Vehicle Weights and Dimensions Study

Volume 13

Heavy Vehicle Braking Systems:

A Review of Available Hardware and Control Systems

Technical Report

DISCLAIMER

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The Technical Steering Committee will be considering the findings of these research investigations in preparing its "Final Technical Report" (Volume 1 & 2), scheduled for completion in December 1986.

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Abstract

The objectives of this study are to review and document available heavy vehicle braking hardware and control systems; and to review past and current research on heavy vehicle braking.

A review of the current state of the art in heavy vehicle brake design and technology will be carried out to assemble technical documentation and provide technical guidance on the applicability of specific braking hardware and compoents to the Canadian trucking fleet. A review will be conducted of research which was carried out in Canada and the United States in the ares of steering axle brakes, air brake timing and anti-skid and anti-lock devices with a view to identifying areas where further research is needed.

Keywords

heavy vehicle braking
steering axle brakes
air brake timing
anti-skid and anti-lock devices

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Supplementary Information

PREFACE

The report which follows constitutes one volume in a series of sixteen which have been produced by contract researchers involved in the Vehicle Weights and Dimensions Study. The research procedures and findings contained herein address one or more specific technical objectives in the context of the development of a consistent knowledge base necessary to achieve the overall goal of the Study; improved uniformity in interprovincial weight and dimension regulations.

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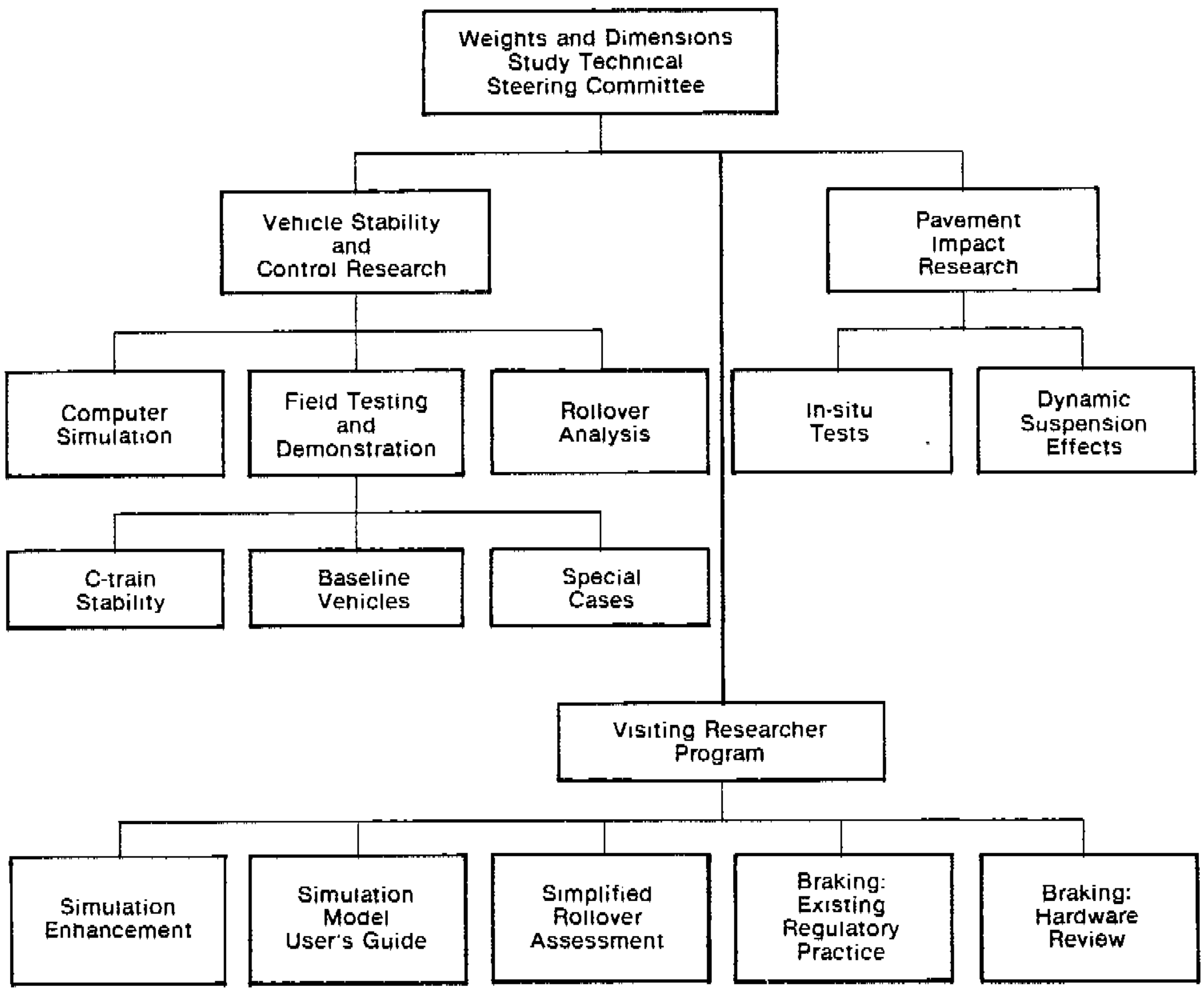

HEAVY VEHICLE BRAKE SYSTEM

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EXECUTIVE SUMMARY

The braking performance of heavy articulated vehicle combinations is a complex subject which is very difficult to address through controlled research, either with computer simulation techniques or in actual field testing. The principal difficulties relate to the highly complex physical nature of braking and the inability to control the factors necessary to obtain consistent and comparable research results. For these reasons, a detailed review of existing research, literature, regulations, and technical documentation is carried out and the results are presented.

The objectives of this study are:

- To review available heavy vehicle brake hardware and control systems; and

- To review the past and current research on heavy vehicle braking.

In order to fulfill the objectives of this study, the following steps were carried out:

1. A number of manufacturers of brake systems and components hardware, and control systems were contacted in order to collect information and data pertinent to the objectives of this study.

2. A number of heavy vehicles manufacturers and haulers in Canada were interviewed in person in order to survey on the brake hardware utilized in canadian trucking fleet and to compile comments and subjective assessments on the use of various advanced braking hardwares.

3. A number of individuals working in the general area of heavy vehicle braking were contacted in order to compile information related to this study.

4. Relevant literature, regulations, and technical documentation in the area of heavy vehicle braking were collected from several sources, and then reviewed and presented in this study in order to meet the objectives.

Based on the methodology outlined above, the detailed description of the findings are summarized as follows:

1. The names of the manufacturer's of brake hardware and control systems who were contacted are listed. A summary of the information compiled based on the survey of canadian trucking fleet manufacturer's and haulers are also presented.

2. The fundamentals of heavy vehicle braking and requirements are presented. Braking functions, requirements of brake sub-systems, comparison of various regulations, performance levels of air brake systems, functions of foundation/service brakes, static and dynamic characteristics of air brake systems, simple analysis of mechanical friction brakes, and air brake system maintenance and troubleshooting are presented.

3. A detailed account of brake system hardware and control systems is presented. Description of an air brake system for a tractor semi-trailer is presented along with the single (pre-121) and dual (121) air brake system circuitry. Several custom built brake circuitry designed for tractor, trailer, and dolly are presented. A detailed description of individual hardware used in brake system, their availability, and specifications is also presented.

4. Heavy vehicle auxiliary braking devices are presented. Engine retarders: engine brakes (Jake or Jacobs brake) and exhaust brakes, are presented in detail which includes operating principle, design considerations, salient features, and available hardware. Driveline retarders: hydrodynamic and electric retarders are also presented in a similar fashion.

5. A detailed review of past and current research on heavy vehicle braking is presented. The review is presented to include braking performance via theoretical analysis and experimental work, commercial vehicle brake hardware, braking system, S-Cam and wedge brake performance, disc brakes, brake lining, proportioning valve, contaminant removal, air brake adjustment, brake testing, and new types of brakes.

6. A detailed discussion on anti-lock/anti-skid brake systems is presented. The operating principle and the advantages of anti-skid systems are presented. The basic components in anti-skid systems are identified, the various hardware manufacturers are listed, and a comparison of the variety of systems available in North America and Europe are presented in a tabular form. A State-of-the-art review of research results on anti-skid devices are also outlined.

7. A discussion on future brake systems is presented.

ABSTRACT

The braking performance of heavy articulated vehicle combinations is a complex subject which is very difficult to address through controlled research, either with computer simulation techniques or in actual field testing. The principal difficulties relate to the highly complex physical nature of braking and the inability to control the factors necessary to obtain consistent and comparable research results. For these reasons, a detailed review of existing research, literature, regulations, and technical documentation is carried out and the results are presented.

The objectives of this study are:

- To review available heavy vehicle brake hardware and control systems; and
- To review the past and current research on heavy vehicle braking.

The presentation in this report covers:

- The function, operating principle, and requirements of heavy vehicle air brake system;
- Comparison of braking regulations;
- Available brake hardware and control systems;
- Various air brake system circuitry for tractor, trailer, dolly, and vehicle combinations;
- Operating principle, design considerations, and availability of auxiliary brake systems/retarders;
- Review of research work in heavy vehicle braking performance and brake hardware;
- Anti-skid brake system hardware-availability, operating principle and past research results; and
- Future direction in air brake systems.

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The canadian heavy vehicle manufacturers and haulers: Fruehauf, Shell Canada, Gulf Canada, Hutchinson, Bulk Carriers, Westank-Willock, Ultramar, Provost, Canadian Liquid Air, and Remtec, who participated in the survey conducted are gratefully acknowledged for providing information and for their time.

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CHAPTER 1

INTRODUCTION

1.1 GENERAL

The braking performance of heavy articulated vehicle combinations is a subject which is very difficult to address through controlled research, either with computer simulation techniques or in actual field testing. The principal difficulties relate to the highly complex physical nature of braking and the inability to control the factors necessary to obtain consistent and comparable research results.

There are, however, considerable research results on the handling of heavy articulated vehicles available in open literature. Although, these research results do not provide answers to all types of heavy vehicle combinations, and variations in vehicle parameters and brake systems, specific issues related to the performance of heavy vehicle braking systems can be derived. A concerted effort of reviewing existing research, literature, regulations, and technical documentation will be helpful in understanding the braking behaviour of heavy articulated vehicles and in establishing criteria for determining acceptable vehicle performance.

1.2 OBJECTIVES

There are two specific objectives in this study. They are:

1. To review the available heavy vehicle brake hardware and control systems, and

2. To review the past and current research on heavy vehicle braking.

The first objective deals with a review of the current state of the art in heavy vehicle brake design and technology in order to assemble technical documentation and provide technical guidance on the relative merits and applicability of specific hardware and components to the canadian trucking fleet.

The second objective concerns in documenting results available from research which has been carried out in Canada and the United States.

1.3 METHODOLOGY

In order to fulfill the objectives of this study, the following steps were carried out:

1. A number of manufacturers of brake systems and components

hardware, and control systems were contacted in order to collect information and data pertinent to the objectives of this study.

2. A number of heavy vehicles manufacturers and haulers in Canada were interviewed in person in order to survey on the brake hardware utilized in canadian trucking fleet and to compile comments and subjective assessments on the use of various advanced braking hardwares.

3. A number of individuals working in the general area of heavy vehicle braking were contacted in order to compile information related to this study.

4. Relevant literature, regulations, and technical documentation in the area of heavy vehicle braking were collected from several sources, and then reviewed and presented in this study in order to meet the objectives.

1.4 DESCRIPTION OF THE FINDINGS

Based on the methodology outlined above, the detailed description of the findings are summarized as follows:

In Chapter 1, the names of the manufacturer's of brake hardware and control systems who were contacted are listed. A summary of the information compiled based on the survey of canadian trucking fleet manufacturer's and haulers are also presented.

In Chapter 2, the fundamentals of heavy vehicle braking and requirements are presented. Braking functions, requirements of brake sub-systems, comparison of various regulations, performance levels of air brake systems, functions of foundation/service brakes, static and dynamic characteristics of air brake systems, simple analysis of mechanical friction brakes, and air brake system maintenance and troubleshooting are presented.

Chapter 3, presents a detailed account of brake system hardware and control systems. Description of an air brake system for a tractor semi-trailer is presented along with the single (pre-121) and dual (121) air brake system circuitry. Several custom built brake circuitry designed for tractor, trailer, and dolly are presented. A detailed description of individual hardware used in brake system, their availability, and specifications is also presented. The hardware includes:

- air compressor
- air compressor governor
- conditioning devices - alcohol evaporator, air dryer;
- reservoir
- service brake chambers
- spring brakes
- drum brakes
- disc brakes
- slack adjuster

- quick release valves
- relay quick release valves
- relay emergency valves
- ratio valves
- service brake valves
- tractor protection valve
- spring brake control valves
- trailer control valves
- other control valves
- check valves
- drain valves
- safety valves

In Chapter 4, heavy vehicle auxiliary braking devices are presented. Engine retarders: engine brakes (Jake or Jacobs brake) and exhaust brakes, are presented in detail which includes operating principle, design considerations, salient features, and available hardware. Driveline retarders: hydrodynamic and electric retarders are also presented in a similar fashion.

Chapter 5 presents a detailed review of past and current research on heavy vehicle braking. The review is presented to include braking performance via theoretical analysis and experimental work, commercial vehicle brake hardware, braking system, S-Cam and wedge brake performance, disc brakes, brake lining, proportioning valve, contaminant removal, air brake adjustment, brake testing, and new types of brakes.

Chapter 6, presents a detailed discussion on anti-lock/anti-skid brake systems. The operating principle and the advantages of anti-skid systems are presented. The basic components in anti-skid systems are identified, the various hardware manufacturers are listed, and a comparison of the variety of systems available in North America and Europe are presented in a tabular form. A State-of-the-art review of research results on anti-skid devices are also outlined.

Finally in Chapter 7, a discussion on future brake systems is presented.

1.5 MANUFACTURER/HAULER SURVEY

Table 1.1 lists the names of various manufacturers of brake hardware, and control systems, who were contacted to collect information and data pertinent to the objectives of this study.

Table 1.2 lists the names of various heavy vehicle manufacturer's and hauler's in Canada who were surveyed during the course of this project. The information received from these manufacturer's and hauler's is listed in Table 1.3. The following lists a summary of the findings:

Table 1.1 List of Manufacturers of Heavy Vehicle Brake Hardware and Control Systems

1. Robert Bosch Corporation
2. ITT Automotive Products Worldwide
3. Bendix Chassis and Brake Components Division
4. Kelsey-Hayes Co.
5. Lucas Industries Inc.
6. Rockwell Industries, Automotive Businesses
7. Eaton Corporation
8. Dana Corporation, Spicer Mobile
   Off-Highway Axle Division
9. Bundy Corp., North America Tubing Operations
10. The Budd Co.
11. Utility Trailer Mfg. Co.
    After Market Division
12. Vernay Laboratories Inc.
13. Wabash Electrical Components Group
14. Wabco Automotive Products Group
15. Wainwright Industries Inc.
16. Goodrich B.F. Canada
17. Aeroquip (Canada) Inc.
18. Berg Division of Echlin Canada Ltd.
19. Aeroquip Corporation
20. Midland-Ross Corp., Midland Brake Division
21. A.C. Spark Plug Division, GM
22. International Transquip Industries
23. ILASA International Marketing Inc.

Table 1.2 List of Manufacturers and Haulers Surveyed

1. Shell Canada
2. Gulf Canada
3. Hutchinson
4. Fruehauf
5. Bulk Carriers
6. Westank-Willock
7. Ultramar
8. Provost Transport
9. Trimac
10. Canadian Liquid Air Ltd.
11. Remtec Inc.

1. All companies surveyed confirmed that the braking systems used on heavy vehicles are of conventional air brake system.

2. Based on comments made by survey respondent's, majority of companies do not use any brakes on the steering axle.

3. Drum brakes with S-Cam and automatic slack adjuster are the most common brake on tractor rear wheels and trailer wheels.

4. Disc brakes are used only by a small percentage of heavy vehicle haulers.

5. The brake linings are of non-asbestos material with dimensions 16 1/2" x 7" and are either bonded or riveted.

6. The brake air chambers have standard diaphragm area of 30 sq.in.

7. Majority of air reservoirs have manual drain valve.

8. Many tractors use retarders. The most common retarder utilized is engine brakes manufactured by Jacobs.

9. Anti-lock brakes are not used by any company.

10. Use of brake proportioning valves are recognized as a means of improving brake balancing.

1.6 SOME DISCUSSIONS ON FRONT STEERING AXLE BRAKES

All provinces in Canada except Prince Edward Island and Manitoba allow heavy vehicles with three axles or more to fit brakes on two axles. In consequence, this vehicle configuration has traditionally been built without brakes on the front steering axle. Although Manitoba legislates brakes on all wheels, the regulation has never been enforced.

In the past, the testing of heavy vehicles without and with brakes have indicated that the use of front wheel brakes does shorten stopping distances and also aids in vehicle stability. However, the brakes are omitted from the front steering axle of the tractor as historically, it was felt that excessive braking on the front axle could cause jackknifing. Based on the compiled results of the survey of canadian manufacturer's and hauler's of heavy vehicles (Table 1.3) and the comments made by survey respondents, clearly shows that no company interviewed uses front steering axle brakes.

Some of the reasons given by experts in favour of front steering axle brakes can be listed as follows:

1. Drive wheel locking is more likely to happen if the front wheels have no brakes.

2. The drive wheel tires are pushed closer to their limit as they

Table 1.3 Air Brake System Survey
(Canadian vehicle manufacturers and owners)

1. Name of respondent company:	GULF	SHELL	FRUEHAUF	BULK CARRIERS	WESTANK-WILLOCK	ULTRAVAN	PROVOST	TRI-VAC
2. TYPE OF WHEEL BRAKES (e.g. drum, disc, etc.)								
2.1 Tractor, steering axle	-	-	-	disc	-	-	-	-
Tractor, rear axle	-	drum	-	disc	-	-	drum	drum
2.2 Semitrailer/Trailer:	drum	drum	drum	drum (2 units with discs)	drum some disc	-	drum	drum
3. SERVICE BRAKES								
3.1 Tractor (front/rear)								
Brake drum diameter and width (in x in)	-	-	-	-	-	-	16 1/2 x 8	16 1/2x7 1/2
Type of brake (e.g. S-Cam, wedge):	S-Cam,	S-Cam, some wedge	-	S-Cam	-	-	S-Cam	S-Cam (2")
Slack adjuster make:	-	Rockwell, Bendix	-	Rockwell	-	manual	Sand automatic	Berg
Slack adjuster length:	-	-	-	-	-	-	-	6 1/2"
Brake lining material:	various	non-asbestos	-	non-asbestos some Kevlar based	-	non-asbestos	-	non-asbestos Kevlar base
Mode of attachment (e.g. rivets, bonded, etc.):	various	bolted	-	bonded	-	bonded	bolted	riveted
3.2 Semitrailer/Trailer								
Brake drum diameter and width (in x in):	-	-	16 1/2 x 7	-	16 1/2 x 7	-	-	16 1/2x7 1/2
Type of brake (e.g. S-cam, wedge):	-	S-Cam,	S-Cam	-	S-Cam	-	-	S-Cam (1")
Slack adjuster make	-	Rockwell, Bendix	proprietary design	-	Midland, Berg, manual	-	-	Berg, others
Slack adjuster length:	-	-	-	-	6"	-	-	6 1/2"
Brake lining material:	-	non-asbestos	non-asbestos	-	non-asbestos	-	-	non-asbestos Kevlar base
Mode of attachment (e.g. rivets, bonded, etc.):	-	bolted	riveted	-	riveted	-	-	riveted
4. PARKING OR EMERGENCY BRAKES								
4.1 Tractor (type and make)	spring brake Maxibrake	spring brake Anchorlock	spring brake Maxibrake	spring brake Maxibrake	-	-	Berg	spring brake
4.2 Semitrailer/Trailer (type and make):	-	-	spring brake Maxibrake	-	spring brake	-	-	-
5. AIR BRAKE SYSTEM								
5.1 Compressor make and model	-	Borgwarner	-	Cummins, Midland Caterpillar	-	-	Bendix Westinghouse	Midland 1600
5.2 Compressor flow:	-	13 1/2 cfm	-	30 cfm	-	-	13 cfm	-
5.3 Compressor delivery pressure:	-	-	-	120 - 125 psi	-	-	110 psi	-
5.4 Air dryer make and model:	-	Borgwarner	-	-	-	-	drawmaster as OK	Bendix
5.5 Number of air tanks								
Tractor	-	no. of axles+ reserve tank	-	-	1 per axle	-	-	-
Semitrailer/Trailer	-	-	(1)*	no. of axles + 1	-	-	1 per axle	-
5.6 Make and model of foot valves:	-	-	-	-	-	-	Bendix Westinghouse	-
5.7 Make and model of relay valves:	-	-	-	-	-	-	-	Sealco
5.8 Make and model of air chambers:	-	-	Anchorlock	-	-	-	Berg (309-10)	Bendix, Berg Midland
Diaphragm dimensions:	-	-	-	-	30 sq. in.	-	30 sq. in.	30 sq. in.
6. AUXILIARY BRAKES (RETARDERS)								
6.1 Type (engine, hydrodynamic, electric):	none	engine	-	engine	-	engine	engine	engine
6.2 Make and model:	none	Jacobs	-	Jacobs, Caterpillar	-	Jacobs	Jacobs-3 Position	Jacobs
7. ANTILOCK SYSTEM (Yes/No):	no	no	-	no	-	-	-	no
8 COMMENTS								
Fruehauf - Retarders are used. Antilock systems are not used because they created too many problems.								
Westank/Willock - Brake proportioning valve would be beneficial for safety and stability during braking.								

*(1) Tandem axle - 3 tanks - 1400 cubic in/tank
3 axle = 4 tanks
4 axle = 5 tanks

compensate for the work the front wheels are failing to do.

3. Limited front wheel braking with anti-lock devices can eliminate front wheel lock-up even in snow and ice.

4. Limited front wheel braking with proper balancing will maintain stability.

5. At high vehicle deceleration during braking, weight is transferred to the front axle. Without front wheel brakes, the vehicle retarding potential must be provided by the tractor drive wheels. This will cause the vehicle to take longer to stop or the drive wheels to lock, causing it to jackknife.

1.7 CONCLUSIONS

In this chapter, the objectives, and the methodology followed to fulfill these objectives are presented. A brief account of what is presented in other chapters is described. Summary of the findings based on the review of information collected from manufacturer's and hauler's of heavy vehicles on brake hardware and control system is also presented. Finally, a brief discussion on front steering axle brakes is outlined.

CHAPTER 2

FUNDAMENTALS OF HEAVY VEHICLE BRAKING AND REQUIREMENTS

2.1 BRAKING FUNCTIONS

Heavy vehicle brakes are fundamentally air brakes which are required to carry out three braking functions:

- Service braking,
- Parking brake, and
- Emergency braking.

Service Brake System

The service brake system applies and releases the braking effort during normal vehicle operation upon driver's command.

Parking Brake System

The parking brake system applies and releases the park braking effort upon driver's command.

Emergency Brake System

The emergency brake system is in reality not a separate brake system, but rather the means by which different portions of the service and the parking brake systems are used to stop the vehicle in the event of a failure somewhere in the supply or control system.

2.2 BRAKE SUB-SYSTEMS

Air brake system can be considered to be divided into three major sub-systems. They are:

- Supply system,
- Control system, and
- Foundation brakes.

Supply System

The supply system provides clean, dry, and moisture free compressed air to control system for actuating the brakes.

Control System

The control system includes all the valving and plumbing between the supply system and the brakes.

Foundation Brakes

The foundation brakes include the brakes, brake chambers, slack adjusters, and the friction pads.

2.3 POINTS TO NOTICE IN BRAKE SYSTEM DESIGN

- Keep it simple.
- Avoid unneeded functions or valves.
- Reliability must be a major consideration.
- Emergency brake control must be the same as the service brake control.
- Avoid implementing separate control lever/button/pedal for the emergency system. Use the service brake pedal to actuate the emergency brake effort.

2.3.1 Parking Requirements

- FMVSS/Canadian 121 regulations require that all parking brakes are mechanically locked (i.e. spring brakes).

2.3.1.1 Tractor-Semitrailer

- Spring brakes on tractor drive axle.
- Spring brakes on trailer axle.

2.3.1.2 Tractor-Trailer Combinations

- Spring brakes on tractor drive axle.
- Spring brakes on semi-trailer axle.

- Dollies:

    . Pneumatically held by the relay emergency valve at dolly front axle.

    . Spring brakes on dolly rear axle.

2.3.2 Emergency Brake System

The emergency brake system is used in the event of brake system failure somewhere in the supply or control system. Such a failure results in the loss of air pressure from either of the service reservoirs. Then, the following series of events take place:

1. The emergency brake system should warn the driver that a failure has occurred;

2. The emergency brake system should not actuate automatically and introduce automatic brake application, surprising the driver;

3. The emergency brake system should provide the driver with necessary means of safely control and stop the vehicle. For this purpose, there must be sufficient air pressure stored in the remaining reservoirs to bring the vehicle to a limited number of safe, controlled stops; and

4. The emergency brake system should not permit unlimited operation of the vehicle with the partially failed system. This is a safety feature. When there is a failure, and if the vehicle is operated continually, ignoring the failure, eventually the parking brakes should come on preventing the use of the vehicle until the problem is remedied.

2.4 REGULATIONS

In order to optimize brake control, brake performance, and brake response on the vehicles combination, vehicle regulations have been proposed. In the United States, FMVSS 121 (FMVSS: Federal Motor Vehicle Safety standards) is the major regulation governing air brake systems. In Canada, CMVSS 121 is the standard that regulates heavy vehicle air brake systems. It should be noted that CMVSS 121 is very similar to the FMVSS 121 requirements in general. Hence, is this report all discussions are made in reference to FMVSS 121.

Further, the United Nations' Economic Commission for Europe (ECE) with headquarters in Geneva, European Economic Community (EEC) with its headquarters in Brussels, and the Swedish National Road Safety Board (NRSB) have established their own regulations for air brake systems. Although, many of the requirements are similar in all these regulations, there are variations to suit the local conditions or their special requirements.

2.4.1 Federal Motor Vehicle Safety Standards (FMVSS) 121

FMVSS 121 is the major regulation governing brakes systems in the USA.

Supply System

The requirements for the supply systems are:

- Minimum reservoir size:
    - . controlled as a ratio of the maximum volume of the brake chambers.
- Minimum compressor size:
    - . controlled by the minimum build-up time from 85 to 100 psi.
- Air pressure gages and low pressure warning devices.

Control System

The requirements for control system are:

- A single control for all parking brakes.
- The service control must be the emergency control.
- Specified minimum brake response times for both application and release.
- Requirements on towing vehicle protection in the event of a failure on the trailer.
- Requirement of a protected reservoir for parking brake release on trailers.

Foundation Brakes

The requirements for foundation brakes are:

- All parking brakes are held mechanically (spring brakes).
- Service brakes are required in all axles.

- Various dynamometer requirements for the various brakes:

    . Fade/Power in all brakes;
    . Effectiveness/retardation (trailers only);
    . Recovery (all brakes except tractor front axle).

Vehicle Performance Requirements

The only vehicle performance requirement in FMVSS 121 is the 20% grade holding requirement for the parking brakes.

Performance requirements of critical components

- Air compressor build-up time (according to FMVSS "121" regulations).

Table 2.1 illustrates the brake system requirements for FMVSS 121, original and new standards.

2.4.2 Other Motor vehicle Safety Regulations in the U.S.A.

In addition to the FMVSS 121 regulations, BMCS has a series of regulations that cover the vehicle in operation on the highways. The BMCS, Federal Motor Carriers Safety Regulation, Part 393, has a section that pertains to brake systems. There is also the New-York Thruway regulations. These regulations are summarized as follows:

2.4.2.1 BMCS - FMCSR Part 393 [2.2]

Requirements on Brake Systems

Many of the requirements are very similar to the FMVSS 121 requirements. However, there are few additional requirements in the BMCS - FMCSR, Part 393. They are:

- The tractor must automatically apply the trailer brakes when the system pressure drops to between 45 to 20 psi in the supply line between the tractor and trailer.

- Single service control for all brakes on the vehicle.

Table 2.1: FMVSS 121 Regulations (Original and New) [2.1].

ORIGINAL	NEW
. Air Compressor . Build up times	* Same
. Reservoirs . Static pressure test . Manual drain valves . Regulated capacity	* Same
. Service Brake Performance . Stopping distance on wet & dry roads . Staying within a 12 ft. lane	* . Some distances removed. . Same
. Application & Release Times . Trucks & tractors . Trailers	. Modified for trucks-tractors-trailers Plus now includes dollies
. Parking Brake Performance . Apply by mechanical means . Ability to hold on 20% grade	. Modified . Hold by mechanical means . Same
. Emergency Braking . Stopping distance requirements .	. Same . New includes emergency regulations for dollies
. Anti-Lock System . All steerable and the two rearmost non-steerable axles cannot lock up.	* ELIMINATED
. Foundation Brake Performance Dynamometer Testing	. Same
. Miscelleneous . Dual system check valves, warning devices, etc.	. Same

- The parameters relative to front brake limiting valve are specified as:

  . No manual front brake limiting valves for 1975 vehicles or newer;

  . An automatic front brake limiting device is needed for limiting front brakes up to a maximum of 50% at pressures below 85 psi.

- An automatic application of trailer brakes in the event of a trailer breakaway.

Foundation Brakes

- Parking brakes must be mechanically held and must hold on all grades.

- Service brakes are required on all wheels except the steer axle of 3-axle tractors.

Vehicle Performance Requirements

- The stopping distances for service brakes under all load conditions on a dry surface from 20 mph and in a 12-ft lane are:

  . Trucks 35 ft;
  . Combinations 40 ft.

- The stopping distances for emergency brakes under all load conditions on a dry surface from 20 mph and in a 12-ft lane are:

  . Trucks 85 ft;
  . Combinations 90 ft.

2.4.2.2 New-York Thruway Regulations [2.2]

The New-York Thruway regulations for doubles are very similar to BMCS requirements. However, it introduces additional requirements. There must be a means at each dolly or at the end of each trailer to accelerate the application and release of the rearward units. Also, the application signal cannot pass directly through more than one trailer and must be dead-ended at dolly or rear of each trailer. This can be achieved by using booster relay valve which takes the control line signal and relays it along the other vehicles behind.

2.4.3 ECE/EEC Regulations [2.3]

The summary of brake system requirements under ECE/EEC regulations are listed below and in Table 2.2. These regulations cover both air and hydraulic brake systems.

Summary of Brake System Requirements - ECE/EEC

1. Road test surface must have good adhesion.
2. No wheel lockup or deviation from path during testing.
3. Type approval certificate includes brake lining "make and type".
4. Divided service brake circuits (after Oct. 1, 1974 for M1-M2-N1-N2).
5. Manual or automatic lining wear adjustment.
6. Adequate reserve travels when brakes are hot or partly worn.
7. Transparent fluid reservoirs or low fluid level warning light.
8. Hydraulic failure warning light (red).
9. Low energy level warning (optical or acoustical) at 65% of normal pressure (in addition to pressure gauge).
10. No "redundant" members normally at rest and only activated by failure.
11. Brake system application time to prescribed performance at least favoured axle not to exceed 0.6 s. (N.B. Additional requirements for towing vehicles and trailers with air brakes).
12. Auxiliary equipment must not deplete service brake energy reservoirs below 65% of normal pressure.
13. Full-power systems must have two independent reservoirs and circuits (each with low pressure warning device), but may use single pump.
14. Energy reservoir capacity must retain emergency brake performance after eight full applications.
15. System pump-up time at maximum engine rpm to 65 and 100% of the service brake pressure must be within 3 and 6 min., respectively, (6 and 9 min. for combinations).
16. Failure of parking brake power-assistance must enable another energy source to be used (also for release).
17. Failure of coupling or trailer brakes must not prejudice tractor emergency brake performance.
18. Trailer air brakes must use at least two lines.
19. Failure of coupling must automatically stop trailer (except single-axle trailers of less than 1.5 t. using chain or cable).

Table 2.2: Summary of ECE/EEC Regulations [2.3].

SUMMARY ECE REGULATION 13 & EEC DIRECTIVE 71/320

No.	BRAKE PERFORMANCE TESTS		Unit	M1	M2	M3	N1	N2	N3	O1	O2	O3	O4
				CARS & BUSES			TRUCKS			TRAILERS			
	FL = FULLY LADEN AL = ANY LOADING	VEHICLE CLASSIFICATION No OF PASSENGER SEATS (+DRIVER) MAX VEHICLE WEIGHT (TONNES)		M1 ≤8	M2 >8 ≤5	M3 >8 >5	N1 ≤3½	N2 3½–12	N3 >12	O1 ≤¾	O2 ¾–3½	O3 3½–10	O4 >10
	SERVICE BRAKE	ACTING ON ALL WHEELS PROPERLY DISTRIBUTED SYMMETRIC A/C VEHICLE											
1	EFFECTIVENESS TEST (TYPE O TEST) NEUTRAL – COLD – AL	INITIAL SPEED STOPPING DISTANCE MIN DECELERATION PEDAL EFFORT	KPH M M/SEC² KG	80 0.1v + v²/150 5.8 50	60 0.15v + v²/130 5.0 70	60 5.0 70	70 0.15v + v²/115 4.4 70	50 4.4 70	40 4.4 70	SERVICE BRAKE IS OPTIONAL	OVERRUN BRAKES PERMITTED EXCEPT FOR SEMITRAILERS	TYPE DRAG ≥ 0.45 OF TRAILER MAX WEIGHT (USING C+3 BARS)	
2	HIGH SPEED TEST (TYPE O TEST) IN GEAR – COLD – AL			EFFECTIVENESS TESTS VARIOUS SPEEDS FROM 30% TO 80% OF V MAX									
3	FADE TEST BY REPEATED BRAKING (TYPE I TEST) IN GEAR – HOT – FL FIRST SNUB AT 3.0 M/SEC² REMAINING SNUBS AT SAME PEDAL EFFORT	V1 = 80% V MAX BUT ≤ V2 = ½ V1 TIME INTERVAL No OF CYCLES	KPH SECS	120 45 15	100 55 15	60 60 20	120 55 15	60 60 20	60 60 20				
4	FADE TEST BY CONTINUOUS BRAKING (TYPE I TEST) HOT – FL									DRAG TEST AT 40 KPH FOR 1.7 KM ON 7% SLOPE			
5	RESIDUAL EFFECTIVENESS AFTER TYPE I TESTS NEUTRAL – HOT – FL			PERFORMANCE TO BE ≥ 80% OF TEST 1 REQUIREMENT AND ≥ 60% OF TEST 1 ACHIEVEMENT									
6	DRAG TEST (TYPE II TEST) IN GEAR – HOT – FL						DRAG TEST AT 30 KPH FOR 6 KM ON 6% SLOPE (OR 0.5 M/SEC² BY ENGINE ALONE)						
6a	DRAG TEST (TYPE II BIS TEST) IN GEAR – HOT – FL (FOR PSV WITH > 8 PASS SEATS AND MAX WEIGHT > 10T)						DRAG TEST AT 30 KPH FOR 6 KM ON 7% SLOPE (OR 0.5 M/SEC² BY ENGINE ALONE) – WITHOUT USING SERVICE EMERGENCY OR PARKING BRAKE						
7	RESIDUAL EFFECTIVENESS AFTER TYPE II TESTS NEUTRAL – HOT – FL N.B. TYPE II TESTS ONLY APPLY TO M3 – N3 – O4						PERFORMANCE TO BE ≥ 75% OF TEST 1 REQUIREMENT						
	EMERGENCY BRAKE	MAY BE OPERATED BY SERVICE OR PARKING BRAKE CONTROL											
8	SERVICE BRAKE FAILURE (TYPE O TEST) NEUTRAL – COLD – AL	INITIAL SPEED STOPPING DISTANCE MIN DECELERATION PEDAL EFFORT HAND LEVER EFFORT	KPH M M/SEC² KG KG	80 0.1v + 2v²/150 2.9 50 40	60 0.15v + 2v²/130 2.5 70 60	60 2.5 70 60	70 0.15v + 2v²/115 2.2 70 60	50 2.2 70 60	40 2.2 70 60				
8a	POWER ASSIST FAILURE AS 8 BY SERVICE BRAKE OR AS 10 IF POWER			N.B. EMERGENCY BRAKES (AND DIVIDED – CIRCUITS) OF TRACTORS FOR O_3 – O_4 TRAILERS MUST ALSO ACTIVATE TRAILER BRAKES GRADUALLY									
	PARKING BRAKE	CONTROL MUST BE INDEPENDENT OF SERVICE BRAKE CONTROL											
9	HILL HOLDING TEST NEUTRAL – COLD – FL N.B. ALL PARKING BRAKES MUST BE CAPABLE OF APPLICATION "IN MOTION"	GRADIENT HAND – LEVER EFFORT FOOT – OPERATED EFFORT	% KG KG	18 40 50	18 60 70	18 60 70	18 60 70	18 60 70	18 60 70	PARKING BRAKE IS OPTIONAL	18 60	18 60	18 60
				N.B. PARKING BRAKE OF TRACTORS MUST HOLD COMBINATIONS ON 12% SLOPE						N.B. OPERABLE FROM OUTSIDE (AND FROM WITHIN ON PSV) – WHEN SEPARATED FROM TRACTOR			
	DIVIDED CIRCUITS	OPERATED BY SERVICE BRAKE CONTROL ON SUFFICIENT WHEELS											
10	CIRCUIT FAILURE NEUTRAL – COLD – AL N.B. NOT APPLICABLE TO TRACTORS FOR SEMI-TRAILERS WITH INDEPENDENT SERVICE BRAKES	MIN DECELERATION (M/SEC²) WITH 70 KG PEDAL EFFORT	FL AL	1.7 1.5	1.5 1.3	1.5 1.3	1.3 1.1	1.3 1.1	1.3 1.1				
				AFTER 1 OCT 1974			AFTER 1 OCT 1974						

20. Normal maximum trailer feed-pressure to be between 6.5 and 8.0 bars.

2.4.4 Swedish Regulations [2.3]

The summary of brake system requirements under swedish regulations are listed below and in Table 2.3.

Summary of Brake System Requirements - Sweden

1. Road tests on 0.8 friction coefficient (interpreted as Skid Number 80).
2. No uncontrolled deviations during testing.
3. Divided service brake circuits.
4. System must resist 100 kg pedal effort.
5. Brake pipe corrosion resistance equivalent to 0.025 mm zinc on steel.
6. Brake fluid to SAE 70R3.
7. Service brake automatic lining wear adjustment or warning indicator when lining wear precludes emergency brake performance.
8. Adequate reserve travels when brakes are hot and partly worn.
9. Transparent fluid reservoirs or low fluid level warning light.
10. Hydraulic failure warning light (red) and activation pressure.
11. Low energy level warning (red light or acoustical) at 65% of estimated pressure (in addition to pressure gauge).
12. No "redundant" members normally at rest and only activated by failure.
13. Brake system application (0-75%) and release (75%-10%) times not to exceed 0.6 s (0.4 s at coupling head of towing vehicles; 0.8 s at trailers in combinations).
14. Pressure tapping (M16 x 1.5) at least favoured air brake chamber.
15. Air reservoirs in accordance with Swedish Pressure Vessel Commission.
16. Auxiliary equipment must not deplete service brake energy reservoirs below 60% of estimated pressure.

Table 2.3: Summary of Swedish Regulations [2.3].

SWEDISH BRAKING REGULATIONS F.18

SUMMARY

BRAKE PERFORMANCE TESTS

FL = FULLY LADEN
AL = ANY LOADING

			KG	MOTOR VEHICLES ≤3500	MOTOR VEHICLES >3500	TRAILERS ≤3500	TRAILERS >3500
SERVICE BRAKE	[illegible]						
[illegible]	EFFECTIVENESS TEST [illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
[illegible]	WHEEL LOCKING TEST [illegible]	[illegible]	[illegible]	[illegible]	[illegible]		
[illegible]	HIGH SPEED TEST [illegible]	[illegible]	[illegible]	[illegible]			
[illegible]	FADE TEST [illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
[illegible]	FADE TEST [illegible]					[illegible]	
[illegible]	RESIDUAL EFFECTIVENESS [illegible]			[illegible]			
7	WATER RECOVERY [illegible]	[illegible]	[illegible]	[illegible]	[illegible]		
8	RECOVERY EFFECTIVENESS [illegible]	[illegible]	[illegible]	[illegible]			
EMERGENCY BRAKE	[illegible]						
[illegible]	CIRCUIT FAILURE [illegible]	[illegible]	[illegible]	[illegible]	[illegible]		
[illegible]	POWER ASSIST FAILURE [illegible]	[illegible]	[illegible]	[illegible]	[illegible]		
PARKING BRAKE	[illegible]						
[illegible]	DYNAMIC TEST	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
[illegible]	HILL-HOLDING TEST [illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]

17. Service brake energy reservoir capacity not less than 12X combined brake chamber volume at 2/3 maximum piston movement (8-12 for trailers).

18. System pump-up time at maximum engine rpm to 65% of estimated pressure within 4 min. (6 min. for combinations).

19. Towing vehicle-trailer compatibility relationships for control pressure versus deceleration.

20. Failure of coupling or trailer brakes must not preclude service brake function on towing vehicle.

21. Trailer air brakes must use two lines and predetermined coupling locations.

22. Maximum trailer feed pressure to be between 6.5 and 8.0 bars.

23. Trailers over 3,500 kg must have automatic lining wear compensation.

24. Trailer axles whose laden-to-unladen weight ratio exceeds 4:3 must have automatic brake force regulators.

25. Failure of coupling or supply pressure must automatically stop trailer.

2.4.5 Comparison of Regulations [2.3]

Oppenheimer [2.3] has presented a comparison of the various brake system testing procedures, requirements, and regulations. These comparisons are presented in Tables 2.4 and 2.5, and Figures 2.1(a) and 2.1(b).

2.5 HEAVY VEHICLE RETARDING CAPACITY [2.2]

When a heavy vehicle weighing approximately, say 80,000 lb traveling at a speed of 55 mph, decelerates, the kinetic energy is dissipated primarily by the foundation brakes as friction. The vehicle kinetic energy is also dissipated by other secondary factors, such as: aerodynamics drag, tire rolling resistance, friction losses in the drive train, and engine braking. Figure 2.2 shows the vehicle retarding capability of a typical 5-axle tractor/trailer combination. It can be seen that with the improved aerodynamics of vehicles, improved tires such as low resistance radials, improved lubricants in the engine and drive trains, improved, more fuel-efficient, and smaller engine cause considerable reduction in aerodynamics drag, rolling resistance, friction loss in driveline and engine brake respectively. This causes an increased effort on the foundation brakes. The energy dissipated must be divided among the various brakes on the vehicle which leads to variations in brake balancing.

Table 2.4: Brake Test Procedures in ECE/EEC, Sweden, and U.S.A. [2.3].

Brake Test Procedures	ECE/EEC	Sweden	USA 105a	USA 121
Instrumentation check	X	X	X	X
Preburnish effectiveness			X	
Burnish procedure			X	X
Cold effectiveness (in neutral)	X	X	X	X
Cold effectiveness (in gear)	X			
High-speed stops (80% V_{max})	X	X	X	
Wheel-locking sequence		X		
Wet road (SN30) effectiveness				X
Partial failure (emergency brake)	X	X	X	X
Inoperative power (assist) units	X	X	X	X
Fade and recovery	X	X	X	X
Water recovery		X	X	
Spike stops			X	
Parking brake (grade-hold)	X	X	X	X
Parking brake (dynamic tests)	X	X		X
Dynamometer brake tests				X

Table 2.5: Brake System Requirements in ECE/EEC, Sweden, and U.S.A. [2.3].

Brake System Requirements	ECE/EEC	Sweden	USA 105a	USA 121
Divided circuits	X	X	X	
System strength		X	X	
Adequate pedal travel reserve	X	X		
No "redundant" members	X	X		
Brake application time	X	X		X
Brake release time		X		X
Warning indicators for:				
Pressure loss	X*	X*	X	X
Low fluid level	X*	X*	X	
Antilock system failure			X	X
Parking brake "applied"			X	
Component specifications for:				
Hydraulic brake hoses		X	X	
Hydraulic brake fluid		X	X	
Brake pipe corrosion resistance		X		
Automatic lining wear adjustment		X		
Transparent fluid reservoir	X*	X*		
Fluid reservoir capacity			X	
Fluid reservoir labeling			X	
Lining material approval	X	X		
Air and vacuum hoses		X	X	X
Energy reservoir capacity	X	X	X	X
Energy source capacity	X	X		X

*Alternatives.

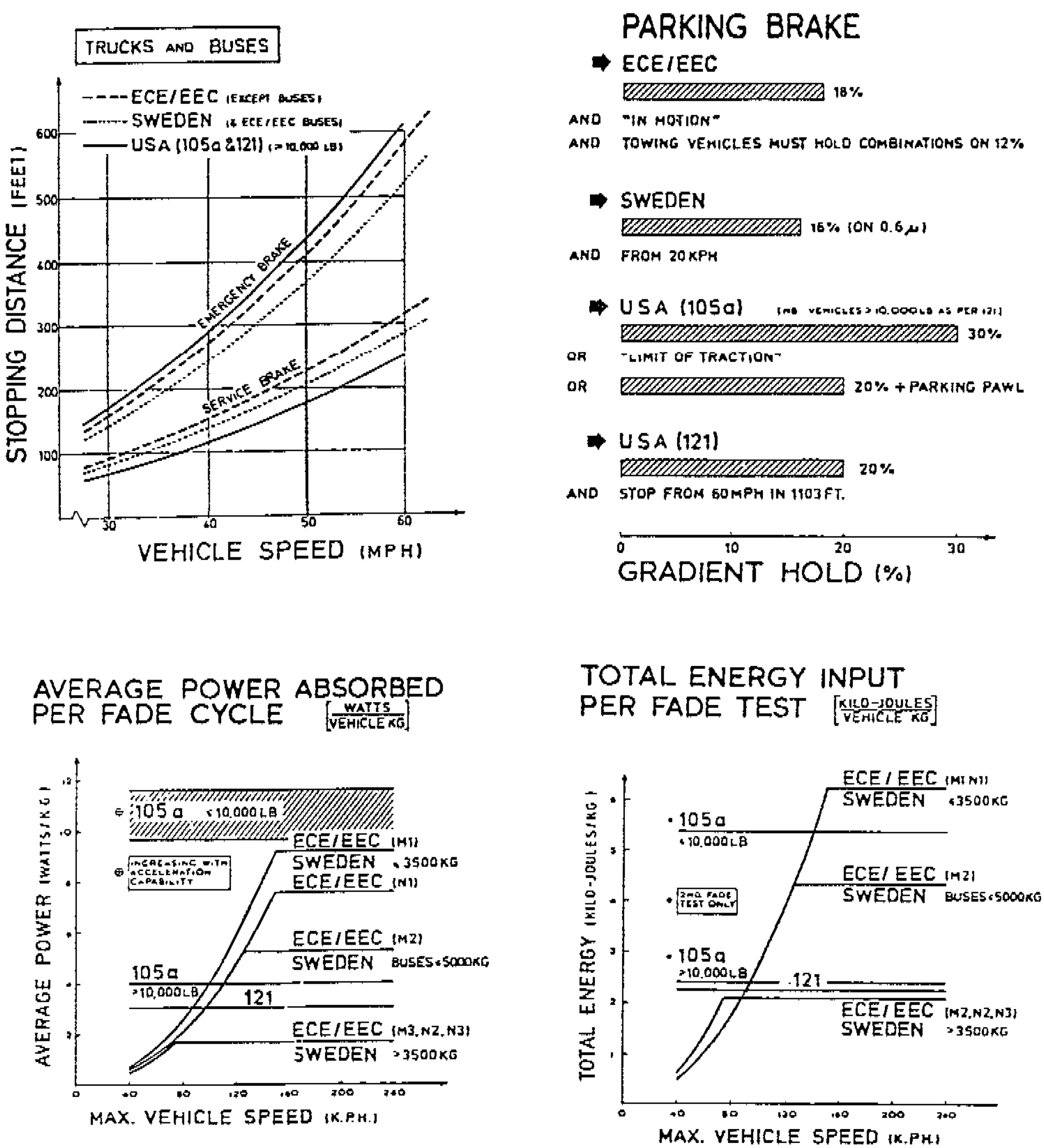

Figure 2.1a: Comparison of Braking Regulations [2.3].

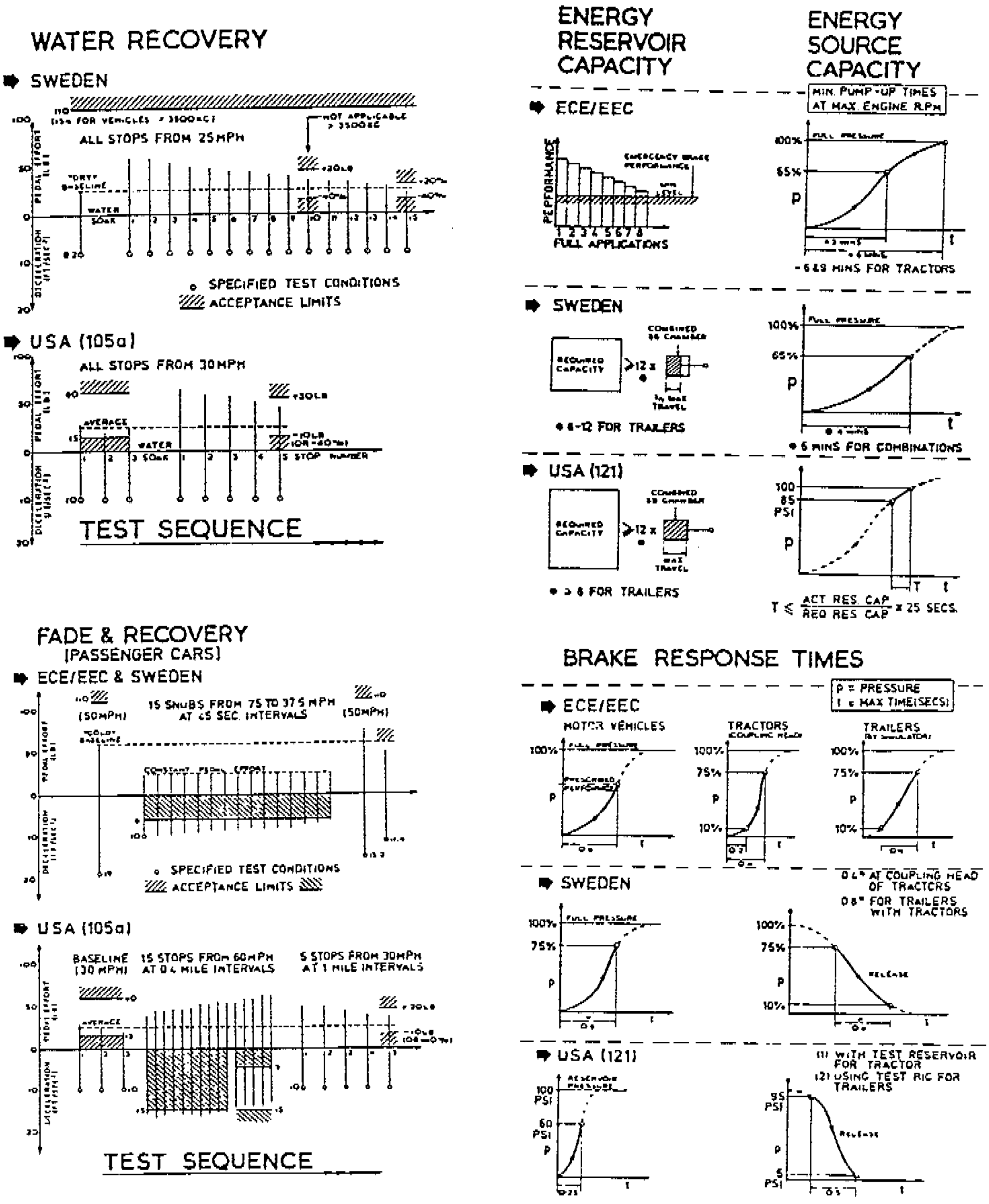

Figure 2.1b: Comparison of Braking Regulations [2.3].

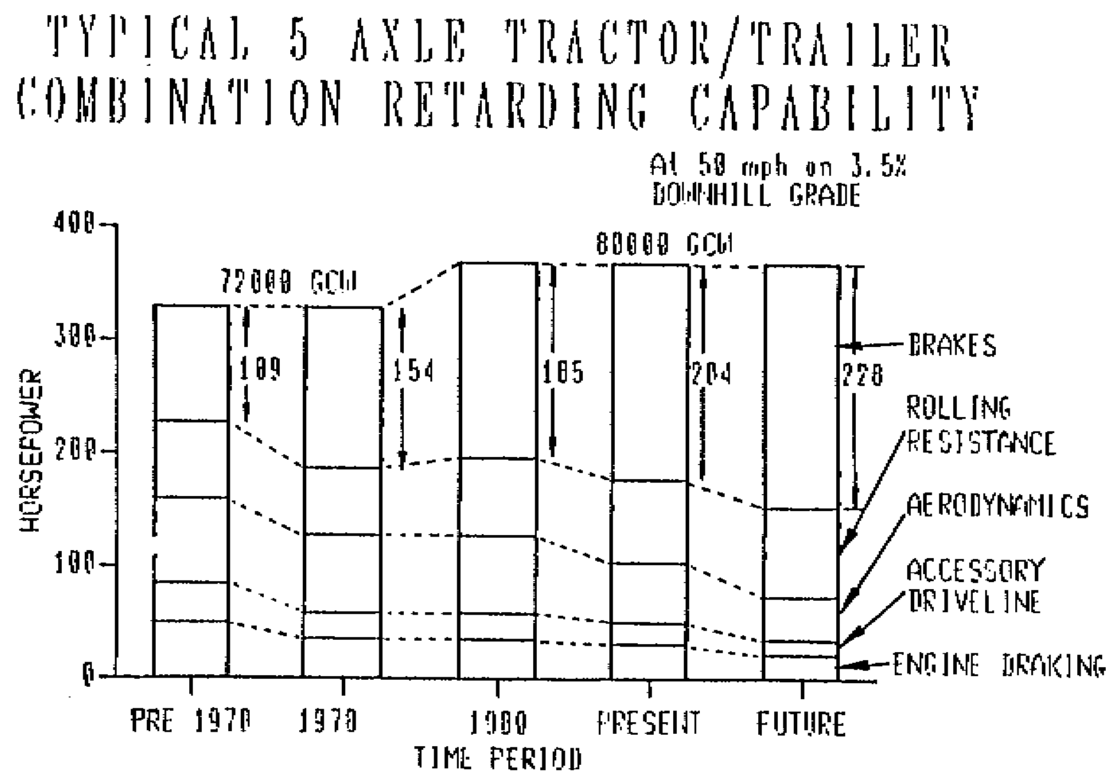

Figure 2.2: Heavy Vehicle Retarding Capability [2.2].

2.6 PERFORMANCE LEVELS OF AIR BRAKE SYSTEMS

A typical brake application in normal stops is represented by the histogram (Figure 2.3). It can be seen that in excess of 90% of the stops are below 25 psi brake pressure.

A typical tractor-semitrailer having a tractor front brake size (15" x 4"), tractor rear brake and trailer brake size (16 1/2" x 7"), absorbs less than 8% energy at the front brakes, about 11% each by the drive axle brakes, and about 12% each by the trailer axle brakes.

Use of front brake limiting valves reduces the energy absorbed by the front axle. Also use of system modifying valve on the drive axle or trailer axles influences brake balance.

The performance levels of air brake systems can be divided into two categories:

- High deceleration performance;
- Low deceleration performance.

High deceleration performance is required during panic type stops. The FMVSS 121 requirements are:

	Tractor	Trailer	Dolly
Application (0 to 60 psi)	0.45 sec.	0.30 sec.	0.35 sec.
Release (95 to 5 psi)	0.55 sec.	0.65 sec.	0.65 sec.

At these low pressures, the brake torque can fluctuate significantly with pressure (Figure 2.4). A variation of 1 psi car cause 30% variation in braking torque. With a 4 psi variation between tractor rear and trailer brake pressures can introduce 120% variation in torque. Currently, in heavy vehicles, a variation of 4 psi and greater exists. Hence, it is essential in the future to maintain this pressure variation below 2 psi during the 10 - 25 psi brake application.

Various major parameters that influence braking performance are:

- Brake inputs;
- Brake and wheel size and design;
- Brake maintenance and adjustment.

The brake input parameters of importance are:

- Pressure delivered to the brake actuators by the control system;
- Size of the actuator and its effectiveness;

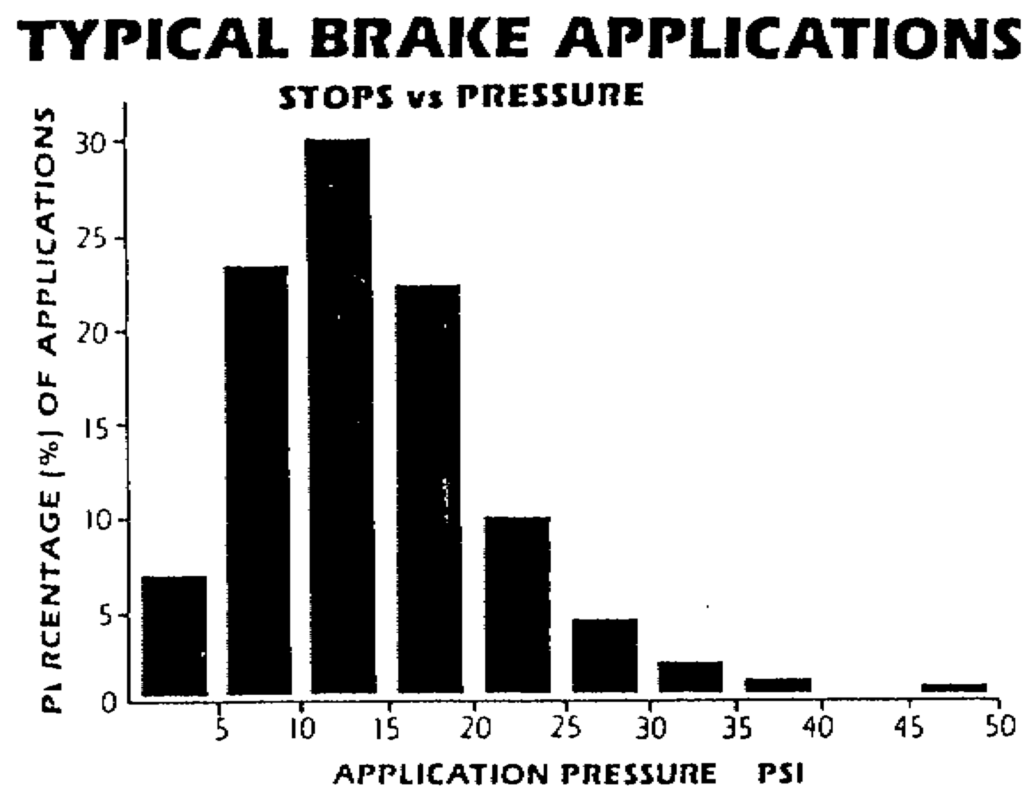

Figure 2.3: Histogram of Typical Brake Applications [2.2].

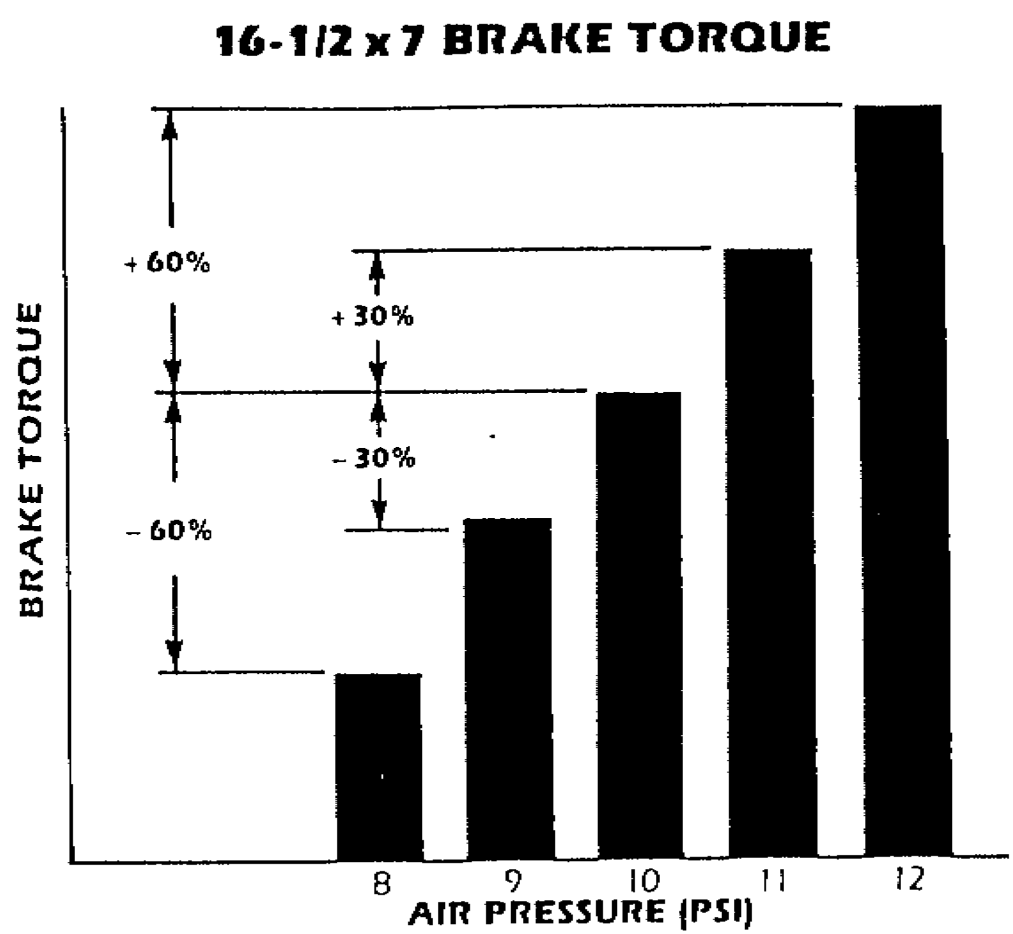

Figure 2.4: Brake Torque Versus Air Pressure [2.2].

- Strength of the return spring in the actuator;
- Slack adjuster length.

The brake and wheel design parameters of importance are:

- S-Cam Brake and size
  - . Example: 16 1/2" diameter x 7" wide
- Wedge-brake and size
  - . Example: 15" diameter x 7" wide
- Types of brake lining
- Tire rolling radius
- Reduced tire rolling radius will give increased energy, increased retardation, and reduced wheel-lock pressure.

2.7 FOUNDATION/SERVICE BRAKES

There are three types of air actuated brakes that are used in heavy vehicles. They are:

- S-Cam Drum Brake:
  - . Used on 90% of vehicles
- Wedge Drum Brake:
  - . Used since 1960's
- Disc Brakes:
  - . Air actuated, mechanical brakes
  - . Hydraulic actuated, hydraulic brake
  - . New development and considered brake of the future

2.7.1 S-Cam Drum Brake

When air is applied to the brake chamber, it causes the slack adjuster to rotate the S-Cam. The S-Cam is a constant rise cam, acting like a half-inch lever between the cam shaft and tip of the shoe. This causes both the shoes to advance the same amount. However, the forces will not be the same on both shoes. The leading shoe will exert more force than the trailing shoe. This results in non-uniform shoe and S-Cam loading, but both the shoes are forced to do the same amount of work in the life of the lining and both wear at the same rate.

Whenever, one lining wears faster than the other, load from the S-Cam is shifted to the opposite shoe. Most S-Cam brakes have tapered lining which utilizes all lining material replacement.

- Advantage:

  . Simple

- Disadvantages:

  . Heaviest brake
  . Requires automatic adjustment
  . Prone to noise and chatter
  . Fade at high temperatures

2.7.2 Wedge Drum Brake

Wedge drum brake uses twin wedges, one for each shoe, and operated by two actuators. Both top and bottom shoes are leading or self-energized shoes. The forces within the twin wedge brake are completely symmetrical; each shoe and wedge loading is the same as is the loading between the shoes and the drum. Automatic adjacement is built into the upper piston. Lining wear is sensed on brake apply and adjustment occurs during release.

- Advantages:

  . Lightest brake
  . Automatic adjustment
  . Smaller chambers/less air requirement

- Disadvantages:

  . More sensitive to lining variation
  . Contamination if seals fail
  . Fade at high temperatures

2.7.3 Air Disc Brakes

The air actuated disc brakes have two shoes which are clamped on to the rotor by a caliper which slides to compensate for outer shoe lining wear. The high clamp loads required on a disc brake is achieved by a mechanical mechanism with a mechanical advantage of 12 or 13 to one which converts 100 psi air to 15 tons of clamp load.

Disc brakes produce a more constant brake torque throughout the stop and thus provides a clear performance advantage over the drum brake.

- Advantages:

  . Stable, uniform brake torque

. Little fade
. Excellent water recovery
. Simple reline

2.7.4 Brake Balance with a Mix of Foundation Brakes

When S-Cam, Wedge, and disc brakes are mixed in vehicle combinations, the following comments can be made regarding the pneumatic balance.

The mix of air disc brakes with S-Cam brakes has a good pneumatic balance and therefore no pressure modification is required.

The wedge brake has a mechanical design difference that requires 4 to 7 psi more pressure to make shoe to drum contact. The brake torque comparison shown in Figure 2.4, shows up to 50% difference in brake torque at actuator pressures less than 15 psi. This will cause the S-Cam or disc brake to do additional braking up to 82% of the time in normal service.

For brake balance at low pressures, pressure modifiers/hold-off springs are required in relay or quick release valves. The brake balance improves at pressures below 20 psi for S-Cam and air disc brakes combination. When wedge brakes are mixed with S-Cam and or disc brakes, pressure modifiers are required in relay or quick release valves.

2.7.5 Brake Rating

Brakes are rated by GAWR of the vehicle. Brakes must have a minimum rating of the lowest load carrying component of the axle.

- Example for the following load rating:

  . 20,000 lb. Spring
  . 23,000 lb. Tires
  . 23,000 lb. Axle
  . Brakes should be rated at 20,000 lb. GAWR minimum

- Vehicle Performance Requirements:

  . 20% grade holding ability in parking

- Dynamometer Performance Requirements:

  . Torque stability
  . Brake torque
  . Fade resistance

- The factors that affect the brake torque and brake balance are:

  . Actuator pressure
  . Brake chamber size and effectiveness
  . Brake chamber pressure spring

. Slack adjuster length
. Mechanical design of brake
. Brake drum diameter
. Brake lining
. Tire rolling radius
. Brake adjustment
. Brake maintenance

A typical brake chamber output curve is shown in Figure 2.5. It shows how increasing stroke affects output force. At point "A", the shoes of the S-Cam brake are in firm contact with the brake drum with a well adjusted brake. At point "B", the output force corresponds to the maximum stroke with a cold, well adjusted brake. Point "C" is the maximum recommended cold brake stroke before adjustment is required. Stroke can increase from "B" to "C" due to wear or brake drum expansion due to heat. The brake starts to fade beyond point "C" and the driver can sense this. This is a mechanical fade and not lining fade.

The cold brake stopping distance increases 25 to 35% at maximum readjustment stroke (point "C"). The hot brake stopping distance increases 75% at maximum readjustment stroke.

- Various surveys show:

  . Half of the air braked vehicles have at least one brake out of adjustment.

  . One fourth of the air braked vehicles have 40% of their brakes out of adjustment.

2.7.6 Brake Adjustment

An S-Cam brake will require 20 adjustments in the life of the lining, unless the brake adjustment is made exactly when the recommended maximum stroke is reached. In practice, 30 adjustments is more likely. However, the operating time or mileage between adjustments will vary widely. With an air brake system, the driver has no warning that brakes are out of adjustment until he senses fade.

Automatic slack adjusters can improve safety and be cost effective. Mixing manual and automatic adjusters on the same vehicle will result in more work and lining wear on the automatically adjusted brakes if the manual adjusters are frequently adjusted.

Poor brake adjustment will cause low brake torque, mechanical fade, noise and chatter, and drum cracking.

2.7.7 Brake Pads

Non-asbestos friction materials are used on tractor and trailers. The non-asbestos materials that are used for friction materials are:

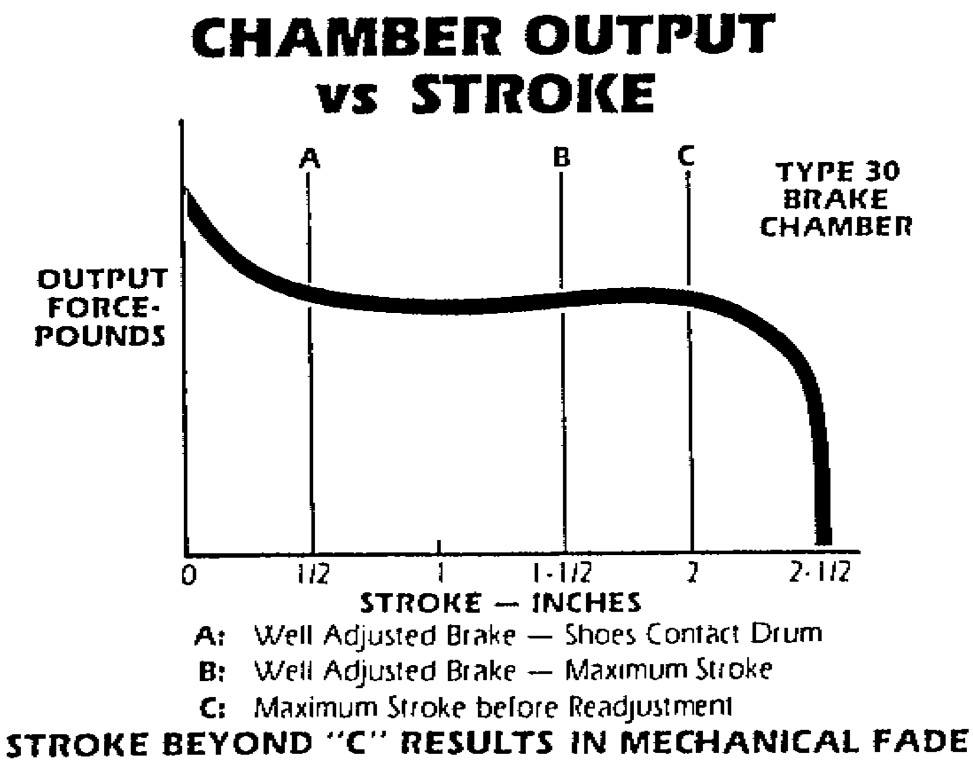

Figure 2.5: Brake Chamber Force Output Versus Stroke [2.2].

- Fiberglass:
  - Provides adequate reinforcement with only 10 - 20 % by volume.
- Kevlar:
  - Amounts below 10% and often less than 5% used in a friction materials.
- Steel Fibers:
  - Add considerable weight to a friction material and require more weight for an equivalent reinforcement.
- Processed Mineral Fiber (PMF)
- Ceramic Fibers:
  - Similar to PMF with a difference in their melting point.

All the asbestos-free fibers listed above are high temperature resistant materials and provide strength equivalent to asbestos blocks. Asbestos-free blocks show similar fade characteristics as asbestos containing blocks.

Since most of the non-asbestos fibers are higher in friction than asbestos, less abrasive is required for a given friction level. Drum wear is in general less than for an equivalent asbestos block. Non-asbestos fiber blocks wear less than asbestos blocks in some temperature ranges. Figure 2.6 shows a comparison of this phenomenon. For non-asbestos materials, wear of both the block and mating surface can be better at the lower operating temperatures but, in general, is no better at the high temperatures.

Non-asbestos materials have similar swell and growth characteristics. Swell is the change which occurs while under heat. Average swell of a 0.750" block for a temperature range of 400° to 700° F. is 0.002" to 0.04" for asbestos-free material and 0.003" to 0.03" for asbestos material. Growth is the permanent change remaining after the block cools down. Average growth of a 0.750" block for a temperature range of 400° to 700° F. is below 0.030" for non-asbestos materials and below 0.020" for asbestos materials.

2.7.8 Brake Block Edge Code

SAE J 866 is a recommended practice, intended to provide a uniform means of identification which may be used to describe the initial friction characteristics of brake lining and brake blocks for use on motor vehicles.

SAE is working on a new methods of classifying friction levels. The block edge code which is used to characterize the initial block friction is established in a laboratory controlled dynamometer with a 11" diameter drum and a one square inch sample. The Table 2.6 shows a comparison of the

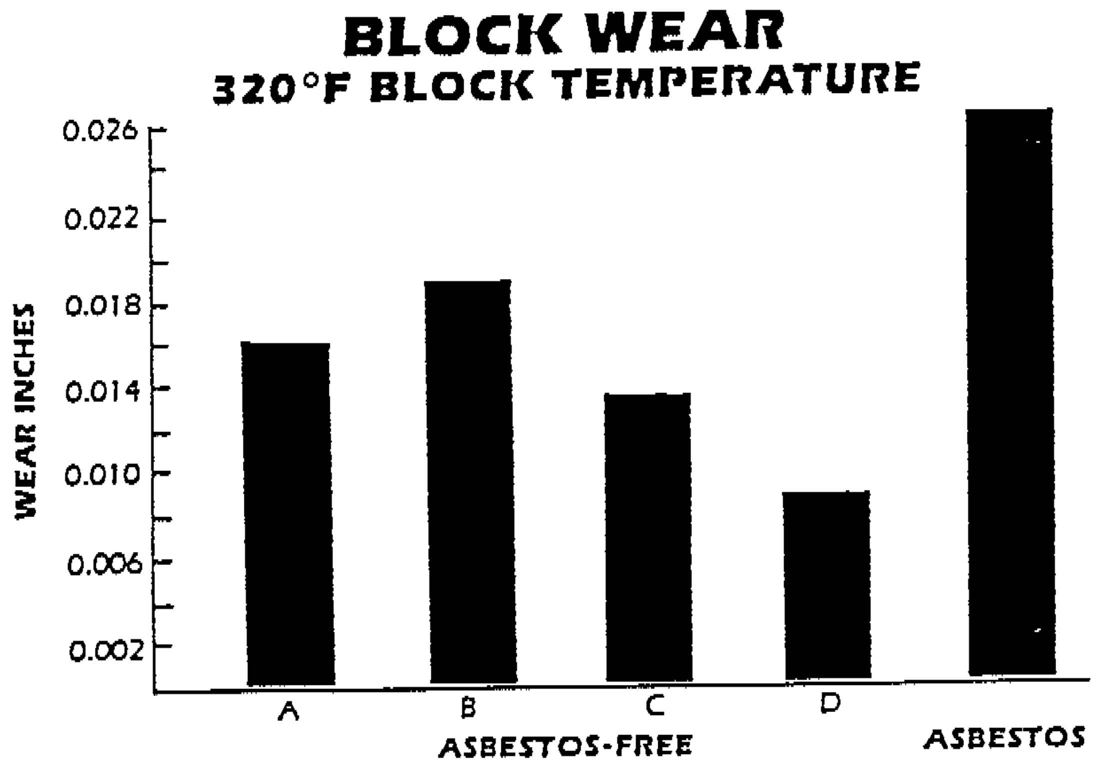

Figure 2.6: Block Wear [2.2].

Table 2.6 Comparison of Edge Code Friction Test and Truck Brake

	Friction Test	Truck Brake (S-Cam : 16 1/2" x 7")	Comments
Surface Speed	1,200 feet/min	1,200 ft/min = 36 mph	Responsable comparison
Unit Loading	Constant at 150 lb/in^2	Variable to a maximum 300 lb/in^2	With pressure sensitive materials this could be a difference
Surface Area	1 in^2	224 in^2	Significant difference
Drum Mass/ Area Ratio	17.66 lb/in^2	.446 lb/in^2	A major difference which affects the rate of heating and cooling of the friction material
Brake Application	Constant Drag, Fixed Input Load, Artificial Heat	Intermittent, Constant Output, Friction Heat	Significant difference

edge code friction test and truck brake. As it can be seen from the comments in the Table 2.6, the block edge codes provide only a reasonable estimate of brake performance fade or wear in service.

2.8 THE STATIC AND DYNAMIC CHARACTERISTICS OF PNEUMATIC ACTUATION SYSTEMS

2.8.1 Static Proportioning

The proportioning of the brake torque generated as a function of axle position is generally achieved by varying the "brake power" at each brake. For an S-cam brake, the brake power is determined by the choice of brake chamber size and the length of the slack adjuster arm. For a wedge brake, the wedge angle and the chamber size determine the brake power. Essentially, static proportioning is adjusted by selecting a value of brake power that will give a desired maximum torque from the brake operated at maximum system air pressure (usually 100 psi).

2.8.2 Dynamics of Actuation Systems

One dynamic phenomenon exhibited by the actuation system is hysteresis. This hysteresis created a time lag in reducing brake torque because the pressure must fall to a lower level than that originally required to obtain a given torque. This effect is important primarily in trying to prevent or remove wheel lockup.

2.8.3 Air Delivery System

An important operational characteristic of the air delivery system is the length of time between the application of pressure at the treadle valve and the achievement of a corresponding pressure in each of the brake chambers. This response time depends upon the "plumbing" of the air system, which is a fairly complicated arrangement of air lines, connectors, relay valves, check valves, and other specialized devices (see Figure 2.7). The response time depends upon the lengths of the air lines, the sizes of the orifices, and the volumes to be filled. As with most vehicle components, a very complicated model could be constructed to study the operation of the air system in detail. Rather than developing a detailed model with the burden of representing a wide variety of air system components, judgement and experience indicate that the input/output characteristics of the air delivery system can be satisfactorily represented using a relatively simple model.

Figure 2.8 shows some time histories of the pressure in trailer brake chamber of some vehicles listed.

These apply times are measured from the instant of pressure increase at the treadle valve. The data are characterized by a time delay for the pressure signal to arrive at the relay valve, a short, slow increase

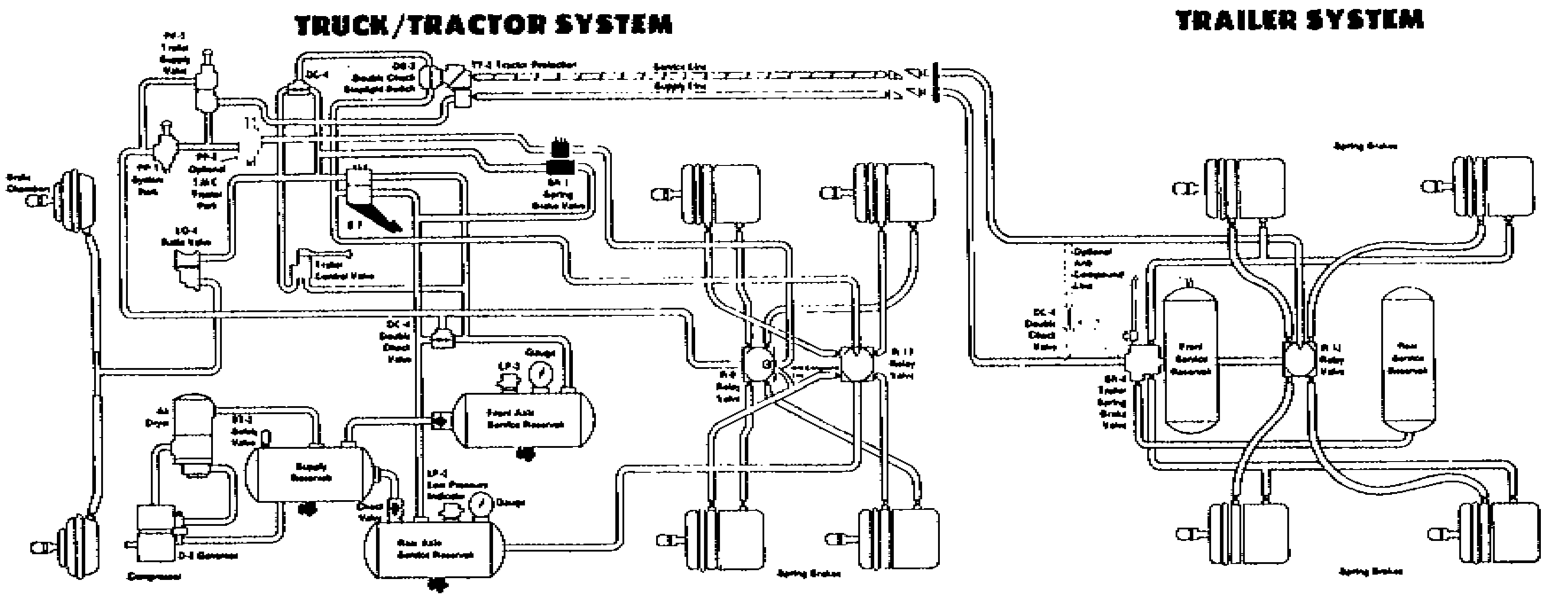

Figure 2.7: Air Brake System [2.2].

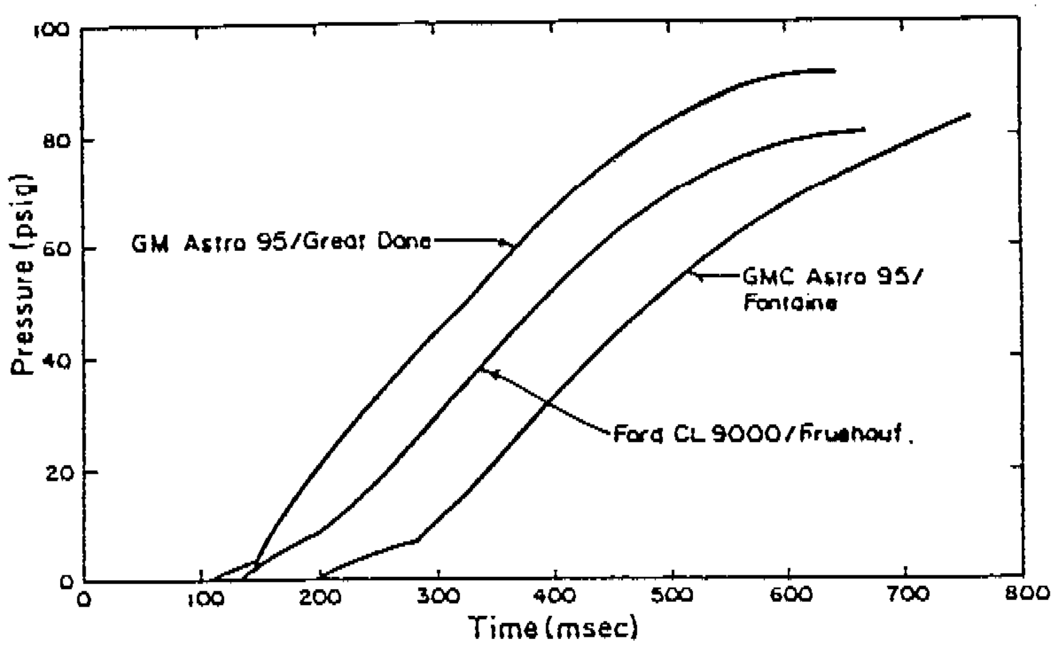

Figure 2.8: Response Time Histories of Trailer Brake Chamber Pressure [2.4].

in pressure as the air chamber is filled, and a rise time as the chamber pressure is increased.

Interestingly, the data obtained show that 3/8 inch diameter air lines may be faster than 1/4 inch or 1/2 inch line diameter lines in typical vehicle systems. Apparently, the 1/4 inch lines are slower due to frictional effects and the 1/2 inch lines are slower due the increased volume when compared to the 3/8 inch line.

2.9 SIMPLE ANALYSIS OF MECHANICAL FRICTION BRAKES USED ON HEAVY TRUCKS

Heavy trucks are equipped almost exclusively with S-cam or wedge-actuated drum brakes. Currently with the development of disc brakes, heavy trucks are being equipped with these.

2.9.1 Disc Brakes

A simple analysis shows the torque output of the disc brake to be related to actuation force by the following equation:

$$T = 2 \ R \ F$$

where

T is torque,

is lining/drum friction coefficient,

R is effective radius based on the pad location and dimension,

and F is the actuation force applied to the pad.

The above equation shows that for simple disc brake operated at a given actuation force, the torque changes linearly (proportionally) with changes in friction coefficient, thereby leading to a "stable" (that is, constant) sensitivity to small changes in friction coefficient regardless of the nominal value of friction coefficient. Disc brakes provide uniform brake torques and have a brake output linearly related to input.

2.9.2 Drum Brakes

Figure 2.9 illustrates the self-energizing principle applicable to leading shoe of a drum brake. This simplified brake configuration shown, consists of a single brake block located on an axis through the center of the drum. (As brakes are typically mounted in trucks, this would be a vertical axis on the truck.) The actuation force, F_I, is applied to the point, B, that is a distance, h, above the pivot point A. The force components between shoe and drum act at point C which is a distance, u, outside the pivot point and a distance, v, below the center line shown. By summing moments on the shoe about point A, the following equation describing the gain of the brake is obtained:

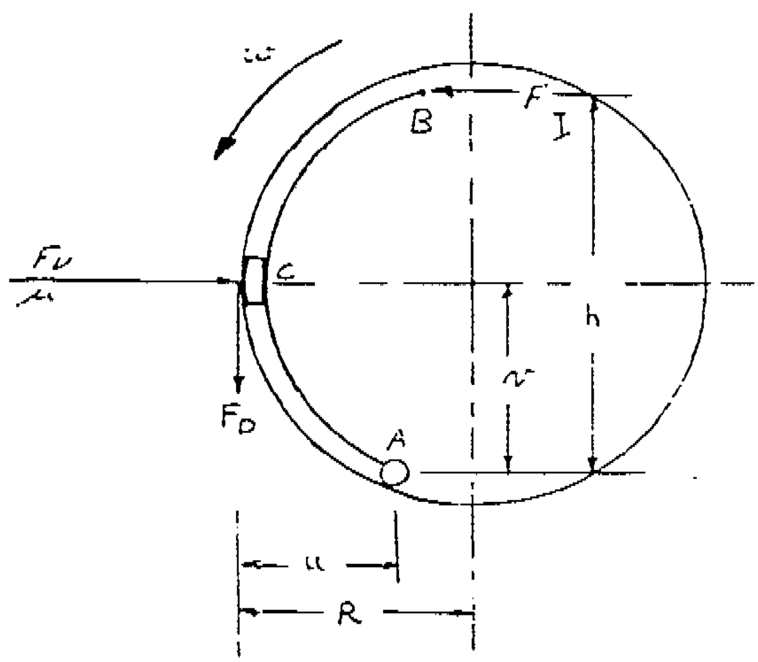

Figure 2.9: Self-Energizing in a Drum [2.4].

$$\frac{F_D}{F_I} = \frac{-h\mu}{v - u\mu}$$

The ratio of the resultant tangent force (F_D) to the actuation force, F_I, is defined as the "shoe" factor. Note that for this direction of rotation of the drum, the force, F_D, produces a moment about the pivot point A aiding the actuation force, F_I, in applying the brake. In this sense, the leading shoe is said to be "self-energizing" or "self-actuating". If $\mu = {}^{v}/_{u}$, the shoe factor becomes infinite and for $\mu > {}^{v}/_{u}$, the wheel is self-locking, that is, the brake "sprags" with rotation of the drum becoming impossible.

In the case of reversal of rotation, the shoe shown becomes a "trailing" shoe. In this case, the drag force, F_D, opposes the actuation force, F_I, in creating a moment rotating the shoe about the pivot point. Equation below, describing the shoe factor for a trailing shoe, indicates that, as u becomes large, the gain of the trailing shoe approaches ${}^{h}/_{u}$. However, for typical brake geometry and values of μ , F_D/F_I is less than one for a trailing shoe.

$$\frac{F_D}{F_I} = \frac{h\mu}{v + u\mu}$$

Pressure Distributions Over the lining

The simple model just employed is clearly not a good approximation to reality in that the linings subtend a large arc equal to more than half of the circumference of the drum. However, equations of the form expressed above can be used to express the shoe factors derived for most drum brakes.

To perform a more sophisticated analysis of drum brakes, the classical approach has been to either assume a pressure distribution or develop a rationale, usually based on wear, that will determine a pressure distribution. For example, assume that:

a) The brake drum, shoe, and shoe pivot are rigid;

b) The lining follows the shape of the drum; and

c) The wear at any point on the lining is proportional to pressure, then the pressure distribution has a sinusoidal shape.

Another approach has been to:

1. Assume a uniform pressure distribution;
2. carry out the calculations for the shape factors; and
3. Compare the results with results obtained using a sinusoidal pressure distribution.

These comparisons show only minor differences between the use of a sinusoidal versus a uniform pressure distribution. Nevertheless, experimental results obtained with (1) an extremely elastic shoe (a condition that might be believed to lead to a uniform pressure distribution) and (2) a very rigid shoe, have shown much larger torque capabilities for the brakes with the elastic shoe.

Figure 2.10 shows an example result indicating the influence of stiffness properties on the pressure distribution. Note the wide variation in the form of the pressure distribution. Clearly, if these predictions are representative, the flexible shoe will have a higher average lining pressure and therefore produce a higher torque than that obtained with rigid shoe.

2.9.3 S-Cam and Wedge Actuation Mechanisms

The input to the actuation mechanism is the force, produced by the brake chambers. The function of the air chamber is to apply rotary motion to a cam or linear motion to a wedge.

In the case of the cam brake, the input force, F_p, acts at a distance, ℓ_s (the "slack adjuster" arm length), to produce a torque on the cam. However, since the cam is fixed, the forces to the two brake shoes need not be equal. Rather, the deflections of the shoes are equal. The moments of the reaction forces acting on the cam are in balance with the torque applied to the cam shaft. From Figure 2.11, the following Equations (1), (2) and (3) can be derived to describe the moment balance on the cam and about both shoe pivot points.

$$T_c = (F_1 + F_2)\, R_c \qquad (1)$$

where R_c = effective cam radius

$$F_{D1} = \frac{h\mu}{v - u\mu}\, F_1 \qquad (2)$$

$$F_{D2} = \frac{h\mu}{v + u\mu}\, F_2 \qquad (3)$$

Note that point B is on the leading shoe and C is on the trailing shoe.

Considering that the displacement of the two shoes are equal, the normal forces F_{D1}/μ and F_{D2}/μ are also assumed to be equal. Therefore Equations (2) and (3) can be substituted in Equation (1) and $F_{D1} = F_{D2} = F_D$. This will give the relationship between the drag force and the cam shaft torque T_c as:

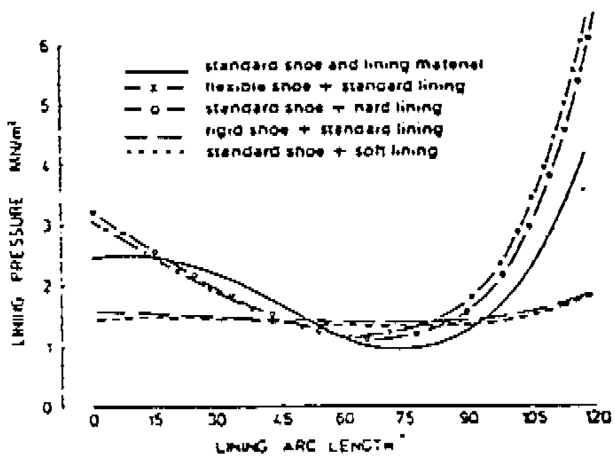

Figure 2.10: Pressure Distribution [2.4].

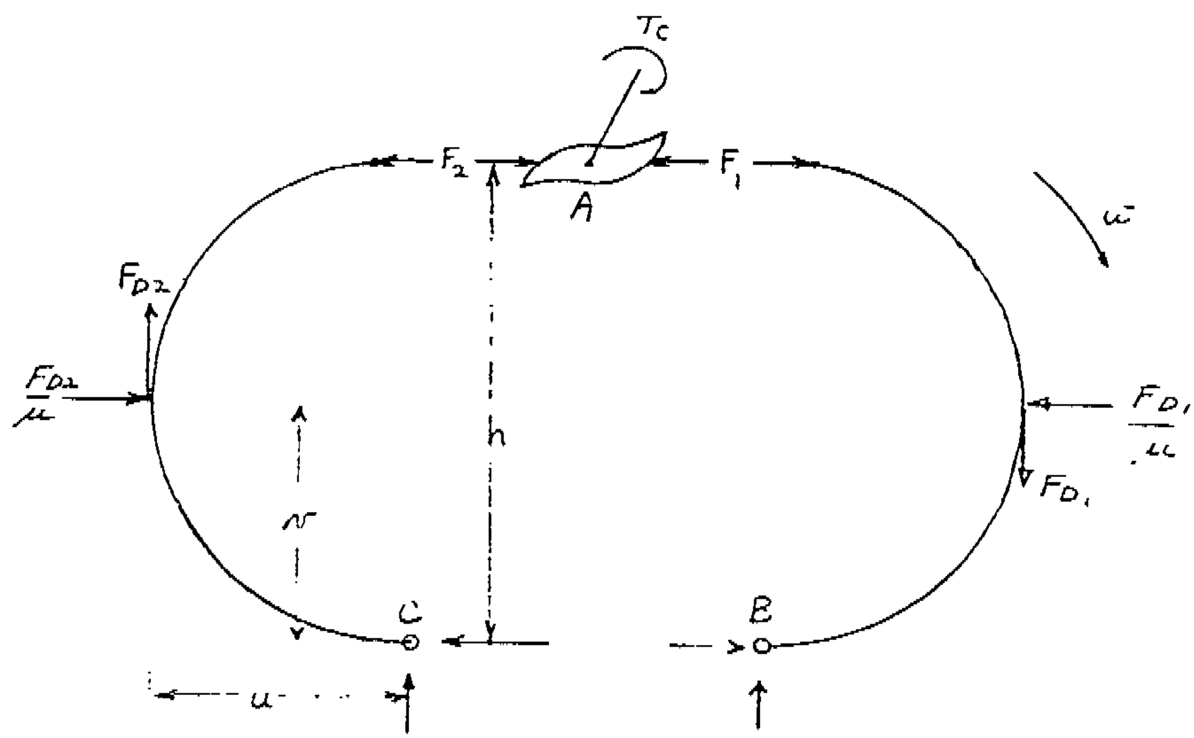

Figure 2.11: S-Cam Free-Body Force Diagram [2.4].

$$T_C = R_C F_D \cdot \frac{2\,v}{h\,\mu}$$

The brake torque: $T = 2\,R_D\,F_D$ where R_D is the drum radius

The cam torque: $T_c = F_p\,\ell_s$

Therefore $$\frac{T}{F_p} = \frac{\ell_s}{R_c}\,\frac{h\mu}{v}\,R_D \qquad (4)$$

The simple result obtained in Equation (4) has several interesting implications. First, the "μ-sensitivity" of the brake is linear even though it is a leading/trailing shoe brake. Consequently, if the friction coefficient changes slightly during a stop, the gain of the brake only changes slightly.

Second, since the two actuation forces are not equal, the classical idea of adding shoe factors to obtain an overall brake factor is not applicable to the S-cam brake. It seems reasonable to use an expression like Equation (4) and not try to employ a brake factor.

Finally, as the cam rotates, the effective radius changes and the gain of the brake decreases as the radius increases at higher levels of input force, F_p.

The wedge actuation mechanism is amenable to employing the brake factor approach. In the case of a dual-wedge brake, one side of a wedge serves to apply the input force to a leading shoe and the other side of that wedge acts against a reactive plunger (see Figure 2.12). For normal values of friction coefficient, the two-leading shoe brake factor for a dual-wedge brake equation.

$$\frac{F_D}{F_I} = \frac{h\mu}{v - u\mu}$$

can be used with:

$$F_I = F_p\,(\tfrac{1}{2}\,\mathrm{Cot}\,\theta_w/2)$$

where θ_w is the wedge angle, and

$$F_T = 2F_D$$

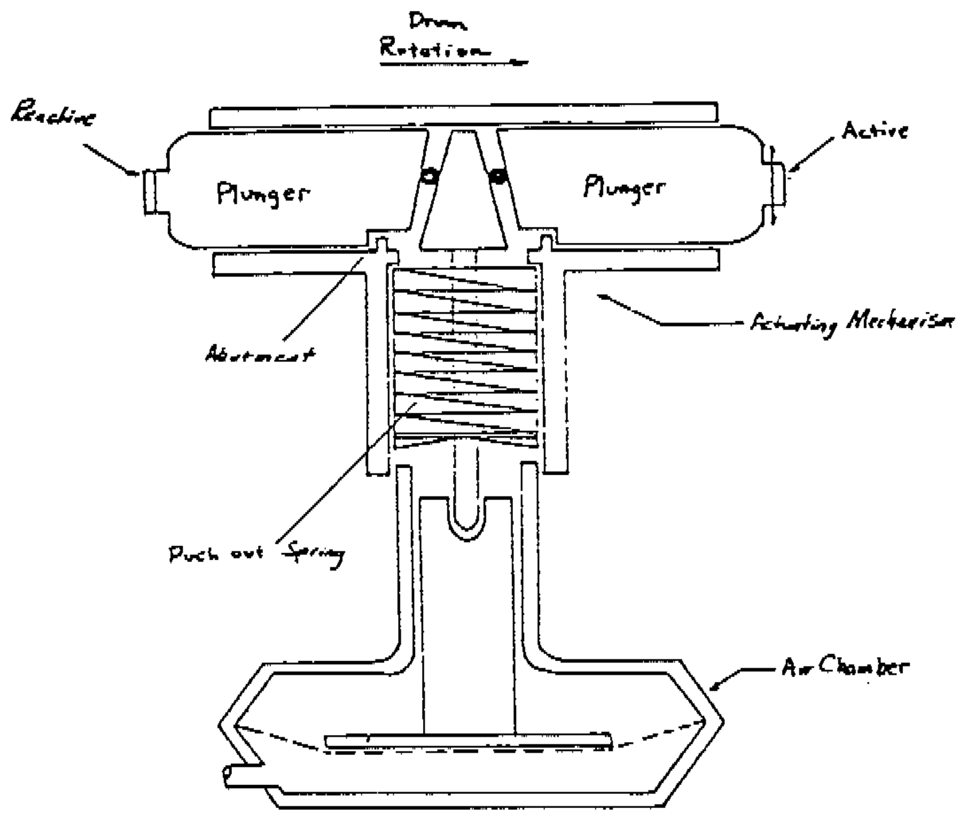

Figure 2.12: Dual-Wedge Brake [2.4].

(i.e., both shoes), the following result is obtained:

$$B_F = \frac{F_T}{F_p} = \frac{h\mu}{v - u\mu} \operatorname{Cot} (\theta_w/2)$$

where F_p is the force from one brake chamber and B_F is the brake factor.

The overall effectiveness is given by:

$$\frac{T}{P} = A \left(\frac{R_D \mu h}{v - u}\right) \operatorname{Cot} (\theta_w/2)$$

where A is the effective area of one brake chamber and P is the air pressure in the brake chamber.

The above equation illustrates that as μ approaches v/u, the μ-sensitivity of the dual wedge brake is greater than that for a cam brake.

2.10 AIR BRAKE SYSTEM MAINTENANCE AND TROUBLESHOOTING [2.2]

Majority of the problems in air brake system are caused by either component leakage or malfunction.

The steps involved in determining the overall system leakage and correcting the component function are as follows:

Objective:

- Isolate the leaking or malfunctioning component.

Steps:

a) Determine whether each component functions the way it should.

b) Determine whether leakage in the system is within allowable limits.

Tests:

1. Verify proper functioning of air compressor, governor, and low pressure warning systems. (Check the pressure build-up).

Table 2.7 Causes and Remedies for Brake System Problems

Cause	Remedy
Air System Contamination	Drain reservoirs frequently (daily). Periodically inspect automatic drain valves. Use air dryers to remove moisture.
Air Compressor Excessive Heat, Carbon Formation in Inlet and Discharge Valves, and Discharge Lines	Proper cooling, lubrication and intake filtration. Minimum coolant line size is 1/2 " tubing; Minimum coolant flow is 2.5 U.S gal/min. at engine governed speed; Coolant temperature should not exceed 200° F. Minimum oil/lubricant supply line size is 1/4" tubing; Minimum oil return line size is 5/8" tubing.

2. Check for leakage in the supply side/reservoir air supply. (Check the time required for the pressure drop and establish whether the leakage is within allowable limits).

3. Check both the front and rear service circuits for leakage. (With brakes applied, estimate the time required for the pressure drop).

4. Check out the manual, driver controlled emergency brake system, and confirm the manual control of the spring brake system.

5. Check out the automatic emergency system which is triggered by the loss of system pressure.

Some of the causes of brake system problems are:

- Air system contamination;

- Air compressor excessive heat.

The Table 2.7 shows the remedies for overcoming the problems.

2.11 CONCLUSIONS

In this chapter, the fundamentals of heavy vehicle braking and requirements are presented. The highlights on FMVSS 121, BMCS - FMCSR Part 393, and New York Thruway regulations are outlined. Heavy vehicle retarding capacity and performance levels of air brake systems are summarized. A detailed discussion on foundation brake, brake balance, brake rating, brake adjustment, brake pads, and brake block edge code is presented. The static and dynamic characteristics of pneumatic actuation systems and a simple analysis of mechanical friction brakes are also presented. Finally, a brief discussion on air brake system maintenance and troubleshooting is provided.

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2.2 "Anti-Lock Braking Systems for Road Vehicles", SAE - MEP 226.

2.3 "Automatic Slack Adjusters", SAE - SP 574.

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2.5 "Brake and Wheel Equipment for Worldwide Truck Application", SAE -SP 562.

CHAPTER 3

BRAKE SYSTEM HARDWARE AND CONTROL SYSTEMS

3.1 AIR BRAKE SYSTEM

Air brake system for heavy vehicles are designed to meet the FMVSS/CMVSS 121 regulations.

A schematic diagram of a tractor-semitrailer air brake system is illustrated in Figure 3.1 [3.1]. The air brake system produces compressed air, stores the air, and makes its use possible by converting its energy into mechanical work used to actuate the wheel brakes of the vehicle.

Referring to Figure 3.1, the compressor (1) takes air from the atmosphere, compresses it, and pumps it into the reservoirs (2) where it is stored for use. The governor, mounted on or near the compressor, controls the compressor so that when maximum reservoir air pressure is obtained no further air is pumped to the reservoir. The reservoir capacity should be no less than 12 times the combined volume of all brake chambers used on the vehicle. Through the compression process, the humidity in the air liquefies and collects in the reservoir. To keep the brake system in good condition, daily draining of the reservoirs is required. To overcome the water vapor problem, standard air brakes use two reservoirs. The first one in line is called the wet (or supply) reservoir. The second one is called the service reservoir and stores the air which is used for brake actuation. In Figure 3.1 a single reservoir with two compartments is illustrated. A reservoir mounted moisture ejector is sometimes used which automatically ejects moisture with each brake application. Finally, an air dryer may be used which removes water from the compressed air before it gets into the reservoir. To protect the air in the reservoir in case of compressor or supply failure, a one-way check valve (3) is installed in front of the reservoir which it protects.

The brake application valve (4) is used to control the flow of air to the wheel brakes and to allow modulation of the braking process. Air at reservoir pressure is constantly supplied to the brake application valve. Brake lines, running from the brake application valve to the front and rear brakes, contain air only when the brakes are applied, and then only at the pressure demanded by the driver.

Air brake systems are equipped with one or more quick release valves to achieve a faster brake release. The valve exhausts brake line pressure at the point of installation, thus supplementing the exhaust at the brake application valve.

The relay quick-release valve (6) is connected into the line leading to the rear brakes as shown in Figure 3.1. The valve helps speed brake application and release.

The compressed air at the wheel brakes is converted into mechanical energy by the brake chamber (7) and slack adjuster (or wedge). Movement of

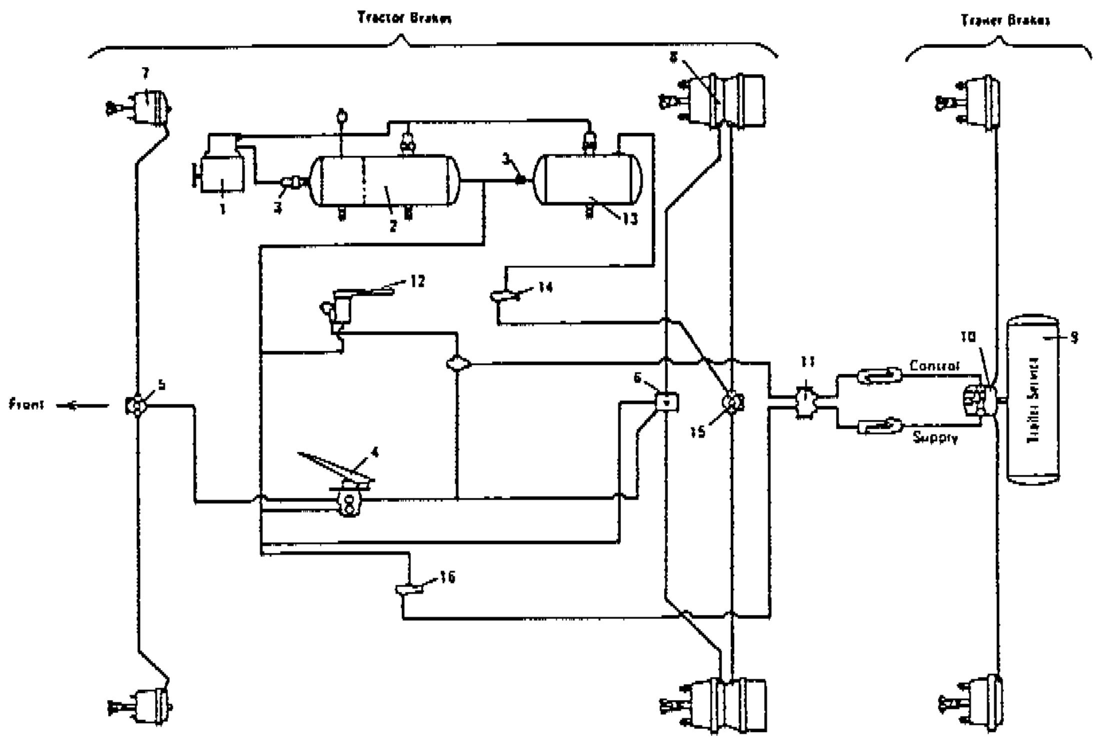


Figure 3.1: Schematic Diagram of a Tractor-Semitrailer Air Brake System [3.1].

the slack adjuster arm causes it to rotate a cam shaft which forces the brake shoes to contact the brake drum.

When a trailer is added to the tractor, special provisions are made to apply the trailer brakes. A trailer reservoir (9) as shown in Figure 3.1 is used to store the compressed air for the trailer brakes. A relay-emergency valve (10) installed on the trailer is used to supply the trailer reservoir with compressed air from the tractor reservoir and to control the brake line pressure and hence the brake force of the trailer - as demanded by the driver. The control line comes from the brake application valve and, when the driver depresses the foot pedal, pressure equal to the tractor brake line pressure opens a port in the relay-emergency valve and allow air at the same pressure level to leave the trailer reservoir and go through the relay-emergency valve to the trailer brake chambers. The relay-emergency valve also acts as an emergency device in case of severe air loss or trailer breakaway. In case of a trailer breakaway the trailer brakes will be applied automatically because the emergency section of the relay-emergency valve will use full trailer reservoir pressure to apply the trailer brakes. If a severe trailer brake leak or trailer breakaway occurs, the tractor brake system is protected by the tractor protection valve (11) as illustrated in Figure 3.1. It is designed to control the service and supply lines to the trailer. It is both automatic and manual. In an emergency the driver can activate it by use of the manual control (12) located in the cab. If the driver does not operate the control, the tractor protection valve automatically will apply the trailer brakes -when the trailer brake line pressure has decreased to between 20 and 45 psi - by venting the supply or emergency line and thereby triggering the emergency section of the relay-emergency valve. The relay emergency valve is combined with a quick-release valve to allow a quick release of the air from the brake chambers when the brakes are released.

Since the tractor-protection valve is easy to use by the driver by means of the control lever in the cab, it is frequently used to apply the trailer brakes for parking the tractor-semitrailer. However, this should not be done. If a leak develops, no more air can be supplied to the trailer reservoir from the tractor since the tractor protection valve has vented the supply or emergency line between the tractor protection valve and relay-emergency valve.

The most widely used parking brake system on air braked vehicles is the spring brake (Figure 3.1). It operates the vehicle service brakes (tractor rear axle brakes) by the energy stored in compressed coil springs. When the parking is not applied, the reservoir pressure is used to compress the coil springs and hold the brakes in the released position. A separate reservoir (13) as illustrated in Figure 3.1 is used for this purpose. A tractor parking valve (14) is used to apply the parking brakes. A quick-release valve (15) is used to exhaust the air of the "parking" chamber in case of a parking brake application.

Frequently, a dash mounted tripping control valve (16) is used to apply the trailer service brake through the action of the relay-emergency valve when the tractor reservoir pressure drops to 55-60 psi.

Air brake systems used in pre-121 systems with single circuit and FMVSS 121 system with dual circuit designed by Bendix are shown in Figures

3.2 and 3.3, respectively.

Several custom designed air brake systems are available for tractor, trailer, and dolly. Figures 3.4 to 3.10, show Bendix custom designed air brake systems circuitry and instructions for:

1. Four or Six Wheel Tractor;
2. Two or Four Wheel Trailer with Service and Parking Brake Control;
3. Single Axle Towing Trailer;
4. Tandem Axle Towing Trailer;
5. Single Axle Dolly;
6. Two Axle Dolly; and
7. Two or Four Wheel Dolly with Emergency Valve.

For Fleets utilizing existing trailers and dollies, and mixing them with new trailers and dollies, the following should be considered [3.2].

1. If the vehicles are pre-FMVSS 121, it is highly recommended that both the towing trailers and dollies be updated per the information provided in [3.2]. Converting non-towing trailers to towing type should be done according to these same guidelines.
2. To meet the New York Thruway Provisions, arrange the equipment so that there is always a booster relay valve on every other unit of a train. The R-8P booster relay valve can be located on either the towing trailer or towed dolly.
3. The most economical arrangement for fleets purchasing new dollies to convert existing trailers to doubles and triples operation is to specify the R-8P booster relay valve on the dolly, because fewer R-8P valves are required. This can only be done when converting existing trailers since the conversion is not covered by FMVSS 121 and the trailers do not have to comply with the requirements.

3.2 AIR BRAKE SYSTEM HARDWARE

In the following section, the various air brake system hardware are described. Their operating principle, requirements, and the names of major manufacturers are presented.

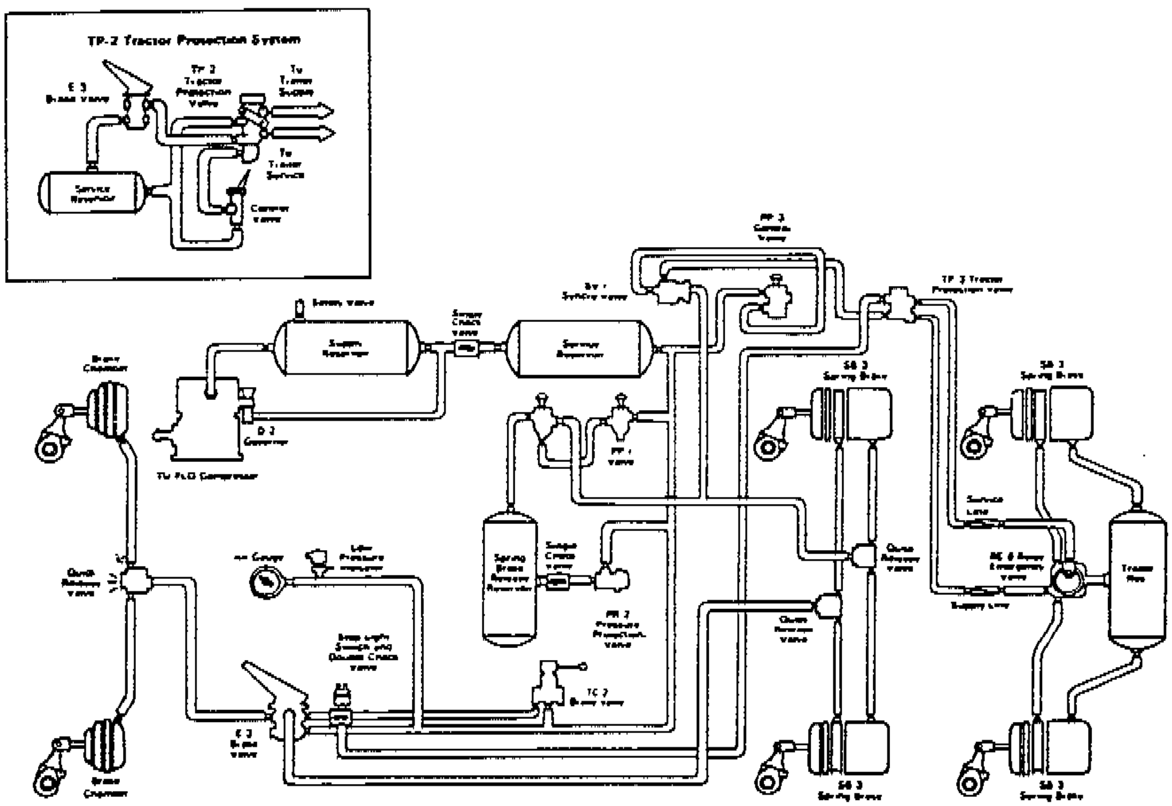


Figure 3.2: Schematic Diagram of a Single Circuit, Bendix-Air Brake System [3.2].

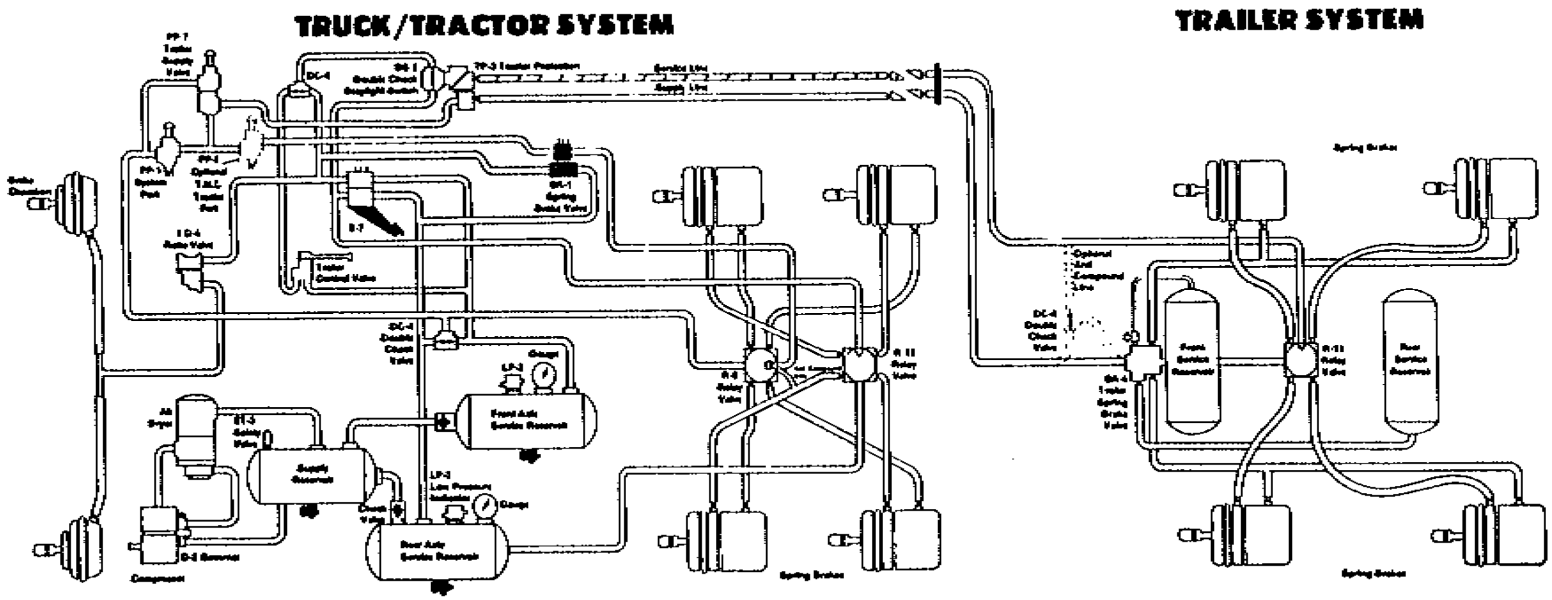


Figure 3.3: Schematic Diagram of a Dual Circuit, Bendix-Air Brake System [3.2].

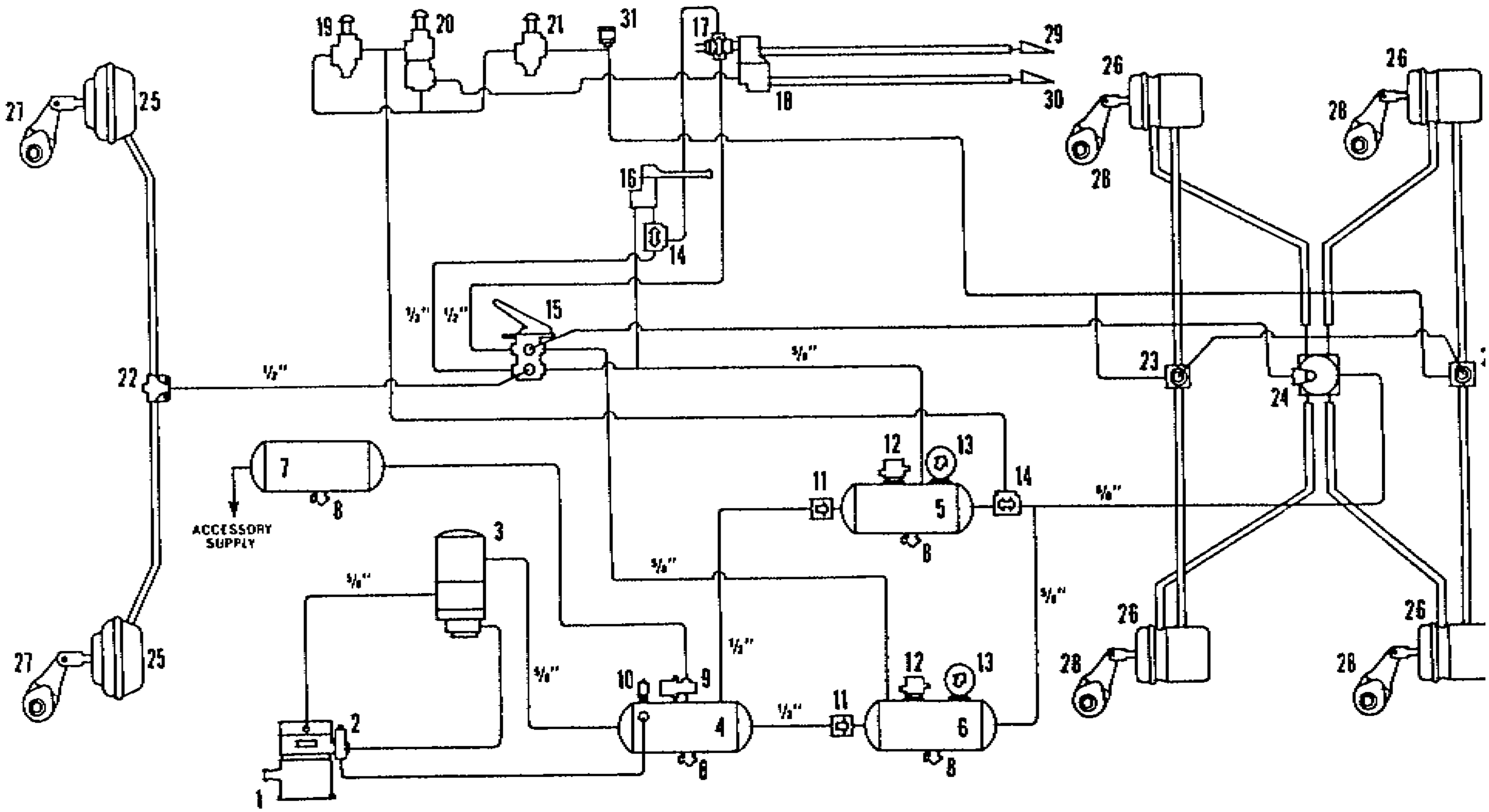


Figure 3.4a: Air Brake System for a Four or Six Wheel Tractor (Circuit Design) [3.2].

ITEM NO.	QTY.	BENDIX MODEL	BENDIX PIECE NO.	DESCRIPTION
1	1			AIR COMPRESSOR ______ C.F.M
2	1	D-2		GOVERNOR
3	1	AD-2		AIR DRYER
4	1			RESERVOIR-SUPPLY
5	1			RESERVOIR-SERVICE-FRONT AXLE
6	1			RESERVOIR-SERVICE-REAR AXLE
7	1			RESERVOIR-ACCESSORY SUPPLY
8	4			DRAIN COCK
9	1	PR-4		PRESSURE PROTECTION VALVE
10	1	ST-3		SAFETY VALVE
11	2	SC-1		SINGLE CHECK VALVE
12	2	LP-3		LOW PRESSURE INDICATOR
13	2			AIR GAGE
14	2	DC-4		DOUBLE CHECK VALVE
15	1	E-6		DUAL BRAKE VALVE
16	1	TC-2		TRAILER CONTROL VALVE
17	1	DS-2		DBLE. CHK. VALVE & STOP LAMP SW
18	1	TP-3		TRACTOR PROTECTION VALVE
19	1	PP-1		PARKING CONTROL VALVE
20	1	PP-7		TRAILER SUPPLY VALVE
21	1	PP-8		TRACTOR PARKING CONTROL VALVE
22	1	QR-1		QUICK RELEASE VALVE
23	2	QR-1C		QUICK RELEASE/DOUBLE CHK. VALVE
24	1	R-12/R-14		SERVICE RELAY VALVE
25	2			BRAKE CHAMBER-FRONT
26	4			SPRING BRAKE ACTUATOR-REAR
27	2	ASA-2		SLACK ADJUSTER-AUTOMATIC-FRONT
28	4	ASA-2		SLACK ADJUSTER-AUTOMATIC-REAR
29	1	HC-2		HOSE COUPLING-TRAILER SERVICE
30	1	HC-2		HOSE COUPLING-TRAILER SUPPLY
31	1	LP-3		PARKING INDICATOR SWITCH

SYSTEM NOTES / FEATURES

1. Maximum recommended air brake reservoir pressures is 125 p.s.i.
2. All tubing to be 3/8 o.d. (copper or synflex nylon spec bw-250-m) unless otherwise specified. Inserts must be used with nylon tubing. Do not use nylon tubing where temperature exceeds 180 degrees F.
3. Air brake tubing must be arranged to eliminate elbows, close bends, reducer fittings, unnecessary line lengths, and etc., wherever possible.
4. Compressor air intake to be connected to the intake manifold or oil bath air cleaner of engine unless air strainer is used.
5. Compressor oil return passage should be of such size and location to prevent compressor crankcase flooding.
6. All accessories must be supplied through pressure protection valve (item 9) & accessory supply reservoir (item 7).
7. Mount check valves (item 11) directly on reservoirs as illustrated. Mount double check valve (item 17) directly on tractor protection valve (item 18).
8. This Bendix system features a control valve (item 21) which permits independent control of the tractor parking brakes. This control valve is a non-automatic type valve and must be located in area remote and detached from parking control and trailer supply valves (items 19 & 20).
9. This is a piping schematic only and is not a construction drawing. O.E.M. manufacturers may deviate for convenience.

Figure 3.4b: Air Brake System for a Four or Six Wheel Tractor (Instructions) [3.2].

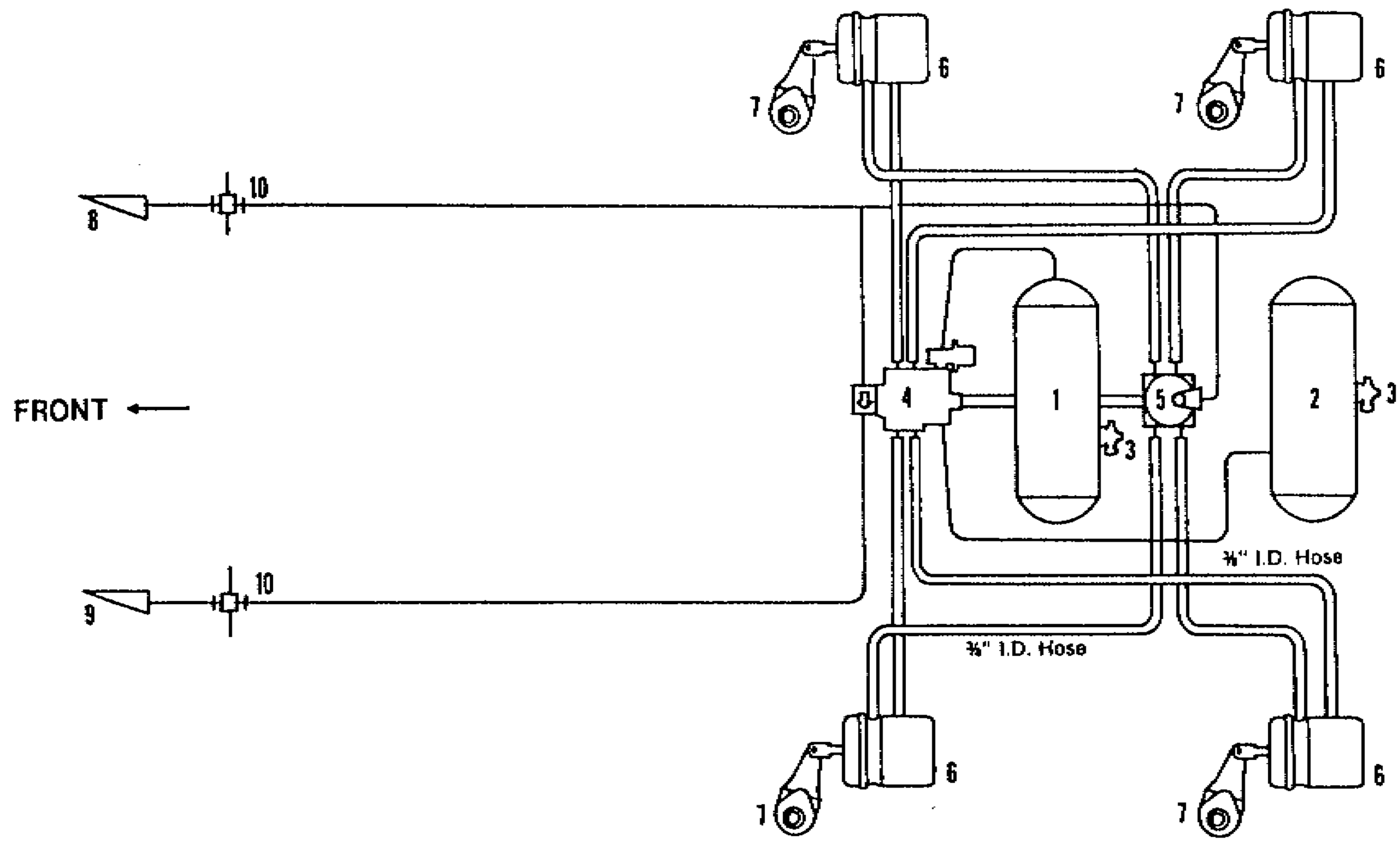


Figure 3.5a: Air Brake System for a Two or Four Wheel Trailer with Service and Parking Brake Control (Circuit Design) [3.2].

BENDIX DEVICES SPECIFIED

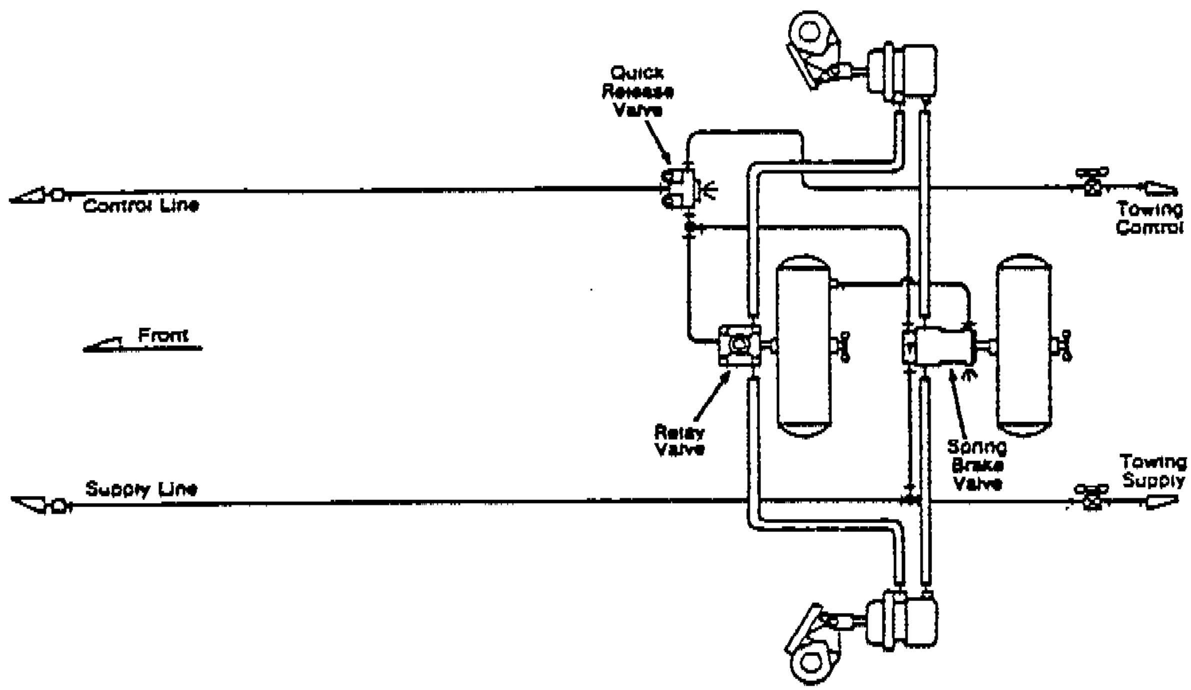
ITEM NO.	QTY.	BENDIX MODEL	BENDIX PIECE NO.	DESCRIPTION
1	1		101127	RESERVOIR-SERVICE-FRONT
2	1		101128	RESERVOIR-SERVICE-REAR
3	2			DRAIN VALVE-MANUAL
4	1	SR-4	101112	TRAILER VALVE ASSEMBLY
5	1	R-12/R-14		RELAY VALVE
6	4			SPRING BRAKE ACTUATOR
7	4	ASA-2		SLACK ADJUSTER-AUTOMATIC
8	1	HC-2		COUPLING-SERVICE
9	1	HC-2		COUPLING-SUPPLY
10	2			ANCHOR FITTING

SYSTEM NOTES / FEATURES

1. Maximum recommended air brake reservoir pressure is 125 p.s.i.
2. All tubing to be 3/8 O.D. (copper or synflex nylon spec. BW-250-M) unless otherwise specified. Inserts must be used with nylon tubing. Do not use nylon tubing where temperature exceeds 180° F.
3. Air brake tubing must be arranged to eliminate close bends, elbows, reducer fittings, unnecessary line lengths, and etc., wherever possible.
4. Reservoirs specified are suitable for four (Type 30) spring brake actuators.
5. This Bendix system features anti-compounding piping that prevents simultaneous application of the service and parking brakes.
6. This is a piping schematic only and is not a construction drawing. O.E.M. manufacturers may deviate for convenience.
7. If this vehicle is to be a two-wheel trailer remove (2) of Items 6 & 7, and related piping.

Figure 3.5b: Air Brake System for a Two or Four Wheel Trailer with Service and Parking Brake Control (Instructions) [3.2].

SINGLE AXLE TOWING TRAILER (UP TO 28 FEET)



DEVICE	MODEL	TYPICAL PC. NO.
Quick Release Valve	QR-N	103999
Relay Valve	R-12	102165
Spring Brake Valve	SR-2	287375

PERFORMANCE FEATURES

1. Valves specified when equipped with proper sized hose and tubing enable compliance with FMVSS '121' timing requirements.
2. Pressure difference between signal and brake chamber pressure is less than 4 psi in the normal braking range.
3. Pressure difference between "In" coupling and "Out" coupling pressure is 1 psi maximum.

DESCRIPTION

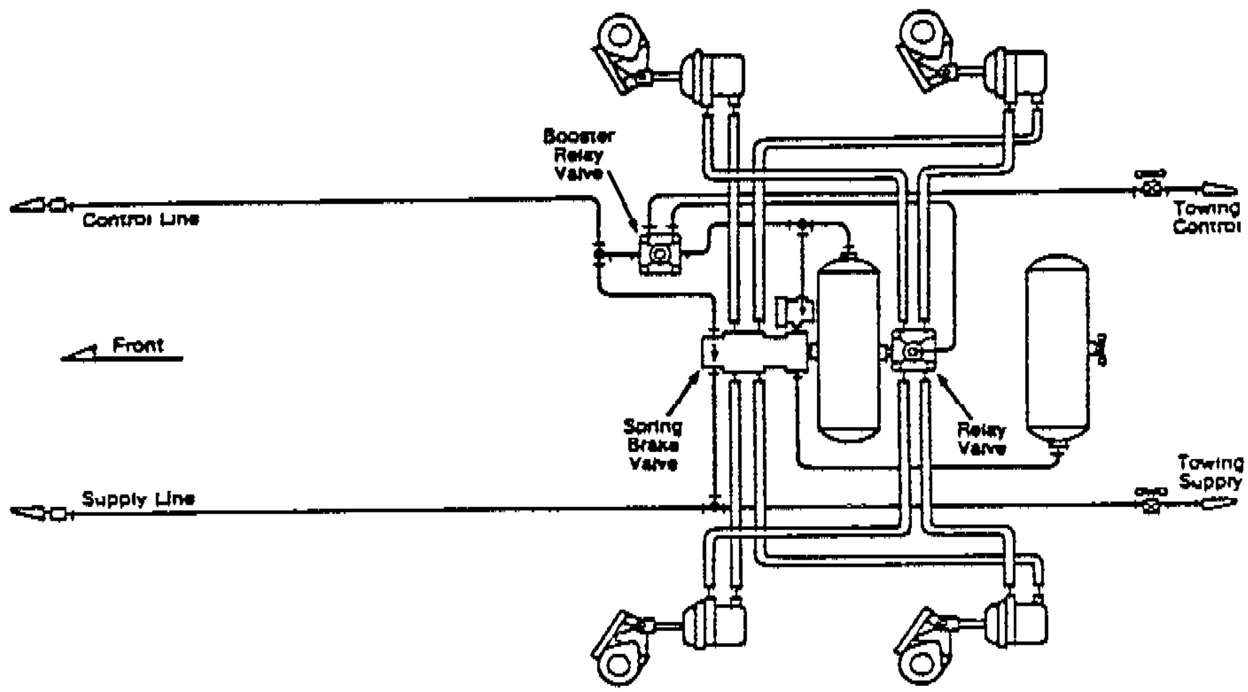
The quick release valve is used on the 28' single axle trailer to comply with brake chamber release times required by FMVSS '121.'

BENDIX HVSD RECOMMENDATIONS

Mount the QR valve as close as possible (two feet desired) to the trailer relay valve. An R-12 relay valve is recommended for the service relay valve.

Figure 3.6: Air Brake System for a Single Axle Towing Trailer [3.2].

TANDEM AXLE TOWING TRAILER (UP TO 48 FEET)



DEVICE	MODEL	TYPICAL PC. NO.
Booster Relay Valve	R-8P	287114
Relay Valve	R-12	102165
Spring Brake Valve	SR-4	101112

PERFORMANCE FEATURES

1. Valves specified when equipped with proper sized hose and tubing enable compliance with FMVSS '121' timing requirements.
2. Pressure difference between signal and brake chamber pressure is less than 4 psi in the normal braking range.
3. Pressure difference between "In" coupling and "Out" coupling pressure is zero psi.

DESCRIPTION

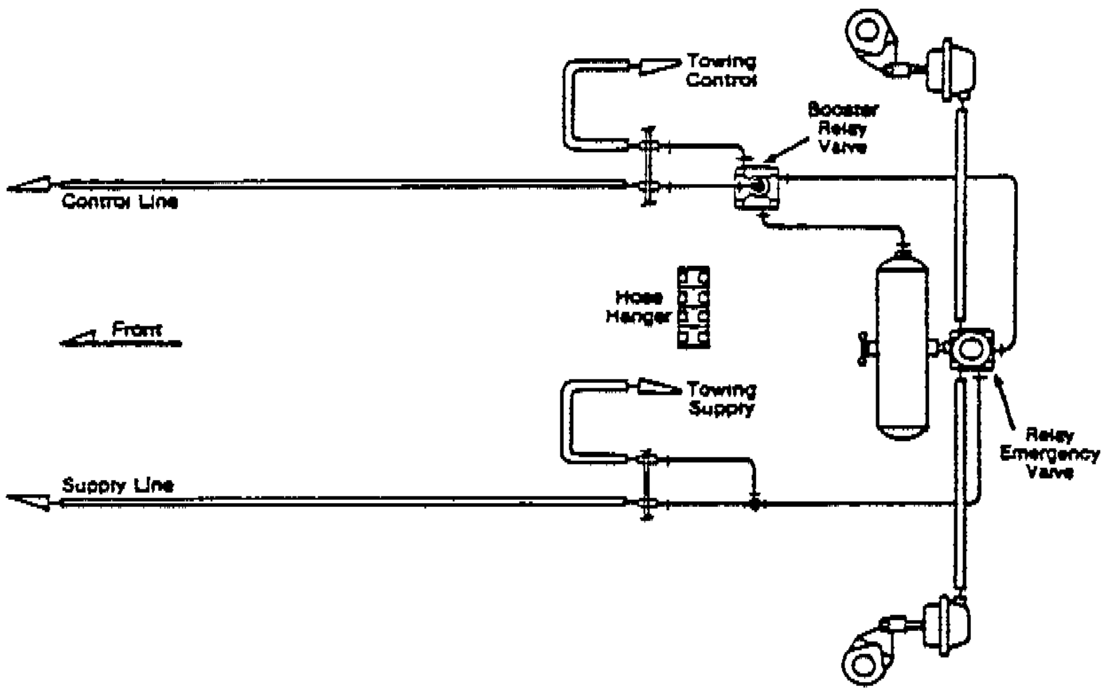
Compliance with timing requirements of the regulations requires a booster relay valve. The R-8P booster relay valve is designed specifically as a retransmission device for multi-bottom vehicles. The R-8P has no pressure differential and transmits the control pressure to the towed trailer and/or dolly without a pressure loss.

BENDIX HVSD RECOMMENDATIONS

The R-8P booster relay valve should be located as close as possible (two feet desired) to the towing control coupling. An R-12 relay valve is recommended for the service relay valve.

Figure 3.7: Air Brake System for a Tandem Axle Towing Trailer [3.2].

SINGLE AXLE DOLLY



DEVICE	MODEL	TYPICAL PC. NO.
Booster Relay Valve	R-8P	287114
Relay Emergency Valve	RE-6	281672

PERFORMANCE FEATURES

1. Valves specified when equipped with proper sized hose and tubing enable compliance with FMVSS "121" timing requirements.
2. Pressure difference between signal and brake chamber pressure is less than 4 psi in the normal braking range.
3. Pressure difference between "In" coupling and "Out" coupling pressure is zero, psi.
4. Can be towed unladen by connecting the towing control line to its hose hanger.
5. Booster relay valve required on New York Thruway (if not on the towing trailer).
6. Improves timing and pressure balance in all towed units (rearward) in the train.

DESCRIPTION

A booster relay valve on dollies is the most economical means of train compliance for New York Thruway Provisions.

The R-8P relay valve is designed specifically as a retransmission device for multiple trailer applications. The R-8P has no pressure differential and transmits the control pressure to the towed trailer or dolly without a pressure loss.

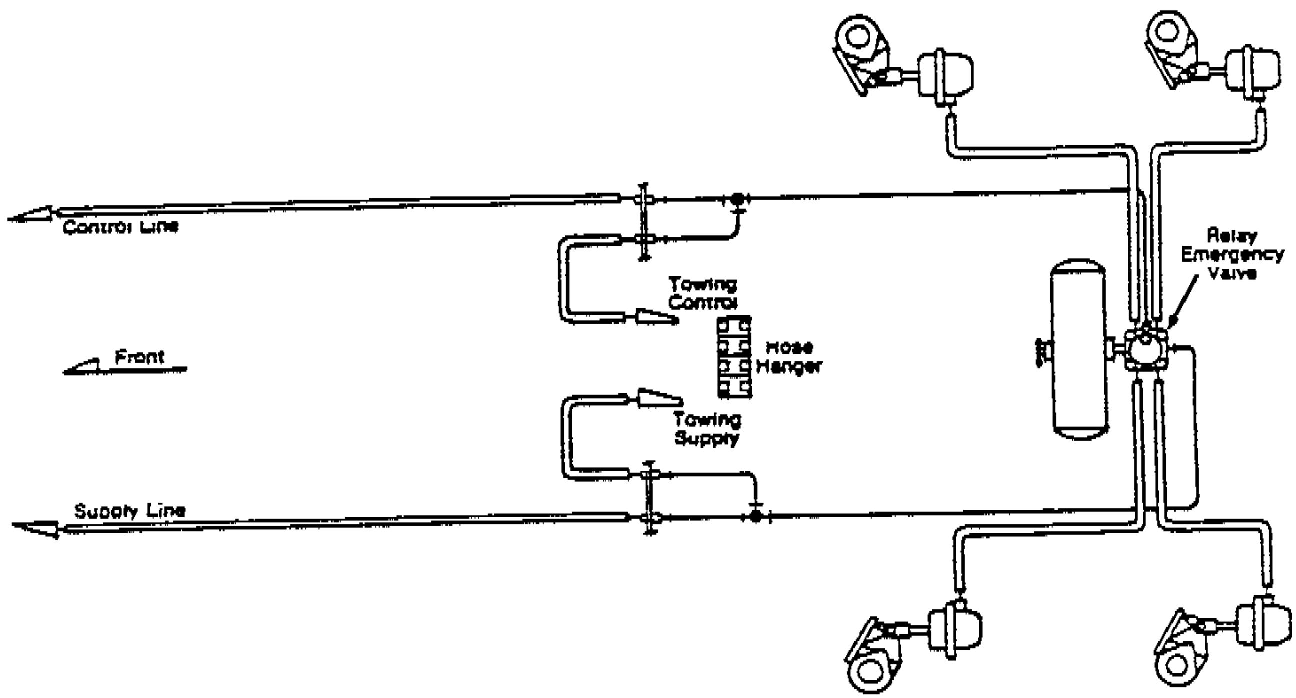
BENDIX HVSD RECOMMENDATIONS

The R-8P booster relay valve should be located as close as possible (two feet desired) to the towing control line. An RE-6 relay emergency valve is recommended for the service relay valve.

NOTE: The use of spring brakes are not required by FMVSS "121" on dollys and most vehicle operators have chosen not to use them. If you desire spring brakes on dollys for your operation, Bendix HVSD has system designs available. Contact Bendix HVSD for further information.

Figure 3.8: Air Brake System for a Single Axle Dolly [3.2].

TWO AXLE DOLLY



DEVICE	MODEL	TYPICAL PC. NO.
Relay Emergency Valve	RE-6	281672

PERFORMANCE FEATURES

1. Valve specified when equipped with proper sized hose and tubing enable compliance with FMVSS '121' timing requirements.
2. Pressure difference between signal and brake chamber pressure is less than 4 psi in the normal braking range.
3. Pressure difference between "In" coupling and "Out" coupling pressure is zero psi.
4. Can be towed unladen by connecting the towing control line to its hose hanger.

DESCRIPTION

This dolly can only be used with trailers equipped with booster relay valves when operating on the New York Thruway. It would generally be used with towing trailers longer than 28 feet or most "doubles" rather than "triples" combinations. These trailers a plumbed per the Bendix HVSD recommendatior would have the R-8P booster relay valve.

BENDIX HVSD RECOMMENDATIONS

An RE-6 relay emergency valve is recommended f the service relay valve.

NOTE: The use of spring brakes are not required t FMVSS '121' on dollies and most vehicle ope ators have chosen not to use them. If yc desire spring brakes on dollies for your oper tion, Bendix HVSD has system designs avail ble. Contact Bendix HVSD for further infc mation.

Figure 3.9: Air Brake System for a Two Axle Dolly [3.2].

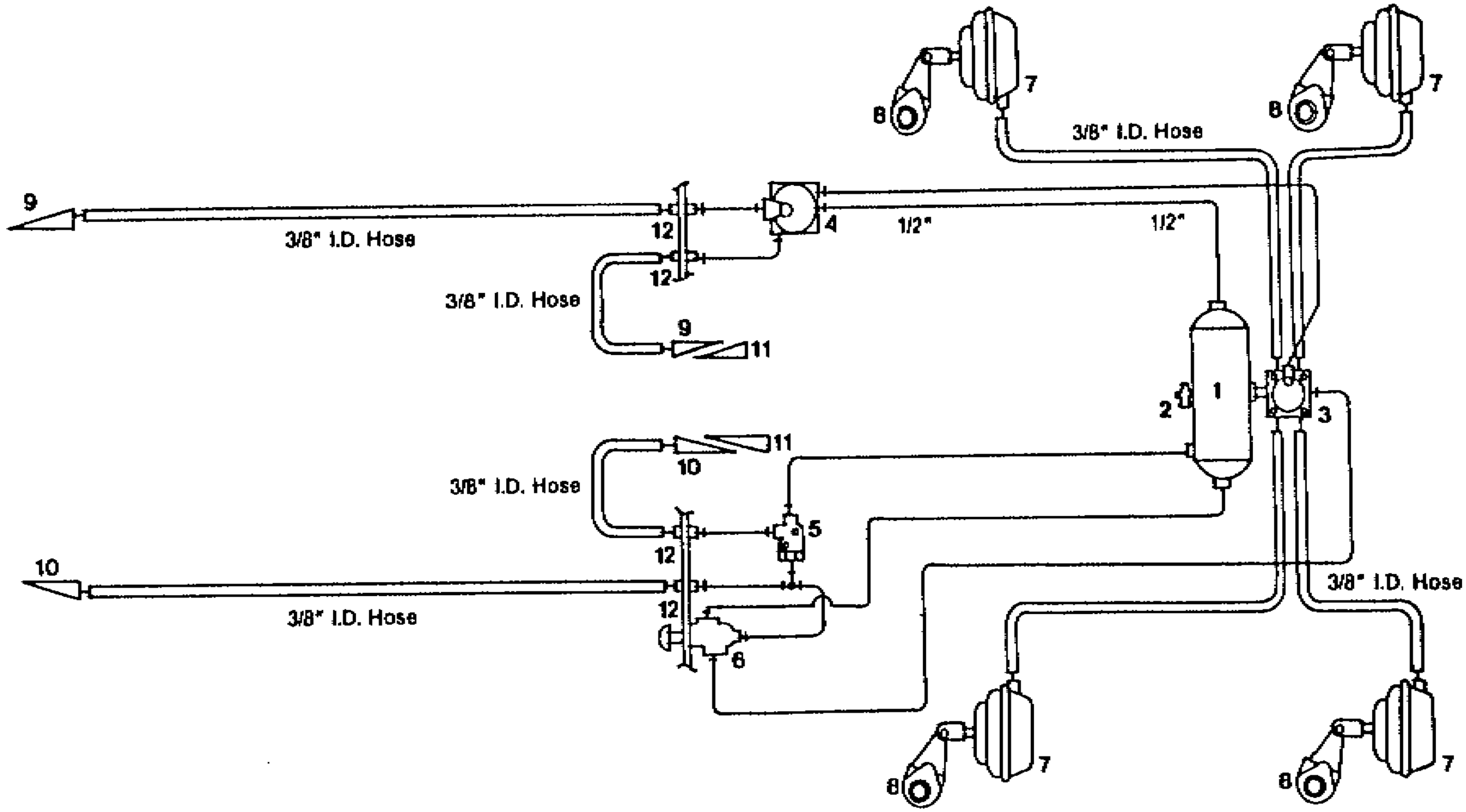


Figure 3.10a: Air Brake System for a Two or Four Wheel Dolly with Emergency Valve (Circuit Design) [3.2].

BENDIX DEVICES SPECIFIED

ITEM NO.	QTY.	BENDIX MODEL	BENDIX PIECE NO.	DESCRIPTION
1	1			RESERVOIR – SERVICE
2	1			DRAIN VALVE
3	1	RE-6	281865	RELAY EMERGENCY VALVE
4	1	R-8P	287114	PILOT RELAY VALVE
5	1	SV-1	283700	SYNCHRO VALVE
6	1	PP-1	282584	EMERGENCY BRAKE REL. VALVE (Optional)
7	4			BRAKE CHAMBER
8	4	ASA-2		SLACK ADJUSTER – AUTOMATIC
9	2	HC-2	278852	COUPLING – SERVICE
10	2	HC-2	278853	COUPLING – SUPPLY
11	2		212227	COUPLING – DUMMY
12	4			ANCHOR FITTING

SYSTEM NOTES/FEATURES

1. Maximum recommended air brake reservoir pressure is 125 p.s.i.

2. All tubing to be 3/8" O.D. (copper or synflex nylon spec. BW-250-M) unless otherwise specified. Inserts must be used with nylon tubing. Do not use tubing where temperature exceeds 180°F.

3. Air brake tubing must be arranged to eliminate close bends, elbows, reducer fittings unnecessary line lengths and etc. wherever possible.

4. Automatic slack adjusters (item 8) must be installed per Bendix instruction sheet BW-S-582.

5. Pipe nipples used to mount brake control valves must be heavy wall type.

6. The optional emergency brake release valve (item 6) must be mounted in a protected location away from wheel splash, mud, snow and ice accumulation.

7. This system provides service and emergency brake control only. If parking brakes are required, refer to T.A.B.S. system BWS-846

8. All air brake equipment must be Bendix.

9. For a single axle two wheel dolly delete two each of brake chamber item 7 and automatic slack adjuster item 8

10. This is a **piping schematic only** and is **not a construction drawing.** O.E.M. manufacturers may deviate from the physical arrangement of components for convenience

Figure 3.10b: Air Brake System for a Two or Four Wheel Dolly with Emergency Valve (Instructions) [3.2].

3.2.1 Air Compressor

Air compressor is the source of energy for the air brake system. The function of air compressor is to provide and maintain air under pressure to operate devices in the air brake system. It is driven by the vehicle engine, either by belt (5%) or drive gear (95%).

The compressor assembly is comprised of three cast iron major sub-assemblies; the cylinder head, the cylinder block and the crankcase. The cylinder head houses the discharge valving and is installed to the cylinder block. The cylinder block houses the cylinder bores and inlet valves and is installed to the crankcase. The crankcase houses the crankshaft and main bearings. The cylinder head and block are cooled by coolant routed to the compressor from the engine cooling system. Lubrication for the internal parts of the compressor is provided by the engine's pressurized oil system.

Operating Principle

The compressor is driven by the engine and operates continuously while the engine is running. Actual compression of air is controlled by the compressor unloading mechanism and the governor. The governor is generally mounted on the compressor and maintains the brake system air pressure to a preset maximum and minimum pressure level.

Air flowing through the engine compartment from the action on the engine's fan and the movement of the vehicle assists in cooling the crankcase. Coolant flowing from the engine's cooling system through connecting lines enters the compressor and flows through the internal passages in the cylinder block and head and is returned back to the engine. Proper cooling is important in maintaining discharge air temperatures below the maximum recommended 400° F. The cool air flow must be minimum 4 m/s as measured at cylinder Head. For water cooled compressors, the amount of water passing through the cylinder head must be at least 2 litres/min and its temperature should not exceed +80° C. measured at the air compressor cylinder head intake.

Requirements

There are four major requirements for the air compressor:

1. The build-up time to charge the supply system;

2. The charge time for the total system;

3. Recovery time from governor cut-in to cut-out; and

4. Air compressor duty cycle;

The build-up time according to FMVSS 121 requirement is to charge the supply system (twelve times the brake chamber volumes) from 85 to 100 psi in 25 seconds maximum.

The typical charge time for the total system from 0 to 120 psi for tractor semi-trailer is 4 minutes, 42 seconds. This is the time required to charge the system for the first time. For multiple trailers, the charge time increases approximately by 70% (8 minutes) for 45' doubles and 35% for 27' triples (6 minutes, 22 seconds). Use of larger air compressor, this charge time can be reduced by 50% (6 minutes 15 seconds/doubles, 4 minutes 55 seconds/triples).

The typical recovery time from governor cut-in to cut-out (typically 15 psi) when the vehicle is in actual service is 25 seconds (tractor semi-trailer). For multiple trailers, the recovery time increases approximately by 70% for 45' doubles (43 seconds) and 35% for 27' triples (34 seconds). Use of larger air compressor, the recovery time can be reduced by 25% (32 seconds/doubles, 26 seconds/triples).

Typical duty cycle or on/off time of air compressor is 5% loaded and 95% unloaded. New vehicles currently increase the duty cycles by 40% for triples (7% loaded) and 20% for doubles (6% loaded). An increase to 9% load cycle rate are within the capabilities of most air compressor designs and hence the trend for using compressor clutch viable [3.3].

Make and Size

There are six major manufacturers:

- Bendix, Borg-Warner, Midland, Cummins, Caterpillar, and Wabco.

There are four ranges of compressor sizes from Bendix:

a) 7 - 10 CFM, small

b) 12 - 13 CFM, medium

c) 14 - 17 CFM, large

d) 24 - 34 CFM, speciality.

Tables 3.1, and 3.2 show the specifications of Bendix and Wabco air compressors.

3.2.2. Air Compressor Governor

Description and Operating Principle

The Air Compressor Governor, operates in conjunction with air compressor unloading mechanism to control air pressure automatically in the air brake or air supply system between the desired, predetermined maximum and minimum pressures. The air compressor runs continually while the engine runs, but actual compression of air is controlled by the governor which stops or starts compression when the maximum or minimum reservoir pressures are reached. The governor has a piston upon which air pressure acts to overcome the pressure setting spring and control the inlet and exhaust valves to

Table 3.1 Air Compressor Specifications

Make: Bendix

	TYPE			
Parameter	BX - 2,150	TU -FLO 501	TU - FLO 700	TU - FLO 1,000
Average Weight (lbs)	33	42	46	79
Number of Cylinders	1	2	2	4
Bore x Stroke (in)	3.375 x 1.468	2.625 x 1.5	2.750 x 1.81	2.500 x 1.688
Displacement @ 1,250 rpm (CFM)	9.5	11.75	15.5	24
Maximum Recommended rpm	3,000	3,000	3,000	3,000 w. cooled 2,400 air cooled
Minimum Coolant Flow (water-cooled) Gpm (U.S.) at Maximum rpm at Minimum rpm	 2.5 0.5	 2.5 0.5	 2.5 0.5	 2.5 0.5
Minimum Coolant Flow/Pressure (air-cooled)	N/A	N/A	N/A	1,500 CFM @ 100 psig disch.
Approximate HP Required @ 1,250 rpm @ 120 psig (naturally aspirated)	1.7	2.3	3.5	4.6
Turbo Charge Limits Maximum rpm Maximum Press.(gauge) (psig) Maximum inlet temp. (°F)	 2,200 15 250	 2,200 25 250	 2,200 15 250	 2,200 25 250
Maximum Discharge Air Temperature (°F)	400	400	400	400
Minimum Pressure Required to Unload (nat. asp.) (psig)	60	60	60	60
LUBRICATION Minimum Oil Pressure Required (psig) Maximum Governed Engine Speed	ENGINE 5 15	ENGINE/SELF 5 15	ENGINE 15 15	ENGINE/SELF 15 15
Oil Capacity of Self-Lubricated Model	N/A	N/A	N/A	0.95 qt. 1.75 qt. #281118 5.0 qt. #288578
Minimum Line Sizes ID (in) Discharge Line Coolant Line Oil Sypply Line Oil Return Line Air Inlet Line Unloader Line	 1/2 3/8 3/16 1/2 5/8 3/16	 1/2 3/8 3/16 1/2 5/8 3/16	 1/2 3/8 3/16 1/2 5/8 3/16	 1/2 3/8 3/16 1/2 5/8 3/16

Table 3.2 Air Compressor Specifications

Make: Wabco

Parameter	TYPE 911004	411040	411041	411042	411043
Average Weight (Kg)	6.5	8.2 - 9.2	10 - 11	11.5 - 12.5	12 - 13
Number of Cylinders	1	1	1	1	1
Bore x Stroke (mm)	65 x 23 (76 cm^3)	75 x 24 (106 cm^3)	75 x 36 (159 cm^3)	90 x 36 (229 cm^3)	90 x 46 (293 cm^3)
Displacement @ 1,250 rpm litre/min	8 @ 16 bar 15 @ 14 bar 21 @ 12 bar	75 @ 6 bar	140 @ 6 bar	air cooled 200 @ 6 bar water cooled 210 @ 6 bar	air cooled 260 @ 6 bar water cooled 265 @ 6 bar
Maximum Recommended rpm	4,000	3,500	3,500	3,000	3,000
Maximum Air Pressure (bar)	-	10	10	10	10
Minimum Coolant Flow (water-cooled) at Maximum rpm at Minimum rpm	-	-	-	-	-
Minimum Coolant Flow/Pressure (air-cooled)	N/A	10 bar	10 bar	10 bar	10 bar
Approximate HP Required Kw/HP @ 2,000 rpm	Kw - bar 0.5 @ 14 0.77 @ 8 0.75 @ 4-10	HP - bar 1.5 @ 6	Kw - bar 1.7 @ 6	Air cooled HP - bar 3 @ 6 3.2 @ 6 water cooled	Air cooled Hp - bar 4.2 @ 6 4 @ 6 water cooled
Turbo Charge Limits Maximum rpm Maximum Press.(gauge) (psig) Maximum inlet temp. (°F)	-	-	-	-	-
Maximum Discharge Air Temp.	-	-	-	-	-
Minimum Pressure Required to Unload (nat. asp.) (psig)	-	-	-	-	-
LUBRICATION Minimum Oil Pressure Required (psig) Max. Governed Engine Speed	ENGINE -	ENGINE -	ENGINE -	ENGINE -	-
Oil Capacity of Self- Lubricated Model	-	-	-	-	-
Minimum Line Sizes ID Discharge Line Coolant Line Oil Supply Line Oil Return Line Air Inlet Line Unloader Line	22 mm 10 mm 13 mm	26 mm 22 mm 22 mm 26 mm	26 mm 22 mm 22 mm 26 mm	26 mm 22 mm 22 mm 22 mm 26 mm	26 mm 22 mm 22 mm 22 mm 26 mm

either admit or exhaust air to or from air compressor unloading mechanism.

Governors can be attached to the air compressor or mounted remotely. They are adaptable to either mounting. Figure 3.11 shows a typical governor with its reservoir, compressor unloading and exhaust ports.

Make

Bendix manufactures (model D-2) air compressor governor.

3.2.3 Conditioning Devices

For a tractor-semitrailer in service for 100,000 miles, the air can be exposed to approximately 30 quarts of water. There are number of alternatives in conditioning the air quality.

The following four alternatives are considered:

1. Wet tanks with manual or automatic drain valves as the minimum level of performance. They are inconsistent and susceptible to contamination.

2. The alcohol evaporator or injector can be installed to reduce the temperature that would cause freeze-ups. This does not remove the water.

3. After coolers to filter and condense water from the air alter the dew point temperature by no more than 10° F. Removes approximately 22.3 quarts of water.

4. Desiccant air dryers are the latest technology for eliminating water and oil from the air brake system. The desiccant material can reduce the air dew point by 30° F to 60° F, when incorporating pre-coolers and filters. This unit removes approximately 29.6 quarts of water.

3.2.3.1 Alcohol Evaporator

The alcohol evaporator serves as an anti-freezing device for air brake system. It introduces vaporized alcohol into the system through the air compressor induction system.

Make

- Bendix manufactures one model: AE-2.

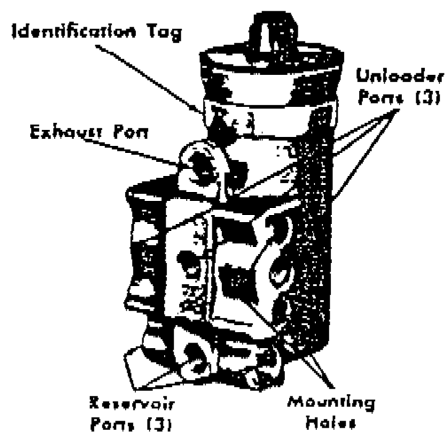


Figure 3.11: Air Governor

3.2.3.2 Air Dryer

Description

The air dryer collects and removes moisture and contaminants before air reaches the first supply reservoir. This provides "dry and clean air" for the braking and other air operated systems. The air dryer must be mounted vertically at the lowest level of the system. Usually it is mounted in a compartment ahead of the left rear wheel and connected in discharge line between cooling coil and reservoir.

Figure 3.12, shows sectional views of a typical air dryer. It consists of a safety valve mounted in the housing assembly which protects against excessive pressure build-up within the housing. A desiccant cartridge and pleated paper oil filter are placed inside the air dryer and are removable for servicing. It also has a heater and thermostat assembly to prevent freeze-up in the purge drain when the dryer is used in severe winter conditions.

Operation

The air dryer operates in two cycles:

- the charge cycle;
- the purge cycle.

Charge Cycle

When the compressor is in its "loaded" or compressing cycle, air from compressor enters the air dryer through the compressor discharge line. When the air, along with water and contaminants, enters the air dryer, the velocity of the air reduces substantially and much of the liquid drops to the bottom or sump of the air dryer. The initial air flow is toward the bottom of the dryer, but air flow direction changes 180^{o} at the bottom of the air dryer, dropping some water and oil. The air now passes through the oil filter which removes some oil and foreign material but does not remove water vapor. At this point, the air remains saturated with water. The filtered air and vapors penetrate the desiccant drying bed and the absorption process begins. Water vapor is removed from the air by the desiccant. The unsaturated "dry and clean air" passes through the ball check valve and purge orifice into the purge volume. From the purge volume, air flows past an inline purge volume reservoir and check valve located outside the air dryer, into the supply reservoir.

Purge Cycle

When desired system air pressure is reached, the governor cuts out, pressurizing the unloaded cavity of the compressor which unloads the compressor (non-compressing cycle). The line connected from the compressor

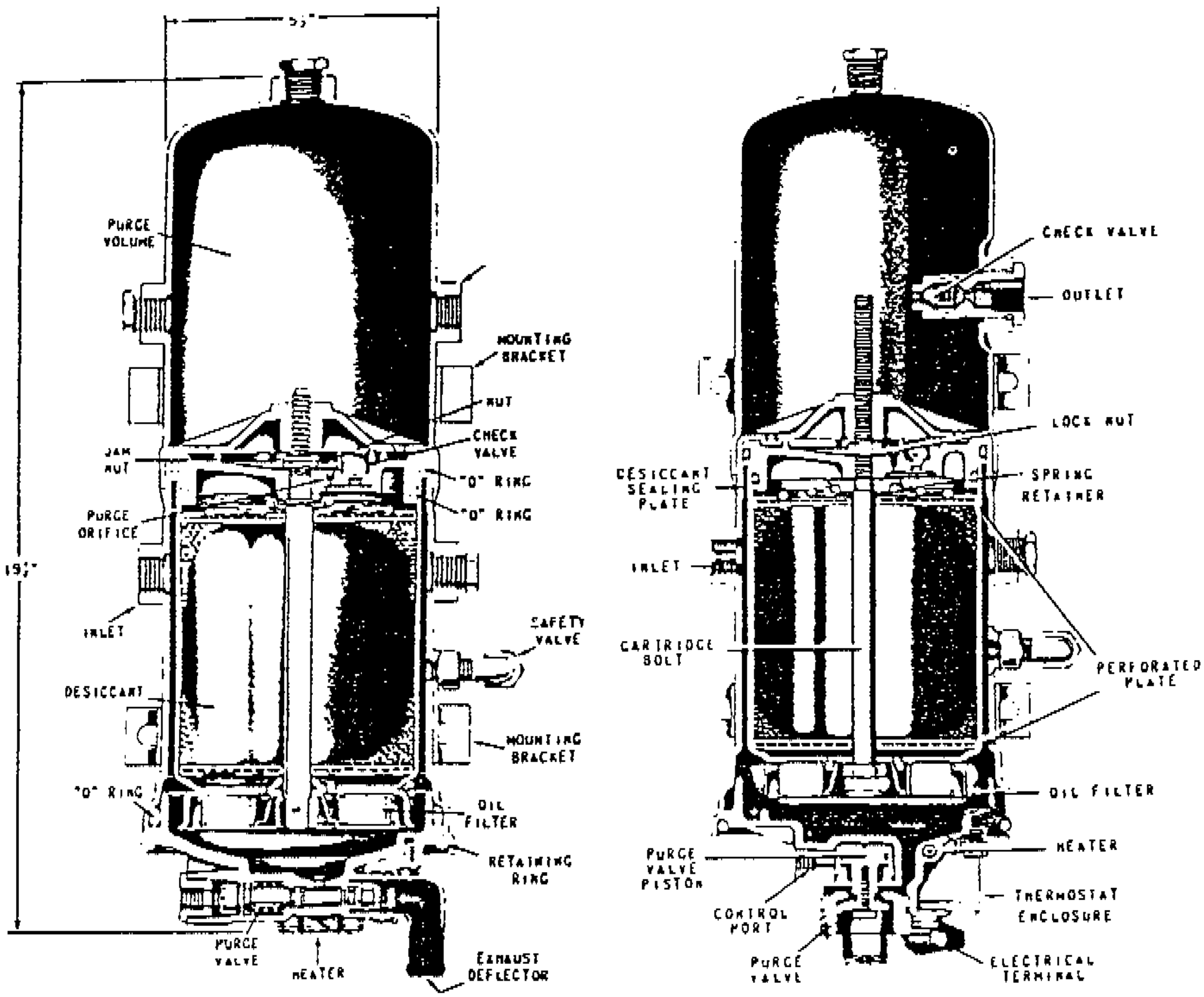


Figure 3.12: Air Dryer [3.2]

unloaded port to the end cover purge valve port (bottom of the air dryer) is also pressurized, moving the purge valve plunger, opening the exhaust of the purge valve to atmosphere. With the exhaust of the purge valve open, contaminants in the discharge line and dryer sump are purged, or forced past the open exhaust out to atmosphere.

The reserve air flow across the desiccant starts the removal process of moisture from the desiccant surface. Dry air flowing from the purge volume reservoir, dryer discharge line and dryer purge volume through the purge orifice and across the drying bed further dries the desiccant.

Installation

The air dryer must be mounted vertically at the lowest level of the system.

Make

There are four major manufacturers:

- Bendix, Anchorlok, Borg-Warner, and Brakemaster.

3.2.4 Reservoirs

The reservoirs serve the air brake system as storage tanks for compressed air. The size of the reservoir, (and the volume of compressed air) is sized by the vehicle manufacturer to provide an adequate volume of air for use by the braking system and auxiliary control devices. Generally, more than one reservoir is used in air-brake systems. A secondary function of reservoirs is too provide a location where the air, heated by compression, may be cooled and the water vapor condensed.

Reservoirs are available in various sizes in both single and double compartment design configurations.

3.2.5 Service Brake Chambers

Description and Operating Principle

Brake chambers are used to actuate the foundation brake mechanism in order to force the brake shoes against the brake drum.

Figure 3.13, shows a schematic diagram of the operation of the brake chamber. There is one independent chamber at each wheel hub. The chamber consists of a flexible diaphragm which separates the chamber into two parts. The pressurized air from brake air tanks enters on one side of the diaphragm by way of the relay valves, and pushes it against a push rod assembly. A return spring is placed on the other side of the diaphragm and is

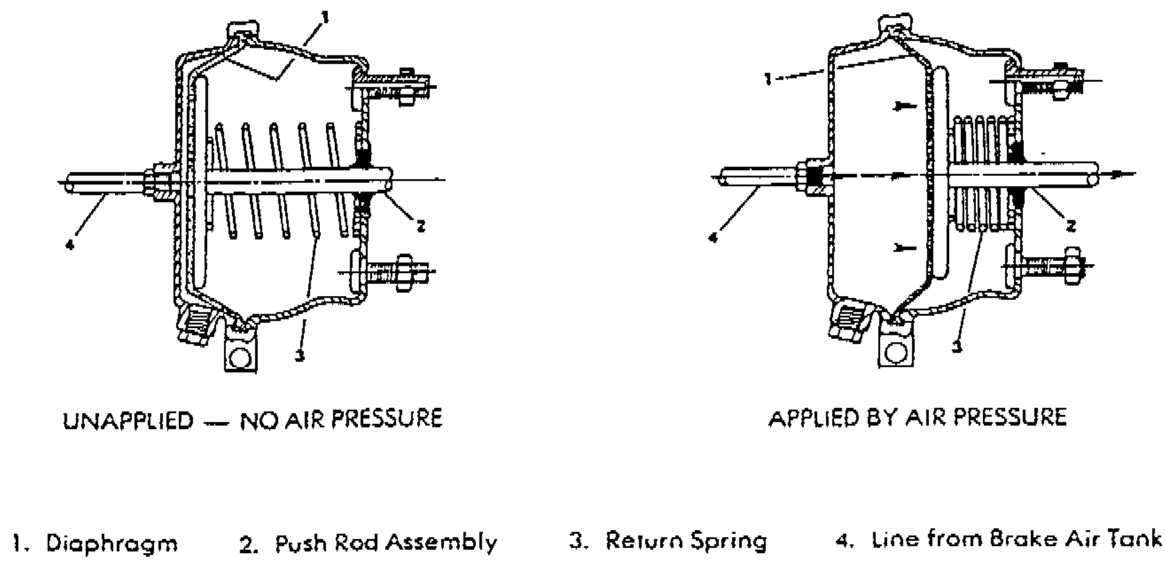


Figure 3.13: Service Brake Chamber

compressed during the actuation of air pressure in the chamber. When air is exhausted, the compressed spring returns to its original length, moving the push rod assembly back.

Make

There are four major manufacturers: Bendix, Aeroquip, Midland/Berg, and Wabco. Both Bendix and Wabco make brake chambers in several sizes, providing a wide range of output forces and strokes (refer to Table 3.3).

3.2.6 Spring Brakes/Brake Chambers with Spring Brakes

Spring brakes or brake chambers with spring brakes are widely used for both as service brakes and as parking brakes. Figure 3.14 shows a schematic diagram. It operates the vehicle service brakes (tractor rear axle brakes) by the energy stored in compressed coil springs. When the parking is not applied, the reservoir pressure is used to compress the coil springs and hold the brakes in the released position. Hence, during normal driving, air pressure cages the spring and holds it ready for parking or emergency braking. During normal service brake operation, the spring brake does not apply. Air pressure keeps the spring caged. For spring brake parking, application of the dash control valve exhausts air from the spring brake chamber, permitting the spring force to actuate the service brake for positive parking. For emergencies, the spring brake can be installed to operate automatically upon loss of air pressure.

Make

There are four major manufacturers:

- Bendix, Aeroquip (maxibrake), Anchorlok, and Berg.

3.2.7 Drum Brakes

In heavy vehicles, drum brakes are actuated via S-Cam and slack adjuster or wedge. Figure 3.15 shows a schematic of a S-Cam and wedge brakes. The following lists the various manufacturers of drum brakes and their specifications.

S-Cam Drum Brakes

1. Manufacturer: Eaton

Model number	Size	Application
EB - 150	15" x 4"	Steer Axle

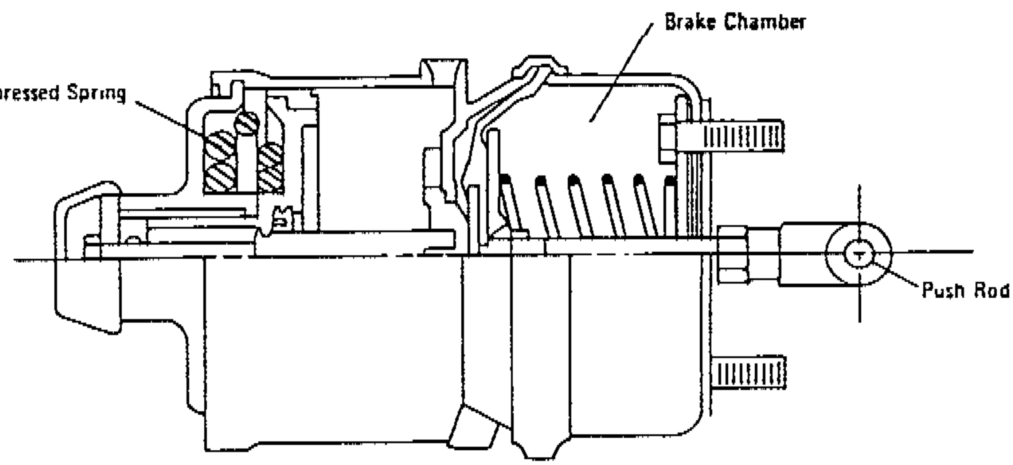


Figure 3.14: Spring Brakes/Brake Chamber [3.1].

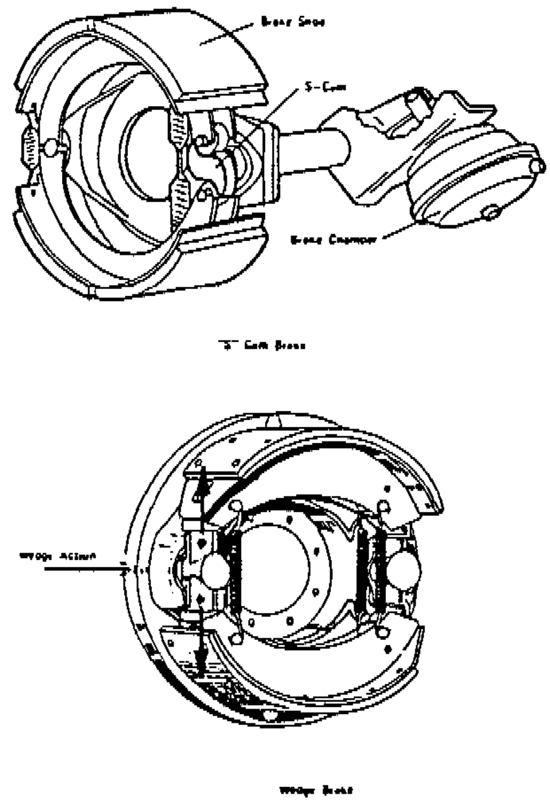


Figure 3.15: "S"-Cam and Wedge Brakes [3.1].

Table 3.3 Brake Chamber Specifications [3.2]

Dimension in Inches
Clamp Type Brake Chamber Data

Type	Effective Area (Sq. In.)	Outside Diameter	Maximum Stroke	Maximum Stroke at Which Brakes Should Be Readjusted
6	6	4 1/2	1 5/8	1 1/4
9	9	5 1/4	1 3/4	1 3/8
12	12	5 11/16	1 3/4	1 3/8
16	16	6 3/8	2 1/4	1 3/4
20	20	6 25/32	2 1/4	1 3/4
24	24	7 7/32	2 1/4	1 3/4
30	30	8 3/32	2 1/2	2
36	36	9	3	2 1/4

Chamber Stroke with Brakes Adjusted Should be as short as possible without brakes dragging.

S-Cam Drum Brakes

1. Manufacturer: Eaton (continued)

Model number	Size	Application
EB - 165	16 1/2" x 5"	Steer Axle
	16 1/2" x 6"	Steer Axle; Drive Axle
	16 1/2" x 7"	Drive Axle; Trailer Axle
	16 1/2" x 8 5/8"	Drive Axle
For Trucks EB - 180	18" x 7"	Heavy-Duty Drive Axle

2. Manufacturer: Rockwell

Model number	Size	Application
Q - Series	16 1/2" x 6"	Drive Axle
	16 1/2" x 7"	
	16 1/2" x 8 5/8"	
	16 1/2" x 10"	
Q - Series	12 1/4" x 7 1/2"	Tractor Axle
	16 1/2" x 7 "	
	16 1/2" x 8 5/8"	
	16 1/2" x 10"	
T - Series	15" x 3 1/2"	Front Steering Axles
	15" x 4"	

Note: "Q" Series is also applicable to extra-heavy duty front axles.

Wedge Drum Brakes

1. Manufacturer: Rockwell

Model number	Size	Application
RDA	15"	Rear axle rating 18,000 lbs; Front axle rating 12,000 lbs.
RDA	12"	-
RDA	15"	Front axles up to 12,000 lbs.

Wedge Drum Brakes

1. Manufacturer: Rockwell (continued)

Model	Designation
R	Stopmaster Brake; Wedge Actuated.
S	Single Actuated (Simplex) Leading/Trailing Shoe.
D	Double Actuated (Duo-Duplex). Two leading shoes directions.
A	Air operated.

Miscellaneous - Drum Brakes

1. Manufacturer: Kelsey-Hayes

   . "Gunite" full cast drums;
   . Hydraulic brake parts.

2. Manufacturer: Midland-Berg

   . Cams for S-Cam drum brakes.

3.2.8 Disc Brakes

There are two types of disc brakes depending on the type of actuation. They are:

- Mechanical - Air actuation
- Hydraulic - Hydraulic actuation

Mechanical - Air Actuation (Floating Type)

There are four manufactures supplying mechanical - air actuation disc brakes:

- Bendix, Kelsey-Hayes, Lucas-Girling, and Rockwell.

1. Manufacturer: Bendix

   Actuation is accomplished with standard S-Cam type air chamber actuation with automatic slack adjuster.

Type I: For front steering axles up to 13,200 lbs G.A.W.R.

Type II: For Rear/drive and trailer axles, up to 13,200 lbs G.A.W.R. single or 40,000 lbs tandem.

Note: Both 15.25" (390 mm) dia, vented rotor, and thickness: Type I - 1.535" (39 mm), Type II - 1.97" (50 mm).

2. Manufacturer: Kelsey-Hayes

   Air Chamber actuation, integral brake automatic slack adjuster.

   Model I: Front steering axles upto 15,000 lbs G.A.W.R.

   Model II: Rear axles upto 23,000 lbs G.A.W.R. single and 40,000 lbs G.W.A.R. tandem.

3. Manufacturer: Lucas-Girling (metric)

   Air Chamber actuation, integral brake adjustment (automatic adjustment), optional integral parking brake, and 36,000 lbs clamping force.

Hydraulic - Hydraulic Actuation (Direct: Floating Type)

1. Manufacturer: Lucas-Girling (metric)

   Model I: Double piston upto 19,704 lbs (87.65 kN) clamping force.

   Model II: Single and double piston upto 22,700 lbs (101 kN) clamping force.

   Note: Optional mechanical parking (spring brake).

2. Manufacturer: ATE (metric)

   Model: Double piston upto 20 kN.m braking moment

   Note: Optional mechanical parking (spring brake).

3. Manufacturer: Kelsey-Hayes

   Model: Front axles - 15,000 lbs to 20,000 lbs G.A.W.R.

3.2.9 Slack Adjuster

Description and Operating Principle

The slack adjuster is a mechanism which converts the rectilinear motion of the air chamber push rod into rotational motion at the S-cam in order to apply the brake shoes against the brake drum.

The adjustment mechanism is operated by a rack that is connected through a clearance notch in the control arm assembly. This control arm must be connected to a stationary or "fix point" member that is attached to the cam shaft housing. Correct installation of the fix point is extremely important for proper operation. There is a special spring-loaded cone clutch system that helps to prevent over adjustment.

The slack adjuster can be manual or automatic. Figure 3.16 shows a typical slack adjuster. Slack adjusters are designed for a maximum torque rating and are available in various arm configurations (straight and off-set), lengths and spline type.

Automatic slack adjusters perform the same function as the manual slack adjusters, except that it automatically adjusts for lining wear. The automatic adjustments may be based on either stroke-sensitive or force-sensitive mechanisms. The units also provide for manual adjustment.

The main features of different makes of slack adjusters: Rockwell, Kelsey-Hayes, Bendix, Wabco and Midland-Berg are presented in Tables 3.4 to 3.7.

3.2.10 Quick Release Valves

Description and Operating Principle

Quick release valve is used to rapidly exhaust air from the controlled device, rather than requiring the exhaust air to return and exhaust through the control valve. Normally it is located adjacent to the controlled device and thus decreases release time.

Air brake systems are equipped with one or more quick release valves to achieve a faster brake release. The valve exhausts brake line pressure at the point of installation, thus supplementing the exhaust at the brake application valve.

A quick release valve is shown in Figure 3.17. Air pressure from the brake application valve enters the quick release valve through the port above the diaphragm and forces the center of the diaphragm to seat tightly against the exhaust port. Air pressure also overcomes diaphragm tension to deflect the outer edges of the diaphragm and air flows through the side ports to the brakes. During release, the pressure above the diaphragm is released quickly and brake line pressure coming from the wheel brakes raises the center of the diaphragm from the exhaust port and permits direct air escape

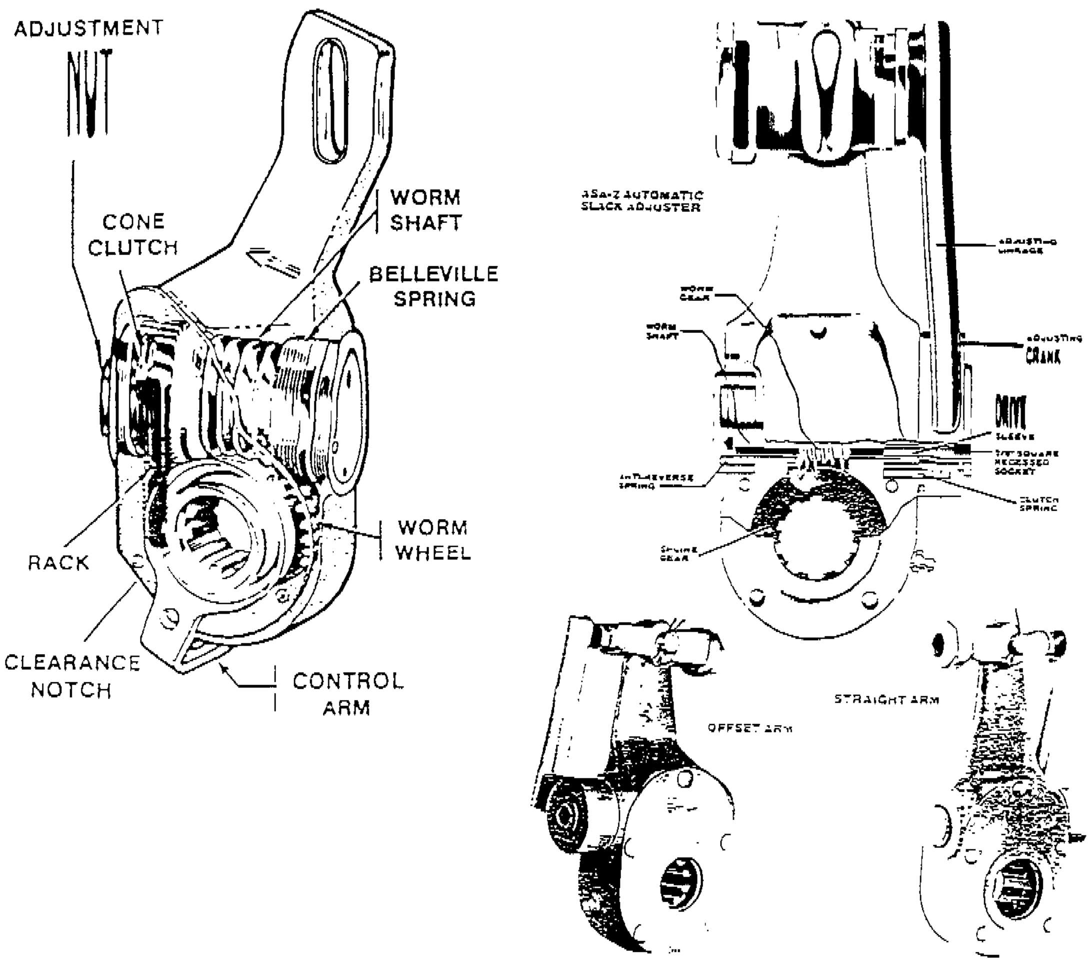


Figure 3.16: Slack Adjuster [3.2]

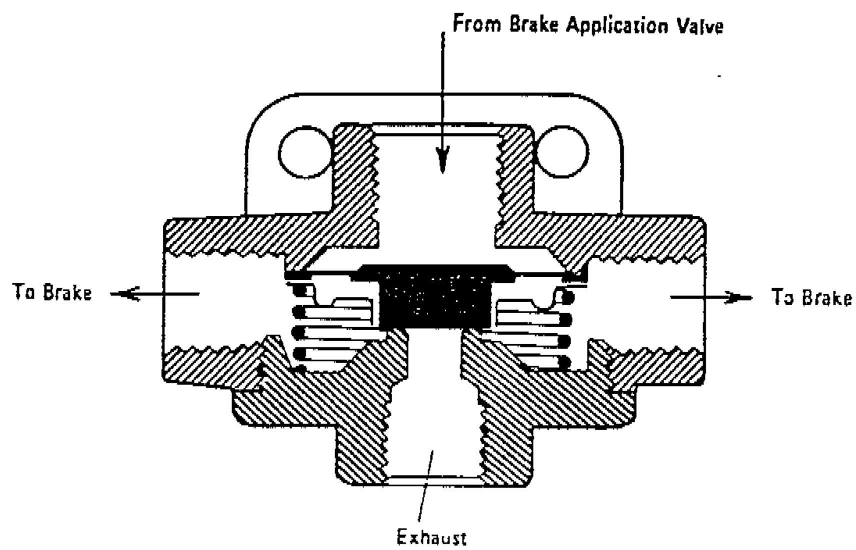


Figure 3.17: Ouick-Release Valve [3.2]

Table 3.4 Slack-Adjusters

Make	Model / Comments
Rockwell	Automatic Front steeling axle: 5", 5 1/2" arms lengths. Drive and trailer axle: 5", 5 1/2", 6", 6 1/2", 7" arms lengths, 30 000 lbs-in. torque capacity.
Kelsey-Hayes	Automatic 5 1/2", 6", 6 1/2", arm lengths.
Bendix	Automatic 5 1/2", 6", 6 1/2" arm lengths. Standard slack adjuster are also available.
Midland-Ross	Manual 4 1/2", 5", 5 1/2", 6", 6 1/2", 7", arm lengths, up to 30 000 lbs-in. torque capacity.
Wabco	Manual 100 mm to 365 mm arm lengths, up to 2 800 N-m torque capacity. Automatic 120 mm to 250 mm arm lengths, straight, off-set.

Table 3.5 Rockwell - Automatic Slack Adjusters

ch: Chamber

Type / Parameter	Steering Axles	Drive and Trailer Axles
Length (in.)	5, 5 1/2	5, 5 1/2, 6, 6 1/2, 7
Camshafts Spline Sizes and Number	1 1/2 - 10 1 1/2 - 28 1 1/4 - 10	1 1/2 - 10 1 1/2 - 28 1 5/8 - 37
Arm Configuration	Straight, .625" offset clevis	-
Clevis Thread	0.5-20 for ch. type 9, 12, 15 0.625 - 18 for ch. type 20, 24	0.625 - 18 for ch. type 20, 24, 30, 36
Maximum Torque Handled	30,000 in.-lbs	

Table 3.6 Kelsey-Hayes/Slack Adjusters

Parameter	Automatic	
Length (in.)	5 1/2, 6, 6 1/2	
Spline Type and Number	10 - 1 1/4",	37 - 1 1/2" 10 - 1 1/2" 28 - 1 1/2"
Arm Configuration	Straight Offset	
Weight (lbs)	N/A	

Table 3.7 Bendix - Slack Adjusters

Parameter	Automatic
Length (in.)	5 1/2, 6, 6 1/2
Camshafts Spline Site	Standard 10C Involute spline
Arm Configuration	Straight, 5/8" offset
Weight (lbs)	N/A

to the atmosphere.

Generally, quick release valves are designed to deliver within 1 psi of control pressure.

Table 3.8 shows the list on available hardware.

3.2.11 Relay Valves/Relay Quick-Release Valves

Description and Operating Principle

Relay valves are used to apply and release rear axle(s) service or parking brakes.

The valve helps speed brake application and release. The relay quick-release valve is connected into the line leading to the rear brakes as shown in Figure 3.18. Until the brake application cycle starts, the relay-inlet valve is closed and the exhaust valve is open to the atmosphere. When brakes are applied, the metered air pressure from the brake application valve forces the relay piston down, closing the exhaust port. Further movement of the piston opens the inlet valve, allowing air pressure from the auxiliary reservoir to enter the valve, pass through the delivery ports, and on to the brake application lines. As braking pressure underneath the piston equals controlling pressure above, the piston balances and allows the inlet valve return spring to close the inlet exhaust valve. When the brakes are held, the decrease in controlling pressure above the piston balances and allows the inlet valve return spring to close the inlet exhaust valve. When the brakes are released, the decrease in controlling pressure unbalanced the relay piston and permits the exhaust port to open, thus releasing braking pressure directly to the atmosphere.

Table 3.9 lists the available hardware.

3.2.12 Relay Emergency Valves

Relay emergency valves are commonly used on pre-FMVSS 121 trailers. It is a dual function valve. Under normal braking conditions, it serves as a relay valve, applying and releasing the service brakes. When the supply line pressure falls below a predetermined minimum, the emergency portion of the valve will automatically apply the vehicle brakes from its own protected reservoir.

Table 3.10 lists the available hardware.

3.2.13 Ratio Valves

The ratio valve is installed in the front axle delivery line. During normal service brake applications, the ratio valve automatically reduces application pressure to the front axle brakes. However, as brake application pressure is increased the percentage of reduction is decreased

Table 3.8 Quick Release Valves

Make	Model / Comments	
Bendix	QR-1	
	QR-1C	Dual function valve. It functions as a quick release valve for the emergency side of the spring brakes and its integral double-check valve prevents simultaneous application of the service and emergency side of the spring brakes, (anti-compounding).
Williams	WM-314	
Midland-Ross	N-12916	
Wagner	AC 15793	
Sealco	2000	
Velvac	32-C	
MGM	QR 318	
Berg	21602	
Aeroquip	KSV 4040737	

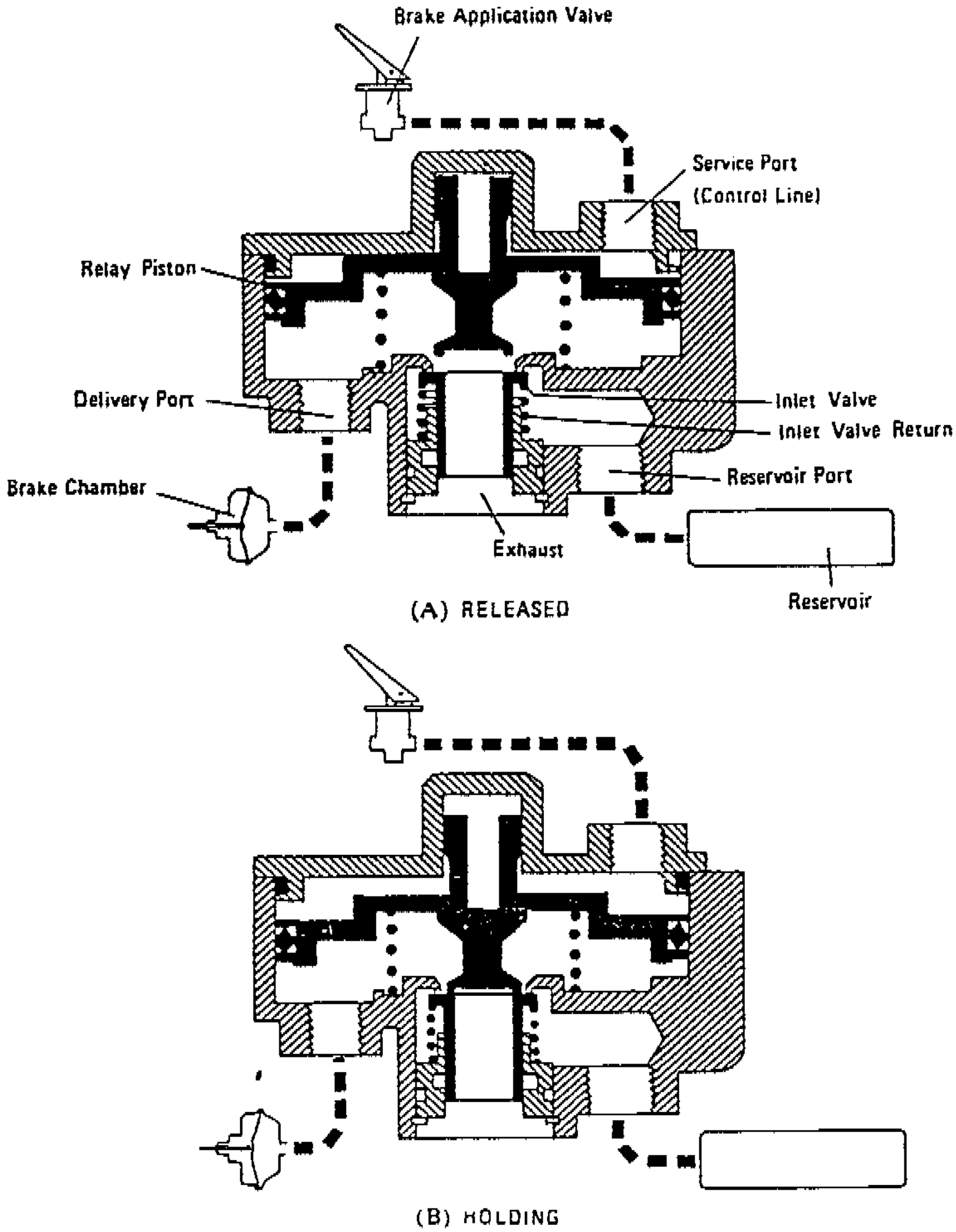


Figure 3.18: Relay Quick-Release Valve [3.1]

Table 3.9 Relay Valves/Relay Quick Release Valves

Make	Model	Comments
Bendix	R-6	
	R-8P	Designed specifically as a control line relay valve in multi-vehicle combinations. Provided balanced pressure to all trailer and dolly brakes. Does not cause a drop in pressure in the service brake signal. The R-8P will replace any dolly booster relay valve.
	R-8 and R-12	Incorporate an integral balance port which provides an anti-compounding feature to control spring actuated parking brakes/service brakes. Serves up to four Type 30 brake chambers.
Williams	WM-227-A	
	WM-227-B	
Midland-Ross	N-20829-C	
	N-2531	
Wagner	AF 37500	
Sealco	A-1100	
Berg	1660	
Velvac	32-B-31	
Aeroquip	KSV 4040846	

Table 3.10 Relay Emergency Valves

Make	Model / Comments	
Bendix	RE-4 RE-6	Both are piston operated; Available in both remote and resevoir mounting configurations.
Williams	WM-101-A	Modulated pressure
	WM-101-B	Modulated pressure
Midland-Ross	N-4171	
	N-3714	
Wagner	AF 37410	
Sealco	A-1000	
Berg	11675, 11678, 11679 11685, 11688, 11689	
Velvac	32-A-31	

until the full pilot pressure (approximately 60 psi) is delivered. The valve is available with several different hold-off pressure which prevents the front brakes from operating until the hold-off pressure is exceeded.

Table 3.11 lists the available hardware.

3.2.14 Brake Valves/Service Brake Valves

Brake valves are used for applying and releasing air brakes on the tractor and vehicles combination. They are available in various mounting configurations, floor, fire-wall mounted, etc. Actuation of the valve can be treadle, pedal and lever/linkage arrangements. The sensitivity of the valve varies depending on the method of actuation and the design. All brake valves are designed to provide a gradual means of applying air in the 5 psi to 80 psi range, with the capability of delivering full reservoir pressure.

Brake valves can be grouped into two types:

- A single circuit valve employed in pre-FMVSS 121 vehicles;

- A dual circuit valve employed in FMVSS 121 vehicles.

In the case of dual circuit, it uses two separate supply and delivery circuits for service and secondary braking. Primary circuit is mechanically operated through the action of the treadle/pedal and plunger. The Secondary circuit operates similar to relay valve, with control air delivered from the primary circuit. In the case of failure of the primary supply (emergency situation), the secondary inlet valve is mechanically opened by a push through mechanical force from the driver's foot via the treadle/pedal, plunger and primary piston.

Table 3.12 lists the available hardware.

3.2.15 Tractor Protection Valves

The tractor protection valve is used to protect the tractor air brake system in the case of either trailer breakaway or where severe air leakage develops in the tractor or trailer. This valve is also used in everyday use, to shut off the trailer service and supply lines before disconnecting the tractor from the trailer. The valve is usually mounted at the rear of the tractor cab and operates in conjunction with a dash mounted control valve.

Table 3.13 lists the available hardware

3.2.16 Spring Brake Valves/Spring Brake Control Valves

Spring brake valves are used on tractor and trailers. They provide emergency braking control on tractor rear axle and control both the parking and the emergency brake functions in the case of trailer.

Table 3.11 Ratio Valves

Make	Model	Comments
Bendix	LQ-4	
	LQ-2	Pre - FMVSS 121 vehicle; limiting quick release valve.
Williams	WM-318 B	Preset
	WM-318	Adjustable
	WM-318 A	Adjustable
Aeroquip	KSV 4040738	

Table 3.12 Brake Valves/Service Brake Valves

Make	Model / Comments
Bendix	Single circuit: E-2, E-3 — Floor mounted, treadle or lever-operated; E-3 has longer plunger travel and softer. E-5 — Similar to E-3, but has a suspended pedal; fire-wall mounted valve within the cab. Dual circuit: E-6, E-10 — Floor mounted; treadle operated. E-7 — Fire-wall mounted; suspended pedal.
Williams	WM-472
Midland-Ross	TCV-3
Wagner	AE 48611
Sealco	6100

Table 3.13 Tractor Protection Valves

Make	Model	Comments
Bendix	TP-2	Used on pre-FMVSS 121 tractors; Requires dash mounted control valves such as TW-1 or PP-1; Houses both the service and emergency valves; Three lines-control line, trailer service line, and trailer supply line, are connected.
	TP-3	Used on pre-FMVSS 121 and 121 tractors; Houses only service line shut-off valve; Controlled by pressure delivered by the control valve in the cab; Uses PP-3/PP-7 control valves; Two lines-trailer supply line, and trailer service line are connected; Will function automatically between 20 and 45 psi as supply pressure is reduced.
	TP-4	Proprietary valve
	TP-5	
Williams	WM-67-A	
	WM-57	Standard
	WM-320	Fast brake application
Midland-Ross	TPV-2	
	N 30073	
Wagner	AC-24912	
Sealco	3700	
	7700	
	9100	
Berg	1501	

In tractor, it is used in dual circuit brake systems and serves two functions; during normal operation, it limits hold-off pressure to the spring brakes via a relay valve or quick release valve, generally 90 or 95 psi. When there is a loss in the rear service brake supply, it will modulate spring brake application at the rear axle proportional to service braking pressure delivered to the front axle

Table 3.14 lists the available hardware.

3.2.17 Trailer Control Valves

The trailer control valves are used for independent control of trailer service brakes. However, the valve can be used for any application where graduated application pressure between approximately 5 psi and end of graduation range is required. They should not be used for parking. The valve employs a cam, cam follower and a graduation spring to control air delivery pressure.

Table 3.15 lists the available hardware.

3.2.18 Other Control Valves

There are several control valves used in the air brake system to control various system components. They are generally dash mounted in the cab.

Table 3.16 lists the available hardware.

3.2.19 Check Valves

Both in-line single check valves and double check valves are used in air brake system. The following manufacturers supply this unit:

- Bendix
- Williams
- Bosch
- Wabco
- Aeroquip
- Midland-Ross
- Wagner
- Sealco.

3.2.20 Drain Valves

Drain valve is installed in air-brake reservoirs for draining the accumulated contaminants collected in the reservoir. Available in both manual and automatic types. The following manufacturers supply this unit:

- Bendix
- Wabco

Table 3.14 Spring Brake Valves/Spring Brake Control Valves

Make	Model / Comments	
Bendix	SR-1	For tractors
	SR-2	Three-reservoir trailer air brake system
	SR-4	Two-reservoir trailer air brake system

Table 3.15 Trailer Control Valves

Make	Model / Comments	
Bendix	TC-2	
	TC-6	
	TC-7	
Williams	WM-606-A1	0-60 psi range
	WM-606-B1	0-90 psi range
	WM-606-C1	0-120 psi range
Midland-Ross	HCV-3	
	N 30060	
Wagner	Type HE	
	AE 25340	
Sealco	5900	
Berg	1163100	

Table 3.16 Control Valves

Make	Model / Comments	
Bendix	PP-1	Used in pre-FMVSS 121 - Single circuit tractor air system; Controls parking and emergency brakes; Works in conjunction with TP-2 tractor protection valve.
	PP-2	Used for parking and emergency brakes; Features an anti-compounding port to prevent simultaneous application of both service and parking brakes.
	PP-3	Used to control the TP-3; Tractor protection valve in pre-FMVSS 121 tractor air system.
	PP-5	Used in conjunction with vehicle torque converter systems, engine speed control systems, and some parking brake systems.
	PP-7	Used in FMVSS 121 dual air circuit system to control tractor protection system; Employs an air operated override control in the lower body which will apply the trailer brakes when the tractor spring brakes are applied.
	PP-8	Used to control the tractor brakes only in the FMVSS 121 dual system; Non-automatic; Remains in the applied (button in) position regardless of delivery or supply pressure.
Williams	WM-672	
Midland-Ross	N-14488; N-20928-A	
Wagner	AD-25077; AD-36130	
Sealco	17600; 9960	
Berg	1550; 1505; 1705; 11558; 11559; 11755-A; 215590	
Western Safety	6000	
MGM	2700	

- Bosch.

Automatic Drain Valve

Description and Operating Principle

The Automatic Reservoir Drain Valve ejects moisture and contaminants from the reservoir to which it is connected.

When there is no air pressure in the system, the inlet and exhaust valves are closed. Upon charging the system, a slight pressure opens the inlet valve, which permits air and contaminants to collect in the sump. The inlet valve remains open until maximum (governor cut-out) pressure is reached. The spring action of the valve guide in the sump cavity closes the inlet valve. The inlet valve and the exhaust valve are now closed. When the reservoir pressure drops slightly (approximately 2 psi), air pressure in the sump cavity opens the exhaust valve, and allows moisture and contaminants to be ejected from the sump cavity until pressure in the sump cavity drops sufficiently to close the exhaust valve.

The length of time the exhaust valve remains open and the amount of moisture and contaminants ejected depends upon the sump pressure and the reservoir drop that occurs each time air is used from the system.

3.2.21 Safety Valves

The safety valve protects the air brake system against excessive air pressure build-up. Installed in the same reservoir where the compressor discharge line is connected. Normally set for the pressure range 140 to 160 psi. Available in both adjustable and non-adjustable types.

Table 3.17 lists the available hardware.

3.3 NEW DEVELOPMENTS TO REPLACE SPRING BRAKES [3.4]

There is a new type of brake chamber called "Mini-Max", a patented brake system manufactured by International Transquip Industries Inc. in Houston, Texas, and now distributed in Canada by the Truckline Parts Division of Hayes Dana Inc. which replaces the conventional spring brakes.

The "Mini-Max" is essentially a basic service brake chamber with a bit of extra valving and ratchet detent. In normal use, it functions about the same as a service brake.

Figure 3.19 shows a pictorial view of a Type 30 "Mini-Max" brake and a cross-sectional view illustrating the basic operation.

The "Mini-Max" brake functions as service, emergency and parking brake. The chamber is bolted on to the mounting bracket and connected to the

Table 3.17 Safety Valves

Make	Model / Comments	
Bendix	ST-1	(Adjustable
	ST-3	(Non-adjustable)
	1/4" or 3/8"	N.P.T.
Williams	WM-342-B WM-342-A	

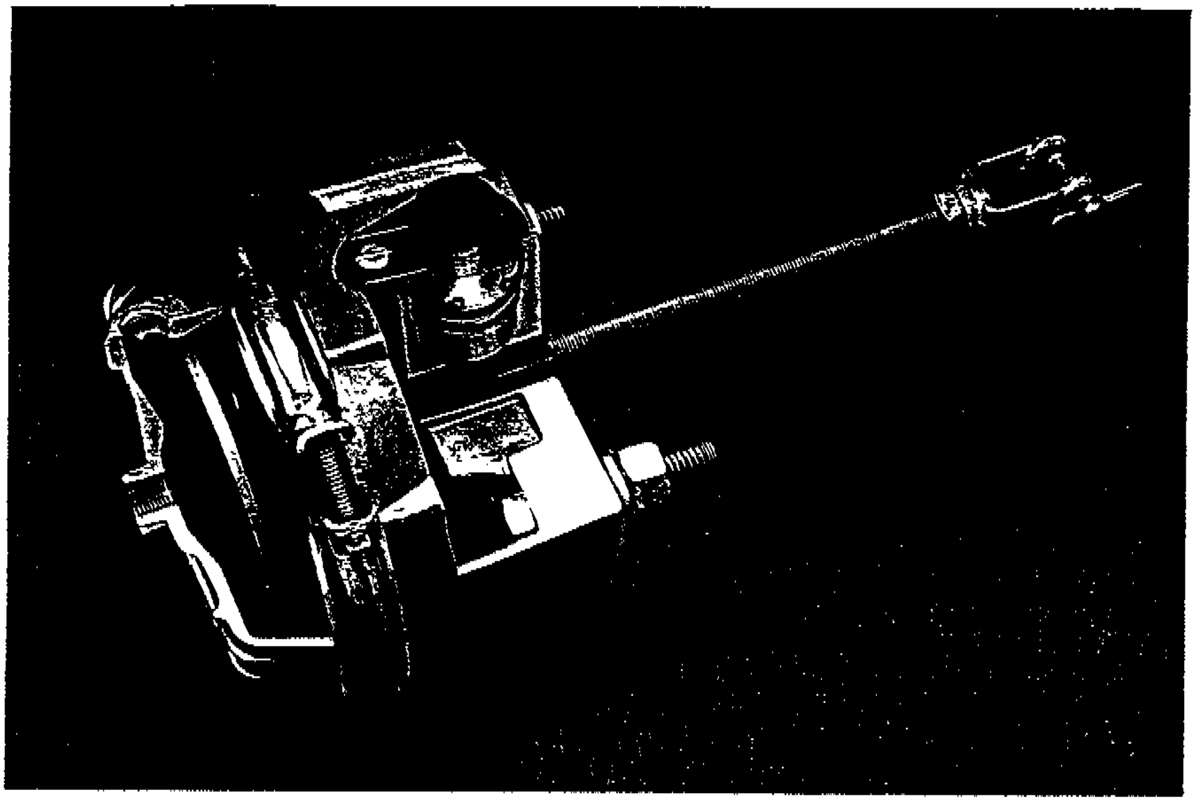


Figure 3.19a: "Mini-Max" Brake [3.4].

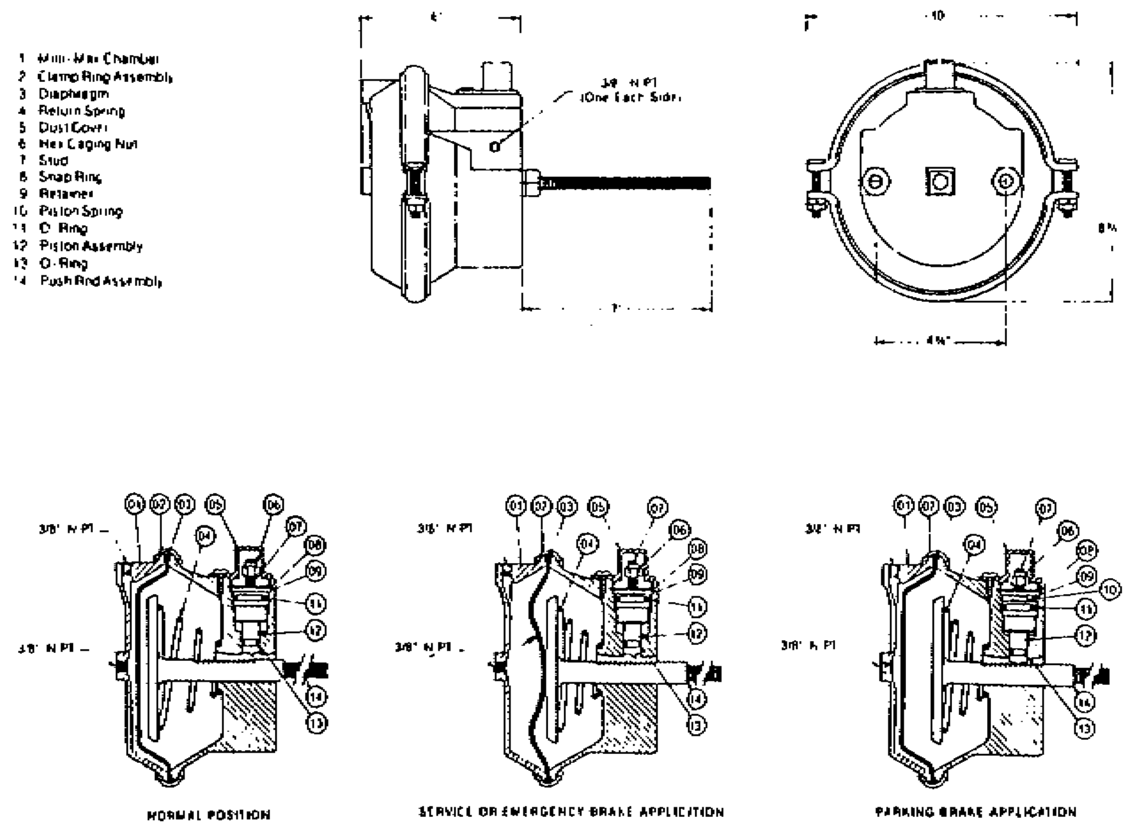


Figure 3.19b: "Mini-Max" Brake [3.4].

slack adjuster in the conventional manner. Supply air is piped to the relay emergency valve and to the piston, keeping it in the raised position during normal vehicle operation.

The service brake operates in the conventional manner. As fair is used for brake applications, it is automatically replaced from the supply line into the tank.

A conventional relay emergency valve is incorporated in the system. With this system there is either a full brake application or over 45 psi tank pressure available for application. Trailer brakes are automatically applied when dash valve is pulled, when pressure falls below 45 psi, or when trailer is disconnected because the supply line is interrupted. Truck tractor brakes are applied when dash valve is pulled or when air pressure falls below 45 psi. When the system is again pressurized to a safe operating level the brakes can be released safely.

The mechanical feature of the "Mini-Max" brake only engages when there is a drastic loss of air in the emergency tank. This feature eliminates unnecessary lock-ups and brake drag. The piston never ratchets along the push rod rack so no wear is ever experienced on these parts. Back force required to shear these parts is over 5,000 lbs compared to 1,200 lbs required to compress the spring on a conventional brake. Brake holding is improved.

Some of the salient features claimed by the manufacturer are:

- Safe and simple maintenance.
- No brake drag.
- Unwanted lockups are eliminated.
- Delayed mechanical parking ensures no compounding.
- Easily released in extreme cold.
- Service applies under 2 psi to allow safe operation.
- Brake life similar to service brake chamber - over 10 years.
- Weight only 12 lbs.
- Brake application force:
  - 3,000 psi @ 100 lbs air pressure.
  - 1,800 psi @ 60 lbs air pressure.
- Caging takes 2 minutes, safe and only cages parking brake, not emergency brake.
- Meets FMVSS 121 regulations.

3.4 CONCLUSIONS

In this chapter, a detailed account of brake system hardware and control systems is presented. Description of an air brake system for a tractor semitrailer along with air brake circuitry is also presented. A detailed description of individual hardware, availability, and specifications is outlined.

REFERENCES

3.1 "Engineering Design Handbook - Analysis and Design of Automotive Brake Systems", DARCOM - P 706 - 358, 1976.

3.2 "Bendix Air Brake Handbook", - BW 5057R 9/83 15M.

3.3 "Bendix Technical Air Brake Seminar" - Notes.

3.4 "Mini-Max" Brake System Brochure - International Transquip Inc.

CHAPTER 4

HEAVY VEHICLE AUXILIARY BRAKING DEVICES

4.1 INTRODUCTION

The performance of the braking system of road vehicles, and heavy vehicles in particular, is becoming more and more an important parameter in the design of safe vehicles. The friction brake, although extensively developed and refined, does not fully satisfy some of the requirements imposed by the modern traffic conditions, especially during steep mountain descents.

During braking, force is generated at the wheels to oppose the motion of the vehicle and to dissipate the associated kinetic or potential energy as heat.

Conventional friction brakes can stop a vehicle fully loaded with high decelerations generally independent of the speed. Under these conditions, the torques generated by the brakes are high and the rates of energy absorption are extremely high for short periods. For example, a vehicle braking from 110 kph at a deceleration of 0.79g. would dissipate 105 kW per tonne [4.1]. This is possible because of the heat reservoir effect of the brake drums and adjacent components. However, these high rates of energy dissipation occur only seldom and for short periods.

As mentioned earlier, the mountainous terrain imposes certain specific conditions and therefore of equal concern is the necessity of absorbing energy at lower rates over longer periods of time. Under these conditions, the brake temperature rises until a temperature gradient sufficient to achieve equilibrium between heat input and heat dissipation to the atmosphere and the adjacent components is reached. At this point, the heat capacity of the brake will influence the time required to attain equilibrium but not significantly its final value. Although, today much progress has been made in lining materials capable of operating of high temperatures, problems of brake fade and high lining wear are still present.

One means of overcoming these problems is to fit a retarder which opposes the vehicle motion at relatively low levels of power dissipation for long periods.

Retarders can be divided into two major categories, either "driveline" or "engine speed" retarders. Driveline retarders, (hydrokinetic or eddy-current type) applies torque to a rotative element connected to the wheel without a transmission in between. An engine speed retarder acts in the engine which will produce the retarding force. The engine speed retarder will produce braking force at the wheels only when the transmission is in gear.

4.2 ENGINE RETARDERS

Engine retarders can be classified in two main classes, i.e. engine brakes and exhaust brakes.

4.2.1 Engine Brakes ("Jake" Brakes)

4.2.1.1 Operating Principle

The engine brake, like the Jacobs engine brake ("Jake" brake), operates on the principle of converting the diesel engine into a power-absorbing air compressor. This is done by opening the exhaust valves near top dead centre on the compression stroke, releasing the compressed air to exhaust. Since the energy of the compressed air is not returned to the piston on the expansion stroke, there is a net energy loss and the work done in compression is not returned during the expansion process.

As shown in Figure 4.1, on engines using a third cam on the main camshaft to provide the necessary motion for the fuel injection, the master piston in the engine brake housing takes the motion of the injector cam via the injector push rod and transfers it to the exhaust valve hydraulically. The master piston A takes the motion of the injector push rod B. The slave (engine brake) piston C transfers this motion to the exhaust valves. This is accomplished when the electric solenoid-operated valve D is actuated by the driver. Oil from the control valve F lifts a return spring and opens the passage to the top of the master piston A and forces the master piston into contact with the injector push rod B. The ball check valve in the control valve traps the oil in the master-slave piston circuit thus forcing the oil under pressure on to the engine brake piston which in turn opens the exhaust valves G. Therefore as long as the main solenoid valve is actuated, there is a hydraulic link between the two operating pistons. When the solenoid valve is switched off, oil pressure is released from below the control valve which then exhausts the oil trapped in the master/slave piston circuit allowing the master piston to return, under the action of its spring, to the rest position without contact with the injector push rod.

On some engines, having a separate injection pump to operate the engine brake, an alternative source of operating the engine exhaust cycle is needed. For example, on a six cylinder engine three cylinders are taken together as a unit and the appropriate exhaust cam and push rod are used to provide the brake engine timing. Table 4.1 shows the arrangement for a Mack 673-engine [4.2].

In recent years, the engine brake was further refined. An example is the Model 404 Jacobs engine brake for Cummins 4-cycle L10 Diesel engines [4.3]. This new retarder was needed to compensate for loss of natural retarding capabilities in modern trucks due to fuel-saving devices such as radial tires, fan clutches, aerodynamic improvements, reduced friction in the truck components, etc. This new version of engine brake is presented in Figure 4.2 which illustrates the interactions of master and slave pistons as well as single-exhaust valve use. The mode of operation is similar to the older model already described. The difference is that the new system uses a

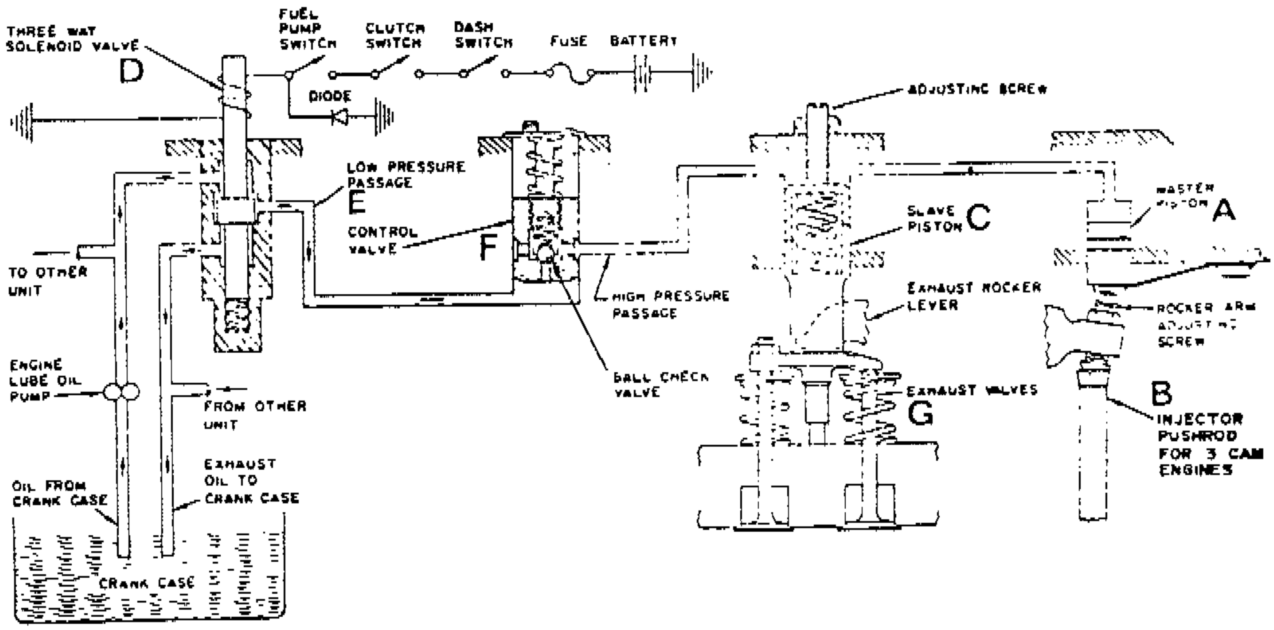


Figure 4.1: Schematic Diagram of Jacobs Engine Brakes (old version) [4.2].

Table 4.1 Master/Slave Cylinder Relationship in Engine Firing Order [4.2]

ACTUATES	
LOCATION OF MASTER PISTON	LOCATION OF SLAVE PISTON
No. 1 Pushrod	No. 3 Exhaust valve
No. 5 Pushrod	No. 6 Exhaust valve
No. 3 Pushrod	No. 2 Exhaust valve
No. 6 Pushrod	No. 4 Exhaust valve
No. 2 Pushrod	No. 1 Exhaust valve
No. 4 Pushrod	No. 5 Exhaust valve

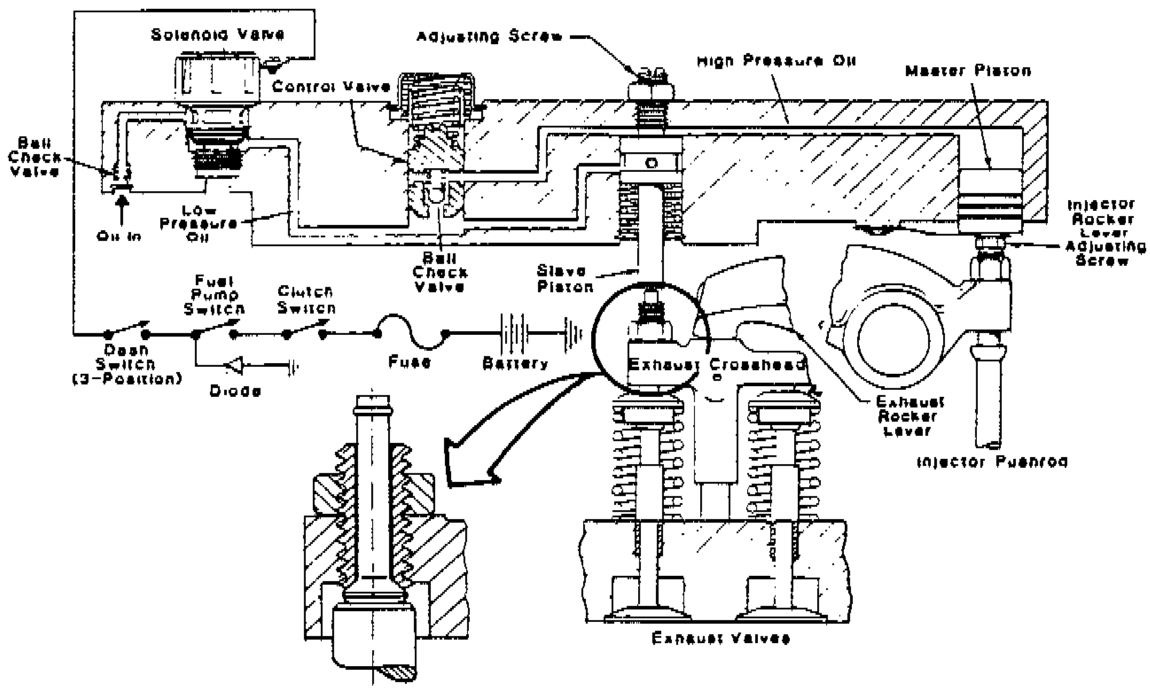


Figure 4.2: Schematic Diagram of Model 404 Jacobs Engine
(New Version) [4.3].

modified injector rocker arm adjusting screw to transfer motion from injector push rod to brake master piston. The L10 engine has four valves per cylinder but the Model 404 Jake brake uses only one of the two exhaust valves. This is shown in Figure 4.3 compared to the older system using both exhaust valves. Jacobs brakes for engines with only two-lobe camshafts (intake and exhaust) use exhaust pushrods from adjacent cylinders to drive the brake master piston.

Among the design considerations for the new brake was the limitation in engine height due to the necessity of the brake housing to be mounted on top of the overhead assembly under the valve cover. This calls for a compact design and also a good distribution of the housing mass in order to obtain an evenly stressed structure.

4.2.1.2 Further Design Considerations

An important aspect of the design of an engine brake are the controls available to the driver. Since the brake must come on and off quickly and without interference with the normal driving habits and the controls must be straightforward simple and foolproof, fast operating electric solenoid valves are used to control the hydraulic circuits with simple automatically actuated electrical switches for the throttle and clutch. A simple dash switch ensures that when the dash switch is "on" and the driver releases the throttle and the brake comes into action simultaneously. As soon as the driver touches the clutch pedal to change gear, the engine brake automatically goes out of action and the engine runs normally.

For the new Jacobs Model 404 engine brake having a single braking valve, requires a hydraulic reset mechanism in the brake system so that normal valve motion is not impaired by the engine brake. The valve motions during one cycle with and without the reset mechanism are shown in Figure 4.4.

The major part of the reset mechanism is mounted in the slave piston adjusting screw. To accommodate this, the slave piston and control valve assembly were slightly modified. The reset mechanism acts in conjunction with the slave piston to obtain the desired valve motion, as shown in Figure 4.5. The reset valve plunger follows the slave piston in its downward motion until its main spring overcomes the hydraulic pressure holding the reset plunger against the slave piston. The plunger then retracts back into the reset mechanism's body and uncovers the hole in the top of the slave piston. This allows the oil from the high pressure circuit to flow out through this hole into the low pressure supply oil. This allows the exhaust valve to return to its seat and resume normal operation.

Because only one valve is used, this valve must lift twice as mush as the valve lift for two valve brake system, in order to provide sufficient exhaust valve flow area, as compared to other engine brake models where two exhaust valves per cylinder are used for braking. Full valve opening occurs near top dead center on the compression stroke. Sufficient clearance was provided by the manufacturer to avoid any interference between the valve and the piston. The brake master/slave piston ratio was selected so as to assure

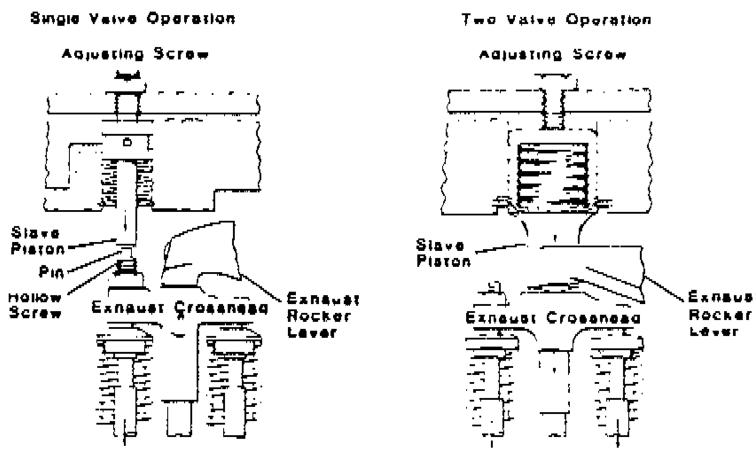


Figure 4.3: Schematic Diagrams of Model 404's Single-Valve Design and Prior Two-Valve Models [4.3].

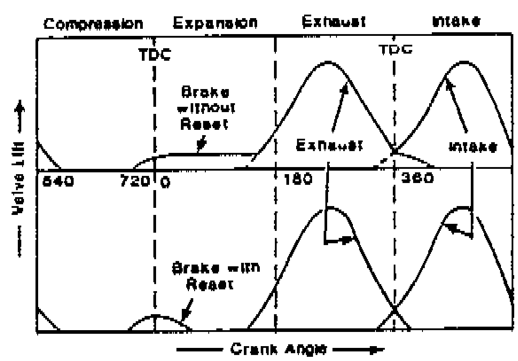


Figure 4.4: Valve Motions with and Without Reset Mechanism for Cummins Four-Cycle L 10 Engine [4.3].

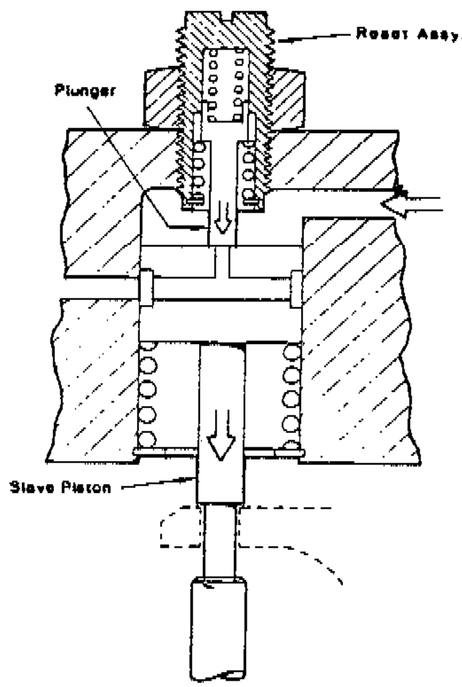


Figure 4.5: Schematic of the Braking Valve Reset Mechanism [4.3].

a safe clearance at maximum brake valve opening.

The Model 404 brake unit has two housing assemblies, attaching hardware and controls. In Figure 4.6, the front housing is presented. It is symmetric about the engine's mid-plane with the rear housing.

Since the engine overhead oil pressure and flow are quite low, oil is supplied externally from the oil filter head to the engine brake, in order to meet the brake activation time and make-up oil flow requirements. This supply oil is routed through a valve cover spacer built into the front housing. There it is distributed to the solenoid valves through a connector and passages incorporated in the front and rear housing. Each housing is controlled by its solenoid valve through a three-position switch, allowing the activation of one or both brake housings for partial or full retarding power. The single exhaust valve in each cylinder is actuated through a unique crosshead adjusting screw assembly.

For the Model 404 brake, the existing engine exhaust crosshead is used. The crosshead adjusting screw is replaced by a special hollow screw containing a pin which slides freely in the vertical direction. When the engine brake is activated, the slave piston pushes down on this pin, opening the exhaust valve.

To compensate for component thermal variation, a clearance between the slave piston and the pin upper portion (slave piston lash) is set during housing installation.

4.2.1.3 Engine Performance

In general, engine brake performance is mainly a function of either cylinder displacement or the intake air consumed. By turbocharging, a small engine can become as efficient as a large naturally aspirated engine. Two-stroke engines performed as well as equally rated 4-cycle engines, since they breath more air than the actual piston displacement requires.

The absorbed power increases if the engine is turbocharged due to the fact that with the engine brake operating, enough exhaust energy is available to drive the turbine and compressor to a high enough speed to produce a significant increase in intake manifold boost. As an example, an engine with an output of 220 HP equipped with a Jacobs engine brake model 20 will absorb 165 HP at 2100 rpm. The same engine, turbocharged, with an output of 250 HP when equipped with the engine brake, will absorb 225 HP [4.2]. The increase in input horsepower is due to an increase in air density due to the increase manifold pressure produced by the turbocharger. Exhaust temperatures are lower with the engine brake and result in lower exhaust energy. This brings lower terminal rotational speeds of the turbocharger than that obtained during full load engine operation.

In a turbocharged engine using an engine brake, the turbine accelerates faster when the engine brake is applied than it does with the sudden application of full fuel load at high engine speeds. At high altitudes, turbine speed decreases instead of increasing as is the case with a normally aspirated engine.

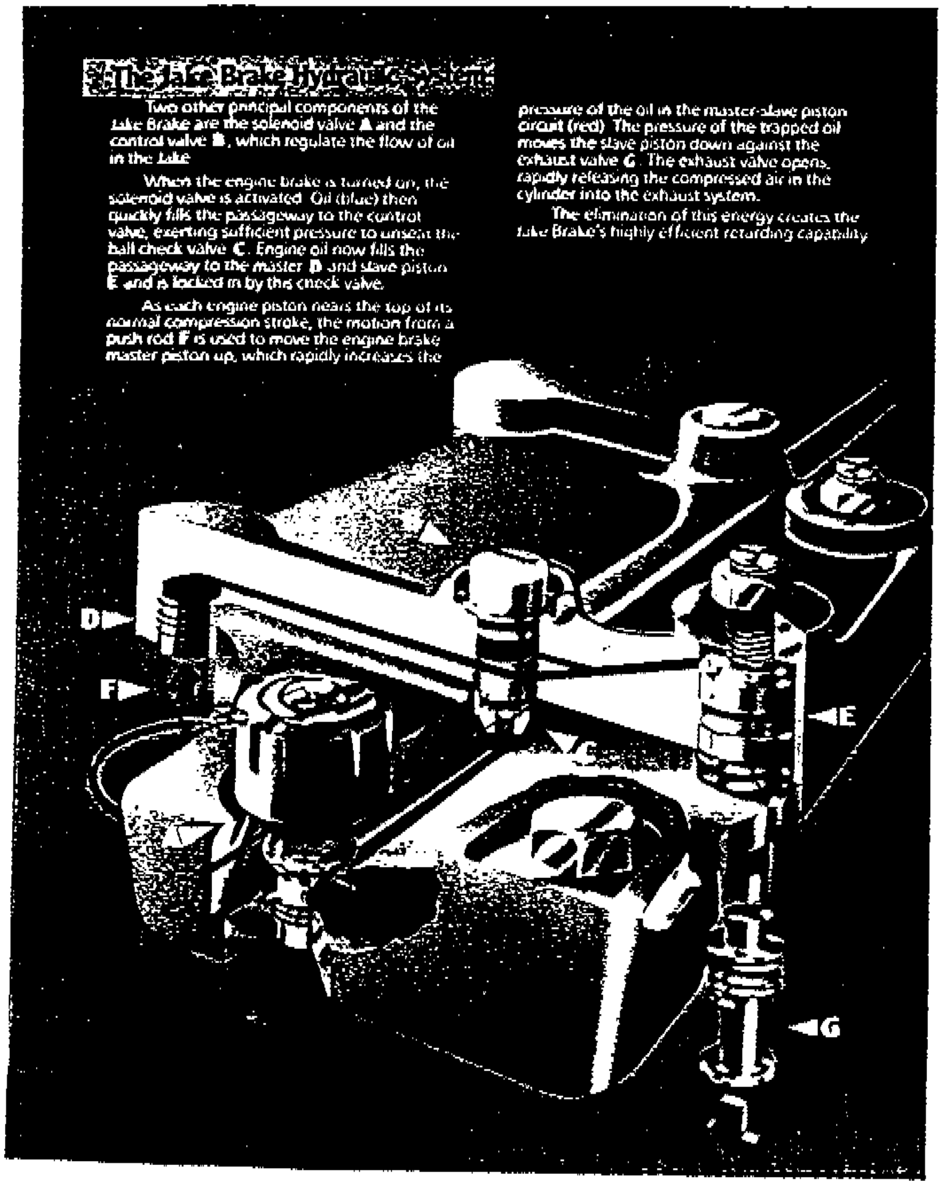


Figure 4.6: "Jacobs" Engine Brake [4.4].

The new engine brake design, like the Jacobs Model 404 installed on the Cummins engine having a rated output of 270 HP produces 295 HP at the flywheel, resulting in about 350 HP retarding power at the rear wheel of a class-8 vehicle. This represents a retarding power up to 130% of the Cummins L10 engine's rated output.

The testing of these new engine brake designs showed that engine brakes were mostly used at higher engine speeds, usually over 1900 rpm. Also, it was found that in the Model 404, there is a nearly straight line relationship between retarding power and engine speed over the operating range.

4.2.1.4 Vehicle Performance

The performance of vehicles equipped with Jacobs engine brake is illustrated in Figure 4.7. This curve shows the horsepower absorbed versus time and it is evident that the vehicle brakes can absorb quite large amounts of power for short periods of time. This older model of Jacobs engine brake based on which the curves were obtained is not capable of high horsepower but can absorb moderate horsepower for indefinite periods of time. Also, the power absorption capability is superior to friction brakes after about three minutes [4.2].

4.2.1.5 Salient Features of Jacobs Engine Brakes (As Per Manufacturer's Claim)

The salient features of Jacobs engine brakes can be listed as follows:

1. Newly-designed Jacobs engine brakes can deliver retarding power up to 130% of the Cummins L10 engine's rated output.

2. Reduces service brake wear with an average service brake life of at least 2 1/2 times longer.

3. Minimizes engine wear due to consistent engine operating temperatures which reduces engine stress. In addition, the removal of carbon build-up from the cylinders by the scavenging effect of the engine braking cycle improves engine maintenance.

4. Minimizes tire wear.

5. Improves vehicle performance and vehicle control.

4.2.2 Exhaust Brakes

4.2.2.1 Introduction

Exhaust brakes are the simplest form of auxiliary brakes on a heavy vehicle. They require only simple modifications to the vehicle and there is

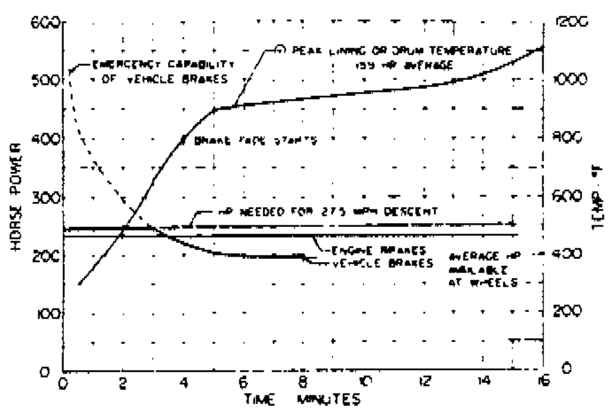


Figure 4.7: Available Retarding Horsepower vs. Time [4.2].

no need of any additional equipment for energy or heat dissipation.

Initially, exhaust brakes were used on gasoline engines but problems were encountered due to the high exhaust temperatures and hard carbon deposits.

The use of diesel engines on nearly all the heavy vehicles improved the reliability of exhaust brakes since the exhaust temperatures are lower and the carbon deposits being also softer, the likelihood of the seizure of the butterfly spindle is less if the exhaust brake is so equipped.

4.2.2.2 Operating Principle

The exhaust brake is a valve which is fitted into the exhaust line between the exhaust manifold and the exhaust silencer. When this valve is closed, air is compressed against it through the open exhaust valve by the piston in its upward motion, on the exhaust stroke. During this operation, the engine becomes a low pressure single stage compressor driven by the vehicle momentum, resulting in a retarding effect being transmitted through the transmission to the driving road wheel [4.5].

The brake torque generated depends on the gearing and engine speed. In general, at moderate and high velocities the primary braking system also must be applied since the generated brake torque is limited to about 70% of the motor drive torque. The major limiting design factor of an exhaust brake is associated with the exit valve spring. Increased pressure in the exhaust system tends to overcome the valve spring, forcing the valve to stay open and consequently limiting the compressor action.

Further improvement in engine brake torque can be achieved by altering the camshaft timing such that the compressor action of the engine is increased. The engine brake torque may be over 100% of the maximum drive torque of the engine. Large retarding torques, however, can only be achieved by using a low gear, which in turn results in undesirably low cruising speeds and thus increased per mile operating costs. No adverse effects on engine wear have been observed with this type of brake. It is claimed that shoe and drum wear can be reduced from 25-50% with the use of exhaust brakes, depending on conditions. Of significance is the effect of engine braking on the thermal state of the combustion engine. Changing thermal conditions (undercooling) may cause premature wear and related problems. Research findings indicate that, when braking on a downhill grade while using the wheel brakes, the combustion cylinder surface temperature decreased from 401° to 167° F. The same test with exhaust brakes showed a temperature decrease from 401° to 302° F, indicating more favorable thermal engine operating conditions [4.6].

Figure 4.8 shows three typical pv-diagrams of the diesel engine. The hatched area represents the indicated work. Indicated power is produced in (a) and (b) whereas in (c) power is absorbed. The calculated cylinder pressure during exhaust and inlet stroke of a 6-cylinder in-line engine is shown in Figure 4.9.

As the exhaust valve starts to open at point 1, the cylinder

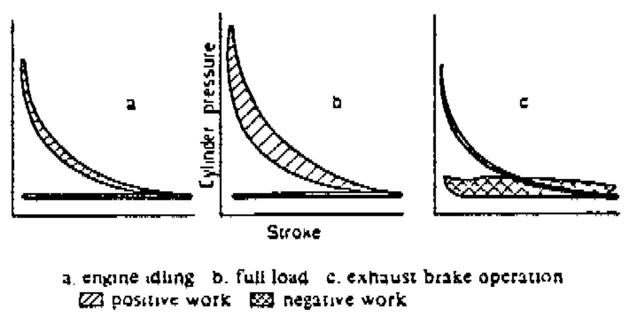


Figure 4.8: Cylinder Pressure Diagram [4.7].

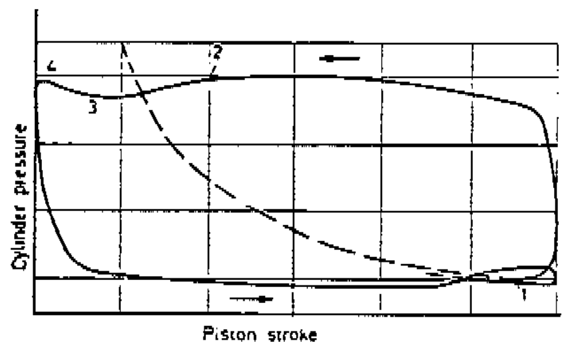


Figure 4.9: Cylinder Pressure Under Exhaust Brake Operation during Inlet/Outlet Stroke [4.7].

pressure is lower than the pressure in the exhaust manifold (see also Figure 4.10). The cylinder is at first filled with compressed air from the exhaust manifold and its pressure rises to the level of the manifold pressure which consequently drops lightly.

During the first part of the exhaust stroke, the air is compressed in the cylinder and in the exhaust manifold. Towards the end of the stroke, between the points 2 and 3, the pressure drops due to the exhaust opening of the next cylinder. At the end of the stroke (3:4), the pressure rises again due to restricted flow through the closing exhaust valve. As the inlet valve opens at point 4, the air expands from the cylinder into the intake manifold.

The intake stroke then follows as under normal operation. A small pressure rise shortly before BDC is due to the uncontrolled opening of the exhaust valve and this allows reverse flow into the cylinder.

4.2.2.3 Exhaust Brake Performance

The performance of exhaust brakes is affected by the design of the engine, gearing of vehicle and, to a lesser extent by the location of the brake relation to the exhaust manifold.

The influence of the engine design consists of three factors:

a) the braking pressure which the engine can develop,

b) the size of the engine, and

c) the engine speed.

The maximum braking pressure is governed by the strength of exhaust valve springs in relation to the size of the valves, and the valve overlap timing of the engine. The pressure build-up in the exhaust manifold and pipe up to the brake is due to the piston compressing air through the open exhaust valve as it rises on the exhaust stroke.

Brake power can be calculated for the 4-cycle engine as follows [4.7].

$$N_B = \frac{(p_{i_B} - p_R)\, n\, z\, V_s}{2}$$

where

N_B - absorbed brake power

p_{i_B} - mean indicated pressure under exhaust brake operation (<0)

p_R - mean pressure resulting from friction and gas exchange

n - engine speed

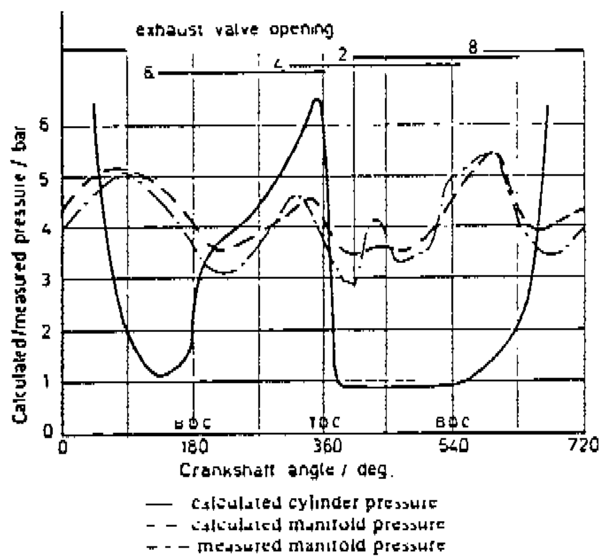


Figure 4.10: Pressure in Cylinder No. 6 and in the Exhaust Manifold of a V8 Engine [4.7].

z - number of cylinders

V_s - swept volume of one cylinder

A certain amount of compression is lost back through the inlet valve due to valve overlap. The braking pressure reaches a maximum at between 1,100 and 1,500 rpm and above this speed, it remains constant due to the effects of valve overlap and the fact that the exhaust valves are acting as relief valves to the pressure in the exhaust manifold.

Maximum braking efficiency is obtained in the higher rpm range. Therefore, the driver should use his gears to keep the engine at high speeds.

Although, different manufacturers will suggest different location for the exhaust brake, the usual practice is to install the brake a short distance from the manifold, the main consideration being the available space. It has been found that when the exhaust brake is installed very close to the manifold, it is usually necessary to install the actuator either remotely or on an extended bracket in order to protect it from excessive heat.

When the brake is installed far from the manifold, the pressure build-up is not as rapid as when it is close to the manifold, but this difference does not create any practical difficulty.

From a metallurgical point of view, the more remote the valve is from the engine, the less stringent are the temperature requirements [4.8]. The remote location increases the tendency of the valve to gum up, therefore the maintenance requirements are also more stringent. Also the remote location raises the number of leakage points substantially [4.8]. By locating the brake close to the manifold, the leakage points can be reduced in number.

Table 4.2 shows the variation of the brake performance as a function of the brake location.

4.2.2.4 Types of Exhaust Brakes

There are four categories of exhaust brakes: butterfly valves, swing valves, poppet valves and slide valves. Of all four, the most common is the butterfly valve type.

The butterfly valve is a simple balanced valve lying parallel to the gas flow in open position and pivoting through an angle from 45° to 90° to closed position. Design variations were limited to refinements of the spindle gland to reduce seizure due to carbon deposits. Also new materials, better suited to high temperature erosion, like stainless steel spindles, were recently introduced in butterfly valve design.

Some typical designs are shown in Figure 4.11 where it is shown that the leakage is different depending on the particular design.

A butterfly valve intended for mechanical controls is presented in

Table 4.2: Variations In Performance with Differing Brake Valve Locations [4.8].

Gradient	1:10	Engine 6 cylinders
Length of gradient	800 metres	5·8 litres
Gross vehicle weight	6800 kg	normally aspirated

Position of brake valve System volume	Maximum vehicle speed kph	Maximum engine speed rpm	Time for descent secs	Exhaust system pressure maximum kN/m^2	Brake valve
Fitted to the exhaust manifold 2·12 litre	59 33·5	3400 2050	58 90	– 290	Open Shut
In exhaust pipe prior to silencer 10·4 litres	59 35	3450 2100	68·5 90	– 290	Open Shut
At rear of exhaust system 81·54	61 34·5	3500 2050	66 85	– 290	Open Shut

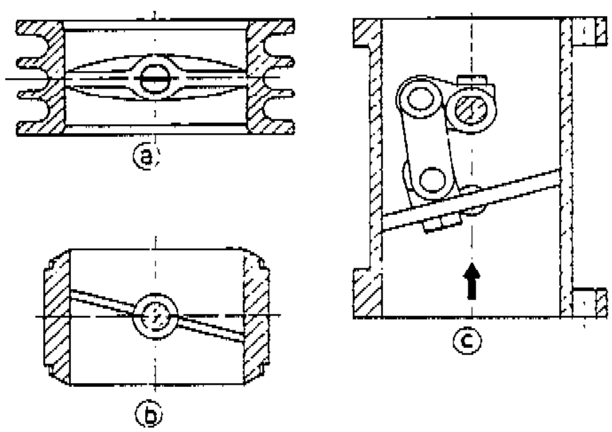


Figure 4.11: Examples of Exhaust Brake Types [4.7].

Figure 4.12. The flap in this design pivots through less than 90° to seat in the bore of the housing. In Figure 4.13 another design is presented in which, in order to overcome problems due to compressed air operation of the brake where it was found that the sudden impact of the hot valve against its seating can lead to premature failure, the valve pivots through a full 90° and leaves a narrow gap between the pivoting flap and the port in the brake housing.

The "Haller" butterfly valve, as shown in Figure 4.14, incorporates a spring loaded pressure relief valve at one end of the spindle.

4.2.2.5 Exhaust Brake Controls

The requirements for the design of the control are as follows:

1. Simultaneous with closing the exhaust brake valve, the injection pump must be set at stop at the idle position, to prevent black smoke after reopening.
2. Stalling of the engine should be avoided when the above condition is met with the clutch disengaged. This is necessary in order to ensure proper functioning of the power steering.

There are four types of exhaust brake controls:

a) Mechanical control;

b) Pneumatic control in 1 or 2 steps;

c) Electro-pneumatic control, e.g. in the brake application valve not adjustable and it needs an rpm protector switch;

d) Electro-magnetic control.

The mechanically operated controls are the ones used exclusively in the earlier years and are still quite popular. The control usually consists of a hand lever in the cab with either rods or a flexible cable connecting it to the brake valve on the exhaust pipe. Then the development of vacuum assisted exhaust brakes showed that it required quite large actuators to provide reliable operation, particularly on exhaust pipes of 4 in. diameter and larger. Electrical actuation of the exhaust brake by means of a solenoid has also been employed. The electrically operated Ashanco exhaust brakes are used in U.K. [4.5].

Use of compressed air operated controls have been used. The original form consisted of hand valve or foot valve control. Foot valve control of the exhaust brake is particularly popular in Germany [4.5]. The synchronization of the exhaust brake with the wheel brakes so that the exhaust brake was applied by the first pressure on the brake pedal are popular in Europe. The method of operation is to actuate the brake by means of a solenoid operated control valve wired to the vehicle's stop light switch. In many of this type of exhaust brake, the brake automatically disengages when the clutch pedal is depressed.

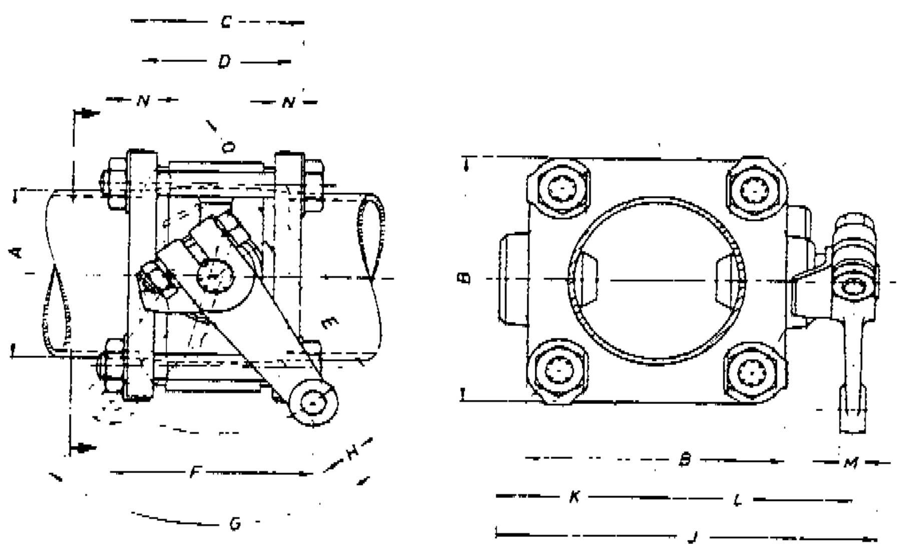


Figure 4.12: Z.F. Exhaust Compression Brake With Full Closing Flap (for operation by a mechanical linkage) [4.5].

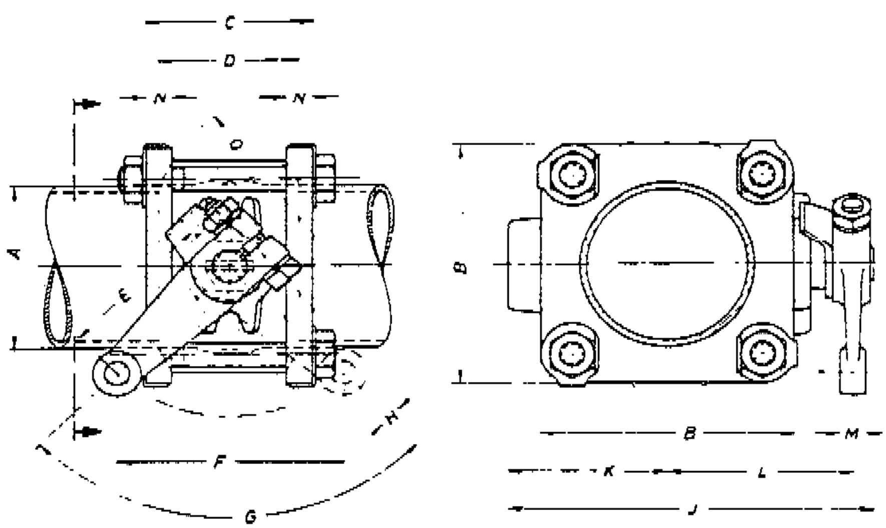


Figure 4.13: Z.F. Exhaust Compression Brake With Full Swivel Flap (for power operation) [4.5].

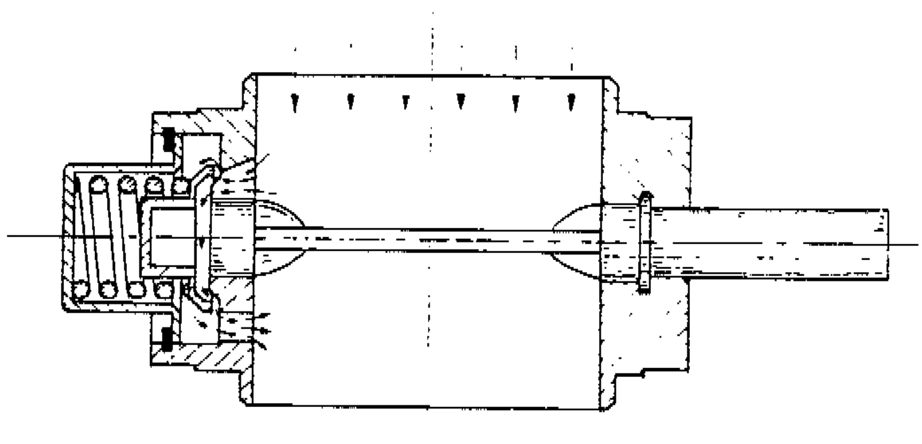


Figure 4.14: 'Haller' Butterfly Valve Exhaust Brake [4.5].

A recent development is the use of accelerator synchronization of the exhaust brake, where the exhaust brake comes into action when the throttle is released. This type of exhaust brake is used in Australia for approximately 75 per cent of total exhaust brake installations and is gaining popularity in the U.K. [4.5]. Such systems are also being used in North America in the recent years. The usual installation incorporates an isolating switch in the cab which enables the driver to turn off the switch if it is required, but in actual practice the exhaust brake is left operative at all times.

Fuel Control

On a diesel engine with a mechanically governed fuel pump, a shut-down device for the fuel pump is not essential. As long as the throttle is closed when the exhaust brake is in operation, the mechanical governor will return the pump rack to the idle delivery position. When this method is used, the exhaust brake must be adjusted to a suitable bypass opening so as to allow the engine to idle while the exhaust brake is applied. If this condition is not met, the combustion of the idle mixture is incomplete and heavy carboning of the valves and injector nozzles will result. The correct adjustment varies from engine to engine but on a typical application, the idle is set at approximately 50 to 74 rpm below its normal idling speed.

On diesels equipped with pneumatically governed fuel pumps, a positive fuel pump shut-down mechanism must be employed. Since there is a leak back of pressure into the inlet manifold while the brake is applied the effect of this pressure will cause the pneumatic governor to "throw" the fuel pump rack to the full delivery position. This results in heavy carboning, negligible braking effect and possible damage to the engine.

For turbocharged engines, the exhaust brake can be fitted either before or after the turbocharger. The installation then becomes similar to that for a normally aspirated engine. For exhaust brakes placed downstream from turbochargers, the major drawback is that the turbine casing is subjected to the back pressure from the exhaust brake and this can result in premature failure of the spindle seals.

4.2.2.6 Make

Williams Compression Brakes, Exhaust/Brakes are available in several models in the U.S.A. These units fit all 4-stroke cycle diesel engines and come either bolt-on or weld-on types for easy installation.

4.2.2.7 Salient Features of Exhaust Brakes [4.6]

1. With use of exhaust brakes, the mean overall vehicle deceleration increases to nearly 0.03g in comparison to 0.015g for engines without exhaust brakes.

2. To maintain a steady speed of about 20mph on down gradients with the vehicle in top gear, the use of foundation brakes are not required on slopes of 1 in 22 or less.

3. On gradients of 1 in 10, a savings of about 33% in usage of the main brakes can be expected.

4. In normal traffic applications, savings of about 20% in usage of the main wheel brakes can be expected.

5. Only a small increase in the maximum braking performance of the vehicle can be expected when using the exhaust brakes in addition to the normal wheel brakes for emergency stop.

4.3 DRIVELINE RETARDERS

Driveline retarders can be classified in two main classes, i.e. hydrodynamic retarders and electric retarders.

4.3.1 Hydrodynamic Retarders

4.3.1.1 Principle of Operation [4.6]

The hydrodynamic retarder is a device that uses viscous damping as the mechanism for producing a retarding torque [4.9 to 4.11]. The viscous damping or internal fluid friction is transformed into thermal energy and dissipated by a heat exchanger. In its design, the hydrodynamic retarder is similar to that of a hydrodynamic clutch; however, its turbine or drive rotor is stationary. The retarding torque is produced by the rotor which pumps a fluid against the stator. The stator reflects the fluid back against the rotor, and a continuous internal pumping cycle is developed. The reaction forces, and hence the retarding torque, are absorbed by the rotor which is connected to the drive wheels of the vehicle. The magnitude of the retarding torque depends upon the amount of fluid in the retarder and the pressure level at which it is introduced into the retarder.

The application of the retarder may result from a hand lever movement or a combined service brake/retarder control such as the foot pedal as shown in Figure 4.15. Depending upon the level of applied control force, compressed air travels over the relay valve to the charge tank and control valve. The compressed air in the charge tank forces the retarder fluid into the hydrodynamic brake simultaneously disconnecting the line between the control valve and the retarder. For a given control input force the control valve allows a constant retarding torque to develop. The degree of fluid application to the retarder determines the amount of fluid and fluid pressure and, consequently, retarding torque. Due to the pumping action of the rotor of the hydrodynamic brake, a pressure difference is produced at the inlet and exit ports, allowing a portion of the service fluid to be circulated through the retarder fluid/water heat exchanger. Hydrodynamic retarders operate independently of the engine, clutch, transmission, or electrical power supply. They are connected to the drive axle and represent an almost indestructible no wear braking element when designed properly. When used on a trailer, a separate cooler becomes necessary without dropping the level below the cooling tubes. A connection to the retarder inlet is taken from the delivery side of the engine cooling pump to supply fluid to the radiator head.

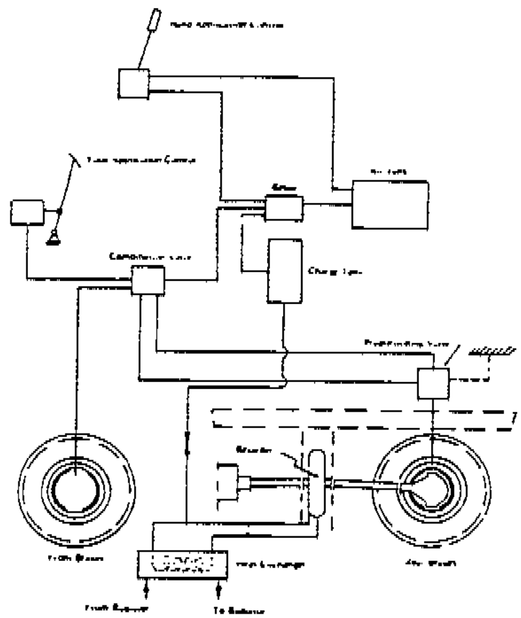


Figure 4.15: Hydrodynamic Retarder Integrated with Foundation Brake [4.6].

4.3.1.2 Typical Hydrodynamic Retarders

"SAMM" Retarders [4.12]

A typical installation of a "SAMM" retarder is shown in Figure 4.16. The radiator head is enlarged, or an auxiliary header tank fitted, to contain sufficient water to fill the retarder.

The braking torque is controlled by regulating the fluid supply to the retarder with an obturator valve in the retarder inlet line. The efficiency of the engine cooling pump is important because the head developed helps considerably in the initial filling of the retarder to reduce the application time lag. To aid the flow to the retarder a throttle valve in the engine to header tank pipe can be linked to the retarder inlet control to simultaneously actuate the retarder and restrict flow through the engine. Flow is further assisted by a centrifugal pump integral with the retarder impeller. In addition to the working fluid connections, a breather pipe is taken from the centre of the torus through the stator vanes to the radiator head. The function of this breather is to vent the centre of the retarder when the control valve is shut to facilitate the discharge of water back to the radiator.

Thompson Retarder [4.6, 4.13, 4.14]

The Thompson retarder fits a transmission-mounted unit, using oil as the working fluid in a closed system. Heat is dissipated to the engine cooling system via an oil-to-water heat exchanger. The schematic layout (Figure 4.17) illustrates the Thompson principle of controlling torque by "injecting" fluid into the retarder with an air-to-oil loading cylinder. Torque is limited by controlling the air pressure acting on the piston in the cylinder. Sufficient air is present in the oil system to permit the retarder to purge itself of oil when no air pressure is acting on the loading cylinder. The torque and power characteristics for this type of retarder fall off at higher speeds. Although this loses the self-stabilizing speed characteristic of the simple hydrokinetic retarder it does alleviate the thermal dissipation problem as the speed increases.

The use of a separate fluid system for the retarder ensures that cool fluid is used for the initial retarder application. This overcomes the problem of the radiator water boiling after a steep ascent when the retarder energy is added to the already hot system.

The property of torque increasing by the square of the retarder speed is frequently exploited to produce a smaller unit for a given retarding force. In consequence, the retarder needs less fluid for the initial application, thus improving the response.

Voith, Fuller, Caterpillar "Brake Saver", Retarders [4.12]

Voith retarder works on the same principle as the Thompson's retarder and uses an integral step-up gear from the final drive. The

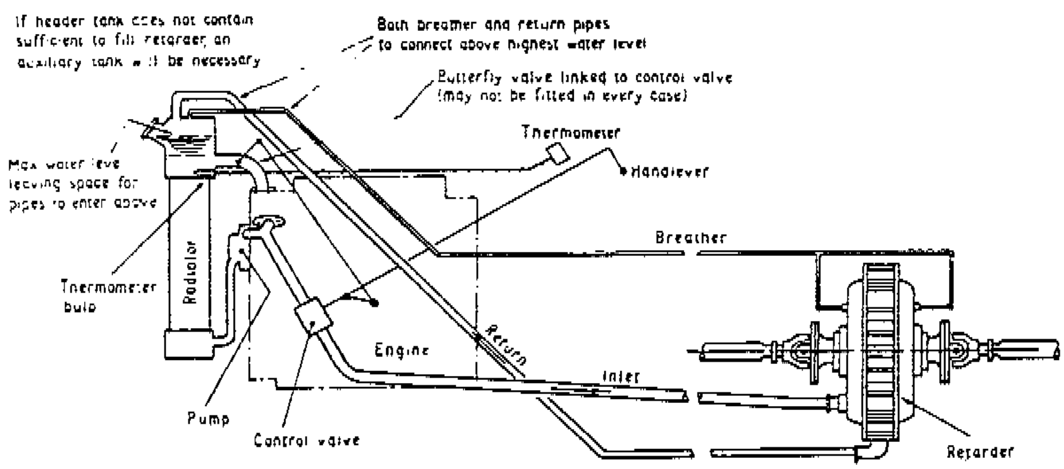


Figure 4.16: 'SAMM' Retarder Installation [4.12].

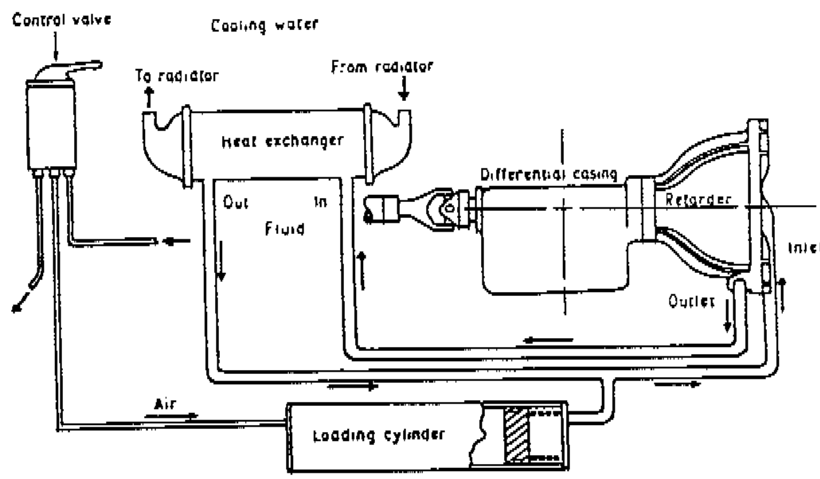


Figure 4.17: Diagrammatic Arrangement of Thompson Axle-Mounted Retarder Installation [4.12].

retarder has a self-contained radiator and a transmission driven cooling fan. By a suitable choice of fluid, this approach permits the dissipation of fluid heat direct to atmosphere at high temperatures, thus minimizing the size of the radiator and making its capacity independent of the engine cooling system. Compared to direct drive retarders which requires one second time lag, the Voith retarder operates with only half a second time lag. Voith retarders, in general, requires high-capacity gearing.

The Fuller retarder and the caterpillar "Brake Saver" are driven off the power take-off and engine crank-shaft, respectively, hence the driven speed and effective retarding torque is dependent upon the gear engaged. This approach enables the retarder output to be matched more closely to the radiator thermal capacity. Also, the effective retarder torque will be reduced by the "high" gear ratios associated with high vehicle speeds. These retarders acts as an exhaust brake, but with superior energy absorption and control and are quieter.

4.3.1.3 Salient Features

1. Operate independently of the engine, clutch, transmission, or electrical power supply.
2. Connected to the drive axle and represents an almost indestructible no-wear braking.
3. Retarding force is greater at higher vehicle speeds and approaches zero retarding torque with decreasing retarder drive shaft speed.
4. Prevents undercooling of the engine below normal operating temperature on long mountain grades by transferring the thermal energy generated through viscous damping in the retarder to the engine cooling system.
5. High power-weight ratio.

4.3.2 Electric Retarders

4.3.2.1 Principle of Operation [4.6]

The principle of the electric retarder is based on the production of eddy currents within a metal disc rotating between two electromagnets which develop a retarding torque on the rotating disc. When the electromagnets are partially energized, the retarding torque is reduced. When the electromagnets are not energized, the retarding torque is zero. The eddy currents result in the heating of the disc. The cooling of the disc is accomplished by means of convection heat transfer with ventilated rotors. Initially, all retarding energy is absorbed by and stored in the rotor material. Only at elevated temperatures does cooling occur. The major problem of the eddy current retarder is associated with the necessity of high brake temperatures for efficient convective cooling capacity - similar to that experienced with friction-type wheel brakes. The high temperatures cause a decrease in retarding effectiveness due to the demagnetizing of the rotor. Depending on the particular material composition involved, this limiting temperature lies near 1350° F.

The maximum retarding performance of an eddy current retarder is limited by the cooling capacity of the ventilated rotor. In order to limit the demagnetizing effects, the operating temperatures should not exceed values of 700° to 900° F. At these levels a reduction in retarding effectiveness of approximately 20 to 30% exists.

4.3.2.2 TYPICAL ELECTRIC RETARDERS

Telma Retarder Type CA [4.15].

Figure 4.18 shows a cross-section of a conventional Telma retarder (type CA) which employs its own transmission shaft.

The stator (3), holds 16 induction coils (6), energized separately in groups of four and central hub (10), housing taper roller bearings (5), equipped with a spring-loaded grease overflow valve (8), and a grease nipple (9). The coils are made up of varnished aluminum wire, moulded in epoxy resin. The stator assembly is supported resiliently through anti-vibration mountings on the chassis frame of the vehicle.

The rotor is made up of two discs (1 and 2), which provide the braking force when subject to the electro-magnetic influence when the coils are excited. Careful design of the fins which are integral with the disc permit independent cooling of the arrangement. The coupling flanges (7), which are solidly mounted on the shaft (4), in turn supporting the discs, provide connection to the propeller shafts of the vehicle.

The air gaps (E) for both discs are adjusted by means of the shims (C).

Focal Type Retarder [4.15]

The Telma Focal retarder is similar to the CA type retarder described above, except that it has no transmission hub. The cross-section of this retarder is shown in Figure 4.19.

The stator (3), attached rigidly to the axle housing (5), or to the gearbox housing through an intermediate support (4), holds eight induction coils (6), energized separately in groups of two. These coils are made up of varnished copper wire moulded in epoxy resin.

The rotor is made up of two discs (1 and 2) which differ from those of the CA type of retarder by the form of their radial arms which are brought back to the centre of the assembly on to a central coupling ring. The adaptor (7), coupled directly to the pinion driving flange of the axle (8), or to the output driving flange of the gearbox, supports the rotor and provides a connection to the propeller shaft of the vehicle.

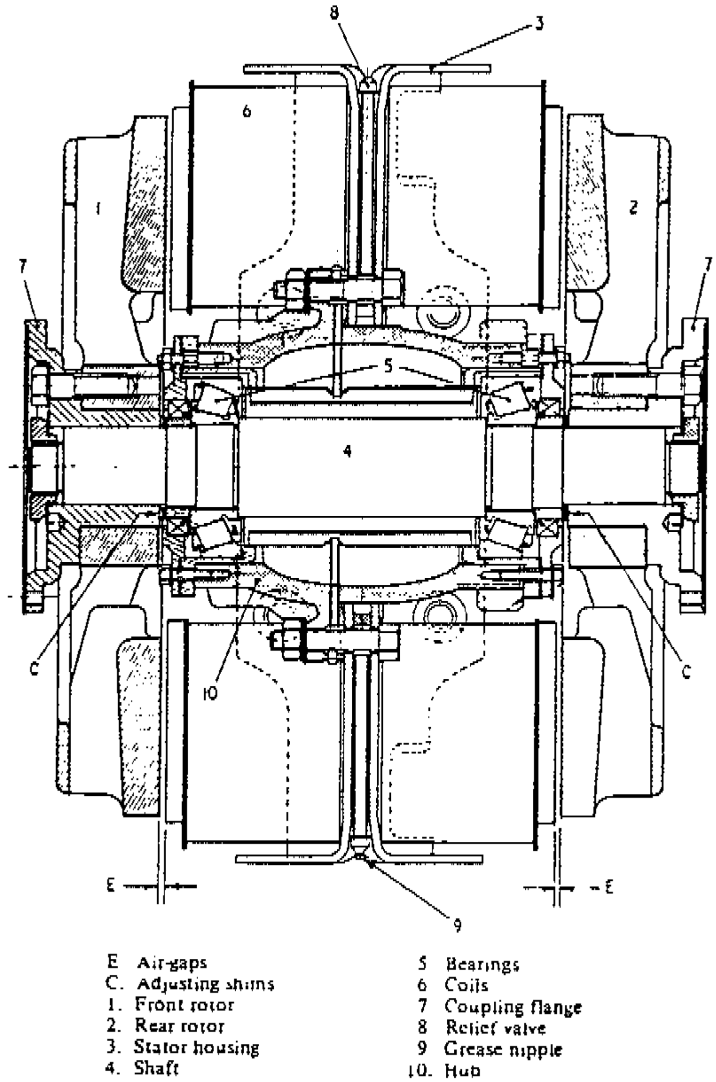
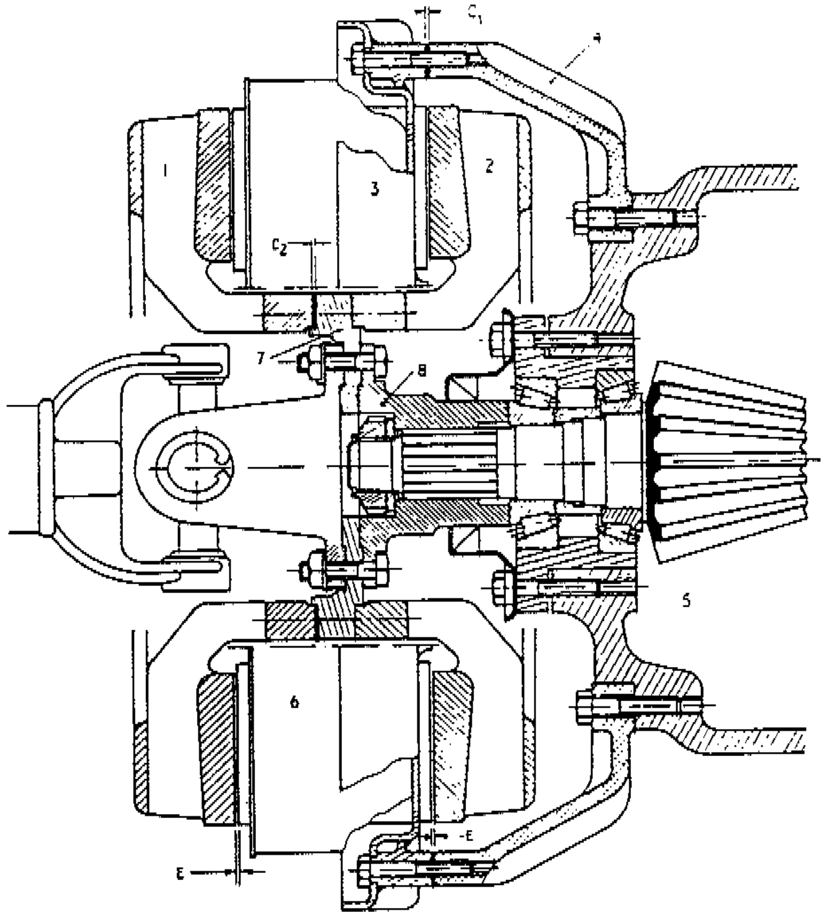


Figure 4.18: Cross-Section of a CA Retarder [4.15].



E. Air gaps
C. Adjusting shims
1 Front rotor
2. Rear rotor
3. Stator housing
4 Stator support
5. Axle housing
6. Coils
7. Adaptor
8. Pinion flange

Figure 4.19: Cross-Section of an Axle Mounted 'Focal-Type' Retarder [4.15].

The air gaps (E) are adjusted on the stator mounting side by the shims (C 1) and on the side of the remote rotor by the shims (C 2).

ILASA Electro-Magnetic Decelerator

ILASA Electro-magnetic decelerator/retarder is an integrated axle model that comes complete with hubs, drums, brakes, "S"-Cams (less slack adjusters), and electrical accessories consisting of: steering wheel mount hand control, and cables. This retarder is available in the U.S.A. and a sample specification of the unit is attached (Table 4.3).

4.3.2.3 Installation of Electric Retarders

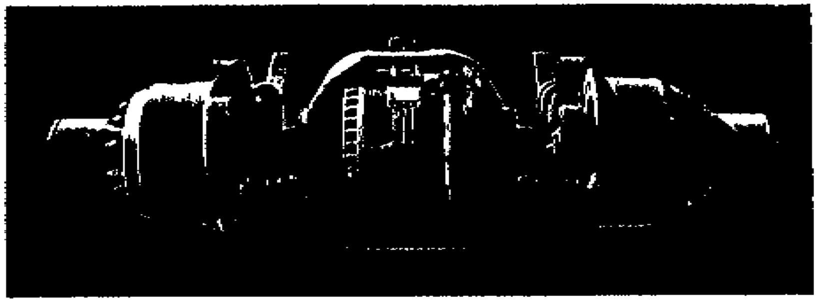
The Telma CA type retarder has been mounted in the transmission line. Its location requires sufficient space between the gearbox and the rear axle. The propeller shaft is divided and fitted with a sliding universal joint and is connected to the coupling flange on the retarder. The retarder is fitted into the chassis of the vehicle by means of anti-vibration mountings.

The Telma "Focal" retarders are combined as a part of the transmission components such as gearbox, or rear axle. This allows installation of retarders on vehicles having very short propeller shafts.

4.4 CONCLUSIONS

In this chapter, heavy vehicle auxiliary braking devices are presented. Engine retarders/brakes, exhaust brakes, hydro-dynamic retarders, and electric retarders are discussed independently. Their operating principle, design considerations, major manufacturers, specifications, and salient features are presented.

Table 4.3: ILASA-Axle with Electro-Magnetic Decelerator (Retarder)



-ILASA-

Axle with Electro-Magnetic Decelerator (RETARDER)

In the early 1960's, ILASA, of Zaragosa, Spain, a leading manufacturer of semi trailer axles, fifth wheels, king pins, landing gear, and couplers, took the electro-magnetic retarder, which had been around since the late 1940's and put it into a trailer axle. Today nearly twenty years and more than 45,000 axles later, the ILASA retarder axle offers the best of all features of heavy duty retarders and more.

ILASA purchases the retarders from three of the four European manufacturers without the shafts, bearings, and supports normally found in the drive line version, as well as making their own. The retarders are available in 16 coil or 2 coil versions.

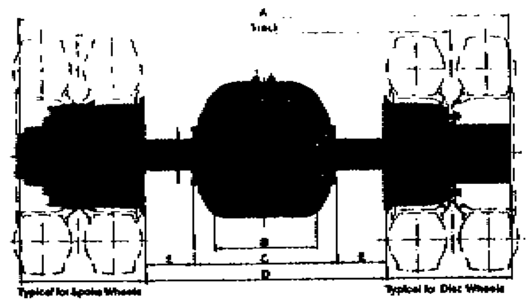
The retarder works on the principal of the generation of eddy currents in soft steel flywheels which are independently driven by the trailer wheels through a planetary speed increaser. These eddy currents are generated as the flywheels pass through magnetic fields created when combinations of four, eight, twelve or sixteen coils are electrically activated by a five position hand control unit located in the cab. The heat produced by these eddy currents dissipates the kinetic energy of the vehicle.

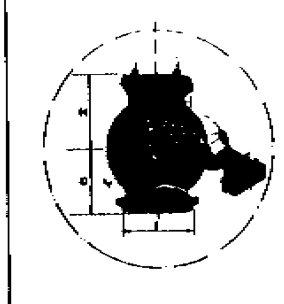
The flywheels are built like fan rotors so that they are self cooling, inducing their own air flow, and requiring no auxiliary cooling system. They are designed so that when they have absorbed the maximum amount of heat, the thermal expansion causes the air gap between them and the coils to increase, thereby reducing the braking effect and preventing overheating or self destruction.

The speed of the flywheels, driven through a 5.5 or 6 to 1 speed increaser is greater for any given ground speed than a similar retarder mounted in the drive line, thereby inducing higher air flows and obtaining better cooling. Further, the flywheels are evenly exposed to cooling air caused by the vehicle motion since they are mounted under the trailer in the air stream.

Mounting the retarder in a trailer axle offers other advantages, the most important being that the braking force is applied at the rear of a combination vehicle, preventing jackknifing. The higher operating speeds of the retarder flywheel, since it is a speed sensitive device, operating in direct proportion to vehicle speed, produces more retarding horsepower for any given land speed. Drive line wear and tear are completely eliminated. There are no reverse "spin-outs" on wet or icy roads. The braking is gentle but powerful — the same every time — there are no surprises. Heat fade is a word of the past.

The ILASA retarder axle minimizes conventional air brake maintenance. It eliminates drive line problems. It saves tire wear. It saves fuel. It decreases trip time. It increases productivity and prevents losses. In time, as it is now in Europe, insurance premium may be reduced and weight allowances made for the extra equipment.





Model	Weight Capacity Rating	Brake Type	Wheel Bolt	Brake Torque	Brake Drum Size	Flywheel Speed	Retarder	Recommended Tire Size	Track Width	A	B	C	D	E	F	G	H	I	J	K	L	Application
[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]

AXLE

RATINGS: 24,000 lb (11 metric ton) and 29,000 lb (13 metric ton) standard. 22,000 lb (10 metric ton) available for lowboys and dropdecks.

AXLE TUBE: 5½" diameter. Fits standard spring seat accessories (5" for lowboy version). Spring seats are not furnished.

DISC WHEEL TYPE: Available for use with all standard and many non standard wheel sizes. Common sizes for wheels are 24.5 x 8.25, 24.5 x 7.5, 22.5 x 8.25, 22.5 x 7.5, 20 x 8.5, 20 x 7.5, steel or aluminum.

SPOKE WHEEL TYPE: Available for use with all standard and many non standard wheel sizes. Common sizes for wheels are 20 x 7.5, 22 x 7.5-5°, 22 x 8.0-5°, 22.5 x 8.25, 24.5 x 8.25, steel or aluminum.

TIRE SIZES: Virtually any common tire size that can be utilized with the above wheels are suitable for use on this axle.

BRAKES: 16½ x 8 brakes furnished as standard. Comes complete with brake drum, and "S" cam lever with 10 splines, 1½" shaft. Slack adjusters and diaphragm operators are not furnished.

BRAKE DRUMS: Cast iron. External mounting permits removal without seal damage or bearing contamination.

LUBRICATION: Splash type. Planetary reduction gears run in 90 weight (E.P.) oil bath. Requires one gallon per side.

STUDS: ¾" — 16 thread, 10 each side. Furnished with inner and outer cap nuts (disc type wheels).

HUBS: Cast iron.

SEALS: Stemco wheel seals. Garlock flywheel seals or equal.

FLYWHEEL DRIVE SHAFT: Field removable for ease of maintenance.

BEARINGS: Sizes vary depending on axle rating. All bearings are a standard number supplied by major bearing manufacturers.

RETARDER

ELECTRICAL: 12 or 24 volt. Field changeable in 16 coil models.

COILS: 16 (standard) or 2 (optional). 16 coil model has four groups of four coils wired in parallel for 12 volt or series for 24 bolt. Two coil model is actually two sets of concentrically wound coils of four independent windings each.

COIL CONSTRUCTION: Aluminum (standard) or copper wiring. Copper is available only in the two coil model. Two coil aluminum model adds approximately 160 lbs. Two coil copper model adds approximately 200 lbs.

TORQUE RATING: 1630 Ft. - lb. (230 MKg) for 16 coil or 2 coil aluminum. Two coil copper with wider flywheels is rated at 2325 ft lb (320 MKg). Slightly higher (plus 15%) ratings are available with air gap change. Torque is highest between 20-25 MPH depending upon tire size.

HORSEPOWER RATING: Standard 600 HP at 50 MPH with 16 coil aluminum retarder. Two coil aluminum is slightly higher. Two coil copper retarder rated at 850 HP at 50 MPH.

ACCESSORIES: Standard with cab control — five position (4 position on). Contactor (relay) box, 15' of five conductor (16 Ga.) cable for use between hand control and contactor box. 50' of four conductor (8 Ga.) with male and female trailer plugs for trailer wiring, and 15' of 1/0 Ga. ground cable, with lugs and connectors for above as required.

FLYWHEELS: Alloy cast steel suitable for sustained high temperature operation without warping. Design is self protecting. High temperature causes rotors to thermally expand in such a way that air gap is increased, reducing heat input, preventing self destruction. Flywheel is built like a fan rotor with cooling fins to dissipate heat caused by retarding action.

SPEED RATIO: 5.5 to 1 flywheel to tire speed for 10 hole disc type wheels, 6 to 1 for spoke type wheels, 4.4 to 1 for lowboy axle. Ratios are normally higher than drive line ratios giving more retarding horsepower and cooling for any given land speed.

-ILASA-

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CHAPTER 5

REVIEW OF PAST AND CURRENT RESEARCH ON HEAVY VEHICLE BRAKING

5.1 INTRODUCTION

The handling characteristic of articulated heavy vehicles are much more complex than the behaviour of single vehicle, since the units affect one another due to inner forces acting at their towing points. Considerable efforts have been dedicated in the past by several researchers through theoretical and experimental works related to the handling performance of commercial vehicle combination. An excellent literature survey on the handling performance of tractor-semitrailer, truck-trailers, and multi-articulated vehicles, had been presented by Vlk [5.1] that includes:

- Directional Performance
- Roll Dynamics
- Brake Performance with and with-out Antilock Device
- Combined Braking and Steering.

This paper includes large number of references that deal with theoretical, experimental, and comparative investigations. Tables 5.1, 5.2, and 5.3 list the various reference numbers and Table 5.4, lists the corresponding references.

In addition to reference [5.1], there are two other State-of-the-Art Review papers available that deal with handling and dynamic performance of articulated highway vehicles [5.2, 5.3]. In the following sections, a detailed account of a review of research on braking performance of heavy vehicles, brake component and system, and brake testing are presented.

5.2 BRAKING PERFORMANCE

5.2.1 Theoretical Analysis

Oetzel [5.4] studied a typical articulated vehicle configuration in terms of balancing of brakes and the shortening of brake response as means of improving stopping distance and stability. He showed that unless brake response times were improved, the practice of balancing brakes to be proportional to axle load would not be effective, since the tractor brakes would reach full power in 1.5 sec. ahead of semi-trailer, which would allow jackknifing to occur if the tractor wheels locked.

Leuzzi [5.5] studied the design of a tandem axle suspension for a semi-trailer. He included the wheel rotational effects along with load transfer on the tandem axle, and the forces and moments associated with suspension geometry. These results indicated that the load on the forward axle of a tandem can be decreased due to action of the equalizing bar during braking, which results in premature wheel lockup. He also proposed a new

Table 5.1 Reference Numbers of Works on the Performance of Tractor-Semitrailers

	theory	experiment	comparison
DIRECTIONAL PERFORMANCE	[2, 16, 17, 18, 20, 21, 22, 26, 32, 36, 37, 39, 42, 44, 45, 46, 47, 49, 64, 68, 70, 72, 74, 75, 76, 77, 78, 82, 89, 110, 134, 135, 136, 139, 140, 152, 153, 154, 155, 169, 170, 172, 173, 174, 177, 182, 183, 186, 188, 192, 194, 197, 198, 200, 201, 202, 210, 211, 212, 214, 234, 252]	[16, 17, 21, 22, 26, 42, 44, 45, 46, 47, 95, 100, 152, 184, 227]	[16, 17, 21, 22, 26, 42, 44, 45, 47, 152]
ROLL DYNAMICS	[19, 39, 54, 55, 56, 63, 67, 83, 84, 85, 129, 131, 165, 175, 176, 193, 213, 219, 221, 222, 223, 235, 240, 241, 243]	[54, 55, 80, 84, 85, 98, 124, 125, 129, 155, 213, 219, 220, 235]	[54, 55, 84, 85, 129, 213, 219, 235]
BRAKING PERFORMANCE	[6, 7, 15, 23, 53, 65, 77, 115, 126, 138, 141, 145, 146, 163, 200, 201, 210, 229]	[15, 24, 73, 95, 97, 106, 107, 108, 109, 127, 133, 143, 145]	[15, 24, 143
BRAKING PERFORMANCE antilock	[6, 8, 15, 69, 113, 119, 149, 150, 195, 196, 207, 211, 212, 214, 231]	[25, 52, 73, 114, 119, 149, 164, 226, 228, 246]	[119, 149]
COMBINED STEERING & BRAKING	[6, 16, 17, 34, 35, 37, 45, 46, 61, 62, 68, 86, 72, 111, 112, 121, 122, 134, 135, 158, 187, 194, 195, 196, 198, 203, 204, 207, 211, 212, 229]	[16, 17, 45, 46, 47, 86, 97, 106, 107, 108, 109, 224]	[16, 17, 45, 46, 47, 86]

Table 5.2 Reference Numbers of Works on the Handling Performance of Truck-Trailers

		theory	experiment	comparison
DIRECTIONAL PERFORMANCE		[11, 12, 27, 57, 59, 75, 76, 77, 90, 91 92, 120, 132, 136, 142, 151, 152, 153, 154, 155, 156, 159, 160, 161, 167, 168, 169, 172, 173, 174, 181, 189, 192, 233, 234, 236, 237, 244, 252]	[10, 11, 28, 29, 57, 81, 120, 151, 152, 168, 216, 232, 233, 234, 244]	[11, 57, 151, 152, 168]
ROLL DYNAMICS		[189, 242, 244, 245]	[155, 244, 245]	
BRAKING PERFORMANCE		[14 51, 77, 94, 141, 163, 166, 180, 237]	[58, 106, 107, 108, 109, 133, 143, 144]	
	antilock	[206, 207, 208]	[53, 58, 164]	
COMBINED STEERING & BRAKING		[181, 203, 205, 206, 207, 208]	[106, 107, 108, 109]	

Table 5.3 Reference Numbers of Works on the Handling Performance of Double and Triple Bottom Configurations (Multi-Articulated Vehicles)

	theory	experiment	comparison
DIRECTIONAL PERFORMANCE	[4, 41, 45, 46, 47, 75, 76, 77, 79, 88, 93, 128, 129, 152, 153, 154, 155, 169, 179, 192]	[3, 4, 30, 41, 45, 46, 47, 79, 128, 129, 152, 169, 216]	[4, 41, 45, 46, 47, 79, 128, 129, 152, 169]
ROLL DYNAMICS	[41, 60, 129]	[41, 60, 129, 230]	[41, 60, 129]
BRAKING PERFORMANCE	[77]	[148, 209]	
COMBINED STEERING & BRAKING	[38]		

Table 5.4 References Corresponding to Tables 5.1, 5.2, and 5.3

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suspension design that, when analyzed, showed satisfactory results.

Fritzsche [5.6] investigated theoretically the influence of static control of the braking forces for truck-trailer vehicles. Bode and Gorge using mathematical analysis [5.7] found that the braking behaviour of articulated vehicles was generally unsatisfactorily and had tendency to jackknife on a slippery surface. They showed that various design and load factors affected braking behaviour. They are wheelbase, centers of gravity, position of the coupling, tractor/trailer weight ratio, and braking force distribution. They also showed that for typical vehicles, large longitudinal forces in the coupling, couplings over or nearly over the rear tractor axle, and high tractor/trailer weight ratio have undesirable effect on braking performance.

Schmid [5.8] investigated the braking effects on the directional stability of the truck-trailer vehicle. He concluded that the greater the proportion of the overall braking force on the combination upon the axles of the trailer, the less the jackknifing. Overbraking on the truck rear axle found to be dangerous. Load sensitive brake force distribution for the truck only reduces the occurance of jackknifing. On the other hand, load sensitive brake proportioning device used on the trailer axle has a favorable influence on stability. However, overbraking of the trailer axle results in trailer axle locking and thus causing the trailer swing.

Koutny [5.9, 5.10] investigated the directional stability of a truck-trailer during braking using scale model tests. The results of straightline braking can be summarized as follows:

- Trailer axle locking did not cause any instability;
- Truck rear axle locking produces jackknifing;
- The brake response times did not exert any influence upon the behaviour of the vehicle combination.

Morse [5.11] investigated the load transfer during braking of a tractor-semitrailer. Since vehicle stability is more likely to be a problem on wet pavement than on dry, he evaluated the brake effectiveness (the ratio of the torque output of the brake to its actuating force) with dynamic axle loads of 0.3g. By means of the calculated deceleration versus brake-line pressure relationship, he demonstrated the difference in braking characteristics between a vehicle whose brakes were statically balanced and one whose brakes are dynamically balanced at 0.3g dynamic load. The results can be summarized as follows:

- The effect of dynamic balancing was to increase the front axle braking effectiveness, reducing braking effectiveness on the trailer rear axle, while not changing the braking effectiveness of the tractor rear axle significantly.

He also demonstrated that the use of load sensitive proportioning valves would improve brake balance even more. He recommended that improvement of pneumatic balance through elimination of pressure and time differentials, and uniform pushout pressures for all brakes on a given vehicle, would aid braking response, especially on low-coefficient surfaces.

Noon, et al. [5.12] investigated the braking performance using a computer simulation which included the thermal effects such as heating and cooling. Variations in brake parameters such as pressure, response times, lining coefficient, and heat transfer characteristics were also studied. Both actual vehicle operations and standard brake tests were simulated. His mathematical model did not include the important suspension effects that lead to spring windup, brake hop, and steering effects due to axle articulation during braking.

North and Oliver [5.13] investigated the brake response time using an in-plane model of an articulated vehicle. They determined the dynamic axle loading for three conditions of vehicle load: empty, laden (low c.g.), and laden (high c.g.); and two braking inputs - a step, F_t and an exponential $F_t\ (1-e^{-10t})$. Based on the results and examining the axle loadings, it was possible to make deductions about the possibility that wheel lock would occur. The study provided the following conclusions:

1. Brake delays increase the possibility of locking of the tractor rear wheels if no load proportioning valve is fitted to the system.

2. If such a valve is fitted to the system, it should respond to transients to minimize the probability of wheel locking.

3. Wheel lockup would be more likely to occur under a step input, since the step input causes more transient load oscillations than the exponential input.

The mathematical model considered by North and Oliver neglected the wheel rotational dynamics and dry friction in the suspension system.

Runge [5.14] investigated the influence of the static (manual) braking control of the trailer depending on the load of the trailer. The effect of the trailer brake pressure reducer and the proportional control of the trailer brake pressure on the braking performance of the truck-trailer vehicles was studied. He concluded that the automatic (dynamic) load sensitive brake proportioning device for the truck and for the trailer are essential for better brake force distribution for all loading conditions.

Bendas [5.15] through theoretical and experimental means investigated the influence of brake pressure reducer devices that were used on all axles of a given truck-trailer vehicle except for the truck front axle. He showed that brake pressure reducing devices improved directional stability and stopping distance.

Limpert [5.16] investigated the brake force distribution on typical tractor-semitrailer vehicles using friction utilization as the criterion. He demonstrated that typical combination vehicles do not achieve braking efficiencies much higher than 60%. He suggested that braking efficiency can be increased as high as 75% by the following:

1. Optimizing the brake force distribution among the axles, and

2. Employing tandem axle suspensions that tend to minimize the inter-axle load transfer and thus prevent premature wheel lockup. For braking efficiency higher than 75%, he suggested the use of load-sensitive proportioning schemes.

Hales [5.17], Fritzsche [5.18], Stump [5.19], Mitsche and Runge [5.20], have shown analytically the use of load - and deceleration-sensitive proportioning valves in improving the stability and response of articulated vehicles.

Shilton [5.21] studied by analytical means the compatibility of truck and drawbar trailer performance. With load sensing equipment, he calculated the adhesion utilization curves for the laden/unladen truck and the laden/unladen trailer.

John [5.22] studied the braking influence on the directional stability of a four axle truck-trailer vehicle. He concluded that the vehicle behaviour during braking depends on the size of the braking ratios of the truck, Z_1, and of the trailer, Z_2. If $Z_1 \quad Z_2$, then compressive force originate at the drawbar, i.e the trailer pushes the truck, and jackknifing occurs. He also suggests the order in which the axles start to lock in truck-trailer vehicles to be:

. 2 - 1 - 3 - 4 (numbering of axles starts with front truck axle).

Hales [5.23] investigated the directional stability of the truck-trailer vehicles braked up to the adhesion limit; i.e. with one or more axle locked.

EEC Regulation 13, covers various regulations in Europe on the distribution of braking between truck and/or trailer axles and the braking compatibility between towing vehicles and trailers (without anti-locking devices). The problems of legislative control of articulated vehicle braking were analyzed by Livingstone [5.24] and Oppenheimer [5.25]. At present the EEC Council Directive on "Braking Devices" (RREC 71/320 EEC) does not lay down the braking efficiency for trailers of type "o" which should be determined. Hoffmann and Rothmann [5.26] have offered a "new" procedure which involves determining the trailer braking ratio by measuring deceleration and the drawbar force.

5.2.2 Experimental Work

Bode and Merz [5.27, 5.28] discussed the brake testing methodology, instrumentation suitable for testing articulated vehicles, and measurement of coupling force. Bode and Gorge [5.29] reported test procedures for articulated vehicles and they correlated their analytical results to the experimental results. They found the values for brake actuator pressure buildup time, response delay, and force buildup time. Factors contributing to these buildup times and delays were discussed along with their effect on forces at the coupling on performance were also studied. Tests were conducted using both a single-axle trailer and a tandem-axle trailer.

Experimental work on synchronizing brake response and minimizing time lags has been reported. Both Underhill [5.30] and Sido [5.31] used brake system mockups closely resembling the vehicle brake system to obtain their results, whereas Guevara [5.32] performed response tests on actual vehicles.

Nelson and Fitch [5.33] reported the well-known Utica tests which used to demonstrate that longer truck combinations (double 40' and triple 27'), when employing properly maintained equipment in current use, could achieve braking, stability and structural characteristics comparable with shorter combinations. The test program consisted of the following:

1. Brake actuation and release timing tests (both bench tests and vehicle tests).

2. Torque balancing according to SAE J880 procedure.

3. Vehicle brake rating tests.

4. Braking performance tests.

5. Stopping distance tests.

6. Directional stability tests.

7. Structural integrity tests.

The tests demonstrated that obtaining the desired results necessitated some modifications of the braking system on the vehicles, as received, including the use of smooth inner wall tubing with streamline fittings, addition of relay and quick release valves, and, in certain cases, the use of load-sensitive proportioning valves.

Kibbee [5.34] and the State of Virginia [5.35] compared the braking performance of a five-axle tractor semi-trailer and a five-axle tractor double-trailer combination. In straight-ahead stops on wet and dry pavement, with the vehicles fully loaded, half loaded, and empty. The results indicate that the vehicles had equivalent performance over a range of speed of 20-55 mph, with the vehicles capable of achieving average decelerations as high as 17 ft/s^2. The results indicate that by appropriate maintenance of brake system, adjustment of the brakes, and proper brake balance, good braking performance can be achieved by multi-articulated vehicles.

Experimental investigations on the effect of brake proportioning have been reported [5.36, 5.45]. Genbom and al. [5.46] carried out experimental work to investigate the effects of axle locking during straighline braking of the four axle truck-trailer vehicle. The results indicate that the axles should lock in the following order:

. 3 - 1 - 4 - 2 (numbering of axles starts with front axle).

Pepoy [5.47] described a commercial vehicle braking simulation that includes user's point of view where economic restraints place additional burden on the simulation beyond just modeling vehicle performance. The aspects of input definition, verification, confidence in results and acceptance criteria are considered as they affect the definition of whether the simulation "works" or not. Recommendations are made which should lead to a more commercially useful tool. The basic factors defined are applicable to other vehicle simulations.

Murphy, Limpert, and Segel [5.48] presented a paper on the developments of braking performance requirements for buses, trucks, and tractor-trailers. Both vehicle testing and analytical techniques, including dynamic modeling and simulation, were used in the program. Performance qualities essential to braking systems are enumerated. The paper presents brake test procedures for:

- Effectiveness;
- Brake failure;
- Minimum stopping distance; and
- Static timing.

Brake testing included vehicles fitted with proportioning valves for tractor rear brakes and trailer brakes, supplied by Borg-Warner, adaptive braking system (antilock device), with a sensor and controller mounted on each of the wheels of the combination vehicle to prevent wheel lockup during braking, supplied by Bendix-Westinghouse; and trailer brake synchronization (syncron) system, which effectively applies the trailer brakes as soon as the treadle valve is depressed, supplied by the Berg Manufacturing Co.

The findings, as derived both from analysis and test, indicate that three major steps will have to be taken to significantly upgrade the maximum braking performance of commercial vehicles.

First, the basic braking systems of the majority of these vehicles will have to be improved by use of more effective brakes, better brake balance, and faster system response on air braked vehicles.

Second, the traction characteristics of tires used on the majority of medium and heavy commercial vehicles will have to be improved so that the advantage of improved brake effectiveness can be fully utilized at the tire-road interface.

Third, advanced brake control systems will have to be employed to allow rapid brake applications without instigating vehicle instability whether the vehicle be loaded or empty, and operating on a dry or slippery surface.

A number of design alternatives exist for achieving these objectives:

1. The effectiveness and fade resistance of the braking systems on medium and heavy trucks can be improved significantly by use of disk brakes.

2. The effectiveness of the braking systems of tractors can be improved by use of large brakes on the front axle of tractors with tandem rear axles (a design configuration in which front brakes are generally absent) and by use of larger brakes on the

front axle of two-axle tractors.

3. The braking efficiency of many trucks and tractor-trailers can be improved by careful distribution of braking effort among the axles of the vehicle.

4. The brake response time of air braked systems can be improved significantly through use of larger hoses, improved connectors and fittings, quick release valves, relay valves on tractors, and trailer brake synchronization.

5. Braking performance can be improved significantly on trucks, buses, and tractor-trailers through use of the advanced brake control systems. These systems, ranked in order of potential for improving braking performance, are: antilock system; dynamic load sensitive proportioning system; static load sensitive proportioning system.

The following points are also suggested for serious consideration:

1. More effective brakes will require stronger suspension and stronger adjacent vehicle structures.

2. Large brakes on the front axles of tractors could require new front axle and steering system designs, and, in many cases, the use of power steering.

3. With increased deceleration capability, methods of cargo restraint will have to be reevaluated. On buses, passenger restraint systems may have to be utilized.

4. The relatively high ratio of center of gravity height above roadway to truck width, that is common to straight trucks, caused vehicle stability problems to be encountered at moderate decelerations. It is expected that the problem will be worse at higher decelerations. This problem may be alleviated by use of antilock systems. However, at this point in time, the problem is not clearly defined and requires much more study before a definite solution can be suggested.

5. If proportioning and/or antilock systems are to be widely used, cognizance should be taken of maintenance and reliability problems associated with each system. Load sensitive proportioning systems require mechanical, pneumatic, or other means of sensing changes in load. Due to wear, corrosion, and other degrading factors, the level of coulomb friction in the suspension system may change, thus requiring periodic inspection and adjustment of the linkage. Since antilock systems for air braked vehicles are still in the developmental stage, reliability is a problem with both mechanical and electronic components. It is mandatory that antilock systems have a high degree of reliability because of the human involved. The test program has pointed out that regardless of load or surface condition, the driver will make rapid, high-level brake

applications if he knows the antilock system is operational, whereas he will be extremely sensitive to load and surface conditions when applying the brakes without the antilock system in operation. Serious stability problems are possible if the driver applies the brakes rapidly thinking that the antilock system is operational where, indeed, it is not due to a component failure.

5.3 COMMERCIAL VEHICLE BRAKE HARDWARE

Slack [5.49] presented a review of commercial vehicle brakes. The review was carried out in three stages:

- The first stage considers the foundation brakes currently in use on commercial vehicles.
- The second stage reviews the selection of brakes against particular application. Some ways of rating brakes are indicated, along with factors involved in selecting friction materials and typical methods of approving the brakes selected.
- The third stage deals with some developments of S-Cam brakes. Specific ways of improving the mechanical efficiency, liner utilization and freedom from vibration are considered.

Thornton [5.50] presented the operation and redesign of axles and brakes to meet FMVSS 121. The purpose of this paper is to discuss the detail known to affect the design of axles, brakes, wheels, and related equipment that will be used on air-braked vehicles under Federal Motor Vehicles Safety Standard (FMVSS) 121. In specific, this paper shows the change that is expected to occur on redesigned axle and braking equipment compatible with the higher levers of vehicle deceleration and controllability to satisfy the Standard. Variables affecting vehicle brake performance and design and application problems related to MVSS 121 qualification are presented.

5.4 BRAKING SYSTEM

Fisher [5.51] investigated brake system component dynamic performance through analysis and experimentation. His analytical investigation included man-machine interaction during braking. Some conclusions drawn by him about the relative speed of response of the components in a typical brake system are:

- The mechanical brake unit, itself, has been found to have very fast response;
- Small masses and large stiffnesses possessed by the brake pedal and the master cylinder indicates that these elements also belong in the fast response category.

Further, conclusions presented in the paper are:

1. The pedal-force/pedal-displacement and brake line pressure/pedal-force characteristics of a typical brake system are highly dependent on the pedal-force application rate.

2. Generalized mathematical models for a brake-pedal linkage, vacuum-assist device, master cylinder, and brake line have been developed for purposes of facilitating analyses of the dynamic performance of brake systems.

3. Analyses and tests have shown that the vacuum-assist device and brake lines are the principal dynamic elements in a typical brake system.

4. Experimental frequency-response data indicate that short sections of brake line exhibit response characteristics similar to those of an underdamped, second-order system.

5. Tests have shown that drum and disc brakes have much higher frequency response characteristics than has been previously reported in the literature.

6. If thermocouples are installed in brake linings for measuring the temperature variations in the brake rotor, they are subject to gross errors caused by the steep temperature gradient in the lining, and the local distortion of the temperature field due to the presence of the thermocouple.

Olsson and et al. [5.52] discussed the applicability of braking control systems for highway vehicles. An evaluation of the applicability of braking control systems for highway vehicles was carried out. Elements of the study included development of a theory of vehicle response in braking maneuvers, design of logic for a braking control system, incorporation of the control in a hybrid computer simulation of a motor vehicle, and evaluation of control system performance. Benefits of braking control system are illustrated in terms of improvement in stability characteristics (rear-wheel control) and in directional control (four-wheel system).

Stearns [5.53] investigated the intermixing of tractors and trailers equipped with pre-121 and FMVSS 121 braking systems. There are a number of factors affecting the braking stability on tractor-semitrailer combination vehicles when intermixing new and existing braking systems. The changes in the stability of combination vehicles causes by the differences in brake performance levels, brake torque utilization, and brake system response of new and existing systems are evaluated. The new and the existing systems are pictorially compared and discussed to clarify the features and performance parameters required on the new equipment which must work in conjunction with pre-FMVSS 121 brake systems.

5.5 S-CAM AND WEDGE BRAKES

Myres [5.54] investigated the interrelationship of the brake lining with other brake components as affected by structural and dimensional variations of the braking components using the "S" cam foundation air brake. His study includes the effect of linings, shoes, cam profile, cam roller, structure, and brake drum. He concludes that a faster wearing material on leading shoe will increase the force to the trailing shoe which improves stability but drops output. A faster wearing material on the trailing shoe will increase the brake output with a drop in consistency. The yield strength of the shoe can be a problem. Dynamometer testing for FMVSS 121 conformance has frequently loaded the trailing shoe of one cam brake above the yield point of the shoe steel; hence, altering the lining loading distribution and absorbed some actuator stroke which, in turn, increases the leading shoe input and brake output torque. He also concludes that there may be some vehicle instability with the stretched shoe. In the case of cam profile, if the cam bushing clearance is excessive when new or due to wear or crushing, the cam can float. This result in reduced stability, increased stroke to maintain clearance, and source of noise. The friction in the cam roller reduces the braking performance.

Marting [5.55] recommended modifications of steering axles cam brakes for FMVSS 121. The increased torque levels required on front axles of air brake vehicles to comply with the 60 miles per hour, 245 foot stopping distance of FMVSS 121 created brake packaging problems not only due to the brake assemblies being larger for a given axle weight, but due to the inclusion of antilock hardware and the larger front axles. To produce the additional torque, increased brake input power was needed, necessitating material and design changes of the basic foundation brake. Many of these design considerations are discussed in this paper.

Newton and England [5.56] presented a two-leading shoe air/spring operated wedge brake for heavy commercial vehicles of up to 13 tonnes static axle weight. They concluded that the dual-wedge design gives a more uniform shoe-to-drum loading pattern than the conventional "S" cam. An auto adjustment mechanism is available which is fully enclosed and functional. The complete brake can be mounted onto the vehicle with a minimum of complication to the vehicle manufacturer. It combines the facility for a service and parking brake together with the actuating mechanism in one compact module using common shoes and only requires connection to the actuation system of the vehicle.

5.6 DISC BRAKES

Birge and Rinker [5.57] presented an excellent paper on the designing of truck disc brakes that meets the requirements of FMVSS 121. The paper presents various design approaches and design considerations dealing specifically on lining area, piston retraction, oil-cooled multiple disc brakes, and air-actuated disc brakes. An actual design of a heavy-duty disc brake for FMVSS 121 is also presented.

Raves [5.58] investigated hydraulic front disc brakes for heavy trucks. The paper addresses the design parameters utilized in the development of the energy absorbing medium, namely, friction material and rotor surface. The study included the evaluation of friction material characteristics in terms of temperature, speed and pressure sensitivity. Truck geometry is also studied along with friction materials to evaluate the combined effects on stopping distances. Rotor thermal stress and interface durability characteristics are also discussed. He concluded the limiting factors on the stopping ability are:

- Relationship of wheel base, to vertical center of gravity;
- Road spring stiffness;
- Suspension damping characteristics;
- Rear drum brake stability; and
- Effectiveness and tire to road adhesion.

The following additional conclusions are also drawn:

1. With all-wheel disc brake installation, increased durability can be anticipated as compared with the drum brake, for the same compliance level.

2. With a heavy duty disc/drum combination, any brake system adversities are manifested in the disc brake, therefore durability cannot be predicted.

3. With a medium duty disc/drum combination, full system durability will be less than pre-121 by virtue of overall effectiveness.

Markert [5.59] described the activities in developing disc brakes at the Bendix Corporation, pointing out the successes and problems encountered.

Stokes [5.60] presented the findings of the testing on heavy truck disc brakes. He outlines various laboratory tests which can be utilized to evaluate heavy truck disc brakes. The methods described include supplemental techniques derived from improved dynamometer procedures which assure that the brake components and assembly will perform the intended function and that a practical and effective design has been obtained. They ensure that the brakes will provide the "toughness" required in the "real world" environment.

Airheart [5.61] discussed various approaches to truck disc brake design. Design of the disc and the effect of its design on brake performance show current designs being proposed may be inadequate. Lining area must be adequate for long life without restricting cooling. Piston retraction and adjustment by mechanical means is more reliable than seal retraction. A multiple disc oil-cooled hydraulic disc brake can provide extended life an high torque in a small diameter package. Air-actuated disc brakes eliminate the need for hydraulics, but introduce problems in force multiplication and brake-to-brake balance.

Soltis [5.62] described the major features of the design of air disc brakes and the development during the laboratory and vehicle tests.

Yamamoto [5.63] described the straight-air disc brake systems for heavy vehicles. The air-over hydraulic systems will be replaced by advanced applications of straight-air brakes by eliminating the complexity of such systems. Mechanisms for heavy duty are discussed while considering its linearity, durability, and dependability.

5.7 BRAKE LINING

Rhee [5.64] discussed various brake materials, test methods, and friction and wear mechanisms. He indicated that ceramic ingredients are used primarily for controlling or stabilizing friction, although they may affect wear in the process. He also concludes that friction increases with increasing Asbestos content and with increasing hardness of ceramic ingredients.

Joshi [5.65] highlights some of the most important characteristics of a friction material in general, and in a disc brake application in particular. He shows how the topic of friction material properties be approached by using a "Friction Material Analysis Chart" developed in the paper for selection of a friction material in a brake application.

Fox [5.66] presented the design considerations for wet wheel brakes. The effect of friction materials, mating members, and groove patterns are discussed. A wet wheel brake's purpose is to retard, stop, or hold a vehicle. It differs from the more conventional dry brake only in being surrounded by a fluid environment which serves to lubricate and cool the friction surfaces. This will improve the braking system's durability, reliability, and smoothness.

Anderson and Knapp [5.67] describes laboratory equipment and test procedures which characterize lining strength and expansion behaviour, using small specimens. A benchtop testing device is introduced which can be used to perform shear and tensile tests on lining samples and singly-riveted lining assemblies. Results are presented for a representative group of production and experimental linings. Applications are discussed.

Aoki [5.68] discussed vehicle road testings of brake linings in Japan. Various methods to evaluate lining characteristics by road test, both governmental and industrial organizations, are reviewed. Japanese Automobile Standards (JASO) are established covering wide range of tests. This standard is widely accepted by vehicle manufacturers, but still there are items not defined in this standard, and some of the manufacturers like to modify the standard for various reasons. It is intended that JASO will cover all aspect of road test procedure and be adopted by all Japanese manufacturers. Further, this standard will serve for the rationalization and harmonization of international standards like ISO and various regulations like ECE, FMVSS, Japanese Regulations, etc.

5.8 PROPORTIONING VALVES

Sivulka and Willi [5.69] studied a dual-mode height sensing brake proportioning valve. They described the performance, design, and operation of this new selective brake proportioning valve developed by Kelsey-Hayes Co. The valve combines positive braking performance with high reliability to achieve significant quality improvements in the brake systems of vehicles with large differences between unladen and laden weights. Use of this new valve provides significant performance improvements over a single fixed mode brake system.

Limpert, Robbins, and Girchrist [5.70], described an analysis, design, and testing of two-way brake proportioning for improved braking in a turn. The theory of two-way proportioning was discussed. The proportioning of brakeline pressure based on braking deceleration and lateral acceleration, permits the production of braking forces while turning which equal, but do not exceed, the dynamic braking capability of each tire, thus ensuring optimum utilization of tire/road friction. The principles discussed in the paper are applicable to combination vehicles.

5.9 CONTAMINANT REMOVAL

Fuerst [5.71] describes methods of removing these contaminants to reduce the number and severity of problems that may develop in the air brake system. The basic mechanical and thermodynamic characteristics of compressed air drying are discussed as they relate to dryers, aftercoolers, and other devices available for moisture removal. An analysis of the benefits accrued through contaminant removal is presented including such topics as reduced corrosion, extension of compressor life, reduced downtime, and elimination of air system freezeup. Moisture expelling devices such as aftercoolers, heat exchangers, and expulsion valves are compared to an contrasted with state-of-the-art air dryers.

5.10 AIR BRAKE ADJUSTMENT

Radlinski, Williams, and Machey [5.72], described the importance of maintaining air brake adjustment in achieving optimum system performance. The conclusions drawn by them can be summarized as follows:

- Increased brake chamber stroke results in degradation of brake torque due to the pushrod force-stroke relationship of clamp ring type service chambers and spring type parking and emergency chambers used on most air brakes.

- Drum brakes are particularly sensitive to adjustment due to drum expansion and self energization.

5.11 BRAKE TESTING

Post, Fancher, and Bernard [5.73] presented the torque characteristics of commercial vehicle brakes. The new over-the-road brake dynamometer was developed to measure commercial vehicle brake torque at nearly constant speed. In this paper, results are presented from the initial test program of this vehicle. The program included tests to assess the influence of initial drum temperature and rubbing velocity on brake torque, as well as the occurrence and nature of brake fade and hysteresis effects. Results of these tests indicated that the rate of energy flow into the brake system had more significant effect on brake torque than initial temperature. Greater hysteresis was found in the S-Cam brakes than in the dual-wedge brakes.

Steis [5.74] presented inertia brake dynamometer testing techniques for FMVSS 121. The need for standardization in inertia brake dynamometer techniques and the basic principles for setting up a "S" cam air brake for evaluation under laboratory conditions are discussed. Specific brake conditions show the need for standardization of brake set-up for obtaining the optimum output from a brake and friction material combination.

Klein and Szostak [5.75] presented description and performance of trailer brake systems with recommendations for an effectiveness test procedure. The operation of electric and surge brake systems were described. Investigations were made to predict stopping distances of combination vehicles as well as those trailers having no brakes.

Hatch [5.76] presented a brake test dynamometer program for Europe and the U.S.A. The dynamometer was used to simulate the behaviors of a brake in a vehicle in service. Examples of the validity of the test procedures are presented including the characterization of cast iron rotors, the effects of surface finish and the direction of machining of rotors, measurement of squeal, and of slight misalignment of the brake.

5.12 NEW TYPES OF BRAKES

Gerbert [5.77] describes and analyzes a newly developed brake mechanism supplied with an integrated automatic slack adjuster for heavy vehicles. The device is based on the crank mechanism principle. The geometric properties are examined and it is shown how some design parameters can be adjusted to meet specific requirements. The relation between applied torque and the brake torque is derived and in an example an inherent brake amplification property of the mechanism is shown. This property persists even when friction in the mechanism is considered. The automatic adjusting system is travel sensing, so the relation between the travel of the sensing device and the clearance is analyzed. Finally, the amount of correction of the clearance is treated and it is shown how the prescribed clearance can be maintained within very narrow limits.

5.13 CONCLUSIONS

In this chapter a detailed review of past and current research on heavy vehicle braking is presented. Summary of simulation and experimental results on braking performance is also presented. Discussions on commercial brake system hardware, proportioning valve, brake testing, etc., are provided.

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CHAPTER 6

ANTI-LOCK/ANTI-SKID BRAKE SYSTEMS

6.1 INTRODUCTION

Systems intended to prevent or minimize wheel lockup during hard brake applications have been known as anti-lock, anti-skid, antiwheel-lock, and adaptive brake control systems. While anti-lock is the term used in FMVSS 121, which perhaps is more accurate and descriptive terminology, anti-skid has been used more frequently.

The anti-lock braking system (ABS) will provide:

- Increased stability of the vehicle on the road during braking.

- Good vehicle steerability while braking.

- Minimize the tractor or semitrailer jackknifing.

- Optimize the vehicle stopping distance.

On slippery roads and more generally in emergency situations, over braking frequently induces wheel locking. A locked wheel transmits practically no lateral force thus the vehicle is rendered unstable or in the case of front wheel lockup, unsteerable. Articulated vehicles in these circumstances are subject to trailer swing and jackknifing of tractor and/or trailer. In the case of smooth or slippery roads, the stopping distance with locked wheels is greatly increased; on rough roads, increase tire abrasion results.

The FMVSS 121 regulation does not explicitly require anti-skid systems. Inspite of this, several air brake anti-skid systems have been developed and promoted with variety of technical approaches, performance claims, packaging techniques, installation requirements, and other factors. The following sections will outline a general review of anti-lock brake systems.

6.2 OPERATING PRINCIPLE

For a vehicle, at the tire to pavement interface, frictional force reaches a peak at a nominal percentage of relative motion (slip) which if exceeded, results in a reduction of friction force, leading to wheel lockup and skidding. In other words, the braking force coefficient (i.e braking force divided by imposed vertical load) produced by a tire reaches a maximum when the peripheral speed of the tire is approximately 15% slower than the linear speed of the vehicle (i.e. 15% wheel slip). After this peak point, the braking force coefficient reduces as the longitudinal slip increases to the locked wheel condition (100% slip). The lateral force coefficient (the ratio of cornering force to imposed vertical load) also varies considerably with slip, reaching a peak earlier, but then falling sharply until, the locked wheel condition at which time it approaches zero. The friction-coefficient (braking force coefficient) varies considerably according to the condition of the road surface (Figure 6.1). By referring to Figure 6.1, it can be seen that the maximum braking force coefficient is significantly higher at 15% wheel slip than with a locked wheel, therefore the function of anti-skid system is to maintain the wheels at a controlled level of slip during a braking application and thereby provide a high lateral force coefficient such that the vehicle will retain its stability. If the vehicle's front axle is also equipped with the system, the driver will be able to maintain steering control and achieve better stopping distances than would be possible with locked wheels.

A schematic diagram of a typical anti-skid control system is shown in Figure 6.2. The major elements of anti-skid system are [6.2]:

1. Wheel speed sensor:

   - Monitor's wheel speed and/or acceleration at least during braking application.

2. Vehicle speed or deceleration reference:

   - Measures directly the vehicle speed or deceleration, independent of wheels.

   or

   - Preset exceedence criterion built into the system logic.

3. Electronic processor/controller:

   - Analyzes wheel speed inputs, performs necessary comparisons with set value, issues proper control commands to the brake modulating valves.

4. Modulator or memory controlled control valve:

   - An electro-pneumatic device which, on commend from the processor, regulates brake chamber air pressure by controlling air input and/or outlet.

5. Failure detector:

   - A sub-system which monitors at least the electrical malfunctioning of the system and provides the required warning in event of failure.

All of the present systems directly monitor wheel speed; one system uses a chassis-mounted accelerometer to directly measure vehicle deceleration. Some systems determine initial vehicle speed from wheel speed at the onset of braking, and adjust internal deceleration references accordingly. Most of the systems studied, however, have a "built-in" maximum wheel deceleration value which will trigger the system into action.

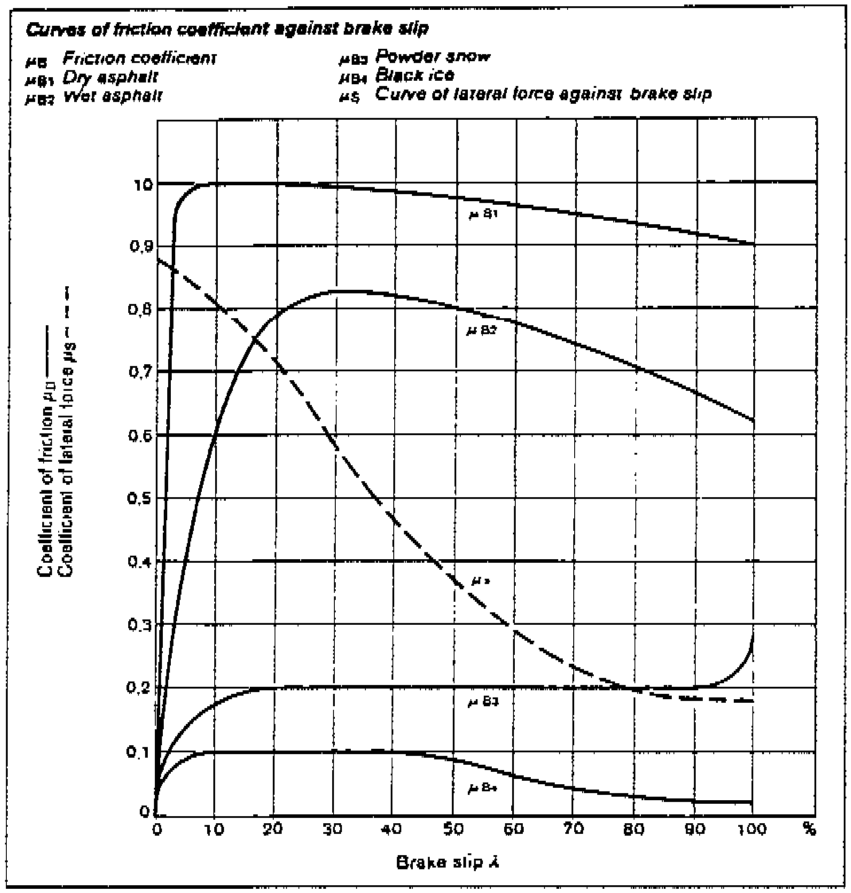
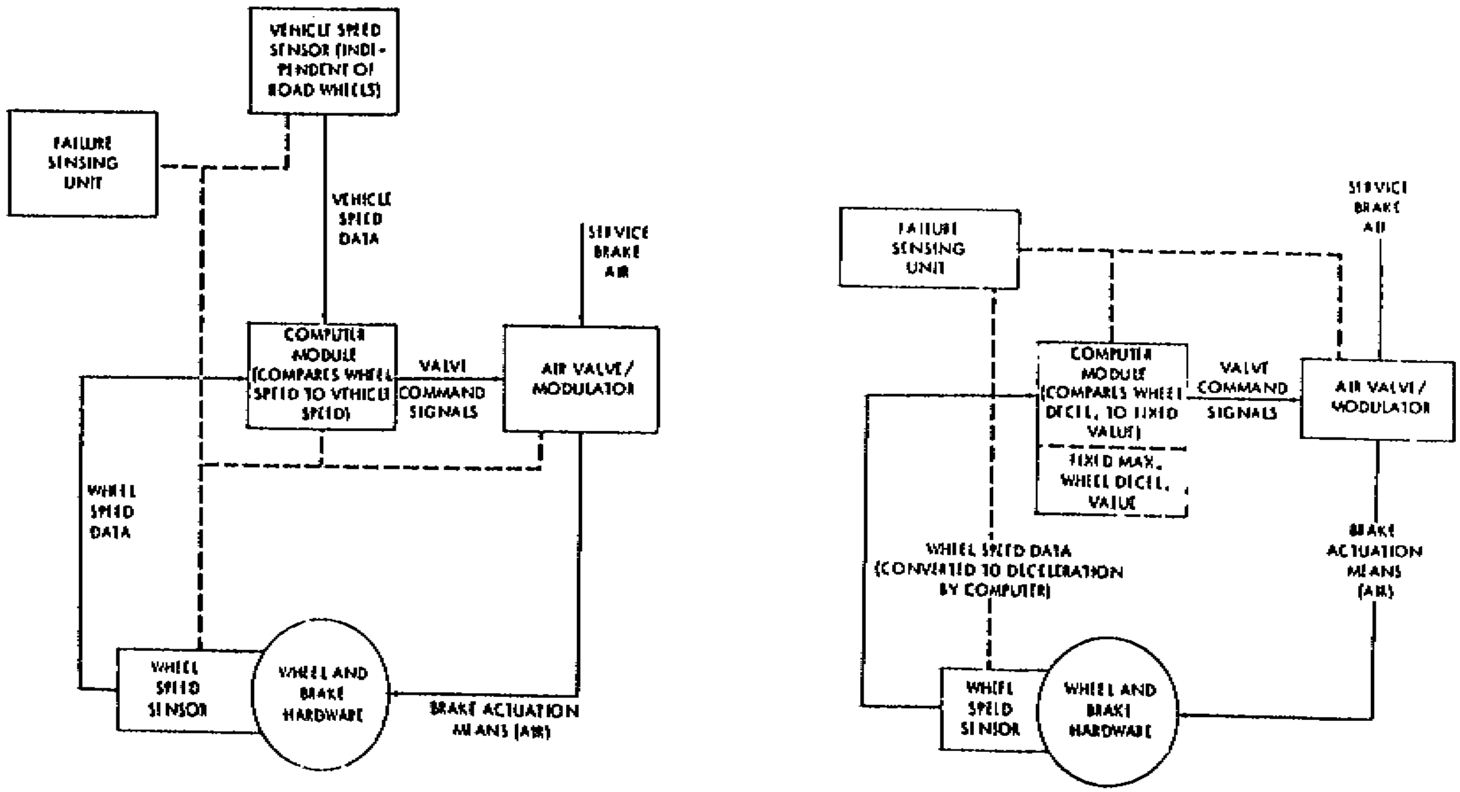


Figure 6.1: Curves of Friction Coefficient Against Brake Slip [6.1].



"Ideal" System

Practical System

Figure 6.2: Block Diagram of Anti-Skid Control [6.2].

Ideally, each wheel and brake set on a vehicle would be an independent subsystem, complete with its own anti-lockup provision. With a steered axle, however, it has been demonstrated that wheel-by-wheel control can be detrimental to vehicle control, particularly if the two wheels are on surfaces of significantly different friction coefficients. The resultant difference in braking effort causes steering to "pull" toward the more heavily braked side, degrading steering control. So, it has become practice to apply axle-by-axle control to steering axles, providing simultaneous modulation of the brakes on both wheels.

While steering "pull" is of little concern for nonsteered axles, economic factors may favor the choice of axle-by-axle control over wheel-by-wheel control. Such a choice dictates consideration of other factors, such as which wheel on a given axle should prevail. "Sense high", and "sense low" are the terms used by skid control manufacturers to describe axle-by-axle systems which are governed by the faster or slower wheel on an axle, respectively. The "sense high" mode has the advantage that it favors shorter stopping distances, but sometimes will allow on wheel to lock and remain locked during an entire stop. The "sense low" mode, however, prevents sustained lockup of either wheel, but can sacrifice stopping distance if one wheel is on a more slippery surface than the other.

Most manufacturers utilize the "sense low" mode, feeling that preventing wheel lockup always is the predominant goal. However, there are some departures from this rule. One system uses "senses low" on steering axles only, because it minimizes steering "pull", and "sense high" on all non-steered axles. Another uses a "biased average" approach, which processes speed sensor signals from both wheels, and applies a built-in proprietary criterion in evaluating the composite signal. And still another uses "sense low" on steered axles, but may use either "sense high" or "sense low" on non-steered axles, depending upon the relative speed between the two wheels. Schemes used by the various manufacturers appear in the table "System Operating Parameters".

At least one further compromise is possible, that of adopting "bogie control" for tandem axle suspensions, a system in which one control unit is designed to modulate all four brakes of a tandem axle installation. This choice, of course, compounds the difficulty in determining which wheel should prevail as the indicator of a lockup condition. However, there has been considerable interest expressed in "bogie" control, largely due to the economics of using only one valve and computer for a pair of axles. Little actual testing has been accomplished with "bogie" control systems.

It is conceivable that a series of considerations and compromises might lead to all of the above configurations being included in a single vehicle combination: axle-by-axle control on the front for good steering characteristics, wheel-by-wheel control on the tandem drive axles for optimum stopping ability, and "bogie" control on the trailer for maximum economy.

6.3 DISCUSSION ON ANTI-SKID SYSTEM'S HARDWARE

In North America, there are nine major manufacturers of anti-skid systems. They are:

- A-C Sparkplug
- Bendix-Westinghouse
- Berg
- Borg-Warner
- Eaton
- B.F. Goodrich
- Kelsey-Hayes
- Rockwell
- Wagner Electric.

Tables 6.1 to 6.7 present a comparison of these anti-skid systems [6.2]. In Table 6.1, the wheel speed sensor system is compared which includes operating principle, signal frequency and wave form, and details on mechanical constructions. Table 6.2, provides a comparison on the required modifications to the running gear, special tools required, adjustment details, and special precautions to be observed. All details pertaining to processor/digital controller: logic type, electronic configuration, rate of update, electronic packaging, "learning" mode, and size are presented in Table 6.3. Information on modulator valve assembly: modulation method, mechanical packaging and disposition of non-antilock valve hardware are outlined in Table 6.4. In Table 6.5, system operating parameters such as preferred control mode, high/low sense mode, vehicle speed reference, system cycle criterion, and minimum operating speed are compared. Table 6.6, provides information on miscellaneous details of the system: packages in 3-axle system, assembly line test procedures, field service equipment, operating temperature and monitoring of failure modes. All requirements for the system electrical parameters: ground, voltage limits, current per axle at 12 v, and transient protection are compared in Table 6.7.

In Europe, there are three major manufacturers of anti-skid systems. They are:

- Lucus-Girling
- Wabco
- Bosch.

Tables 6.8 to 6.10 present a detailed specification of these systems.

From the various anti-skid systems presented in Tables, it can be observed that there are several design solutions. Most systems presented, have the wheel speed sensor rely upon the principle of variable reluctance magnetic sensing. It is a permanent magnet sensor (stator) arranged in close proximity to a toothed or slotted rotor producing a signal whose frequency is a function of the number of teeth or slots in the rotor, and the speed of wheel rotation.

Most wheel speed sensors fall into the following three basic categories:

- Axial pickup, mated with stamped slotted rotor ring (Figure 6.3).

Table 6.1: Air Brake Antilock System Comparisons: Wheel Speed Sensor System [6.2].

Manufacturer (System Name)	Operating Principle	Signal Frequency and Waveform	Mechanical Construction
AC Spark Plug (Wheel Lock Brake Control)	Variable reluctance magnetic Self-energized	60/90 cycles/rev sinewave	Teeth cast in brake drum; axial pickup mounted to air chamber support bracket; other types available
Berg/Fiat (Electronic Braking System)	Variable reluctance magnetic Self-energized	120 cycles/rev sinewave	Radially-toothed wheel and radially oriented magnetic pickup; two types available: factory assembled outboard type, or inboard hub mounted rotor and spider mounted pickup
Eaton (Skid Control)	Variable reluctance magnetic Self-energized	60 cycles/rev sinewave	Stamped perforated ring mounted on hub; axial pickup mounted on brake spider
B. F. Goodrich (Skid Control)	Variable reluctance magnetic Self-energized	60 cycles/rev sinewave	Stamped toothed ring mounted on hub or brake drum; radial-type bipolar pickup mounted on brake spider
Kelsey-Hayes (Computer Brake Control)	Variable reluctance magnetic Self-energized	60/120 cycles/rev sinewave	Stamped perforated ring mounted on brake drum; axial pickup mounted on spider
Rockwell-Standard (Skid-Trol)	Variable reluctance magnetic	120 cycles/rev	Stamped perforated ring mounted on hub oil slinger; axial pickup mounted on spider or backing plate
Wagner Electric (Skid Control)	Eddy current generator	60 cycles/rev sinewave	Concentric toothed rotor and stator with 360 deg "active" faces; hub-mounted rotor and spindle-mount stator

Table 6.2: Air Brake Antilock System Comparisons: Wheel Speed Sensor-Installation [6.2].

Manufacturer (System Name)	Running Gear Modifications Required	Special Tools Required	Adjustment	Special Precautions
AC Spark Plug (Wheel Lock Brake Control)	Special notched brake drum Additional bracket on chamber bracket	None	Manual, after final assembly of wheel end hardware	Sensor should be mounted and adjusted after assembly
Berg/Fiat (Electronic Braking System)	Machined step on hub (inboard mount) Drill housing for wire (outboard mount)	Gage for adjusting inboard type sensor	Inboard—manual using specified gage Outboard—factory preadjusted	Avoid damage to inboard type unit during hub/drum assembly
Eaton (Skid Control)	Remove slinger in some cases	None	Automatic	None
B. F. Goodrich (Skid Control)	Remove slinger in some cases	Pickup location fixture	Fixed, for each given installation	Use care in aligning hub and drum during assembly procedure
Kelsey-Hayes (Computer Brake Control)	Drill hole for sensor wire	None	Automatic	Automatic adjust feature must be preset
Rockwell-Standard (Skid-Trol)	Drill hole for sensor wire	None	Automatic	Remove sensor retainer pin before assembly
Wagner Electric (Skid Control)	Machined step on hub for rotor	None	None required	Use care in treatment of sensor parts (treat as any oil seal)

Table 6.3: Air Brake Antilock System Comparisons: Computer Modules [6.2].

Manufacturer (System Name)	Logic Type	Electronic Configuration	Data "Update" Rate	Electronics Packaging	"Learning" Mode	Size, Height x Width x Depth, in
AC Spark Plug (Wheel Lock Brake Control)	Hybrid—essentially analog	Hybrid	Each brake system cycle	Etched circuit board, mounted in valve assembly cavity	NA	(In valve body)
Berg/Fiat (Electronic Braking System)	Analog	Present—hybrid Future—LSI	Continuous analog (each 7/8 in of ground travel, typical)	Etched circuit board in protected housing	None	2.5x6.5x7 (part of valve package)
Eaton (Skid Control)	Analog	Present—discrete Future—hybrid	Continuous analog	Encapsulated circuit board in housing attached to valve body	Compares with previous cycles	(Part of valve package)
B. F. Goodrich (Skid Control)	Hybrid—essentially analog	Present—discrete Future—LSI	Continuous analog	Within modulator valve body	From repetition of valve cycling	(In valve body)
Kelsey-Hayes (Computer Brake Control)	Analog	Present—IC discrete Future—LSI	Continuous analog	Potted circuit in plastic housing	Adjusts vehicle speed data through successive measurements	2.5x5.5x8 (integrated-type unit also available)
Rockwell-Standard (Skid-Trol)	Digital	LSI	20 ms	Protected chamber in valve housing	"Remembers" conditions of immediately preceding cycle	(In valve body)
Wagner Electric (Skid Control)	Analog	Hybrid	35-40 ms	Sealed separate module	From wheel acceleration rates	1.67x6x7.5

Table 6.4: Air Brake Antilock System Comparisons: Modulator Valve Assembly [6.2].

Manufacturer (System Name)	Modulation Method Used	Mechanical Packaging, in	Disposition of Nonantilock Valve Hardware
AC Spark Plug (Wheel Lock Brake Control)	Single solenoid, on/off cycling	6x6x7.25, includes computer	Replaces existing relay and quick release valves
Berg/Fiat (Electronic Braking System)	Single solenoid plus pneumatic logic	9x6.5x8.5, includes computer	Replaces existing relay and quick release valves
Eaton (Skid Control)	Single solenoid modulating relay valve. Pneumatic logic controlling build rate	6.32x8.30x5.50, includes computer	Replaces existing relay and quick release valves
B. F. Goodrich (Skid Control)	Dual solenoid system: 1/3, 2/3, 100% dump, depending upon stopping conditions	7x6.25x5.63, includes computer	Replaces existing relay and quick release valves
Kelsey-Hayes (Computer Brake Control)	Single solenoid, on/off cycling	4.75x2.25x3.75	Two options: replaces existing valves; installed in addition to existing valves ("down stream")
Rockwell-Standard (Skid-Trol)	Single solenoid, 25 Hz pulse—width modulated	7.17x8.10x7.45, includes computer	Replaces existing relay and quick release valves
Wagner Electric (Skid Control)	Single solenoid, two exhaust and one apply rate	4x3x3	Replaces existing relay and quick release valves

Table 6.5: Air Brake Antilock System Comparisons: System Operating Parameters [6.2].

Manufacturer (System Name)	Preferred Control Mode	Sense Mode (High/Low)	Vehicle Speed Reference	System Cycle Criterion	Operating Speed, min, mph
AC Spark Plug (Wheel Lock Brake Control)	Axle by axle	Combined average	NA	High wheel decelera- tion or differential wheel speeds	5, trucks 10, trucks
Berg/Fiat (Electronic Brak- ing System)	Axle by axle, or tandem	Sense low (high deceleration wheel for release, low accelera- ation wheel for apply)	Chassis mounted "decelerometer"	High wheel deceleration dependent upon vehicle speed and deceleration	4
Eaton (Skid Control)	Axle by axle	Adapts to wheel speed differential	Synthesizes from wheel speed data	High wheel deceleration	6
B. F. Goodrich (Skid Control)	Axle by axle	Sense low (high wheel deceleration)	Internally derived	High wheel deceleration	3-7
Kelsey-Hayes (Computer Brake Control)	Axle by axle	Sense low (slower wheel)'	Computed from suc- cessive wheel speed measurements	High wheel deceleration	3-5
Rockwell-Standard (Skid-Trol)	Axle by axle	Sense low (slower wheel)	Computed internally	High wheel deceleration	3
Wagner Electric (Skid Control)	Axle by axle	Sense low standard; user may specify other modes to be built-in at factory	Computed internally	High wheel deceleration (1.2 g)	6-7

Table 6.6: Air Brake Antilock System Comparisons: System Electrical Parameters [6.2].

Manufacturer (System Name)	Ground	Voltage Limits	Current/Axle at 12 V	Transient Protection	RFI and EMI Protection?
AC Spark Plug (Wheel Lock Brake Control)	+ or −	10-16	0.06 A idle 3.0 A average 6.0 A max	Diode suppressors; no limits specified	Yes. Extensive RFI rejection tests performed
Berg/Fiat (Electronic Braking System)	+ or −	9-16 (fuse blows at 32 V)	0 A idle 0.2 A average 3.0 A max	Zener diode protected; no limits specified	Yes. Filtered inputs to computer
Eaton (Skid Control)	+ or −	9-16	0.2 A idle 0.9 A average 2.2 A max	Protected against typical truck induced transients	Yes
B. F. Goodrich (Skid Control)	+ or −	8-16	0.1 A idle 2.0 A average 4.0 A max	−300 V, 5μs +140 V, 5μs	Yes
Kelsey-Hayes (Computer Brake Control)	+ or −	8-15	0.1 A idle 2.5 A average 3.0 A max	± 300 V, 100μs	Yes Can withstand 100 W of on-board RF transmit power
Rockwell-Standard (Skid-Trol)	+ or −	9-16, protected to 32 V max	0.5 A idle 3.0 A average 5.0 A max	± 300 V, 1 ms	Yes Filtered inputs
Wagner Electric (Skid Control)	+ or −	9-16, protected to 22 V peak	0.05 A idle 2.5 A average 2.5 A max	± 50 V (no time limit specified)	Yes

Table 6.7: Air Brake Antilock System Comparisons: System-Miscellaneous [6.2].

Manufacturers (System Name)	Packages in 3-Axle System	Assembly Line Test Procedures	Field Service Equipment	Operating Temperature Limits, F	Failure Modes Monitored
AC Spark Plug (Wheel Lock Brake Control)	3 total	Special test module, dynamometer test	Special test module	-40-+185	Open or shorted sensor, power loss; open or shorted solenoid
Berg/Fiat (Electronic Braking System)	3 total	Special test module	Special test module	-40-+180	Open sensor lead; open or shorted power circuit, open solenoid
Eaton (Skid Control)	3 valve/logic 1 failure mode 4 total	Spinning wheel for sensor check Completed system self-checking	VOM is adequate. Special tester available	-40-+180	Electrical shorts or opens; high or low input voltages, logic malfunctions
B F Goodrich (Skid Control)	3 total	Special test module being developed	VOM is adequate. Special tester available	-40-+160	Open or shorted sensor; power loss; excessive release time
Kelsey-Hayes (Computer Brake Control)	3 valves* 3 logic* 6 total	Voltmeter for sensors; internal self test plus 1 min road test	VOM is adequate. Special tester available	-40-+160	Open sensor; power loss, open or shorted solenoid
Rockwell-Standard (Skid-Trol)	3 total	Special test module to be available	Special tester available, but not required	-40-+180	Sensor signal loss; power loss; open solenoid; bad power transistor; sustained zero pressure command
Wagner Electric (Skid Control)	3 logic 3 valves 1 failure sense 7 total	Harness continuity checker to be available	Special testers to be available	-40-+180	Open or shorted harness wires; excessive or improper solenoid operation

*Combined mounting available: also available with integrated valve/computer module.

Table 6.8 Specifications of Lucus-Girling Anti-Skid Systems (Skidchek) [6.3]

Parameter	Feature
Wheel Speed Sensor	Variable reluctance magnetic sensing. Self-energized. Stamped, perforated ring fixed to the hub. Automatic adjustment of sensor. Sensor size: 18 mm dia. Sensor weight: 0.115 Kg. Exciter weight: 0.9 Kg.
Processor/ Controller	Solid state analog computer. Integrated circuitry for control and continuous monitoring. Hermetically sealed in an aluminum casing. Suppressed against external interference. Detects damaged perforated ring. Suitable for 12 or 24 Volt electrical supply. Installed on the inside of the chassis frame, adjacent to the appropriate axle. Size: 312 mm x 155 mm x 98 mm. Weight: 2.44 Kg.
Modulator Valve Assembly	Single solenoid on-off cycling. Single unit consisting of a relay valve base, a memory chamber, a solenoid, and a latch valve. Size: 165 mm x 104 mm x 127 mm Weight: 3.75 Kg.
System Operating Parameters	Automatically reinstates anti-lock after wheelspin. Fault shut down uses non-redundant components. Axle control with individual wheel sensing. Detects wheel lock before it happens. Sense low (lower wheel speed). Suitable for use in conjunction with brake load-proportioning valves if fitted to vehicles. System monitor size: 102 mm x 44 mm x 32 mm System monitor weight: 0.148 Kg. System operating range: - Temperature: -40°C to +82°C - Speed: 5 to 75 mph - Surface coefficient: 0.1 to 1.0 - Tire rolling radii: 18 in to 22 in - Power consumption/axle: 0.1 amp/72 watts (max).

Table 6.9 Specifications of Wabco Anti-Lock Braking Systems (ABS) [6.4]

Parameter	Feature
Wheel Speed Sensor	Inductive sensor. Self-energized. Toothed pole wheel rotating with the wheel hub. Automatic adjustment of sensor. Sensor size: 16 mm dia.
Processor/ Controller	LSI fully digital electronics. Sealed unit. Suitable for 24 V ± 25% Installed on the chassis.
Modulator Valve Assembly	Two solenoid on-off cycling. Modulator valves quickly adjust the brake pressure (up to 5 times per second) to prevent wheel locking. Size: 102 mm x 117.5 mm x 59 mm
System Operating Parameters	Fail-safe circuit signals a fault to the driver (red warning light) and then switches off the defective half of the system. Wheel by wheel control. System operating range: - Temperature: -40°C to +82°C - Speed: 5 Km/h (min) - Surface coefficient: 0.1 to 1.0

Table 6.10 Specifications of Bosch Anti-Skid Systems (ABS) [6.1]

Parameter	Feature
Wheel Speed Sensor	Variable reluctance magnetic sensing. Self-energized. A.C. voltage excitation. A 100 tooth sensor ring fitted to the wheel.
Processor/ Controller	Micro-processor incorporating special processor components-groups.
Modulator Valve Assembly	Three solenoid operated, 2 channel pressure-control valve. One pressure-control valve is required for each axle. In case of tandem axles, the control of the service-brake cylinders can be combined on each side so that only one pressure-control valve is required for both axles.
System Operating Parameters	Vehicle reference speed is obtained from each pair of diagonally opposite wheels. Sense low. Automatic fault shut down. Warning lamp to driver when ABS switched-off.

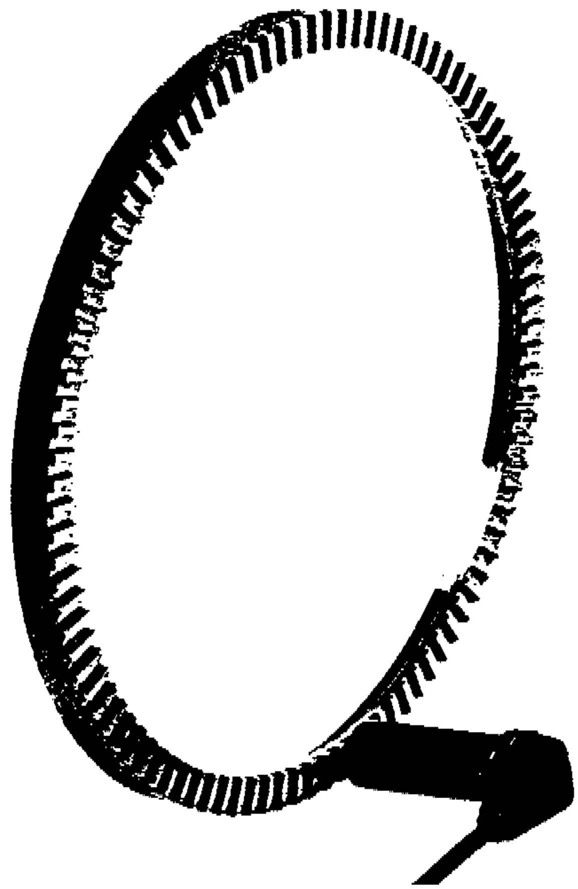


Figure 6.3: Typical Slotted Rotor/Axial Pickup Wheel Speed Sensor [6.1].

- Radially oriented pickup paired with a toothed rotor (Figure 6.4).

- Multitoothed rotor and stator pair operating concentrically (Figure 6.5).

Some ABS systems have a discrete vehicle speed reference were a longitudinally oriented accelerometer is used to obtain a deceleration signal which is integrated to determine vehicle speed. Most systems function by comparing a computed wheel deceleration value to a reference value stored in each system's memory.

Each of the ABS systems depend upon a logic center (processor) to determine if and when a brake release should be initiated to avert wheel lockup. For example, a comparison is made between the value from a wheel speed sensor and stored reference value and this comparison prompts the processor that a wheel lockup is about to occur which will then force the valve in the ABS system to cause a brake release.

The modulator control valve accepts signals from a processor and this signal is information about whether a wheel lockup will occur. The modulator valve then will perform the required brake modulation on the brake chambers. For example, if the wheel is about to lockup then the processor after comparison will sent a signal to the valve which will reduce the air pressure in the brake chamber. This is done by exhausting the air in the chamber at the same time blocking the inlet air line and then reestablishing air pressure after wheel lockup has been averted.

The failure sensing system has no active role in the ABS system but only provides failure indication at a level much lower than "total electrical failure".

Sensing systems can be in many form such as:

- A subsystem included in the prime skid control computer module for each axle.

- A separate "monitor module" wired to several axle systems on a vehicle and capable of sensing and indicating failure of any or all axles.

For anti-skid systems to become standard equipment on air braked vehicles, they will require new designs or modifications for Heavy Vehicles:

- Finding space in which to mount additional components to the chassis.

- Protection of the ABS system from the operating environment the truck undergoes.

- Obtaining components for the ABS that are damaged or defective for their particular ABS system installed.

Although, there are variations between manufacturers, the basic

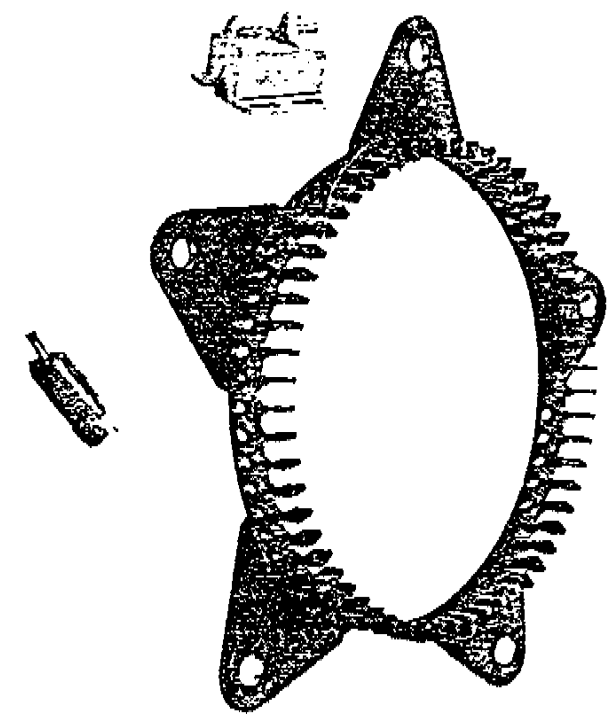


Figure 6.4: Wheel Speed Sensor with Radially-Oriented Pickup [6.2].

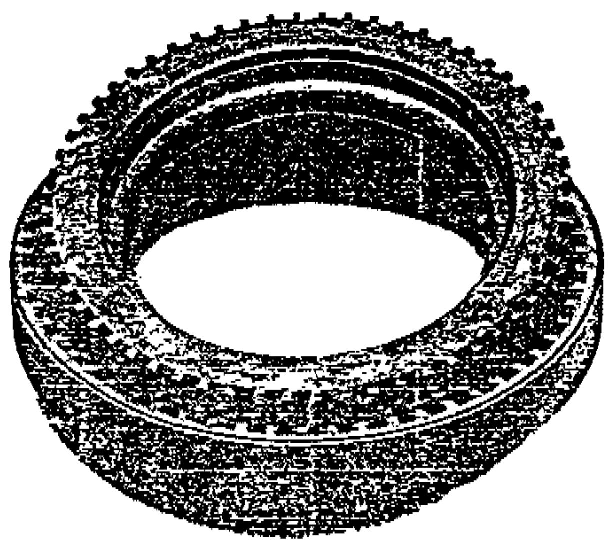


Figure 6.5: Wheel Speed Sensor with Multi-toothed Rotor and Stator [6.2].

concept of operation remains the same. These systems can be made available at the time of vehicle manufacture or retrofitted to give single axle or tandem axle control, in either or both of the tractor and semitrailer. Since they operate off the standard electrical system and air supply, they will function in any vehicle combination; for example, trailer with anti-lock and a tractor without.

To meet the FMVSS 121 regulation with anti-lock systems, more aggressive brake linings and larger air reservoirs are required. The requirement for larger air reservoirs is due to the increased demand for air during the air brake modulation and to meet the air brake timing specification. Because of this and the complexity of retrofitting, anti-lock systems are generally only purchased for new equipment.

6.4 TANDEM ANTI-LOCK BRAKING SYSTEM [6.5]

Tandem control differs from other systems by having a singular modulator/controller for both axles. Several options may exist for the wheel sensing such as:

- Four Wheel Sensing.
- Two Wheel Sensing on One axle.
- Two Wheel Sensing with a Sensor on each Axle.
- Remote Sensing of Driveline.

Bendix heavy vehicle systems have investigated this Tandem ABS by analyzing two popular types of truck suspensions: the four spring and walking beam suspension. In their analysis of Tandem ABS, they used the four wheel sensing and the two wheel sensing on one axle as the best choice for wheel sensing. The four wheel sensing because at incipient lock of any one wheel will modulate chamber pressure at all four wheels of the tandem pair. While the two-wheel sensing on one axle is obtained by placing the wheel sensor at the axle pair which is determined to be more prone to lock than the other axle (Figure 6.6).

To determine which axle was closer to locking, a reference parameter known as the wheel lock coefficient (WLC) was established. The WLC is directly proportional to brake effectiveness and chamber pressure and inversely proportional to vehicle loading and tire rolling radius. An increased WLC was investigated and represents a greater tendency to produce a lockup of the wheels.

Bendix investigated the behaviour of tandem suspensions under braking conditions to obtain a Tandem Antilock System by devising a computer model to establish which axle of a tandem was most likely to go to lock under any given set of circumstances.

After reviewing the results of the computer models and vehicle testing, some conclusions are drawn.

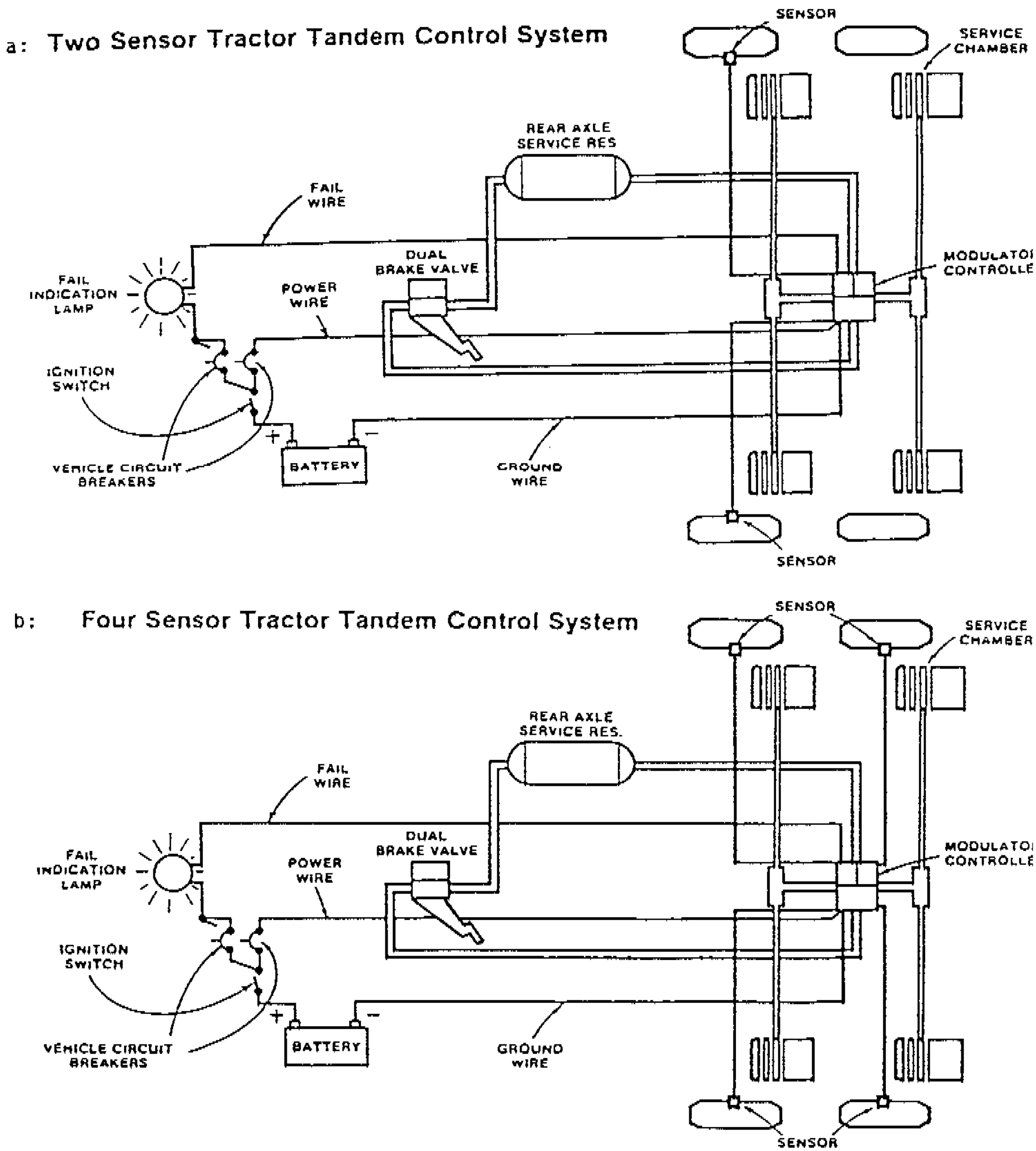


Figure 6.6: Tractor Tandem Anti-Lock System [6.5].

Vehicles equipped with four spring suspensions generally are, with some reservation, adaptable to the use of two sensor tandem systems. When applying a two sensor system, the vehicle manufacturer must consider the possible increase in stopping distance compared to individual axle control, take into account the loss in braking efficiency caused by a high gain wheel on the sensed axle, and must consider frame deflection under dynamic load transfer.

Brake chamber pressure traces taken when testing axle control systems indicate that walking beam suspension equipped vehicles tend to have closely balanced axle loadings during braking and thus are more compatible with four sensor tandem systems. Vehicle test results taken when using a two sensor system on the walking beam vehicles were successful in preventing wheel locks only when the brakes were carefully kept in balance. Introduction of an unsensed high gain wheel frequently resulted in high-speed wheel locks.

6.5 STATE-OF-THE-ART REVIEW OF RESEARCH RESULTS ON ANTI-SKID SYSTEMS

A reasonable amount of research has been carried out in the past to study the braking performance of heavy vehicles fitted with anti-lock devices. Tables (5.1) to (5.4) show some of the relevant references in this area.

First attempt at anti-skid systems was in 1948 for the aircraft industry by the Hydro-Air Division of Crane Co. The system was known as HYTROL. It was then acquired by Kelsey-Hayes Co. and modified for heavy vehicles.

Latvala and et al. [6.6] presented an adaptive braking system. The theory of operation and the specifications of the system and performance under several typical conditions were discussed.

Harned and et al. [6.7] discussed the measurement of tire brake force characteristics as related to anti-lock brake system design. The correlation of the measured data was established with anti-lock system performance. Experimentally measured slip curves were given for a large number of tire/road pairings. These measurements cover a wide range of commercial tire types on dry and wet road surfaces and glare ice. It was shown how wet road characteristics are affected by road construction, water cover depth, and tread wear. The measuring system used to obtain these data were described.

Hickner and et al. [6.8] used a 17 degree-of-freedom hybrid computer simulation to investigate 4-wheel adaptive braking system (ABS). The derivation and verification of the ABS model, the form of the integrated vehicle/ABS model, and future plans for validation and utilization of the integrated hybrid simulation were presented.

Rixmann [6.9] conducted full-scale field tests with a MAN truck-trailer vehicle having individual anti-skid control on all wheels of the vehicle combination. The test were carried out on a very wet road surface during braking in a straight line and in a turn.

Fritzsche and Reinecke [6.10] presented the development of an anti-skid control system for commercial air-braked vehicles. Field testing were also conducted.

Muller, Kling, and Schneider [6.11] compared the handling behaviour of commercial vehicle trains during emergency braking with and without anti-lock devices.

Guntur [6.12] presented some design considerations of adaptive brake control systems. In this paper, some of the design aspects of adaptive brake control systems were studied, especially the interaction of the software with the hardware of the system. Two modes of operation of the brake pressure modulator were considered; the software changes are effected to modify further the mode of operation of the system. The effect of the rate of rise of wheel cylinder pressure and the effect of rate of decay of pressure on the effectiveness and the maximum wheel slip in the first cycle had been studied. The hardware and the software were so modified as to give satisfactory performance of the wheel and the vehicle for four different forward speeds and for three different road conditions.

Grimm [6.13] investigated wheel lock control braking system. This presentation discussed a system for preventing continuous rear wheel lock-up of an automobile during maximum braking stops. Included was a description of the control system components, tire and road characteristics, brake and vehicle dynamics, and an analysis leading to the requirements for optimum control.

Grimm and et al. [6.14] evaluated a vehicle installed with an anti-lock device. A real-time hybrid computer simulation which interfaced with an air brake truck was described. The simulation interconnects with the wheel lock electronic controller-modulator hardware and air brake system of a multiple axle truck to provide a laboratory tool for simulating vehicle braking performance. This technique provided controllable vehicle and road characteristics for evaluating the actual wheel lock and vehicle pneumatic system hardware. A set of comparative data was given and the merits of the simulation technique were discussed.

Cardon and et al. [6.14] presented the development and evaluation of anti-lock brake systems. Six anti-lock system configurations involving individual wheel and axle control were discussed. Also discussed were techniques for evaluating the performance of anti-lock systems; included are straight line braking, the use of a split coefficient surface and braking in a turn. The results of computer simulation and vehicle tests conducted to evaluate the performance of the various anti-lock system configurations were presented. It was concluded that the best anti-lock system configuration for a particular vehicle requires a trade off among vehicle design characteristics, desired level of braking, and vehicle handling performance and cost.

Adams and Spencer [6.16] described practical aspects of testing anti-lock systems on commercial vehicles.

Fancher and Macadam [6.17] analyzed using computer simulation the braking performance of commercial vehicles equipped with anti-lock systems. A technique for measuring the pressure modulation characteristics of anti-lock systems was presented. A subprogram for synthesizing anti-lock system performance within the framework of a large-scale digital computer simulation of the braking of trucks and tractor-semitrailer vehicles was described. Comparisons between measured and calculated vehicle performance were made to indicate the validity of the simulation results. Sensitivity of the braking performance of a truck to tire, brake, suspension, and anti-lock properties were discussed.

Lam and et al. [6.18] evaluated the braking performance of a tractor-semitrailer equipped with two different types of anti-lock systems. A digital computer simulation was used. A commercially available anti-lock system and a proposed system were investigated. The merits and disadvantages of the proposed system were also examined. Based on the results of the simulation, guiding principles for the development of control logic of anti-lock brake systems were suggested.

Uffelmann [6.19, 6.20, 6.21] studied the directional behaviour during braking of four-axle truck-trailer vehicles equipped with anti-lock devices. The computer model included the air-brake model and a model of an anti-locking device. The dead time, the throttling of the lines and brake valves, and also the hysteresis of the cam brakes were also modelled.

Srinivasa and et al. [6.22] evaluated the performance of anti-lock brake systems using laboratory simulation techniques. Two commercially available anti-lock systems were tested and a comparison of their performance were presented.

Snelgrove, Billing, and Choi [6.23] presented the performance evaluation of several jackknife control devices. The Kelsey-Hayes skid control system was used in the evaluation. The results indicated that the wheel anti-lock system provided the best means of maintaining vehicle control under severe adverse braking conditions. The vehicle stayed within a lane during braking and the jackknifing phenomenon did not occur.

6.6 CONCLUSIONS

In this chapter, anti-lock/anti-skid brake systems are discussed. A brief outline on the operating principle and major elements of anti-skid systems are presented. A detailed discussion on the available hardware in North America and Europe is made. A comparison of these different hardware units is presented in a tabular form. A State-of-the-art review on the available research results is described with salient points.

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CHAPTER 7

FUTURE SYSTEMS

There are a number of new developments for the future systems [7.1]. They are:

1. Compressor Clutch
2. Long Stroke Brake Chambers
3. Automatic Slack Adjusters
4. Large Front Brakes
5. Disc Brakes
6. Brake Proportioning
7. Simplified Controls
8. Retarders
9. Antilock Braking

During a normal highway operation, the compressor pumps less than 10% of the time. An unloaded compressor uses only 0.9 to 1.8 H.P. Because of the above facts, the compressor can be used for other secondary functions without affecting its primary role of air compression. One such secondary role is its use in compressor clutch. With compressor clutch, the compressor life increases almost 100% and the oil consumption reduces significantly.

Long stroke brake chambers have a definite advantage of providing greater margin of safety.

Large front brakes will provide improved stability and adhesion utilization.

Disc brakes provide better side-to-side brake balance and thus maintain stability, uniform brake torque, little fade, excellent water recovery and simple reline.

Brake proportioning/load proportioning system reduces the braking effort on the rear axle(s) of a tractor in the bobtail mode, thereby, reducing the tendency for lock-up. This will reduce the stopping distance, and improve vehicle stability and safety. Figure 7.1 shows a typical tractor brake proportioning system. The inversion valve picks up the signal from the supply line indicating whether the tractor is hauling a trailer. If there is no trailer, the supply line has no air pressure, the inversion valve senses that fact and sends a positive signal to the rear proportioning valve which cuts the pressure to the rear brakes to 30 to 50% of the front axle pressure.

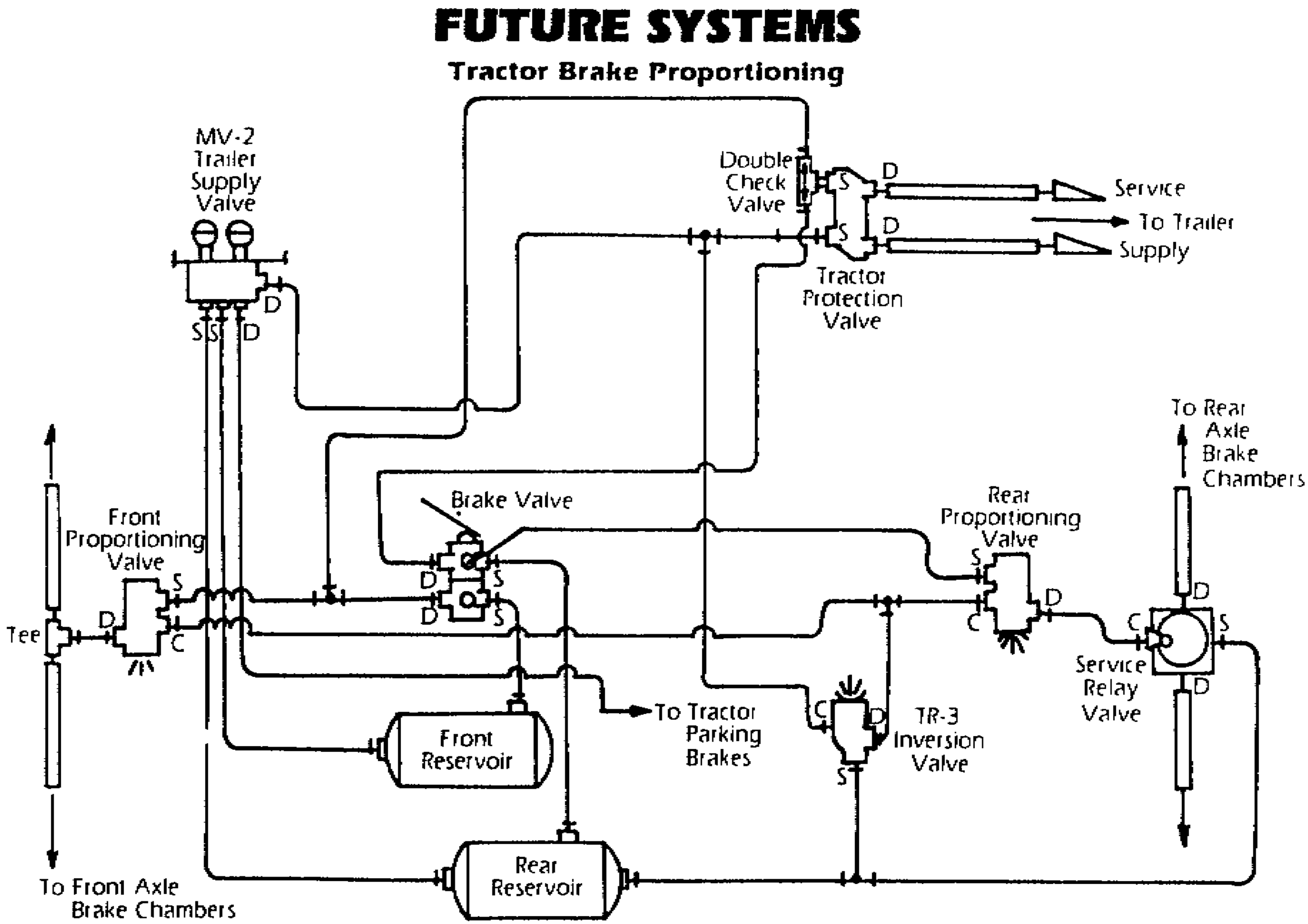

Figure 7.1: Tractor Brake Proportioning System [7.1].

The system is fully automatic thus insuring that the improved stability and braking capabilities will be available when needed. Figures 7.2, and 7.3, respectively, show the improvement in stopping ability using proportioned brakes in low and high coefficient surfaces.

Various types of retarders - engine brakes, exhaust brakes, hydrodynamic and electric retarders, are available. Demands depend on the region - flat areas have less demand. Disc brakes may be an alternative in mildly hilly areas. Use of retarders, saves wear on brakes and thus reduces cost. Some of the concerns in the use of retarders are:

- Stability;
- Initial cost;
- Weight;
- Engine life; and
- Noise.

Future systems trend will include more electronics:

- Increasingly on dashboard displays.
- On board diagnostic capability with output from sensors to monitor:
  - Rates of change of pressure buildup in the air brake system vs engine speed (check for compressor performance and acceptable system leakage).
  - Vibration on the engine to detect bearing wear or compression loss.
- Engine and transmission with electronic control.
- Electronic accelerator pedal eliminating mechanical linkage between the pedal and an electronically controlled engine.
- Use of on-board recording systems for maintenance and management.
- Antilock braking:
  - Seriously considered; acceptance depends on economy and safety.

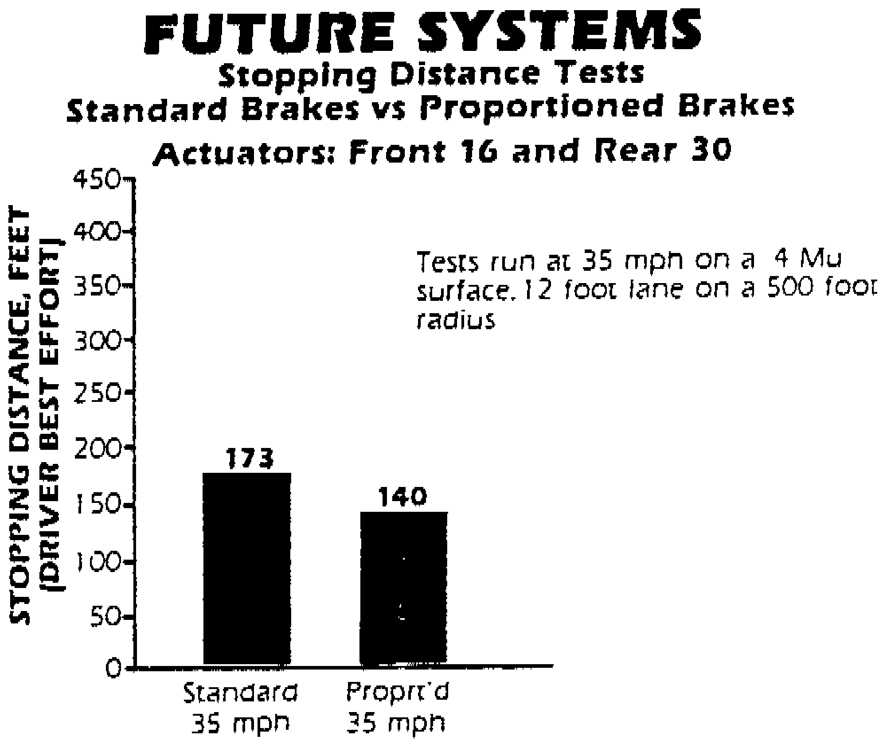

Figure 7.2: Stopping Distance of Standard Brakes Vs Proportioned Brakes [7.1].

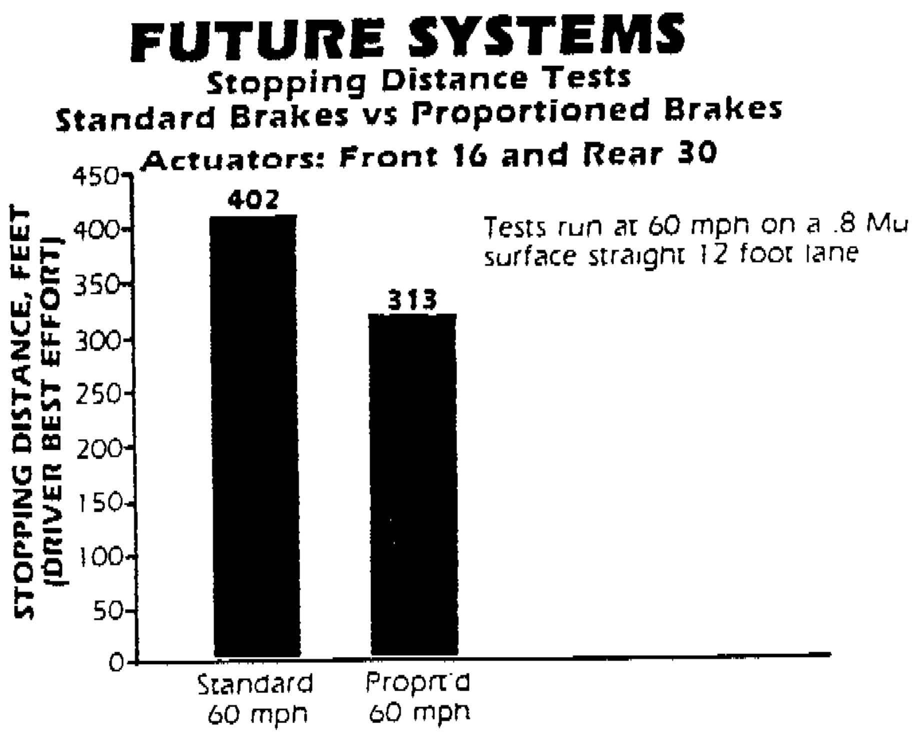

Figure 7.3: Stopping Distance of Standard Brakes Vs Proportioned Brakes [7.1].

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