important is the selection of brake rigging that provides proper brake shoe alignment.

5.3.1.4 Wheel Metallurgy - Although the metallurgical content of wheels is typically selected based upon other considerations (in particular, shelling, spalling, and wear characteristics), from the standpoint of wheel thermal capacity, wheels of low carbon content are the most resistant to thermal cracking.

5.3.1.5 Brake Application Cycles; Speed in Grade Operations - Temperature "shock" effects have been demonstrated to be damaging to wheels, as have severe brake applications over long time periods. Train operations should be managed such that cyclic braking of severe nature should not be required, and so that descending-grade train speeds are limited to a safe level of horsepower dissipation (See Figure 5-10).

5.3.2 Safe Energy Dissipation Rates - The purpose of the above summary was to present favorable conditions for obtaining the freight car wheels using ontread braking techniques.

The safe energy dissipation rate for freight car wheels has been tentatively determined (for 33-inch Class U wheels) by Wetenkamp and Kipp (1975) in research performed at the University of Illinois, sponsored by Griffin Wheel Company. The tentative safe energy dissipation-versus-time curve is shown in Figure 5-11. It should be noted that these results do not include the physical, mechanical wheel loadings that would be experienced in service.

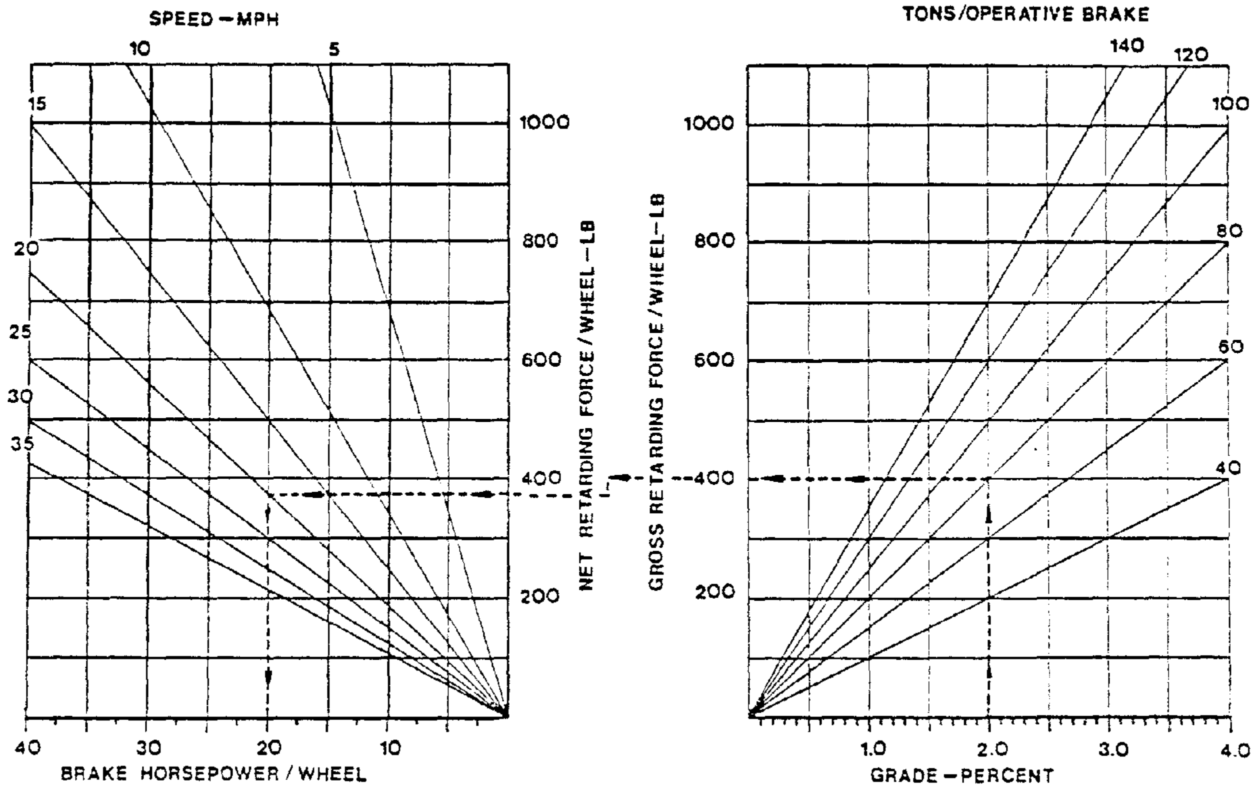


Figure 5-10

NOMOGRAPH FOR CALCULATING BRAKE HORSEPOWER PER WHEEL FOR VARIOUS FREIGHT TRAIN DRAG OPERATIONS

Source: Cabble (1973). Used by permission of copyright owner.

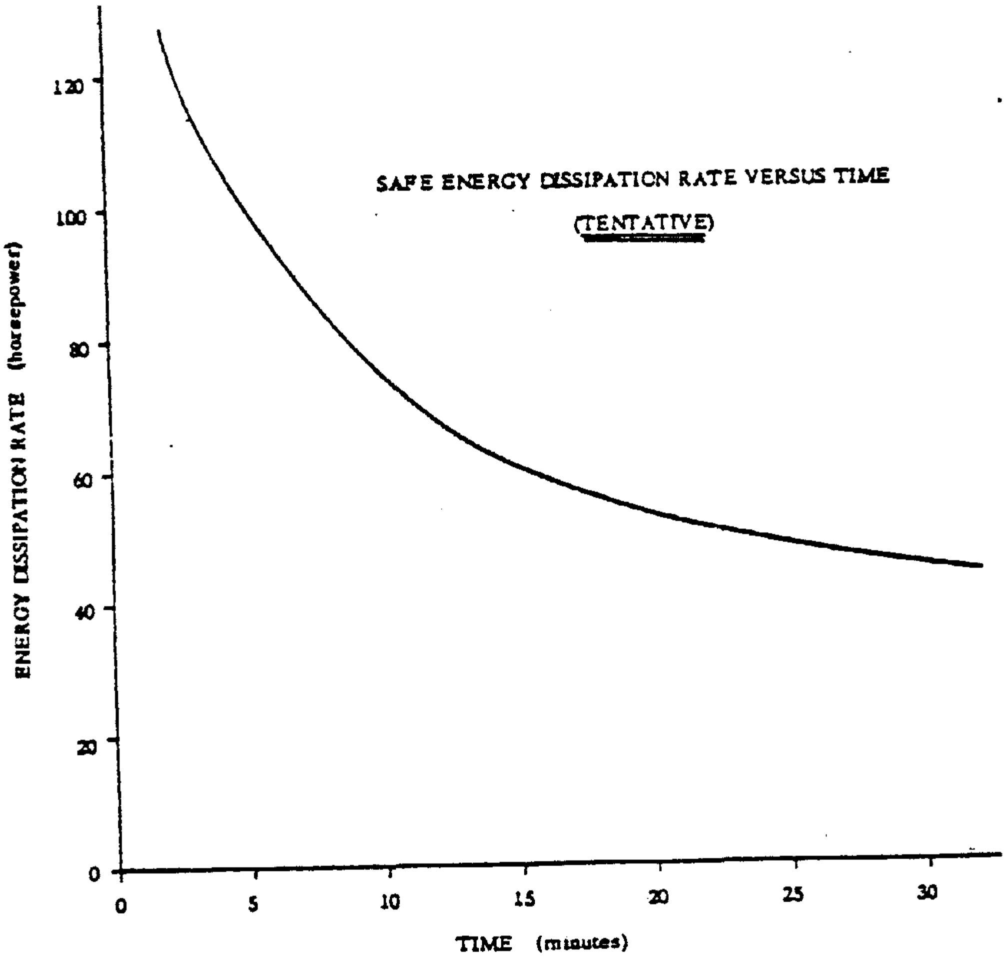


Figure 5-11

TENTATIVE SAFE ENERGY DISSIPATION RATE VERSUS TIME

Source: Wetenkamp and Kipp (1975). Used by permission of copyright owner.

The tentative safe energy dissipation rate curve asymptotically approaches the 40 bhp level with time, being substantially greater than 40 bhp for short time periods.

Various discussions with researchers and brake system professionals have generally revealed that the curve presented in Figure 5-11 is considered to be conservative. This is especially true if the guidelines presented previously are followed.

APPENDIX A - GLOSSARY OF TERMS

AB VALVE (Including related valve schedules such as ABD, ABDW and Z1A) - the pneumatic control valve on each car that controls the brake operation of the car.

Accelerated Emergency Release - A brake release feature where each car assists in recharging the brake pipe by permitting emergency brake cylinder pressure to flow into the brake pipe for the initial portion of the brake pipe recharge.

Accelerated Service Release - A brake release feature designed into the ABD brake valve which functions to assist brake pipe recharging after a service application by permitting air under pressure from the emergency reservoir of each car to flow into the brake pipe.

Adhesion - The coefficient of friction between the wheel and the rail. Normally expressed as a percentage in terms of maximum tractive force of a rail vehicle divided by its gross weight.

Air Brake System - All of the devices and components included on a trail that can act to limit train speed.

Air Hose - A flexible hose attached to the end of the brake pipe on each car which is manually connected to a mating hose of an adjacent car to form an effectively continuous brake pipe.

Air Reservoir - A twin compartment (auxiliary and emergency) tank mounted on each freight car that stores compressed air for use in brake applications.

Angle Cock - A manually operated valve located at each end of locomotives and cars used to open or close the brake pipe.

Automatic Air Brake - An arrangement of brake equipment which stores energy for brake application. For the standard air brake system, the braking energy is stored as compressed air in reservoirs on cars and locomotives. A reduction in brake pipe pressure, from whatever cause, tends to cause a brake application. An increase in brake pipe pressure tends to release the brakes.

Automatic Brake Valve - A manually operated device on the locomotive which can be positioned by the engineman to: (1) control the flow of air into the equalizing reservoir and brake pipe for charging them and releasing a brake application, and (2) provide a reduction of equalizing reservoir and brake pipe pressures to effect a service or emergency rate of brake application.

Automatic Slack Adjuster - An appliance to restore automatically the brake cylinder piston travel to a predetermined distance and thereby, compensate for brake shoe wear.

Auxiliary Reservoir Compartment - A storage volume for compressed air which is charged from the brake pipe and which provides air pressure for use in service and emergency brake applications.

"B" End of Car - The end of a freight car on which the hand brake assembly is located.

Bleed or "Bleed Off" - A term commonly used for the venting of air pressure to the atmosphere, as in the venting of the air pressure from the brake cylinder of air reservoir of individual cars by manual manipulation of a release valve. The operation of the release valve depends on the type of brake equipment installed on the car.

Brake Beam - A structure located on the freight car truck that transmits the braking force from the brake cylinders to the wheels.

Brake Cylinder - A cylinder within which compressed air from the air reservoir acts on a piston which transmits the force of the compressed air to the associated brake rigging which in turn forces the brake shoes against the wheels.

Brake Cylinder Release Valve - A manual device for quickly releasing the air from a brake cylinder without depleting the air stored in the reservoir on the car. This action is called bleeding.

Brake Hanger - The part of the foundation brake rigging that holds the brake head or beam in position.

Brake Head - A holder attached to or a part of the brake beam which carries the detachable brake shoe.

Brake Pipe (Train Line) - The interconnected air brake piping of cars and locomotives(s) which acts as a supply pipe for the reservoirs on each car and is the sole connecting means by which the train brakes are controlled by the locomotive engineman. Flexible air hoses provide connections between the cars. When a train is made up and all air hoses are connected, the entire pipeline comprises what is commonly called the brake pipe.

Brake Pipe Gradient - A term used to express the difference between air brake pipe pressure at the head-end and rear-end of a train.

Brake Pipe Pressure - The pressure of the air in the brake pipe expressed in pounds per square inch.

Braking Force - The total force (in pounds) pressing the brake shoes against the wheels.

Braking Power - A term used to describe the ability to either control train speed on a descending grade, or to bring a moving train to a controlled stop.

Braking Ratio - The ratio obtained by dividing the total braking force between the brake shoes and the wheels by the weight of a car or locomotive. It is usually expressed as a percentage value. Gross braking ratio is the ratio of the theoretical (without efficiency losses) total brake shoe force divided by the car weight in its loaded or empty condition. Net braking ratio includes efficiency losses of the rigging and is the (actual) brake shoe force divided by the weight of the car.

Brake Shoe - A replaceable friction element secured to the brake head for the purpose of producing a retarding force when forced against a wheel tread.

Brake Shoe Key - A key by which a brake shoe is positioned and fastened to a brake head.

Brake Shoe Force - The total force in pounds exerted by the brake shoe on all wheels when the brakes are applied.

Brake System - Includes all brake apparatus working harmoniously on railway vehicles. (Same as Air Brake System.)

Buff - Compressive coupler forces.

Center Sill - The central longitudinal member of the underframe of a car which transmits longitudinal forces from one end of the car to the other.

Clasp Brake - A brake arrangement having two brake shoes per wheel.

Control Stand - The upright column to which the locomotive throttle control, reverser handle, transition lever and dynamic braking control are mounted in a locomotive cab within convenient reach of the engineman. Air gauges and some other control switches are also included on the control stand.

Cyclic Braking - A term used to describe the alternative brake application and release used by the engineman to control slack-action forces when operating trains over undulating terrain.

Direct Release - The normal total release functioning of freight car brake equipment. When the control valve is moved to release position by increasing brake pipe pressure, brake cylinder pressure is exhausted, thereby removing the retarding forces.

Draft - A term used to describe tensile coupler forces.

Draft Gear - A shock cushioning unit installed at each end of a car or locomotive, to which the coupler is attached, which transmits compression (buff) and tension (draft) forces between the coupler and the center sill of the car or locomotive. The draft gear is within the coupler yoke.

Driver - A locomotive wheel connected to the propulsion system capable of providing tractive effort.

Dynamic Brake - Dynamic braking is an electrical means used to convert some of the energy of a moving train into an effective retarding force by causing the traction motors to function as generators, the energy so generated being dissipated as heat from resistors.

"Dynamiter" or "Kicker" - A slang term for a car with a defective brake valve which spontaneously creates an emergency brake application throughout the train. Common causes are stuck control valve, or the opening of these valves due to the vibration and/or slack-action of the car.

Emergency Application - A rapid exhausting of air pressure from the brake pipe which exceeds the service rate of reduction and trips the brake pipe vent valves on each car, resulting in a controlled stop of the train in the minimum safe distance.

Emergency Reservoir Compartment - A storage volume of the air reservoir for compressed air to provide air pressure for use in emergency brake applications and for certain recharge features (for example, accelerated service release feature of ABD valve).

End-of-Car Cushioning Device - A unit installed at both ends of a car for the purpose of absorbing energy through a hydraulic piston arrangement supplemented by springs. These are used to minimize or prevent damage to lading.

Equalizing Reservoir - A small reservoir on the locomotive which is connected to an equalizing piston of diaphragm chamber for use in automatic air brake applications. The equalizing reservoir lends stability by drawing air at a controlled rate from the brake pipe.

Failsafe - A term used to describe the characteristic of the braking system in which a train separation can cause an emergency brake application automatically.

Feed Value (Regulating Valve) - The valve that reduces air pressure from the reservoir of the locomotive to the pressure desired in the brake pipe. The valve automatically maintains that pressure when the automatic brake is in running position.

Flat Spot - Loss of roundness of the tread of a railroad wheel caused by wheelsliding. Also called a "Slid-Flat."

Foundation Brake Rigging - The levers, rods, brake beams, etc., by which the piston rod of the brake cylinder is connected to the brake shoes in such a manner that when air pressure forces the piston out the brake shoes are forced against the wheels.

Full Service Application - A term used to define the application of the automatic air brake to the point that the auxiliary and brake cylinder pressures are equalized. This is the normal method of stopping a train.

Graduated Release - A feature designed into brake equipment, whereby brake cylinder pressure may be reduced in steps proportional to increments of brake pipe pressure buildup. This feature is not used in North American rail freight operations.

Hand Brake - An arrangement of levers, rods, and gears which are actuated manually to force the brake shoes against the braking surfaces to hold a car stationary.

Hot Journal Detector - A wayside located device that monitors the axle bearing temperatures of a passing train.

Independent Brake Valve - A brake valve that provides independent control of the locomotive brakes regardless of the automatic brake valve handle position.

Lever Ratio - The mechanical advantage obtained through a system of levers; generally in air brake practice, the ratio of the braking force pressing shoes against the wheels to the cylinder force.

Live Levers - Levers not having fixed or nonmovable connection at any point.

L/V Ratio - Defined as the ratio of the lateral force to the vertical force of a car or locomotive wheel on a rail. It is an important indicator of wheel climb, rail turnover and/or derailments.

Main Reservoir - A reservoir on the locomotive for storing and cooling compressed air.

Net Braking Ratio - (See "Braking Ratio.")

Overcharge - A condition wherein car air reservoirs have been charged to a high brake pipe pressure, and then placed in a train operated at a lower pipe pressure, thereby preventing the brakes from releasing.

Over-Reduction - A service brake pipe reduction to a pressure lower than that at which the reservoir and the cylinder equalize.

"P" Wire - An electronic system controlling braking in a electro-pneumatic brake system.

Power Braking - A term used to describe the operation of a freight train in which locomotive power is used in conjunction with light train brake application to stretch the train and control slack-action. Also called "stretch braking."

Pressure Maintaining Feature - A system designed to overcome normal brake pipe leakage and maintain the brake pipe pressure at the desired level during a service reduction.

Pressure Retaining Valve - (See "Retaining Valve.")

Propagation - The serial action of transmitting a brake application from car to car through a train, such as in quick action, emergency or quick service.

Quick Action - The feature whereby the emergency brake pipe reduction is passed rapidly from car to car throughout the train.

Quick Service - A feature which provides a controlled service rate of brake pipe reduction through a local reduction of brake pipe pressure at each car.

Quick Service Limiting Valve - A portion of a brake valve that nullifies further quick service activity when the brake cylinder or displacement reservoir pressure reaches approximately 10 psi.

Quick Service Valve - A device auxiliary to the brake valve to assist in reducing brake pipe pressure, providing continuous quick service regardless of brake cylinder or displacement reservoir pressure. Each brake pipe reduction will cause the quick service valve to respond.

RCE-1 Unit - A locomotive unit or consist placed in the rear of the train and operated remotely, usually by radio link.

Reduction (of the brake pipe) - A decrease in brake pipe pressure at a rate and of an amount sufficient to cause a train brake application to be initiated or increased.

Reduction Relay Valve - A combination of an emergency vent valve and a continuous quick service valve mounted on a common pipe bracket, and designed for application to long freight cars to offset the effect of the increased volume and length of brake pipe per car by promoting quick service activity as well as ensuring the transmission of an emergency application through a train of long cars.

Release Rod - A small rod situated at the side sill of a car, for the purpose of manually releasing the air brakes at terminals to allow switching. (See "Bleed.")

Remote Consist - Designation for a locomotive consist and RCE-1 radio equipment that is placed in the body of a train and is controlled via radio by the engineman in the lead locomotive (or control) consist.

Remote Control Car (RCC) - A vehicle usually a locomotive shell or boxcar in which the remote RCE-1 equipment is installed. The RCC is then connected in multiple-unit mode to the motive power of the remote consist to control its operation. Both the RCC (if one is used) and the remote motive power make up the remote consist.

Remote RCE-1 Radio Equipment - Electrical equipment used to translate radio commands into control operations and to telemeter the status of the remote consist to the control unit. This equipment is permanently installed in either a remote control car, or a remote locomotive.

Retaining Valve - A manually operated valve through which the brake cylinder air is exhausted completely or a predetermined brake cylinder pressure is retained during the brake "release."

Rigging - The system of rods and levers which amplify and transmit the braking force from the brake cylinder to the brake shoes of a car.

Running Release - Release of a service application while the train is in motion.

Service Application - A gradual reduction of brake pipe pressure at a rate and amount sufficient to cause the brake valve to move to service position.

Service Brake - A brake application at a service rate and limited to a brake cylinder pressure less than emergency; i.e., the normal train stopping condition.

Service Rate - The rate, slower than emergency, which the brake pipe pressure reduces to cause the brake valves to assume service position.

Slack - There are two kinds of slack: one is termed " free slack" and is the accumulation of clearances and wear in the associated parts of the couplers. The other type of slack is called "spring slack" and results from extension of the draft gears.

Sliding Center Sill Cushioning Devices - Equipment installed between a fixed center sill and an auxiliary sliding sill that absorbs shock to the car. The sliding sill travels longitudinally through the fixed sill and acts as a single unit throughout the car.

Split Reduction - A term used to describe the process of making an initial brake pipe reduction to a lesser degree than the fully desired reduction, followed by further reductions until the desired total amount is reached. A smoother slowdown or stop is the principal advantage of this method, if properly performed.

Stretch Braking - (See "Power Braking.")

Tare Weight - The weight of the empty car.

Thermal Cracking of Wheels - Cracks in a railroad wheel due to excessive heat.

Tons per Operative Brake - The gross training tonnage of the train divided by the total number of cars having operative brakes (typically, 100%.)

Undesired Emergency - That situation whereby the train brakes apply in emergency (air brake application) from causes other than the engineman's actions. (See also "Dynamiter.")

Undulating Terrain - A track profile with grade changes such that a train passing over the track has some cars on three or more alternating ascending and descending grades.

Vent Valve - The name applied to a valve or valvular portion of a car or locomotive brake system which responds to an emergency rate of reduction of the brake pipe and in turn vents the brake pipe locally at each vehicle, thereby propagating serially the emergency application throughout the train.

Yard Plant - A system of piping and fittings installed in a classification yard between the tracks to provide an air supply at convenient locations for charging and making tests on cars without a locomotive being present.

APPENDIX B - REFERENCES AND BIBLIOGRAPHY

Air Brake Association, 1972. Management of Train Operations and Train Handling, Chicago, IL, Sept.

Air Brake Association, 1975. Engineering and Design of Railway Brake Systems, Chicago, IL, Sept.

Archibald, R. H., and G. M. Cabble, 1973. "Disk Assist, the Simultaneous Use of Tread and Disk Brakes", presented to the American Society of Mechanical Engineers, Detroit, MI, Nov. 11-15, 1973 (73-WA/RT-11).

Association of American Railroads, 1973. "Track Train Dynamics to Improve Freight Train Performance," Research and Test Department - Technical Center, Chicago, IL, Report No. R-122.

Association of American Railroads, 1978a. Manual of Standards and Recommended Practices, Section D: Trucks and Truck Details, Operations and Maintenance Dept., Mechanical Division, Washington, D.C.

Association of American Railroads, 1978b. Manual of Standards and Recommended Practices, Section E: Brakes and Brake Equipment, Operations and Maintenance Dept., Mechanical Division, Washington, D.C.

Association of American Railroads, 1978c. Field Manual of the AAR Interchange Rules, Washington, D.C.

Association of American Railroads, 1978d. "Installation Specifications for Freight Car Brake Equipment," Specification No. 2518, Washington, D.C., January 1, 1975.

Association of American Railroads, 1978e. Office Manual of the Interchange Rules, Operations and Maintenance Dept., Mechanical Division, Washington, D.C., January 1, 1975.

Association of American Railroads, 1978f. Wheel and Axle Manual, Operations and Maintenance Dept., Mechanical Division, Washington, D.C.

Berg, N. A., and R. H. Alber, 1972. "Tread Braking versus the Wheel," Proceedings of the 64th Annual Convention of the Air Brake Association, Chicago, IL, September 18, 1972.

Berg, N. A., and W. J. Kucera, 1970. "A Review of Thermal Damage in Railroad Wheels," Proceedings of the 62nd Annual Convention of the Air Brake Association, Chicago, IL, September 15, 1970.

Blaine, D. G., 1967. "Braking High-Speed Freight Trains and Effects on Railroad Capacity," presented to the American Society of Mechanical Engineers, Pittsburgh, PA, November 12-17, 1967.

Blaine, D. G., 1969. "Determining Practical Tonnage Limits and Speeds in Grade Operations," presented to the American Society of Mechanical Engineers, Los Angeles, CA, November 16-20. (69-WA/RR-6.)

Blaine, D. G., 1975a. "The Importance of Being Able to Stop," Trains, October, pp. 3.

Blaine, D. G., 1975b. "Load-to-Tare Ratios vs. Braking," Trains, December, 1975 pp. 3.

Blaine, D. G., 1976. "Post-Zephyr Braking," Trains, January, 1976 pp. 40-46.

Blaine, D. G., and M. F. Hengel, 1971. "Brake-System Operation and Testing Procedures and Their Effects on Train Performance," presented the American Society of Mechanical Engineers, Washington, D.C., November 29-December , ASME Paper 71-WA/RT-9.

Cabble, G. M., 1972. "The Effect of Wheel Diameter on Tread Temperature in Grade Operation," presented to the American Society of Mechanical Engineers, New York, NY, November 26-30, 1972.

Cabble, G. M., 1973. "Wheel Temperature Performance Data in Grade Service," Proceedings of the 65th Annual Convention of the Air Brake Association, Chicago, IL, September 11, 1973.

Cabble, G. M., 1975. "A Compilation of the AAR Rules Affecting the COBRA and Composition Brake Shoes," Proceedings of the 67th Annual Convention of the Air Brake Association, Chicago, IL, September 14-17, pp. 181-196.

Cassidy, R. J., 1971. "Nonfriction Braking Methods for Railway Vehicles," presented to the American Society of Mechanical Engineers, Washington, D.C., November 28-December 2. (71-WA/RT-3.)

Central Air Brake Club, 1971. "Freight Car and Train Brake Tests - How and Why?," Proceedings of the 63rd Annual Convention of the Air Brake Association, Chicago, IL, September 15.

Central Air Brake Club, 1975. "I. M. T.-Handbrakes," Proceedings of the 67th Annual Convention of the Air Brake Association, Chicago, IL, September 14-17, pp. 83-97.

Grejda, F. J., 1972. "Computer Simulation of Train Braking Performance," Proceedings of the 64th Annual Convention of the Air Brake Association, Chicago, IL, September 13, 1972.

Hart, J. E., 1974. "Train Braking Analysis Through Computer Simulation," Proceedings of the 66th Annual Convention of the Air Brake Association, Chicago, IL, September 17, 1974.

Hart, J. E., and F. J. Grejda, 1972. "Computer Simulation of Rail Vehicle Braking Performance." ASME

Holloway, R. M., 1955. "Fundamentals and Designs of Freight Car Brake Riggings," Proceedings of the Annual Convention of the Air Brake Association, Chicago, IL, September.

International Union of Railways, 1960. "Air Brakes for 'Goods' and 'Passenger' Trains," Leaflet No. 540, Second Edition, Paris, France, January 1.

International Union of Railways, 1963. "Air Brake - Standard Programme of Tests," Leaflet No. 547, Third Edition, Paris, France, January 1.

International Union of Railways, 1966. "Braking Power," Leaflet No. 544-1, Third Edition, Paris, France, January 1.

International Union of Railways, 1967. "Regulations Relative to the Equipment and Use of Variables," Leaflet No. 543, Eighth Edition, Paris, France, January 1.

International Union of Railways, 1973. "Regulations Concerning the Construction of the Various Brake Components," Leaflet No. 541, Fifth Edition, Paris, France, July 1.

Janke, J. C., 1972. "Upgraded Train Performance Resulting From 1972 Braking Ratio Definitions," Proceedings of the 64th Annual Convention of the Air Brake Association, Chicago, IL, September 19.

Johnson, M. R., Welch, R. E., and Yeung, K. S., 1975. "Analysis of Thermal Stresses and Residual Stress Change in Railroad Wheels Caused by Severe Drag Braking," presented to the American Society of Mechanical Engineers, Houston, TX, November 30-December 5, 1975. (75-WA/RT-3.)

Kirk, W. B., 1968. "Freight Train Braking Level," Proceedings of the 60th Annual Convention of the Air Brake Association, Chicago, IL, October 1.

Knorr-Bremse, GmbH, 1973. "Knorr Air Brake KE for Passenger and Goods Trains," Munich, West Germany.

Knorr-Bremse, GmbH, 1975. "Magnetic Rail Brake High Suspension with Spring-Loaded Cylinder," B. 1259, EN, Munich, West Germany.

Knorr-Bremse, GmbH, 1973. "Brake Engineering Terms and Values for Rail Vehicles," Munich, West Germany.

Knorr-Bremse, GmbH, 1974. "Triple Value Type Fe 115", D33342lEN, Munich, West Germany.

Knorr-Bremse, GmbH, 1974. "Weighing Value Type W4," D34115EN, Munich, West Germany.

Knorr-Bremse, GmbH, 1974. "Weighing Value WB," D341159EN, Munich, West Germany.

Laplaiche, M., 1962. "The Electro-Pneumatic Brake for Railway Rolling Stock," Bulletin of the International Railway Congress Association, November, pp. 31 - 33.

Laplaiche, M., 1963. "The Electro-Pneumatic Brake for Railways Rolling Stock," Bulletin of the International Railway Congress Association, November, pp. 17 - 18.

Laplaiche, M., 1973. "The Disc Brake: A Contribution Towards The Study of Its Rational Use on Rail Vehicles," in French Railway Techniques, Federation des Industries Ferrociairies, pp. 75-99.

Manhattan Air Brake Club, 1971. "A Combined Tread-Disc Braking System for Multiple Unit Commuter Cars," Proceedings of the 63rd Annual Convention of the Air Brake Association, Chicago, IL, September 14, 1971.

Manhattan Air Brake Club, 1972. "In-Service Testing of Disc-Assist Brake Applications," Proceedings of the 64th Annual Convention of the Air Brake Association, Chicago, IL, September 19.

McLean, L. A., 1972. "Engineering for Maintenance-Free Freight Cars," presented to the American Society of Mechanical Engineers ASME Paper No. 72-RT-4.

McLean, L. A., 1973. "A Re-Evaluation of Freight Car Truck Performance Requirements," Proceedings of the 1973 Railroad Engineering Conference, Dresser Industries, Inc., DePew, NY, September 5-7, pp. 38-47.

Moore, I. G., and O. J. Wicham, 1974. "Improvements in the Design of Air Brake Control Valves," presented to the American Society of Mechanical Engineers, Pittsburgh, PA, April 3-4.

New York Air Brake Company, 1964. "The 26-L Brake Equipment Instruction Pamphlet No. 74," Watertown, NY, June, 1964.

New York Air Brake Company, 1970. "Breakaway Tests - Brake Performance Tests, C&O/B&O Railroad, Huntington, West Virginia," Watertown, NY, August.

Novak, G. E., and B. J. Eck, 1975. "Cyclic Distortions and Stresses in 36-Inch Wheels Under Combined Service Loads," presented to the American Society of Mechanical Engineers, Houston, TX, November 30-December 4, 1975 (75-WA/RT-5.)

Novak, G. E., and L. P. Greenfield, and D. H. Stone, 1975. "Simulated Operating Stresses in 28-Inch Diameter Wheels," presented to the IEEE-ASME Joint Railroad Conference, San Francisco, CA, April 15. (75-RT-10.)

Novak, G. E., L. P. Greenfield, and D. H. Stone, 1975. "The Effects of Wheel Design and Service Environment on the State of Stress in 28-Inch (0.712m) Diameter Freight Car Wheels," presented to the Fifth International Wheelset Congress, Tokyo, Japan, October.

Office of Safety, 1973. United States Safety Appliance Standards and Power Brake Requirements, U.S. Dept. of Transportation, Federal Railroad Administration, Washington, D.C., January. Catalog No. TD-3.106:AP5.

Palmer, D. E., 1975. "Brake System Leakage - 1975," Proceedings of the 67th Annual Convention of the Air Brake Association, Chicago, IL, September, pp. 38-53.

Pittsburgh Air Brake Club, 1975. "Wabcopac Brake Assembly - An Idea Whose Time Has Arrived," Proceedings of the 67th Annual Convention of the Air Brake Association, Chicago, IL, September, pp. 53-83.

Railroad Products Group, 1974. Train Attendant's Instruction Manual for the Friction Brake System of the Washington Area Transit Authority, Abex Corp., Mahway, NJ.

Sillcox, Lewis K., 1955. Mastering Momentum, Second Edition, Simmons-Boardman Publishing Corp., New York, NY.

Simmons-Boardman Publishing Corp., 1970. Car and Locomotive Cylopedia, New York, NY.

Simmons-Boardman Publishing Corp., 1973. Car Wheels, New York, NY.

"SP Makes Remote Air Brake Tests," Railway Locomotives and Cars, March, 1974, pp. 11-14.

St. Louis Air Brake Club, 1972. "Explanation of Net Braking Ratio," Proceedings of the 64th Annual Convention of the Air Brake Association, Chicago, IL, September, pp. 1-8.

Wabco, 1969. "Freight Brake Equipment With the ABD Control Valve," Westinghouse Air Brake Division, Wilmerding, PA, August, Instruction Pamphlet G-g-5062-16.

Wabco, 1974. "The Meaning and Value of Train Braking in Railroad Operations," Westinghouse Air Brake Division, Wilmerding, PA, June.

Wabco, 1975. "Wabcopac Brake Assembly - Technical & Economic Analysis," Westinghouse Air Brake Division, Wilmerding, PA, July 1, G-b-9013-12.

Wabco, 1976. "Freight Train Brake Systems as They Contribute to Railroad Productivity," Westinghouse Air Brake Division, Wilmerding, PA.

Wabco, 1973. "Combination Friction Brake System - Technical and Economic Analysis," Westinghouse Air Brake Division, Wilmerding, PA.

Wabco, 1974. "Basic Air Brake Fundamentals," Westinghouse Air Brake Division, Wilmerding, PA.

Wardrisco, J. M. and F. J. Dewey, Jr., 1960. "Study of the Defects that Originate and Develop in the Treads of Railroad Wheels During Service," presented to the American Society of Mechanical Engineers, ASME Paper No. 60-RR-L.

Weaver, G. R., P. A. Archibald, E. B. Brenneman, and G. M. Cabble, 1969. "Investigation of Thermal Capacity of Railroad Wheels Using COBRA Brake Shoes," presented to the American Society of Mechanical Engineers, Los Angeles, CA, November 16-20, 1969.

Westcode, Inc., 1975. "Eddy-Current Wear-Free Brake for High Speed Rail Demonstration," December.

Westcode, Inc., 1975. "The Z1A Control Valve Description," Publication A795/29.

Wetenkamp, H.R., and R. M. Kipp, 1975. "Safe Thermal Loads for a 33-Inch Railroad Wheel," Proceedings of the 67th Annual Convention of the Air Brake Association, Chicago, IL, September 14-17, pp. 142167.

Wilson, J. T., 1975. "Braking Systems Research Needs," presented to Railroad Research Study, Woods Hole, MA, July 22, 1975.

Wilson, J. T., R. C. Rolston, and M. J. Ash, 1972. "Investigation of In-Train Forces During Freight Train Brake Applications by Computer Simulation," CN Technical Research Center, St. Laurent, Quebec, Canada, December 11.

Wright, C. D., 1971. "Contribution of Empty and Load Brake Equipment To Increased Braking Level," Proceedings of the 63rd Annual Convention of the Air Brake Association, Chicago, IL, September 16.

110 copies