An Investigation of Brake Application Delays in Australian Train Brake Systems

By

Ian Ripley

Submitted in fulfilment of the requirements
for the degree of Master of Engineering

at the

**James Goldston Faculty of
Engineering and Physical Systems**

Central Queensland University

Rockhampton, Qld.

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Abstract

An investigation of brake application delays in Australian train brake systems began with a literature review of pneumatic train braking systems. Data located in the review gave examples of brake application delays of pre 1990 designs from the U.K., India and North America. Information on application delays on later Australian designs was scarce. Reading of literature has shown a difference between the Australian and North American control valves in the way the propagation of the pressure reduction rate in the brake pipe is maintained. Control valves of the North American style allow the brake pipe air to be connected for a short time to a small cavity or quick service volume of each valve. The quick service volume is then released to atmosphere. The action of exhausting a small amount of air from the brake pipe helps to ensure a propagation of an adequate pressure reduction rate as it travels to the next valve. Australian control valves rely on the ratio of the volume of brake pipe between control valves and the size of the quick service volume or 'bulb' to ensure the propagation of an adequate pressure reduction as it travels to the next valve. The air in a bulb of an Australian valve is not expelled to atmosphere until a brake release is made.

The research explored possible reductions in application delays by utilizing an experimental pipe test rack that included 4 control valves and 120 meters of brake pipe. Experiments with different configurations of exhaust orifices or chokes, valves and branch pipe lengths that supplied the valves gave a record acquired by data acquisition of the timing of each valve and the local pressure drop from a valve or each valve for comparison.

Experiments with exhaust chokes that gave a reduction drop rate in the brake pipe that approached the minimum required to operate a control valve resulted in instability of the application operation of the control valve. The quick service volume of different sizes was included in the experiments to give comparisons in the propagation of the pressure reduction toward the end of a long train. Further increases into the size of the bulb of a control valve to enhance the propagation features toward the end of a long train are discussed. The branch pipe with different diameters from 12 mm to 20 mm and lengths from 160 mm to 800 mm when fitted to an adaptor pipe bracket were investigated and results show that larger diameters gave larger gulps in the brake pipe.

Other components that were studied included the pipe bracket that is fitted on some control valves. The pipe bracket and isolation cock was found to add 282 mm of additional length to the air path and while not changing the operation of the valve, the results showed a smaller drop in local pressure in the brake pipe to assist the pressure reduction rate than shown in valves without pipe brackets.

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Glossary of Terms

Abbreviations

AAR	Association of American Railroads
AAV	Accelerated Application Valve
AB	North American Westinghouse triple valve (1936)
AF2	Australian freight triple valve (1960)
ABD	North American Westinghouse triple valve (1965)
ABDW	North American Westinghouse triple valve (1976)
ABDX	North American Westinghouse triple valve (1989)
BCP	Brake Cylinder Pressure
BP	Brake Pipe
DB-60	North American triple valve by New York Air Brake Company
ES	United Kingdom triple valve by Davies and Metcalf
EX	Normal exhaust position, grade control valve
IP	Intermediate pressure position, grade control valve
ITV	Australian improved triple valve (1950)
HP	High pressure position, grade control valve
K	North American triple valve (1902)
KE	European triple valve by Knorr-Bremese
kPa	Kilopascal SI unit of pressure.
I.D	Internal diameter
NB	Nominal bore (pipe size)
m/s	Velocity, meters per second
P.B.	Pipe Bracket
PSI	Pounds per square inch unit of pressure
PSIA	Pounds per square inch atmosphere
PSIG	Pounds per square inch gauge
SW4	Sab Wabco distributor
Q6	Queensland rail permanent restrictor grade control valve
Q.A	Quick Action
Q.S	Quick Service bulb
Q.S.1	Preliminary Quick Service bulb
Q.S.2	Secondary quick service bulb
UK	United Kingdom
UIC	International Union of Railways
VSH	Queensland Rail 100t mineral hopper wagon
VSAL	Queensland Rail 104t lead tandem mineral hopper wagon
VSAS	Queensland Rail 104t slave tandem mineral hopper wagon
WF	Australian Westinghouse ‘W’ series diaphragm and poppet valve triple valve (1967)

Glossary of functions

Accelerated Release	Each control valve connects a reservoir of initial BP air to the BP in a brake release application to assist BP build up
Accelerated Application Valve	Vents air from the BP whenever a pressure reduction is initiated
Accelerated release reservoir	Supplies air to the brake pipe on a brake release
Auxiliary reservoir	Holds brake pipe air for triple valve differential operation
Bulb	Term for control valve quick service volume
Dummy brake cylinder volume	Stores air in a brake application
End-of-Train	Electronic operated valve to exhaust the brake pipe air at the rear of a train
Gulp	Term for the sharp local drop of air pressure seen in a brake pipe in a control valve operation
Inshot	Allows an initial quick build up of BCP, then with chokes restricts auxiliary air to the brake cylinders
Supplementary reservoir	Used in conjunction with the dummy brake cylinder to supply air to the brake cylinders
Locomotive brake controller	Controls the reduction and charge of the train brake pipe
Local gulp	Term for drop of air pressure seen at the control valve
Quick action	Assists reducing BP in emergency applications by connecting the BP to a small chamber
Quick service	Connects the brake pipe air to the brake cylinder or to atmosphere so each control valve assists the transmission of brake pipe air reduction in a service application
Release ensuring	Connects the auxiliary to atmosphere only on cars where the pressure differential in the BP exceeds the pressure in the diaphragm in the release ensuring valve
Retarded recharge	Restricts flow into the emergency and auxiliary reservoirs on a brake release application
Reduction ensuring	Connects air from either the quick service bulb or BP to the brake cylinder and ensures a sufficient initial reduction of BP is produced through out the train.

Publications list

Unpublished Industry Report (Confidential)

Cole. C and Ripley. I *Investigation of Applications Delays in Australian Train Brake Systems* October 2004

Unpublished Industry Report (Confidential)

Ripley. I *Development of the Railway Air Brake Triple Valve in America and Australia* June 2003

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Further thanks must go to my brother Robert and my sister Nanette for their personal help over the period of this project.

Declaration

I declare to the best of my knowledge this thesis does not contain any material previously published or written by another person except where due reference is made in the text. The contents of this thesis have not been included in any other work submitted by the author for another degree or diploma at any other tertiary institution.

Signed: ……………………   Dated:………………………

1. Introduction

1.1 Aims and Objectives

The aim of this project is to research the possible ways to decrease the brake pipe pressure reduction propagation times. It is well known that the longer the train has become the more problems there are with the operation of the control valves especially on the rear of a head end train. The addition of distributed locomotive power and/or End-of-Train devices has overcome some of the performance issues of the brake system for some rollingstock operators. The number of cars behind a locomotive or between locomotives and End-of-Train systems is determined by the performance of the propagation rate of the reduction in the brake pipe.

Methods and modifications for reducing the propagation times in Australian systems are to be evaluated.

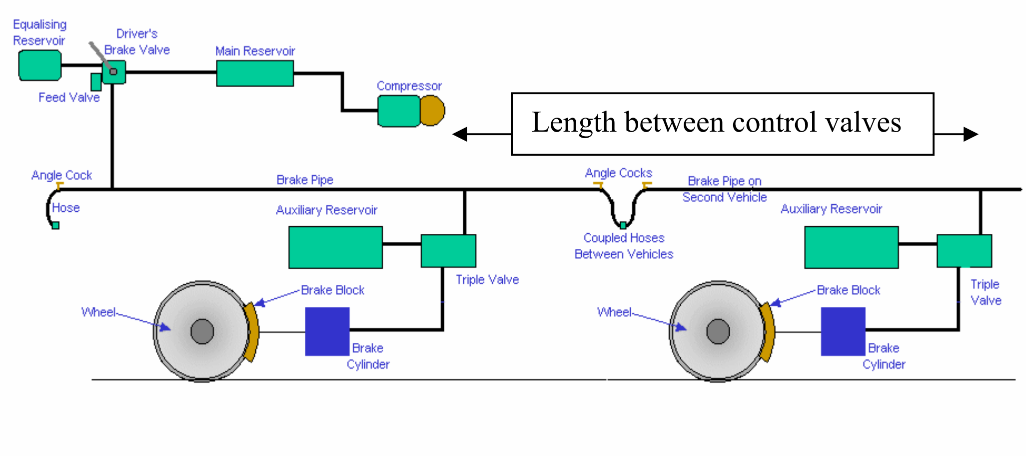


Figure 1: Schematic of a brake system.

This research thesis is limited to the Westinghouse Australian brake valve systems predominantly used within Australia. The objective is the investigation is to establish performance in terms of the propagation of brake pipe pressure reduction and to examine the size of the local reduction in brake pipe pressure in various existing systems. Modifications to parts of the standard Westinghouse vehicle control valve system are to be examined that may provide performance improvements.

1.2 Outline of the thesis

The arrangement of this thesis is as follows:

- Chapter 1 is a brief introduction with an overview of each chapter.
- Chapter 2 is a literature review of past and present American and Australian systems.
- Chapter 3 includes the theory on pipe equations, sonic velocity and pressure loss of pipes and also fittings Chapter 3 also includes the details of the equipment used in the experiments.
- Chapter 4 presents the five laboratory test programs completed on the test rig with results and discussions on each.
- Chapter 5 gives a general discussion exploring issues that have arisen from the previous Chapter 4 in more depth.
- Chapter 6 provides on the conclusions.
- Chapter 7 gives recommendations for future areas of research.

2. Literature Review

The methods of reducing brake application delays in triple valves and propagation rates in brake pipe systems on trains, have improved many times since the invention of the first simple triple valve with the patent granted to George Westinghouse in 1873. This review will look at the development of the triple valve and the improvements that have been made to the triple valve and brake pipe system.

2.1 Brake Systems

From the references in his book Railway Safety, Brakes, Macfarlane outlines three of the most common forms of automatic train brake (Macfarlane 2000).

- Automatic air brake direct release
- Automatic air brake gradable- release
- Vacuum automatic air brake.

2.1.1 The automatic air brake with direct release system

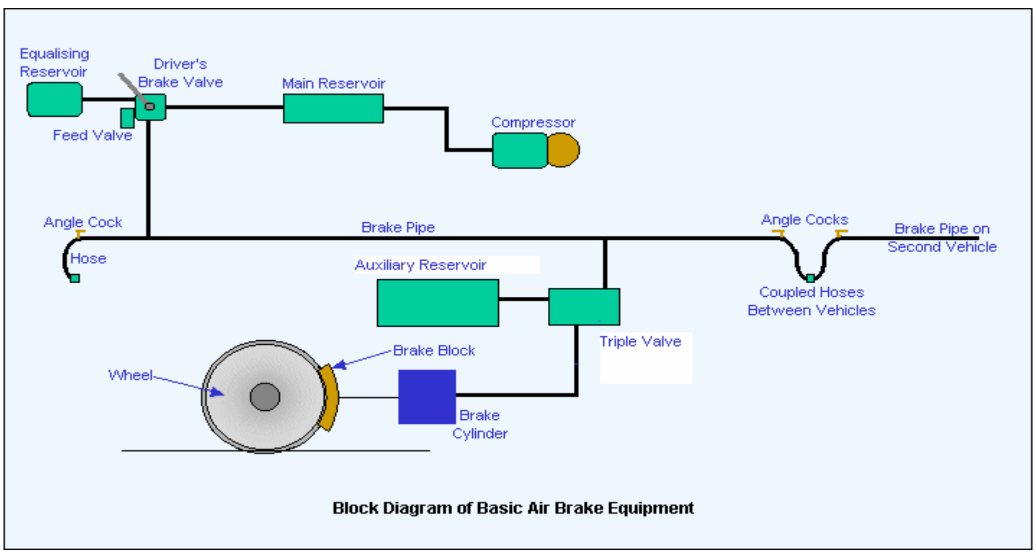


Figure 2: Automatic air brake direct release system.

Air brakes on trains utilise pneumatic cylinders to provide the force to push brake shoes onto wheels. The components in Figure 2 comprise a supply pipe, which provides compressed air along a train through a brake pipe. The brake pipe is connected between wagons by a flexible hose. Connected to the brake pipe at each wagon is a triple valve, this valve is also connected to the auxiliary reservoir and brake cylinder. The triple valve can be constructed as either a slide valve or diaphragm system, which responds to the difference between the pressures in the brake pipe and in the auxiliary reservoir. This pressure difference causes the triple valve to either:

1. Recharge the Auxiliary Reservoir and open the brake cylinder exhaust to atmosphere. [Brake release]
2. Close the brake cylinder exhaust and allow the Auxiliary reservoir air to feed into the brake cylinder. [Brake application]

2.1.2 Automatic air brake graduated - release

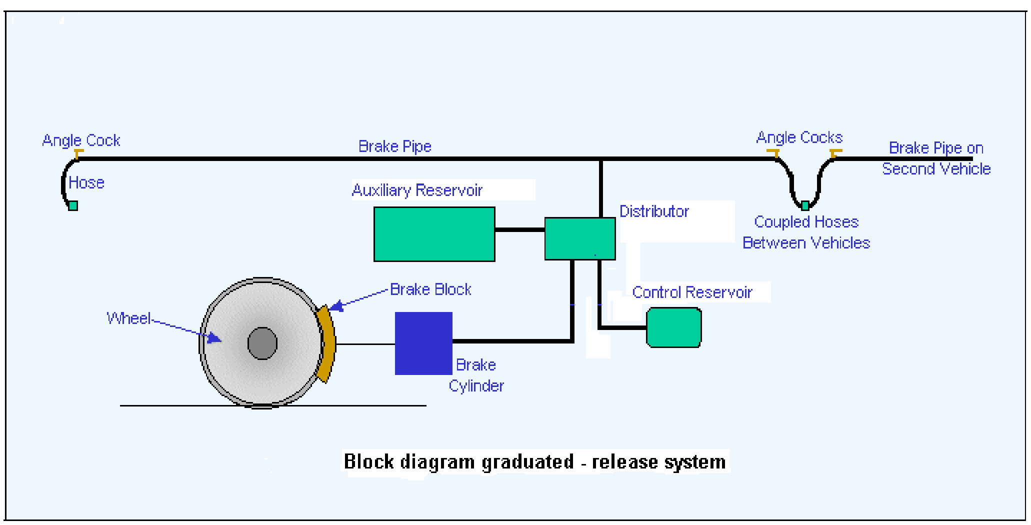


Figure 3: Automatic air brake graduated - release system.

A graduated - release function can be added to the triple valve system. Valves equipped with graduable release are used mainly in European countries that have many descending grades. In a graduated release system, as shown in Figure 3 compressed air is again provided along a train through the brake pipe. Connected to the brake pipe at each wagon is a distributor, which is piped to an auxiliary reservoir and a control reservoir. The distributor uses the pressure difference in control reservoir to the brake pipe as a reference to control the brake cylinder pressure during a release. The change of pressure in the brake pipe causes the triple valve to either:

1. Recharge the auxiliary reservoir and open the brake cylinder exhaust to atmosphere and allow a partial release followed by a holding of the lower application rate by using the control reservoir as a reference point. [Brake release]
2. Close the brake cylinder exhaust and allow the Auxiliary reservoir air to feed into the brake cylinder. [Brake application]
3. Charge the control reservoir

2.1.3 Vacuum automatic brake.

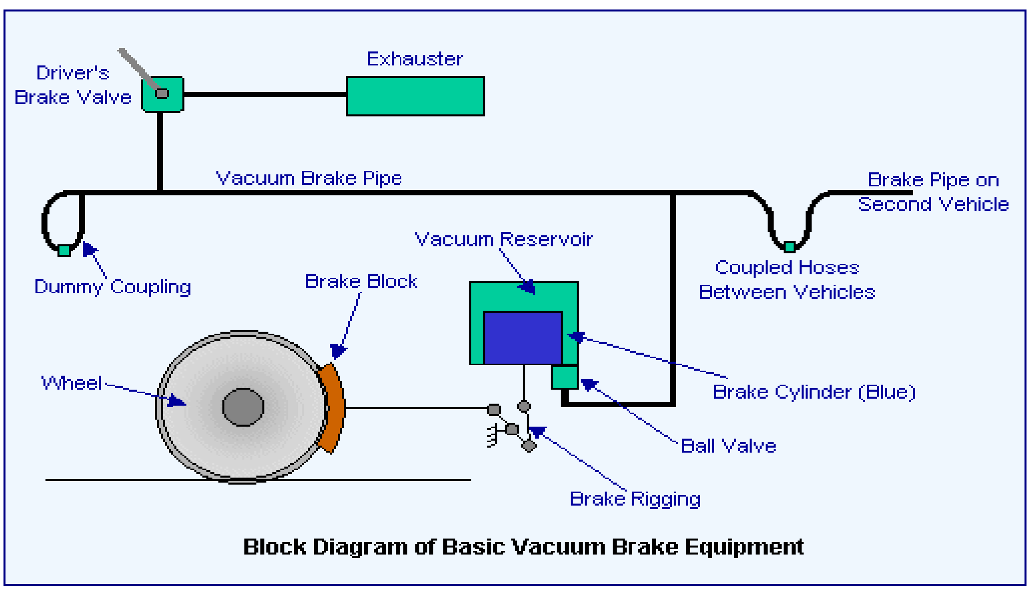


Figure 4: Vacuum automatic air brake system.

The vacuum brake system is controlled through a brake pipe connecting a brake valve in the driver's cabin with braking equipment on every vehicle. The vacuum brake equipment is shown in Figure 4 where each wagon has a combined brake cylinder and vacuum reservoir. The movement of the brake cylinder piston is dependant on the condition of a vacuum created in the brake pipe by an exhauster. The exhauster removes air from the brake pipe to create a partial vacuum. At maximum vacuum, or when there is no pressure difference on both sides of the brake cylinder piston, the brakes are released. When there is no vacuum, i.e. normal atmospheric pressure in the brake pipe, the brake cylinder piston has a pressure difference of atmospheric on one side and a vacuum on the other and the brake is fully applied. These systems have both:

1. A graduable application function
2. A graduable release function

2.2 Valve Terminology

2.2.1 Triple Valve

The name ‘triple valve’ came from the three valves in the chambers in the original design. The three chambers or compartments had internally, three poppet valves A, B, and C and being connected by the same shaft, all the valves are synchronised in operation. As shown on Figure 5 with this triple-valve arrangement, through an open valve (A) the auxiliary reservoir is charged with compressed air from the brake pipe, and as valve (C) is open and valve (B) is closed the brakes are either released or kept released. When air is discharged from the brake pipe, valves (A) and (C) close and the brakes are applied by a flow of air from the auxiliary reservoir through open valve (B) (Westinghouse 1873). The pipes that were connected to the triple valve are the brake pipe, the auxiliary reservoir pipe and the brake cylinder pipe. Reference is made to a 'fourth connection', which was an exhaust to atmosphere from the brake cylinder through the triple valve (Macfarlane 2000).

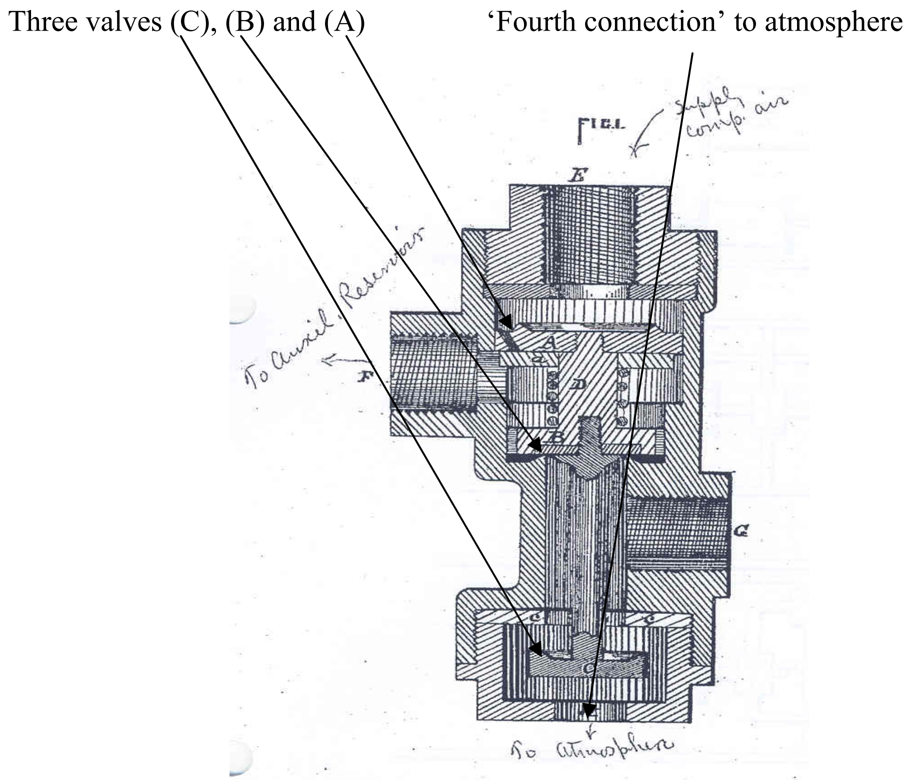


Figure 5: First triple-valve valve of Westinghouse. (Reproduced in whole from patent No. 144006 Westinghouse 1873)

2.2.2 Service Portion and Emergency Portions.

In America (Farmer 1936) was granted a patent for the invention of a replacement of the single triple valve with a new combined valve. The new combined valve shown in Figure 6 consisted of two valve portions bolted to a common pipe bracket, these valve portions were named a Service Portion and an Emergency Portion with the Pipe Bracket in between. A later patent (Hewitt 1939) relates to the new combined valve of Farmer's as the 'AB' brake equipment. This valve and later variations of the valve is the most widely used valve in the North American market.

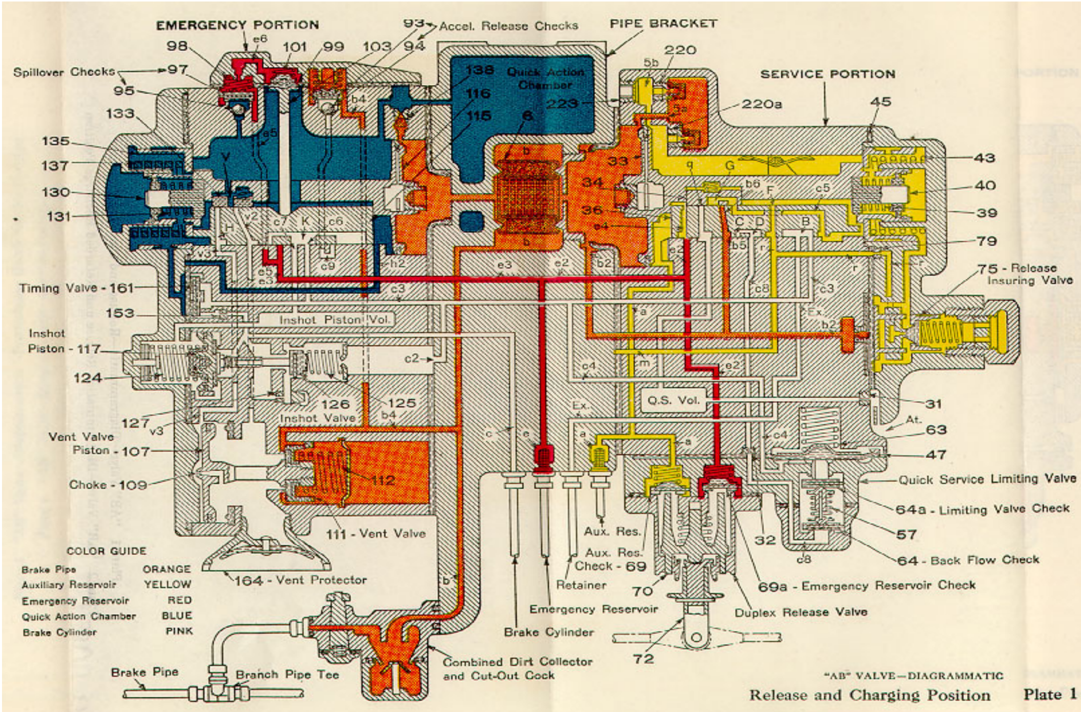


Figure 6: "AB" triple valve. (Reproduced in whole from Westinghouse Air Brake Company 1945)

2.2.3 Distributor

In other regions in the world, which included Canada, England and Europe, the Service portion or triple valve is called a distributor. The name 'distributor' first appeared in Europe, when the European railways wanted a valve that would have a graduable release feature. The addition of a graduable release feature into a triple valve had made it more complicated than the traditional direct release Westinghouse triple valve. The triple valve (Macfarlane 2000) became known as a distributor in those countries that used the automatic gradable release system.

2.2.4 Control valve

The code of practice for the defined interstate rail network in Australia has proposed, in a draft of Volume 5 Part 1, a standard for terminology for rollingstock. The section 4.4 names a triple valve as a control valve (Australia 2002).

2.3 The Development of the Railway Air Brake

2.3.1 First Air Brakes

A patent to George Westinghouse was issued on April 13th 1869 in the United States of America, for a “straight-air” brake (Westinghouse 1869). It consisted of an air-compressing pump, operated by steam from the locomotive boiler, by which air was compressed into a reservoir. The locomotive reservoir was known as the main reservoir. The main reservoir was connected to a train pipe or brake pipe via a control valve. The control valve was located in the operator’s cabin as shown in Figure 7 and regulated the supply of air to the brake pipe. The compressed air was conveyed down the cars by piping, and flexible hoses with metal couplings, making the brake pipe continuous from the locomotive to the last car. The brake pipe of each car had a branch pipe that was connected to the end of a brake cylinder. The brake cylinder contained a piston and the stem of the piston was connected with the brake rigging of the car. When the brakes were to be applied, the control valve in the cabin is opened. The stored compressed air in the main reservoir was allowed to flow into the train-pipes and then to the brake cylinders. To release the brakes, the control valve in the cabin is moved to the closed position of the valve. The compressed air in the brake pipe flowed forward along the train pipe to the escape port of the control valve in the operator’s cabin, and then into the atmosphere. The release of the compressed air in the brake pipe allowed the pistons of the brake cylinders to be retracted by means of return springs, releasing the brake shoes.

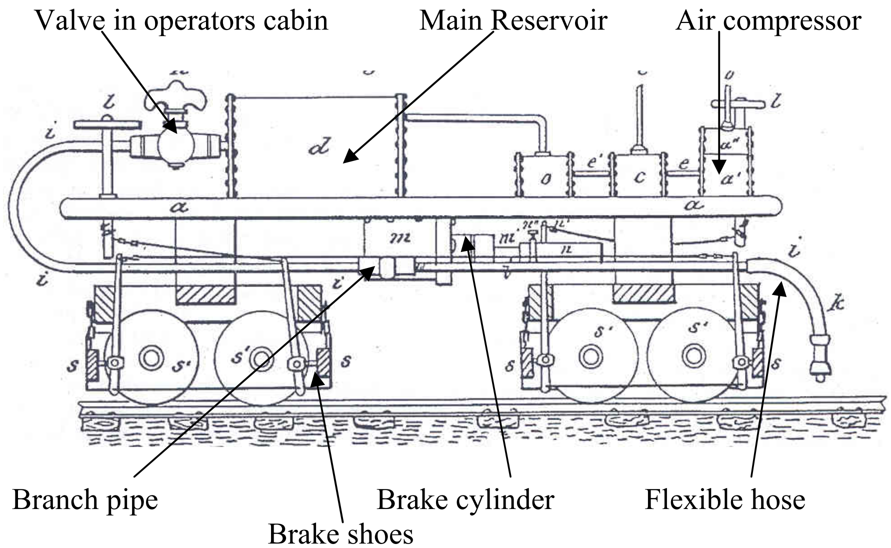


Figure 7: The Westinghouse straight air brake. (Westinghouse 1869)

Improvements to the brake system led to the automatic air brake system (Westinghouse 1872). The first improvement in this automatic system was to provide an auxiliary reservoir on each wagon. The automatic brake system employed a pressurised brake pipe and the auxiliary reservoir was filled through a connection from a valve (as shown in Figure 5) on each wagon, when the brake pipe was being charged with air. The second improvement was that this same valve also opened ports to allow the brake cylinder air to escape to atmosphere, when the brake pipe was being charged with air giving faster brake release. When the brake pipe pressure was lowered or a disconnection of the brake pipe occurred, the opening of other ports of the valve connected the auxiliary reservoir to the brake cylinder, which then applied the brake pad onto the wheel. [Note: this gave a degree of failsafe operation] This failsafe operation was significant, as there was no other means of brakes on a rolling wagon if it was detached from the manned locomotive. The valve in this system was improved and by 1873, was called a triple valve (Westinghouse 1873). Westinghouse in America, applied for a patent, titled ‘An improvement in air-valves for power-brakes’. The improvement to the valve also included a slide valve attached and a differential pressure piston. The piston was connected to the brake pipe on the full piston side and to the auxiliary on the annulus side and the difference in pressure moved the piston. The operation of this slide-valve gave porting, so that the valve would either charge the auxiliary reservoir and release air from the brake cylinder or would charge the brake cylinder from the auxiliary reservoir (Westinghouse 1875). The triple valve from the Westinghouse invention of 1875 and as shown in Figure 8 had the slide valve arrangement and a 4-way cock that made the triple valve inoperable for use with a straight-air brake system.

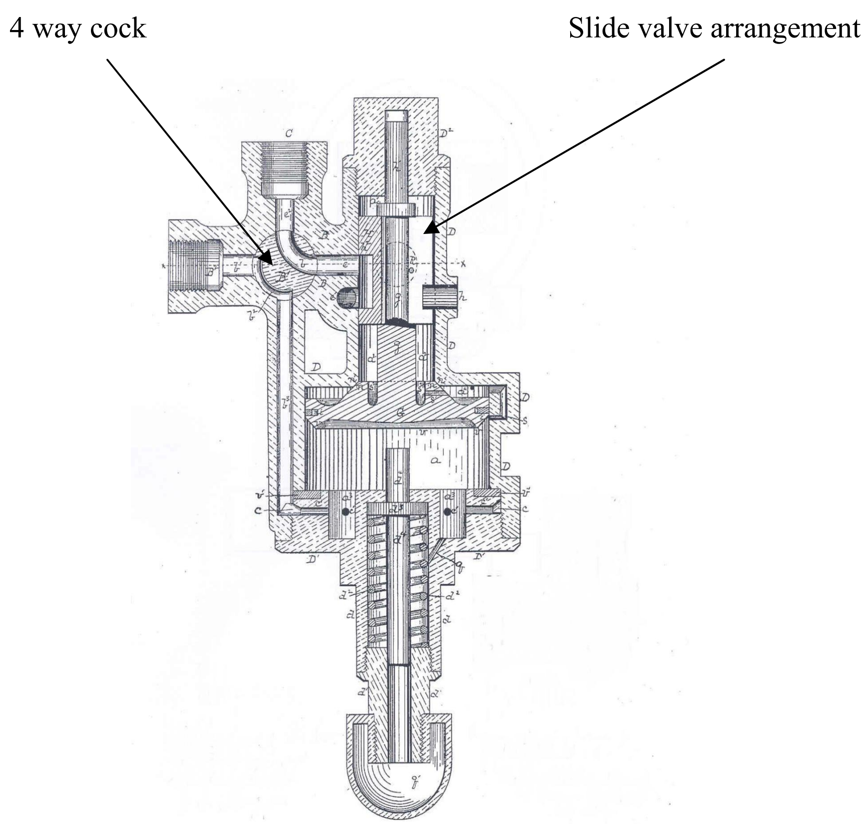


Figure 8: First triple valve with a slide valve. (Westinghouse 1875)

2.3.2 American Westinghouse Triple Valve Developments and Improvements.

The chart shown in Figure 9 shows from patents granted to Westinghouse and the development of the railway brake valve from 1869 to 1989 in America.

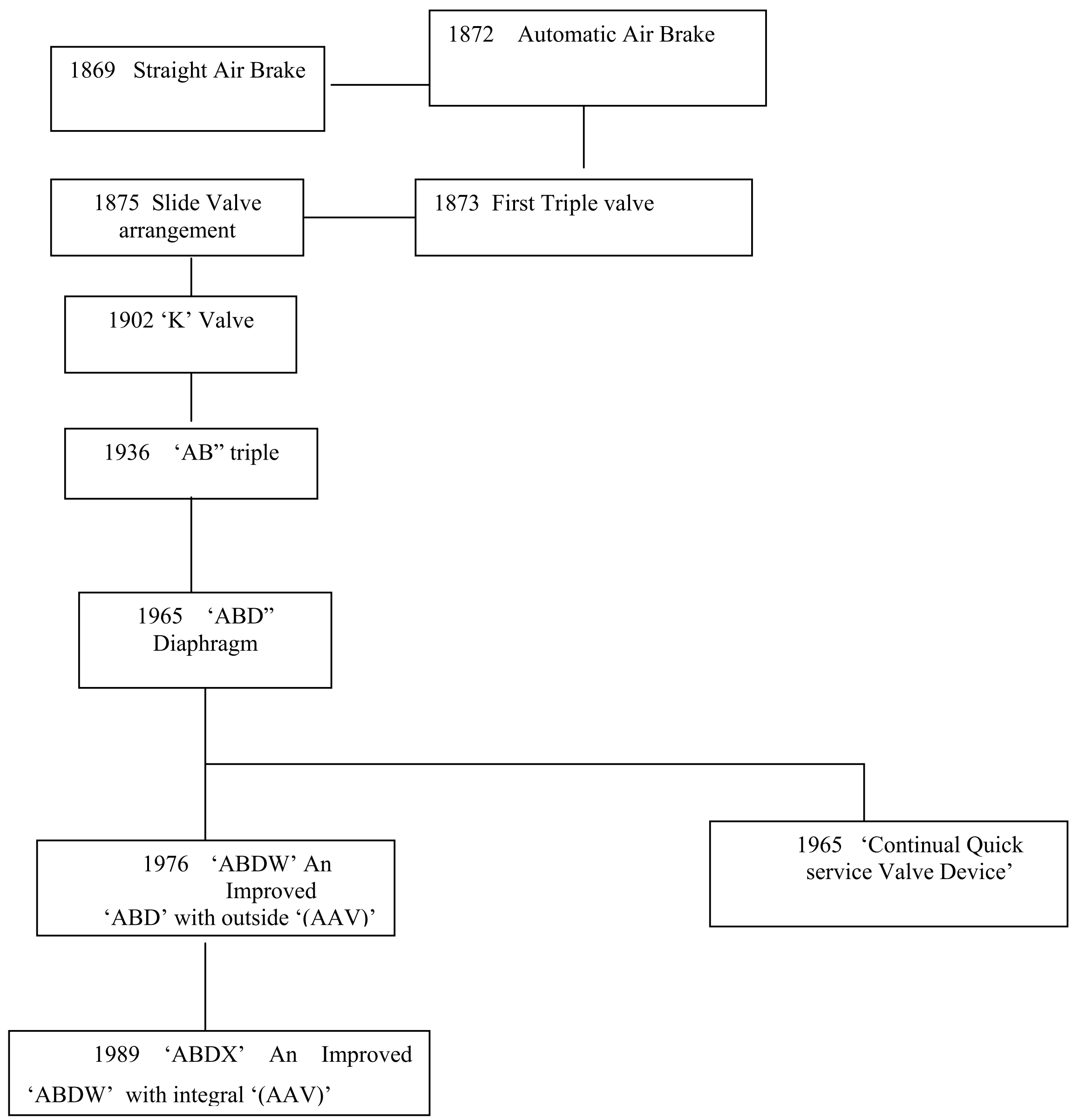


Figure 9: The Westinghouse American development tree.

The introduction of the improved ‘K’ triple valve as shown in Figure 10 was granted a patent in 1902. The ‘K’ triple system had a combination triple valve, brake cylinder and a auxiliary reservoir with a supplementary emergency air reservoir attached by piping. The ‘K’ triple was an automatic air brake system valve and could not be used with the straight-air brake system. A supplementary reservoir was included to supply air to the brake cylinders in an emergency brake application in addition to that supplied by the auxiliary reservoir. The service applications only used the stored air in the auxiliary reservoir and in repeated applications, this stored volume could become consumed, lowering available pressure and braking effort (Westinghouse 1902).

In 1936 the ‘K’ type brake system was superseded by the “AB” system, which split the ‘K’ brake's integral triple valve into two separate valves, and had a two-compartment air tank under each car (Farmer 1936). The two-compartment tank stored the supplementary or emergency air and the auxiliary air. The ‘AB’ valve still used the technology of the triple valve Westinghouse patented in 1875 having inside the triple valve, a piston connected to a slide valve, aligning or blocking ports to make the valve function.

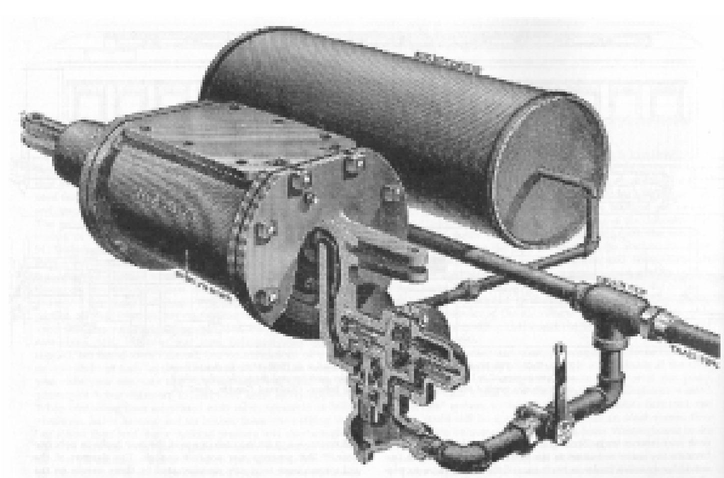


Figure 10: The ‘K’ triple valve. (Westinghouse 1902)

Although the 'AB' brake system was a vast improvement over the 'K' triple valves, Westinghouse added further improvements.

Westinghouse in 1965 designed the 'ABD' brake valve range (Kirk 1965). The 'ABD' brake valve utilised rubber diaphragms in place of pistons with sealing rings, although the slide valves were still connected to the diaphragms.

It was also shown by 1965 that a need for a greater local reduction in brake pipe pressure during a brake application (Wilson 1965). The valve that was patented arose because of the increasing length of cars. In the 1930's, the length of freight cars did not exceed 18.3 m (60 ft). By 1965 the length varied from 9.4 m to 27 m (30 to 90 ft). It had been shown that a higher rate of propagation of a brake pipe pressure wave than the standard 'AB' freight brake equipment could supply was needed. The new valve that Wilson designed in 1965 was located separate from the 'AB' valve and was known as the 'continual quick service valve device'. The new continual quick service valve operates on the difference of pressure on a diaphragm with one side connected to the brake pipe and the other side connected to a reference chamber with stored brake pipe pressure. When brake pipe pressure was lowered, the diaphragm moved to allow a local release or exhaust of brake pipe air to atmosphere. When the pressure equalizes on both sides of the diaphragm the local release of brake pipe air is then stopped. The valve can then repeat its operation if the brake pipe pressure lowers again. Wilson reported that it made a more nearly simultaneous initiation of a service application of brakes on all cars in a train.

A new triple valve (Wilson 1973) also had a 'quick service valve device'. The new device was attached to the outside of the emergency portion of the triple valve and operated with the same principle as the first continual quick service valve, the device now used a chamber in the emergency portion of the new 'ABDW' valve as a reference chamber. The 'W' in this design was for the first letter in the inventor's name. The continual quick service valve became known as the Accelerated Application Valve (AAV). A patent for a 'Freight Brake Control Valve having an emergency piston slide valve arranged to provide an accelerated brake application function' (Hart 1987) was granted. This design put the 'quick service valve device' now called the Accelerated Application Valve, inside the emergency portion of new 'ABDX' control valve. The 'ABDX' is still in use at the present time.

2.3.3 Operation of an American Control Valve

The American control valve is characterised by separate emergency and service valve portions. The emergency portion responds to the rate of increase in pressure differential across the diaphragm within the portion and activates only when the brake pipe is being completely exhausted. A reservoir, larger than the auxiliary reservoir is connected to the emergency portion. The capacities of the reservoirs are .041 m^3 (2440 cubic inch.) for the auxiliary reservoir and .057 m^3 (3500 cubic inch) for the emergency reservoir. These volumes were for use with a standard piston diameter of 254 mm (10 inches) with a 203 mm (8 inch) stroke (Bureau 2002). Difference in pressure across the diaphragm in the emergency portion will allow the diaphragm to move the slide valve connected to it. In the case of an emergency application passages under the slide valve connect both the emergency reservoir and the auxiliary reservoir to the brake cylinder resulting in a pressure that is approximately 20% higher for emergency applications.

Reducing the brake pipe pressure at a controlled rate not fast enough to trigger an emergency application gives a service brake application via the service portion. The service portion piston in response to the reduction in the brake pipe pressure is moved to the application position by the pressure differential across the diaphragm within the portion. The movement toward the application position also allows the graduating slide valve that is connected to the piston, to align ports of the slide valve.

The first alignment of ports initiates the preliminary quick service of a two-phase quick service function. The alignment of the ports establishes a communication between the brake pipe and a quick service volume that is vented to atmosphere via a quick service exhaust choke. In this manner, a local reduction of brake pipe pressure is obtained at each valve on a wagon of a train to supplement the pressure reduction propagation. When the service portion piston has moved to the application position, the preliminary quick service is terminated by cutting off the brake pipe pressure from the quick service volume at the service piston slide valve, and the secondary phase of the quick service commences. During this secondary phase of quick service, the service piston slide valve connects the brake pipe to the wagon brake cylinder concurrently with the auxiliary reservoir volume until approximately 55 - 82 kPa (8-12 PSI) brake cylinder pressure develops. When the brake cylinder pressure has been reached from the two-phase quick service function, the quick service-limiting valve closes and further air to the brake cylinder is only from the auxiliary reservoir.

In a brake release, the brake pipe is charged back to maximum pressure and the difference between the pressures in the brake pipe and the auxiliary reservoir moves the service portion piston allowing ports on the slide valve to connect passages to give flow from the brake pipe to the auxiliary via a choke. The slide valve also connects ports allowing the brake cylinder air to flow to atmosphere through a choke and a retainer valve.

2.3.4 American Reservoir Sizes.

Maximum Brake cylinder pressures are dependant on the auxiliary reservoir volume and the maximum brake cylinder volume. The equalizing or final pressure P_f in the brake cylinder is calculated by the equation:

$$P_f = \frac{(P_1 V_1) + (P_2 V_2)}{(V_1 + V_2 + V_3)} \tag{2.1}$$

Where:

- P_1 is auxiliary pressure (PSIA)
- V_1 is auxiliary volume
- P_2 is brake cylinder pressure (+ atmospheric pressure at beginning of an application) (PSIG)
- V_2 is maximum brake cylinder volume
- V_3 is the volume in the pipe to the brake cylinder and brake cylinder clearance volume.

An equalizing pressure in the brake cylinder of 352 kPa (51 PSIG) can therefore be obtained using an auxiliary pressure of 583 kPa (84.6 PSIA), a auxiliary volume of .041m^3 (2440 cubic inch.) and a brake cylinder volume of .0103 m^3 (630 cubic inch) (Blaine 1979). When certain brake designs require the use of larger than normal brake cylinder volumes, different size auxiliary reservoirs are used. The brake system that has 4 cylinders with a bore of 228 mm (9 inch) and a 76.2 mm (3 inch) stroke requires a larger auxiliary reservoir of .057m^3 (3500 cubic inches) to achieve desired maximum brake cylinder pressures. In larger brake cylinder volume systems, where 254 mm (10 inch) bores are used, a relayed system with a dummy brake cylinder reservoir as shown in Figure 11 is used and the size of this dummy reservoir is .013 m^3 (800 cubic inches). The dummy brake cylinder reservoir is matched with the auxiliary side of the combined reservoir to achieve a maximum pressure that would have been used in the larger brake cylinder volumes. The relay valve in this system uses pressure from the dummy brake cylinder reservoir as a reference and matches this pressure, using a supply reservoir of 0.168 m^3 (10,230 cubic inches) to deliver air to the brake cylinders (Bureau 2002). The brake cylinders are filled to the same pressure as the dummy brake cylinder reservoir.

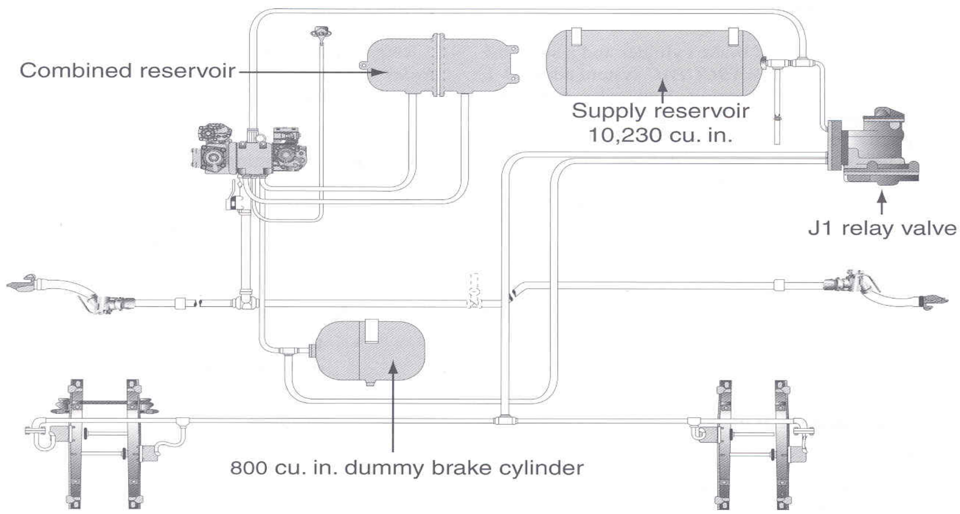


Figure 11: American relayed brake system. (Bureau 2002)

2.3.5 Retainers or Grade control valves.

The retaining or grade control valve is a valve to hold back or retain some air pressure in the brake cylinder after the train brakes have been released. Retaining valves are used on steep grades and allow the brake pipe to recharge while at the same time controlling its speed with air pressure retained in the brake cylinders, it was first introduced in 1883 (Jones 1979). The retaining action is achieved by connecting a three way selector cock as shown in Figure 12 to the exhaust port of each triple valve on the train and then selecting a suitable restrictive time to exhaust the brake cylinder. The operation of these valves involves the manual positioning of a lever to the desired position of each valve in the train before descending a grade. The positions available will give: 1) EX. an open exhaust to atmosphere 2) I.P. a restricted exhaust to atmosphere 3) H.P. a restricted exhaust to atmosphere until a pressure of between 275 kPa (40 PSI) and 48 kPa (7 PSI) is reached in the brake cylinder, the valve then holds the pressure in the brake cylinder (Bureau 2002). Different springs in an internal check valve determine these holding pressures. When the train has descended the grade, then the valve lever is brought back to its normal position EX on each wagon.

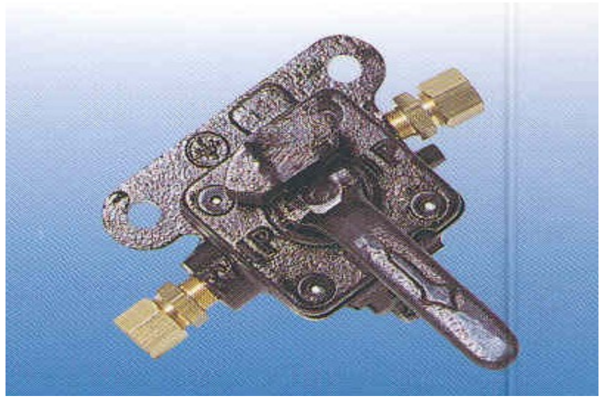


Figure 12: Grade control valve. (Westinghouse Brakes Australia)

2.3.6 Australian Triple Valve Developments and Improvements.

The chart in Figure 13 from granted patents show how the Triple Valve evolved in Australia from beginnings in 1931 to the present ‘W’ series valve.

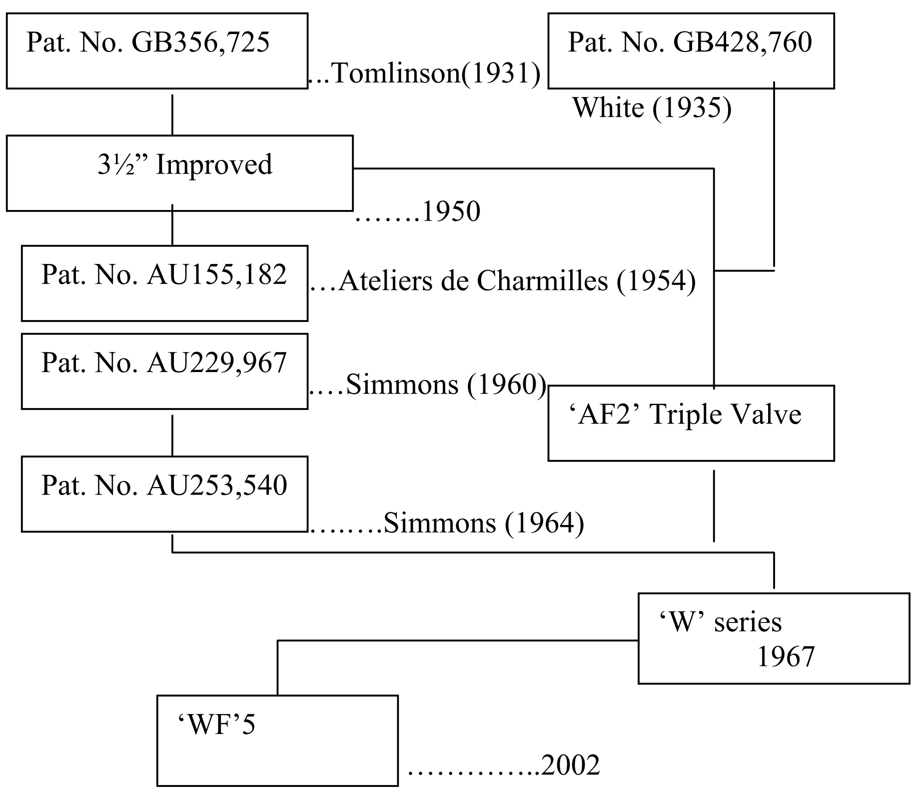


Figure 13: The Westinghouse Australian development tree.

The Westinghouse 3 ½” Improved Triple valve first started to be supplied to Australian railway operators in 1950 and was the last of improvements of a design invented by Tomlinson (Tomlinson 1931). The 3 ½” Improved Triple Valve is now being phased out of use, and like in North America, improvements followed the 3 ½” Improved Triple Valve, with the ‘AF’ type Triple Valve which was derived from the design invented by White (White 1935). Improvements to this design led to the introduction of the ‘AF’2 Triple Valve in 1960. The ‘AF’2 Triple Valve had several new features incorporated, which included a retarded recharge and inshot valve, allowing these triples to be more sensitive to the pressure difference across the diaphragm in both application and release. A new design of a triple valve (Simmons 1964) led to the introduction of the ‘W’ Triple valve in 1967. The new design followed the European style of Distributors, where rubber diaphragms and rubber faced poppet valves replaced the old style piston and slide valves. These valves also had an inbuilt quick service valve. The design was granted Australian patent No. 253540, the applicant was Westinghouse Brakes Australia, succeeding earlier patents issued to Simmons (Simmons 1960) and Ateliers Des Charmilles (Ateliers Des Charmilles 1954).

A significant difference in the Australian system shown in Figure 14, was that it did not include the emergency brake system that the American system used (as described in section 2.3.3). The Australian system achieved a faster propagation rate for emergency brake applications as shown in Table 3.

Since 1950, Westinghouse Australia used the improved 3 ½ "Triple Valve, Slide type, usually called an ITV as shown in Figure 15. The slide type triple valve (known as the Improved 3 ½” Triple) was similar to the American ‘K’ type triple valve (as described in section 2.3.2).

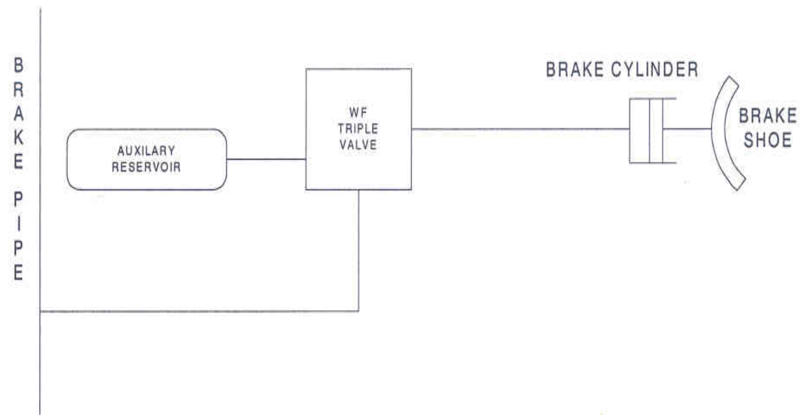


Figure 14: Australian basic control valve system

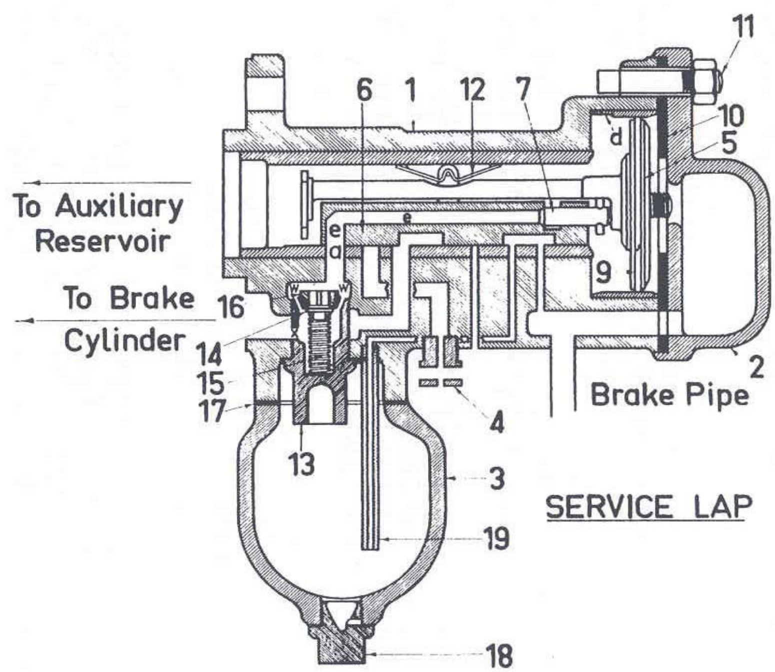


Figure 15: The 3 ½ " improved triple valve. (QR_Manual 1995)

The slide type triple valve (known as the Improved 3 ½" Triple) was similar to the American K type. The next improvement to the triple valve was the 'AF' type Triple valve. The last valve in this range was released in 1960 and was known as the 'AF' 2 as shown in Figure 16. The last major improvement to the triple valve is the

'W' type, which is still in use. Since the first 'W' type valve was introduced and it has evolved from the basic valve as shown in Figure 17. The following variations and enhancements have been added for particular types of services and vehicles, made possible by the addition of devices to form a modular construction shown in Figure 18.

- Inshot valve
- Reduction ensuring valve
- Retarded recharge valve
- Accelerated release valve

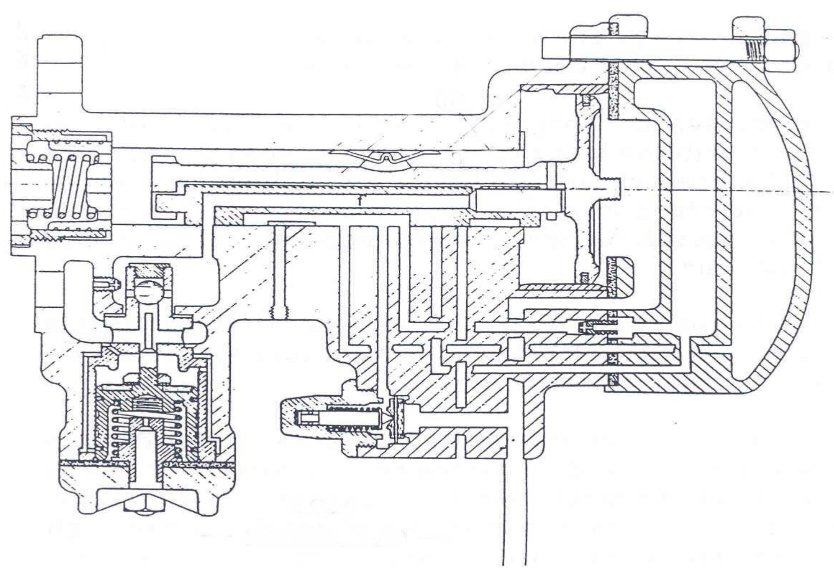


Figure 16: The AF 2 type triple valve. (QR-Manual 1995)

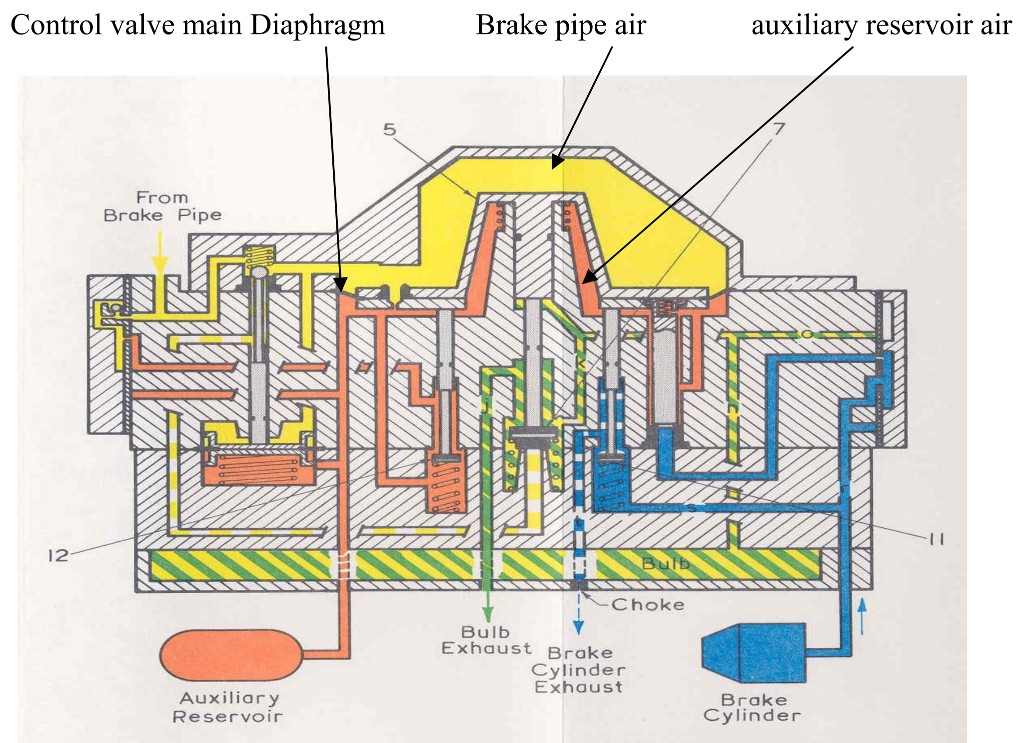


Figure 17: The basic ‘W’ type triple valve. (Westinghouse Brakes Australia)

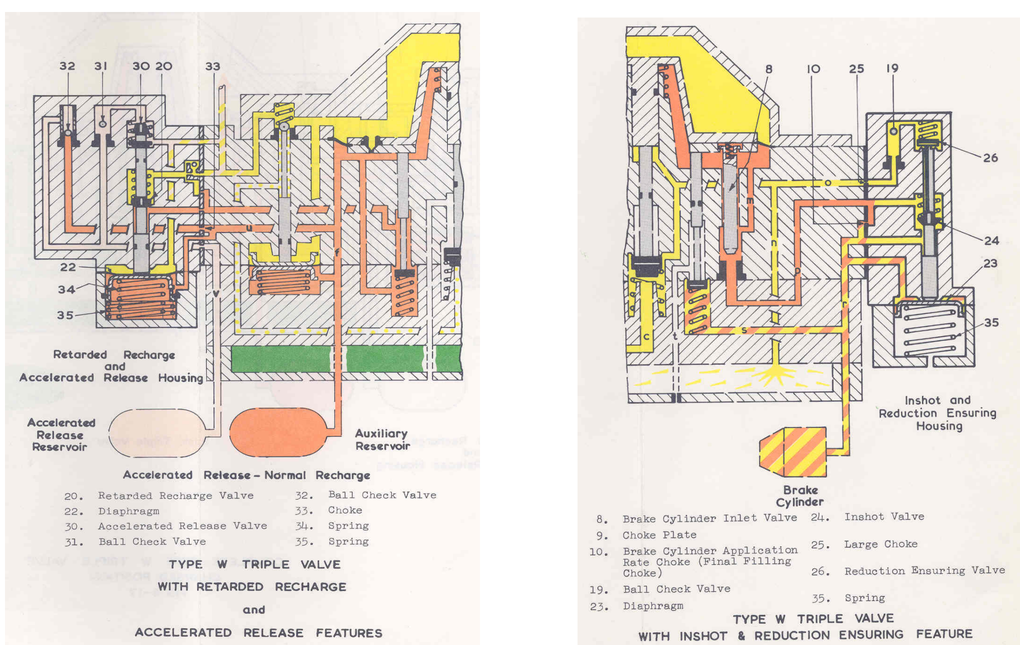


Figure 18: Added devices to the ‘W’ triple valve. (Westinghouse Brakes Australia)

A comparison on page 24 in Table 1 shows six major control valve manufactures in Australia, America and Europe.

Table 1: Brake Control Valve Comparison.

Valve Feature (2)	**WF** 5	**ABDX** (1)	**ES** 500WD1	**DB**-60	**KE** Distributor	**SW**4
Quick Service Volume	X	X	X	X	X	X
Retarded Recharge	X	X	X	X	X	X
Inshot	X	X	X	X	X	X
Reduction Ensuring	X		X	X	X	X
Accelerated Release	X	X	X	X		
Slide Valve		X				
Poppet Valves	X		X	X	X	X
Diaphragm	X	X	X	X	X	X
Graduated brake release					X	X
Accelerated Application		X				
Interchangeable with WF5			X			

Note: (1). 'ABDX' valves can be used in a train where 'WF' valves are used on other wagons, when the emergency portion of the ABDX is removed and blanked off.

Note: (2). Valve features:

Quick Service Volume: To assist in the application propagation, a valve connects the brake pipe pressure to a Quick service volume and produces a local small, sharp drop in brake pipe pressure. The local reduction is repeated in all control valves propagating a fast brake application on all cars in the train.

Retarded Recharge: Uniform charging throughout a train is accomplished by a retarded recharge valve, which provides two stages of charging and recharging. When the pressure differential is high at the front of a train between the brake pipe and auxiliary reservoir, only one charging choke is used. At the rear of the train, where the pressure differential would be low both charging chokes will be used.

Inshot: Allows for a rapid initial build up of brake cylinder pressure to approximately 70 kPa and ensures that braking action is initiated rapidly to help control train slack action.

Reduction Ensuring: During initial application brake pipe air pressure is permitted to flow at a controlled rate to the brake cylinder. The reduction ensuring valve closes with the inshot valve.

Accelerated Release: The accelerated release feature is to help propagate the brake release response. It increases the initial rate of brake pipe charge by momentarily connecting a stored volume of air to the brake pipe.

Slide Valve: Terminology used for the valve that opened and closed feed groves. The various positions that the valve provided was a means to charge the auxiliary reservoir and applying the brakes and releasing the brakes.

Poppet Valves: These valves were designed to replace the slide valves to give longer service life between overhauls of the control valve.

Diaphragm: A rubber flexible disc that is sealed at the outer edge to separate two volumes of air pressure. A valve is attached that will move when there is a difference in pressure across the diaphragm.

Graduated brake release: A release of the brakes in steps or graduations. The control valve is able to use the pressure in a control reservoir as a reference to control the brake cylinder pressure on a release and provide an inexhaustible release function.

Accelerated Application: This is a valve designed to decrease brake application time, and achieves this by venting the brake pipe pressure locally at each vehicle of the train

2.3.8 The Major Valve manufactures Worldwide

- Westinghouse Brake Australia (**WF** series Triple Valves) now marketed by Knorr-Bremse; most widely used valve in Australia.
- Westinghouse (**ABDX** Control Valves) now marketed by Wabtec used in North Western Australia on the Iron Ore trains and on the Weipa line in far northern Queensland.
- Davies and Metcalfe (**ES** style Distributors) supplied by Sab Wabco and used in New South Wales and Queensland.
- New York Air Brake Company (**DB** – 60 Control Valve) now controlled by Knorr-Bremse, not used in Australia.
- Knorr-Bremse (**KE** Distributor), not used in Australia.

Sab Wabco (**SW**4), not used in Australia.

2.4. Australian Control Valve

2.4.1 Operation of the Australian Control Valve

The control valve has cavities on either side of the main diaphragm connected to the brake pipe air pressure and auxiliary reservoir air pressure. If the brake pipe pressure is 12 kPa higher than the auxiliary reservoir pressure, the control valve main diaphragm moves to the release position. In this position it vents any brake cylinder air to atmosphere via an exhaust choke, and releases the brakes. It also connects the brake pipe through internal passages to the auxiliary reservoir, so brake pipe air pressure can recharge the auxiliary reservoir. When the brake pipe is fully charged, both auxiliary reservoir and brake pipe stabilise at a pressure of 500kPa. The control valve main diaphragm will now have equal pressure on both sides and moves to a lap position, in this lap position air is blocked from moving to the brake cylinders.

When the brake pipe pressure is reducing, as in a brake application, and the brake pipe pressure becomes lower than 10 kPa below the auxiliary reservoir pressure, the control valve main diaphragm moves to the apply position. In the apply position, the flow of air from the brake pipe is connected to the quick service volume (bulb) and a sharp reduction in pressure is made in the brake pipe. This sharp reduction is termed as a 'gulp'. The sharp reduction propagates along the pipe to the next valve and helps to ensure an application of that valve. As the air is connected to the quick service volume, the reduction ensuring valve in the inshot portion, allows air from the bulb to flow through a choke and build up in the brake cylinder, this gives the effect of a second bulb filling at a controlled rate until 70 kPa is reached in addition to the main bulb filling rapidly. The bulb pressure eventually equalises with the brake pipe pressure and then follows the brake pipe pressure. The bulb will then stay at brake pipe pressure until a brake release is made.

At the same time that the bulb is being filled, a passage is opened from the auxiliary reservoir to the brake cylinder. Air flows from the auxiliary reservoir to the brake cylinder through a second choke. Pressure in the auxiliary reservoir will lower until it equals the brake pipe pressure. When the pressure on the top and bottom of the control valve main diaphragm becomes equal, the control valve main diaphragm returns to its lap position and prevents any further increase in brake cylinder pressure. If a further reduction to the brake pipe pressure occurs the control valve main diaphragm will again move to the apply position and air from the auxiliary reservoir will add pressure to the brake cylinder. The pressure attained in the brake cylinder is limited by the condition where the pressure in the auxiliary reservoir and brake cylinder becomes equal and is the maximum brake cylinder pressure.

Emergency brake applications exhausts all brake pipe pressure, and as with a minimum service application the control valve main diaphragm will move to the apply position. The brake cylinder pressure will become equal to the auxiliary reservoir pressure, this again will be the maximum brake cylinder pressure. (Westinghouse Brakes Australia).

2.4.2 Australian Reservoir sizes

In Australia, as in America, the modern brake setup can use 4 brake cylinders on one wagon and therefore could require larger brake cylinder volumes to be filled on a brake application. A relay valve is used to supply air from a large supplementary reservoir to the brake cylinders and allowing maximum brake cylinder pressure to be achieved for a range of different cylinder volumes. Australian brake systems are manufactured, with a multi-compartment combined reservoir, as shown in Figure 19 allowing the control valve to be bolted directly to the end of the combined reservoir.

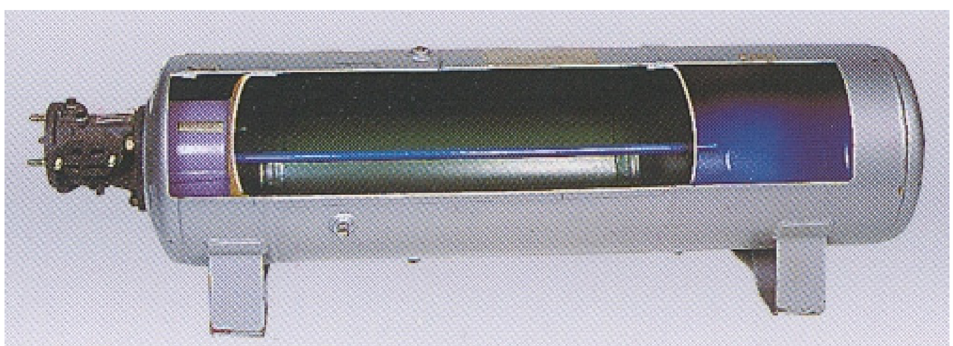


Figure 19: A combined Reservoir showing separate chambers.

The operation of the relay system is the same as the American relay system and is described in section 2.3.3. The sizes of auxiliary and dummy compartments within the combined reservoir shown in Figure 20 are smaller than the American brake system as described in section 2.3.4, although the volume ratios are the same. This multi-compartment reservoir (Figure 20) houses four separate reservoirs, which include:

1. Accelerated release Reservoir (0.039 m^3)

2. Supplementary Reservoir (0.085 – 0.113 m^3)

3. Auxiliary Reservoir (0.014 m^3)

4. Dummy Brake Cylinder Volume (0.00465 m^3)

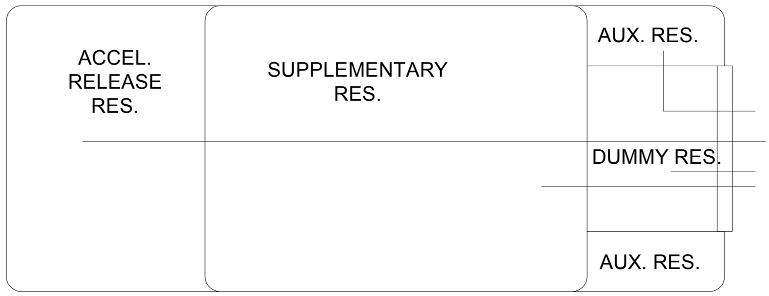


Figure 20: Multi-compartment reservoir.

2.4.3 Australian bulb sizes

The Australian train line brake system relies on the bulb volume to propagate the brake pipe pressure reduction along the train length. The bulb volume is tuned to the pipe length between following control valves. The manufactures information show recommendations for the size of the bulb to the brake pipe length between control valves. Manufactures information show that from 1989 onwards there are only two bulb sizes that are used and these are the medium (33 $x10^{-5}$ m^3) and the large (59 $x10^{-5}$ m^3) type (Westinghouse 1989).

The control valve shown in Figure 21 is made from sandwiching components together and within three of these components are the four bulb cavities. The bottom component has the first two of the bulb cavities (joined by a choked drilling) of which one is connected to the next cavity in the next sandwich component, this in turn is connected to the last bulb cavity in the third sandwich component. The connections are by 5 mm diameter holes in gaskets between the sandwich plates.

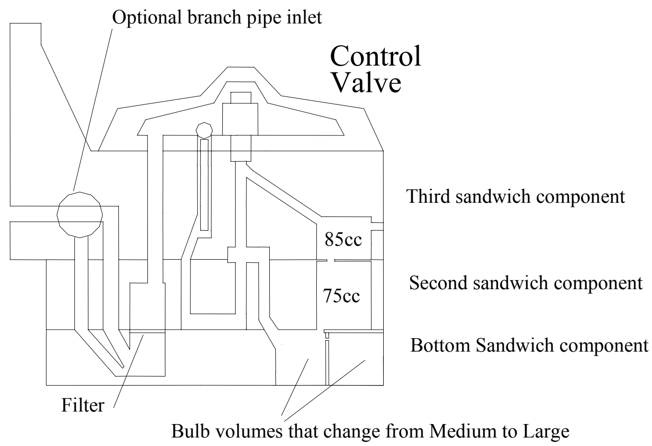


Figure 21: Control valve with bulb cavities placement

The top two sandwich components are of fixed volumes and are at 8.5 x10^{-5} m^3 and 7.5 x10^{-5} m^3, in the bottom sandwich component there are two types and the cavities in each are of two different sizes which when combined with the other two fixed volumes make the medium and long bulb volumes.

2.4.4 Retainers or Grade Control Valves

The retainers or grade control valve used in the Australian rail industry, work in the same way as the American type of retainer and is described in section 2.3.5. The Australian railway industry is phasing out the retainer or grade control valve and fitting a moderately restrictive exhaust orifice on all control valves. The change over has come about because stopping the train each time to set the retainers to the desired setting takes time, causes delays and costs money (Macfarlane 2000). Queensland Rail uses a 60 second permanent restriction (Q6) as seen in Figure 22 on all its coal wagons in various different coal corridors. The use of different sized drillings, each suitable to a type of wagon and in some instances also tuned for a specific track section has resulted in many different sizes of orifice in use (Macfarlane 2000).

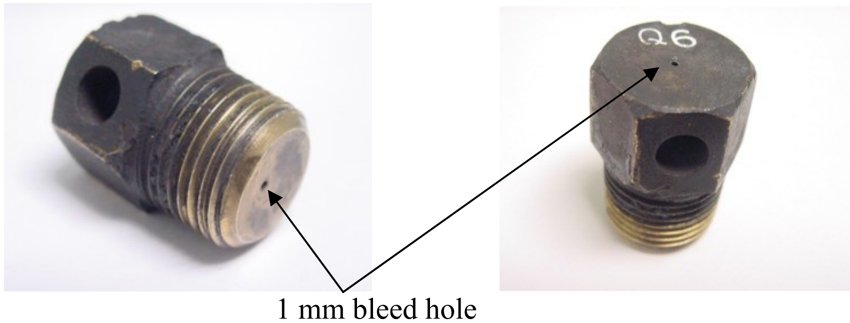


Figure 22: Q6 exhaust choke used on mineral freight wagons.

2.5. Brake pipes and Propagation Rates

2.5.1 Brake Pipes

The major change to the direct release air brake pipe has been the increase in the internal diameter from 25 mm (1 inch) to 32 mm (1 ¼ inch). In America the first known use of 32 mm pipe was in 1888 and this move up in size, was to increase the transmission speed of the air in the brake pipe (Jones 1979). The size of brake pipe on UIC freight trains was 25 mm up until the mid 1950's when the 32mm diameter pipe was adopted (Leigh 1990). The results of tests in the UK indicated that release times following a full service application is much longer at 360 seconds with a 25mm brake pipe. From these results in 1969 the UK also changed the brake pipe diameter from 25 mm to 32 mm to give 184 seconds under the same conditions for single pipe systems (Roberts 1979). The tests in the UK also indicated that the increase from 25 mm to 32 mm had almost no effect on the brake pipe application time.

Figure 23: 32mm brake pipe with 90^0 300mm bends.

The type of wagon used determines the length of the brake pipe and this can range from 9 m to 37 m per wagon. The number of 90^0 bends in an average Australian gondola wagon as shown in Figure 23 brake pipe tends to be between 4 and 6 with radii between 180 and 200 mm.

The brake pipe transmission speed has been shown to be the length of the brake pipe of the train divided by the time between the movement of the drivers brake valve and when air begins to enter the brake cylinder of the last vehicle (Roberts 1979). Brake pipe propagation rates have been studied with respect of time between the initial valve operation and the last valve operating to the length of brake pipe used. By using empirical equations Murtaza indicates that by using a method of equivalent length a substantial saving in computation time is possible by avoiding fitting loss factor computations when dealing with Railway Air Brake Simulation (Murtaza 1993). The equivalent length that Murtaza uses is 3.38 times the brake pipe length and as shown in Figure 24 makes an actual train pipe much longer, this is with 4 bends plus the friction factor of the length of pipe.

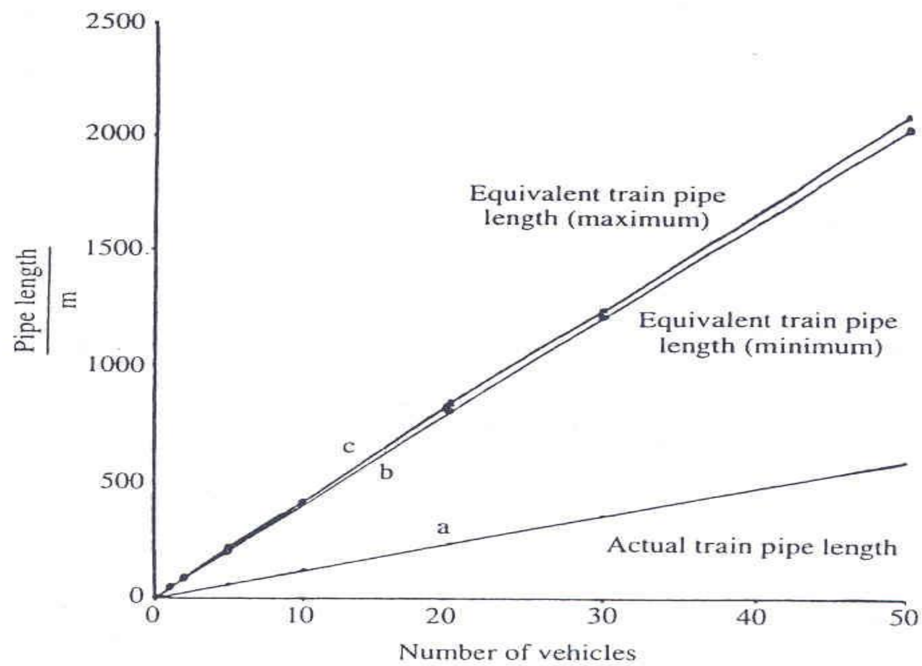
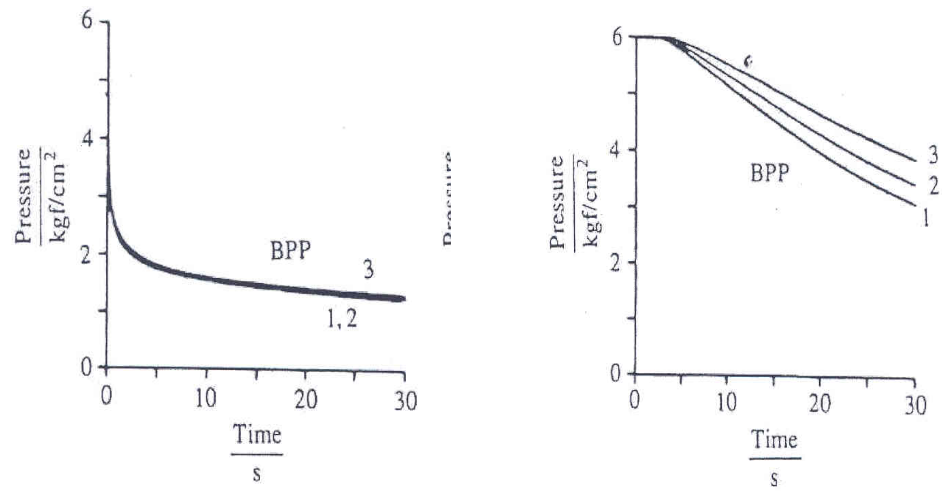


Figure 24: Brake pipe with equivalent and actual lengths. (Murtaza 1993)

The effects of the diameter of the brake pipe are shown in Figure 25 by (Murtaza 1989). The results are obtained from a numerical analysis and considered changes to the diameter *d*. Variations of brake pipe pressure on the 1st and 57th cars are presented in Figure 25. Results have shown that the speed of propagation of the brake pipe pressure drop to the 57th car increases as the brake pipe diameter is increased from 24 mm to 32 mm to 40 mm. Figure 25 shows the comparison in a 30 second period of an emergency pressure drop from 590 kPa with the first car dropping to 117 kPa and then on the 57th car where the pressures are 360 kPa, 320 kPa and 295 kPa. These results shown in Figure 25, which the larger pipe has little effect at the first car, but the effect increases on cars further from the locomotive.



Pipe Diameters 1 = 24 mm, 2 = 32 mm, 3 = 40 mm

Figure 25: Comparison of brake pipe diameters over 57 cars. (Murtaza 1989)

The equations for corrections (Murtaza 1990) has shown that friction losses for bends and valve fittings, to obtaining the equivalent length, are needed in pressure drop rate calculations. The equivalent length does not include the flexible hose fittings between each wagon. Murtaza approximates the number of bends on each wagon to 4 and uses a r/d of 5.906. These approximations infer a bend radius of 187 mm for 31.75 mm diameter pipe. The equations for the total increase in friction length are shown (Murtaza 1990) below:

$$L_e = L_b + L_j + L_c \tag{2.2}$$

Where L_b is each bend, L_j is each joint and L_c is each cock.

The equivalent length that would account for additional losses is given by:

$$L_f = \left(\frac{d}{4f}\right)\left\langle 0.106\left(\frac{DR}{d}\right)^{-2.5} + 2000(4f)^{2.5}\right\rangle \tag{2.3}$$

Where DR = radius of bend, d = internal diameter of pipe, $4f$ = friction factor.

The net equivalent length (L_t) increase in piping is given by:

$$L_t = L_e + L_f \quad \textbf{(2.4)}$$

Using the governing equations of continuity and momentum are respectively

$$\frac{\partial \rho}{\partial t} + \frac{\partial(\rho u)}{\partial z} = 0 \quad \textbf{(2.5)}$$

and $$\frac{\rho}{g} \cdot \frac{\partial u}{\partial t} + \frac{\rho u}{g} \cdot \frac{\partial u}{\partial z} = -\frac{\partial \rho}{\partial z} - \frac{4\tau}{d} \quad \textbf{(2.6)}$$

Where g = acceleration due to gravity (m/s^2), ρ = density of air (kg/m^3), τ = wall shear stress, t = time, u = velocity, z = spacial coordinate and d = diameter.

With the braking demand model shown in Figure 26, the results obtained by Murtaza are shown in Figure 27 . The graphs show a full service application, with the initial brake pipe pressure at 588 kPa (6.00 kgf/cm^2) and a final pressure of 440 kPa (4.5 kgf/cm^2)

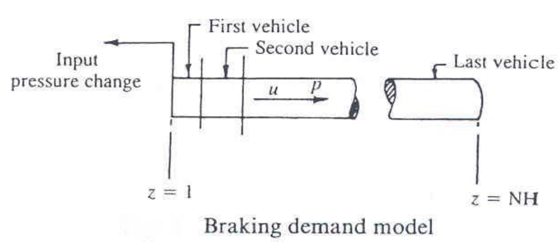


Figure 26: The braking demand model. (Murtaza 1989)

Murtaza has developed different equations from the continuity and momentum equations. Wall shear stress τ is approximated by friction calculations. The equation governing pressure distribution is shown in equation **(2.7)** and the equation governing velocity distribution equation is shown in equation **(2.8)**.

$$P_{t+\Delta t,z} = P_{t,z}\left\{\frac{1 - n \cdot \Delta t\left(u_{t,z} - u_{t,z-1}\right)}{\Delta z}\right\} - \frac{u_{t,z} \cdot \Delta t\left(P_{t,z} - P_{t,z-1}\right)}{\Delta z} \quad \textbf{(2.7)}$$

Where: P = pressure in the brake pipe, t = time, z = spacial coordinate (see **Figure 26**), u = velocity of flow in the brake pipe, n = polytropic index

$$
\begin{aligned}
u_{t+\Delta t,z} = u_{t,z} \{1 - \Delta \cdot \Delta z (u_{t,z+\Delta z} - u_{t,z})\} \\
- \frac{g\Delta t}{k\Delta z} \frac{(P_{t,z+\Delta z} - P_{t,z})}{(P_{t,z})^{1/n}} \\
- \frac{4f}{2d} \cdot \Delta t (u_{t,z})^2
\end{aligned}
\tag{2.8}
$$

Where k = ratio of specific heats (1.4 for air)

The correction factor/segment

$$C_f = \frac{L_t + H}{H} \tag{2.9}$$

Where H = length of each segment. Therefore the term $4f/2d$ in equation **(2.8)** is replaced by $(4f/2d)C_f$. the value of $4f$ is computed from the following relations:

Re < 2000	$4f = 64/\text{Re}$
2000<Re < 4000	$4f = 0.0027/(\text{Re})^{0.222}$
Re > 4000	$4f = 0.316/(\text{Re})^{0,25}$

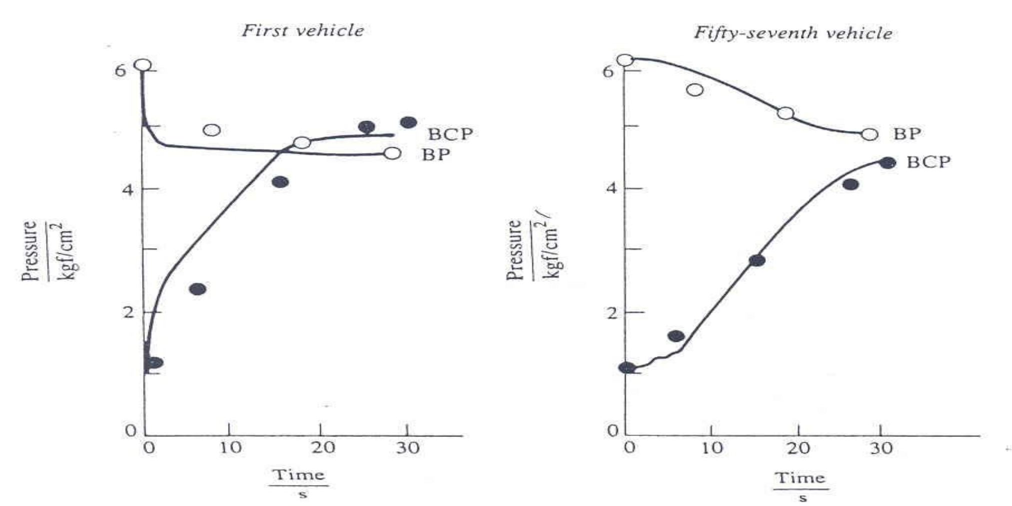


Figure 27: Results from a train pipe length of 690 m. (Murtaza 1989)

The 25 mm brake lines are suitable to relatively short trains of under approximately 300 meters (Leigh 1992). The increasing lengths of trains has made the control of the brake system more difficult due to the time taken to raise the brake pipe pressure at the rear of the train during the release of the brakes, plus the effect of leakages in the system. This fact was identified in Europe and wagons in these places have been fitted with 32 mm (1¼ inch) brake pipe, which allowed up to 480 meters to be handled. Leigh explains that improvements are only achieved at the expense of downgrading the performance with regard to exhaustibility. The direct releasing system is exhaustible in that successive applications and release of the brake, without allowing time for the auxiliary reservoirs to recharge, will result in the brake application on the individual wagons becoming less and less effective.

Computational models can closely relate to experimental results (Abdol-Hamid 1986), although there is no given size of brake pipe used and no equations are shown. The author states that for a longer train, the brake pipe pressure drop at the rear of the train is slower than at the front of the train.

Graphs from brake pipe rack tests on 50 wagons with 15.25 m of brake pipe between each valve (Leigh 1990). These tests were of minimum reduction, 70 kPa reduction, full service and emergency applications and releases with ABDW valves.

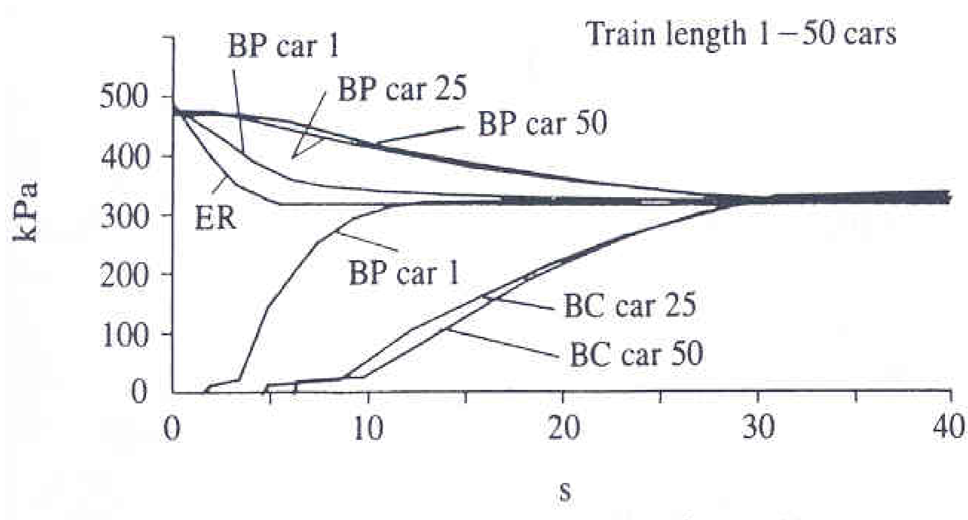


Figure 28: Full service application with an unrestricted exhaust choke. (Leigh 1990)

The results shown in Figure 28 were obtained by using an unrestricted exhaust choke of 25 mm diameter and in Figure 29 a restriction of 6.35 mm is used in the brake pipe exhaust. The comparison of the results indicates a quicker rise in pressure in the BC (brake cylinder) on car 1 when using the unrestricted exhaust passage and a quicker fall in pressure on car 1in the BP (brake pipe).

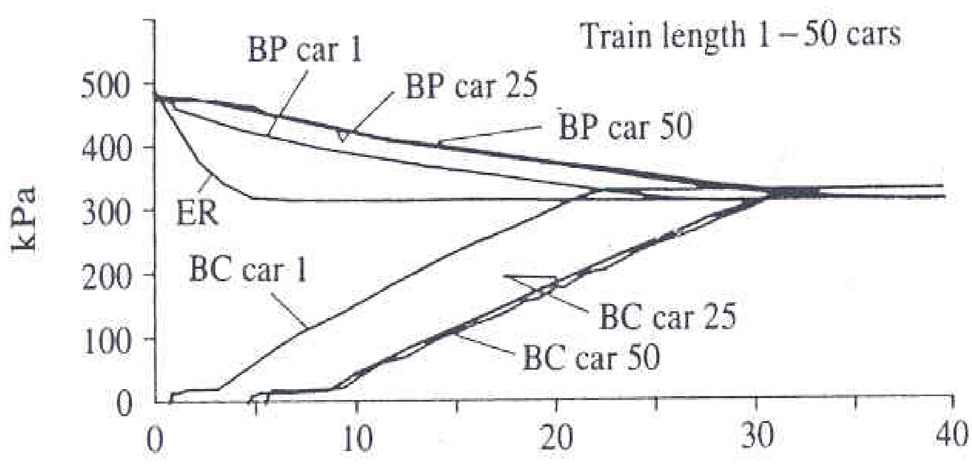


Figure 29: Full service application, with a restricted exhaust choke.(Leigh 1990)

2.5.2 American brake pipe lengths.

Inventor Richard L. Wilson (Wilson 1973) noted the significant importance of pipe length, in the United States Patent for a Quick Service Valve Device in 1973. Wilson indicated this over 30 years when prior the AB air brake control valve device came into general use, and the length of freight cars at that time did not exceed over 18 meters (60 feet). In 1973 freight cars that were conventionally equipped with an ABD air brake control valve device, which is an upgraded AB brake control valve device, varied from approximately 9 to 27 meters (30 to 90 feet). Wilson's invention was for a 'quick service' valve device to be used in conjunction with the ABD air brake control valve device, this combination became known as the ABDW air brake control device, (Wilson 1973).

Longer brake pipe lengths were used and by 1989, lengths of over 30.5 meters (100 feet) were being used. Because of the extra length and the need to have an acceptable propagation rate along the brake pipe, a further valve in the brake pipe was used and was called a continual quick service valve device and later called an accelerated application valve. This provided an additional reduction in the brake pipe pressure at the individual car during a service brake application.

Knorr-Bremse in 1978 was issued a patent for the work of Josef Hintner (Hintner 1978) for a 'Brake Accelerator'. In the description of the patent, he relates to the previous patent from Westinghouse, Pat. No. 3,175,871 described as a 'Brake Accelerator' where compressed air is drawn off from the train line air of an air brake system of railway vehicles.

In 1984 a patent was issued to Knorr-Bremse for a 'Service Brake Accelerator' for Rail Vehicle Brake Systems for the work of inventor Herbert Eder (Eder 1984). This valve worked along the same functional lines as the 'Continual Quick Service Valve Device' patented previously by Westinghouse. Both the Knorr-Bremse and

Westinghouse valves exhausted the brake pressure in response to a rapid drop in brake pipe pressure. Neither valve was directly connected to the triple or ABD valve.

2.5.3 Australian brake pipe lengths

The wagons built in Australia now are using the 32 mm diameter brake pipe (Westinghouse Brakes Australia). Older wagons that were equipped with 25 mm brake pipe, when being refurbished are also being converted to 32 mm diameter brake pipe. Wagons range in length from 9 m and up to 32 m when used in coupled form. [It is to be noted that the pipe being used is known as 32 NB heavy galvanised pipe, which has an internal diameter of 34.5 mm.]

2.5.4 Brake Cylinder fill times and brake pipe propagation rate charts

It has been difficult to locate a complete series of charts that would combine the brake cylinder and brake pipe charts, which would reflect the present practice from either Australia or U.S.A. In the U.S.A., Wabco presented a paper to the International Heavy Haul Conference in Montreal, Canada, in 1996 with graphs from the Westinghouse Air Brake Division. These graphs range from between the years 1975 to 1986 and show the performance of the ABD, ABDW and ABDX triple valves when applying a service reduction of 103 kPa (15PSI) reduction to 150 wagons of 15.24 m (50 ft) in length. The following graphs show how the improvements of the American valves have improved propagation times for minimum service reduction of the brake pipe and for full service reduction.

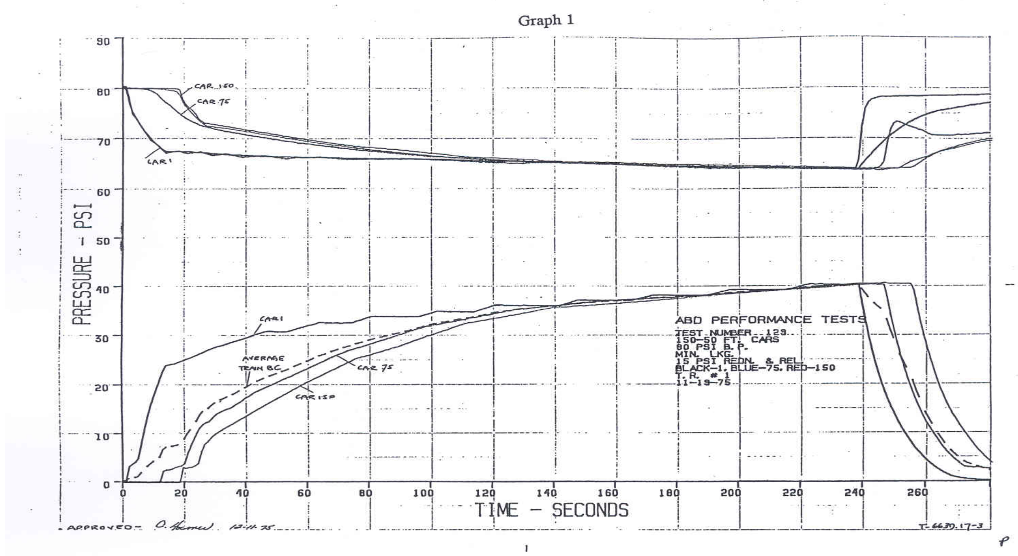


Figure 30: ABD tests for 150 cars of 50ft each in length. (WABCO 1996)

The results from Westinghouse in Figure 30 show a brake service application, a reduction in brake pipe pressure of 103.4 kPa (15 PSI) with stabilised brake pipe pressure initially at 551 kPa (80 PSI). The build up of pressure to 241 kPa (35 PSI) in the last vehicle brake cylinder takes 135 seconds on a 150 car train.

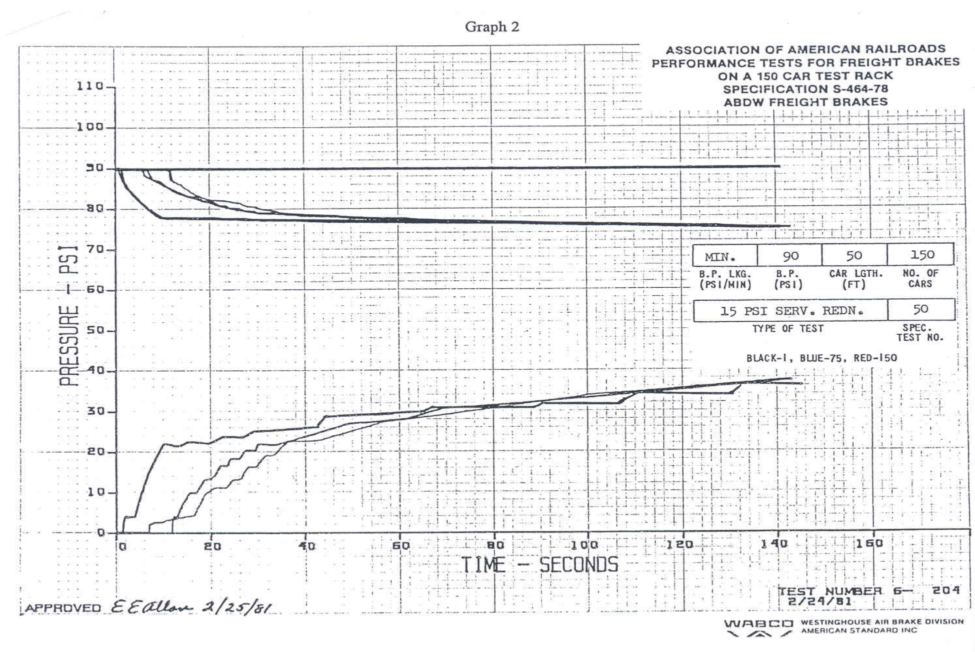


Figure 31: AAR tests of ABDW in 1981. (WABCO 1996)

In 1981, the ABDW valve was tested on a 150 car test rack and as shown in Figure 31. The results show that a higher stabilised pressure, compared to the previous pressure of 551 kPa (80 PSI) is used. From the initial stabilised brake pipe pressure of 620 kPa (90 PSI) and with a brake service application, a reduction in the brake pipe of 103.4 kPa (15 PSI) is made. The improved last car brake cylinder fill time which is now down from 135 seconds to 115 seconds. The valve used had a continuous quick service function or AAV as explained in section 2.3.2.

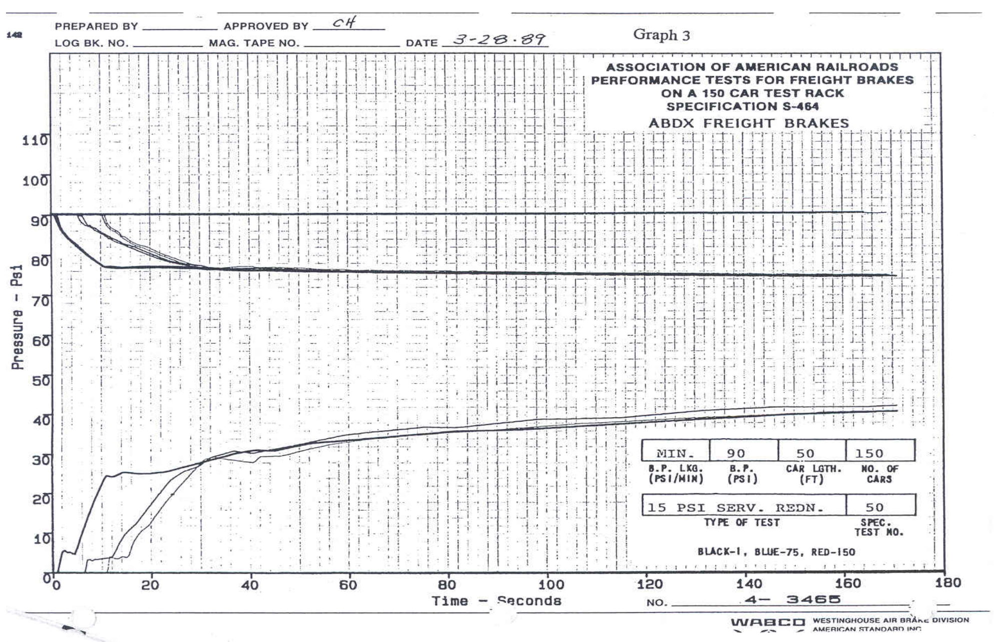


Figure 32: AAR tests of ABDX in 1989. (WABCO 1996)

In 1989, Wabco carried out similar tests on the ABDX type valves, which provided a better continuous control of the quick service function. Figure 32 shows that the valve improved the application time of the brake cylinders on the 150th wagon to 74 seconds. The last three graphs from Figure 30 to Figure 32, show how the improvements of the American Valves have resulted in faster propagation times for a service reduction of the brake pipe by improvements to the quick service function.

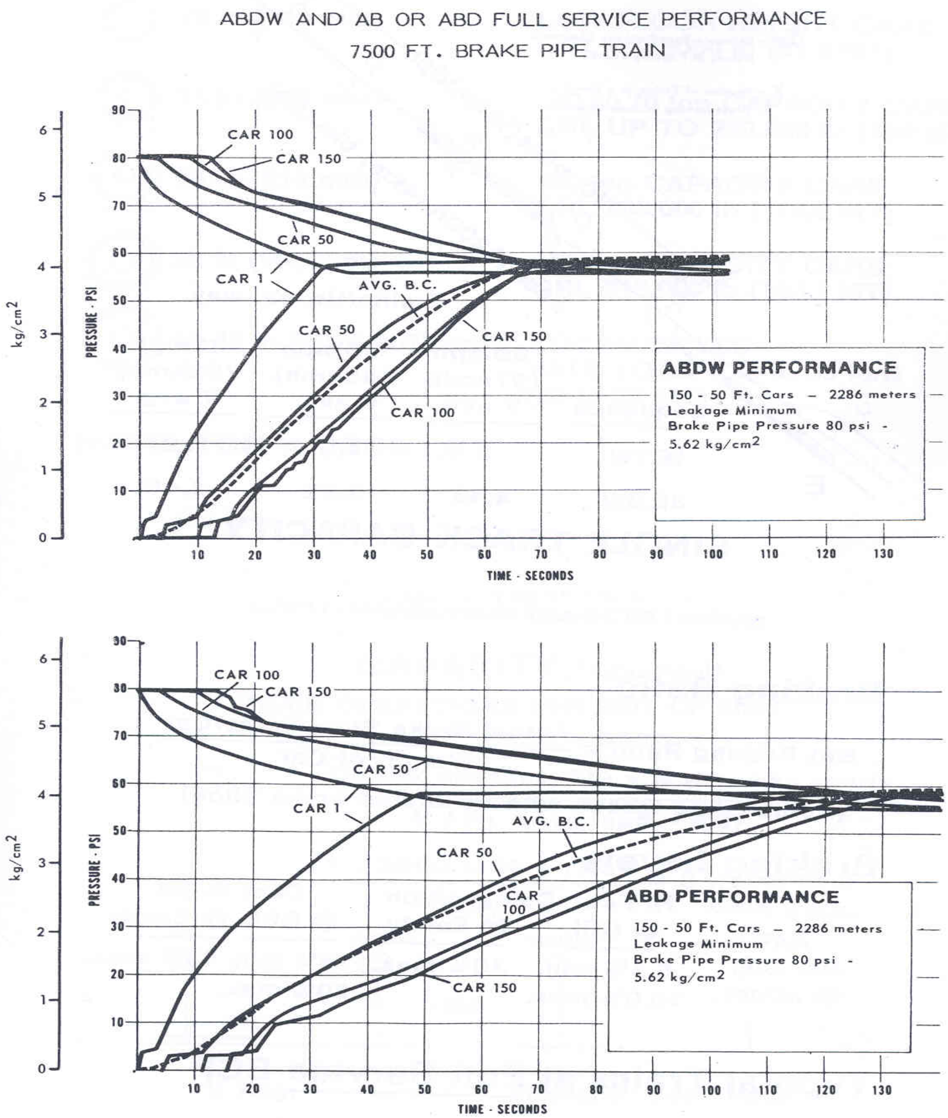


Figure 33: ABDW and ABD tests on a full service reduction.

The graphs shown in Figure 33 are a comparison of an ABD valve and an ABDW valve with the accelerated application function(Jones 1979). The ABDW shows a much faster build up of pressure in the brake cylinder. These tests started with a brake pipe initial pressure of 551 kPa (80 PSI).

The comparison between a digitised test from Westinghouse Air Brake Division test rack and a simulated model is shown in Figure 34, these graphs show that modelling can produce results close to the tests results (Johnson 1986).

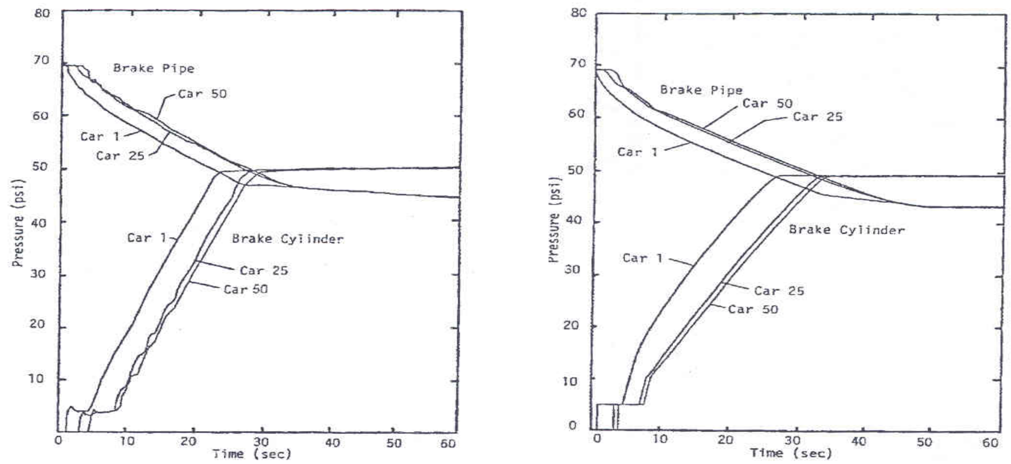


Figure 34: Results from a test rack and a simulated model. (Johnson 1986)

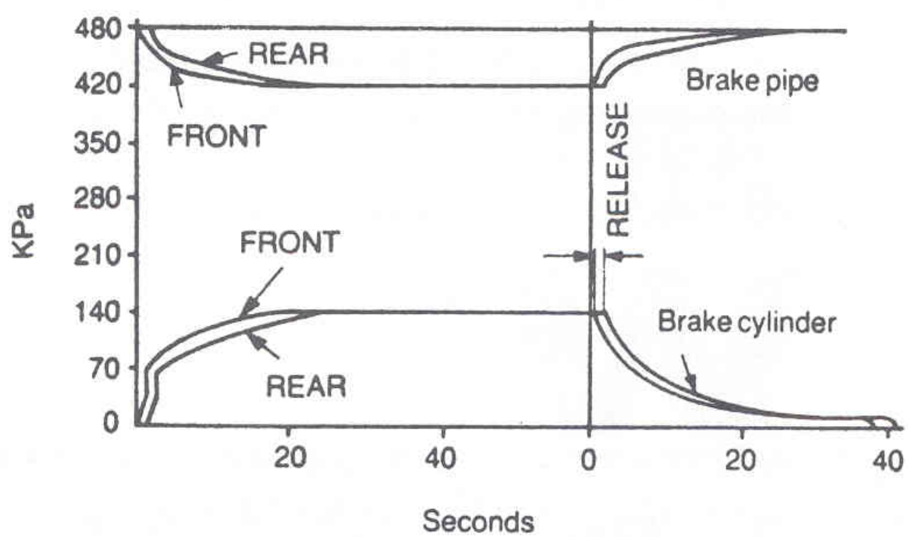


Figure 35: Service application on a 25 car train (Westinghouse Australia)

The graph shown in Figure 35 is provided by Westinghouse Air Brake Australia, this shows a partial service application. The brake pipe diameter was 32 mm and the tests were carried out on a train test rack of 25 cars. [Note: The lower right hand side part of the graph shows results of an accelerated release]. The results show that for a reduction of brake pipe pressure from 480 kPa to a pressure of 420 kPa, the rear of the train at 25 cars, the lower pressure is reached within 25 seconds. The graph also shows that in the release mode the brake cylinder pressure of 140 kPa has dropped to zero in just over 40 seconds. Comparing this Australian graph to the American graphs shown from Figure 30 to Figure 33 is difficult as the Australian test is on 25 cars, each 13.2 m long and the American tests are with 150 cars each 15.3 m long. The first three American graphs were for a service application (103 kPa) the next graph was for a maximum service application (150 kPa). The Australian graph was for a partial service application (60 kPa).

2.6 Comparison of American and Australian triple valves

From the charts in the previous section of the American triple valve data, times can be extracted to show the progression of the improvements that have been made to these valves. Table 2 shows the times in seconds and the pressure in the brake cylinder reached at various car numbers. The last three columns show how the inclusion of the AAV in the ABDW and ABDX give a quicker time to reach the same pressure.

Table 2: Times of American Valves.

	$ABD_{(1)}$	$ABDW_{(2)}$	$ABDX_{(3)}$	$ABDW_{(4)}$	$ABDW_{(5)}$
137 kPa	49	23.5	22	21	20
206 kPa	88	68	34.5	30	28
255 kPa	165	143	94	36	35

Explanatory notes for the above table

(1) Initial pressure of 550 kPa and a reduction of 103 kPa car 75 on a 150 car train.
(2) Initial pressure of 628 kPa and a reduction of 103 kPa car 75 on a 150 car train.
(3) Initial pressure of 628 kPa and a reduction of 103 kPa car 75 on a 150 car train.
(4) Initial pressure of 628 kPa and a reduction of 150 kPa car 50 on a 50 car train.
(5) Initial pressure of 628 kPa and a reduction of 150 kPa car 50 on a 150 car train.

In Table 3 the speed of the signal to start to operate a triple valve located at the 75th car on some American valves, and at the 25th car and 71st car on Australian valves and 50th car on an American valve. The table shows the improvements made to the American valves ABD, ABDW and ABDX when the AAV is used. The comparison to the 'W' triple at a minimum service application of 50 kPa reduction, show that the propagation rate of the 'W' triple is as good, if not better, than the American valves at a service application of 100 kPa.

Table 3: Propagation rates of American and Australian triple valves.

	ABD	ABDW	ABDX	'W' triple	'W' triple	ABDX
Triple starts to operate	12 seconds at 75th car	8 seconds at 75th car	8 seconds at 75th car	2.5 seconds at 25th car	4 seconds at 71st car	3 seconds at 50th car
Length of brake pipe	1143 m	1143 m	1143 m	330 m	1065 m	754 m
Brake application mode	Service	Service	Service	Minimum Service	Emergency	Full Service
Calculated propagation rate	95 m/s	142 m/s	142 m/s	132 m/s	266 m/s	254 m/s

Using all the shown graphs and combining the results, a table has been made and as shown in Table 4 the propagation rate of various brake pipe reductions at the position of the 25th car can be seen. The results show that the Australian brake pipe propagation rate is faster than the American counterpart.

Table 4: Propagation rates of triple valves at the 25th car.

	ABD	ABDW	ABDX	'W' triple	'W' triple	ABDX
Triple starts to operate	5 seconds at 25 cars	4.5 seconds at 25 cars	4 seconds at 25 cars	2.5 seconds at 25 cars	1.4 seconds at 25 cars	2.5 seconds at 25 cars
Length of brake pipe	381 m	381 m	381 m	330 m	375 m	381 m
Brake application mode	Service	Service	Service	Minimum Service	Emergency	Full Service
Calculated propagation rate	76 m/s	84 m/s	95 m/s	132 m/s	267 m/s	152 m/s

Explanatory notes for the above table

- The number of wagons the American valves used was 150 whereas the Australian tests were from using 25 wagons.
- The American valve as explained in 2.3.3 allows the first stage of the quick service volume to expel to atmosphere, whereas the Australian valve as explained in 2.4.1 holds the volume of compressed air in the quick service until a release is made.
- The American valve also utilises an AAV as explained in 2.3.2 to help ensure the propagation of the local reduction of pressure in the brake pipe.

2.7 Conclusion

The literature review has shown evolving improvements to the control valve and has shown improvements to the application delays especially in North America. The review shows that since 1989, little has been researched into improvements especially in Australia, when the wagon brake pipe length per control valve has increased and the train length has also increased up to 2840 m (Sismey 2000). The improvements, by the use of a 'Continual Quick Service Valve Device' or AAV and the 'Brake Accelerator' or 'Service Brake Accelerator' or similar apparatus have not shown to have been researched for the Australian rail industry.

3. Theory and Equipment

3.1. Pipe Equations

To analyse the speed of a pressure wave a control volume that moves with the wave is chosen as shown in Figure 36 (Gerhart 1985)

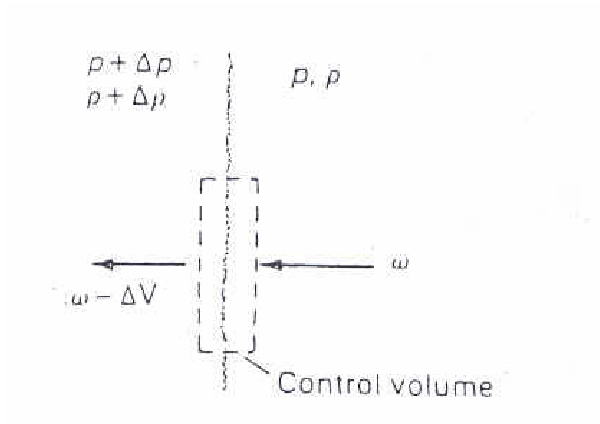


Figure 36: Compression wave in a fluid (Gerhart 1985)

Since the wave is very thin, it is assumed that it is has no thickness so all the areas of the front and back faces of the control volume are equal and the volume is zero. Then the continuity equation for a control volume is:

$$\dot{m} = \rho A \omega = (\rho + \Delta\rho) A (\omega - \Delta V) \qquad \textbf{(3.1)}$$

Where $\dot{m}$ = mass flow rate, ρ = density, A = cross sectional area, ω = velocity of the wave, ΔV = velocity of the fluid behind the wave and p = pressure of the fluid. Then cancelling A and solving for ΔV gives:

$$\Delta V = \omega \left(\frac{\Delta\rho}{\rho + \Delta\rho} \right) \qquad \textbf{(3.2)}$$

Applying the linear momentum equation to the control volume as the ‘sides’ of the control volume are vanishing small, the only force on the control volume is due to the pressure on the inlet and outlet faces.

$$\sum F_x = pA - (p + \Delta p)A = \dot{m}[(\omega - \Delta V) - \omega] \quad \textbf{(3.3)}$$

Substituting for $\dot{m}$ and solving for Δp will give:

$$\Delta p = \rho \omega \Delta V \quad \textbf{(3.4)}$$

An expression for wave speed is obtained by substituting **(3.2)** into **(3.4)**

$$\omega^2 = \frac{\Delta p}{\Delta \rho}\left(1 + \frac{\Delta \rho}{\rho}\right) \quad \textbf{(3.5)}$$

An isentropic (adiabatic) process is assumed because there is minimal opportunity for heat transfer to or from the fluid as it passes through a thin wave. The speed of sound is therefore defined by:

$$c^2 \equiv \left.\frac{\partial p}{\partial \rho}\right)_s \quad \textbf{(3.6)}$$

Where c = speed of sound

For an ideal gas, an isentropic process obeys the equation:

$$\frac{p}{p_{ref}} = \left(\frac{\rho}{\rho_{ref}}\right)^k \text{ and so } \left.\frac{\partial p}{\partial \rho}\right)_s = k\left(\frac{p}{\rho}\right) = k\mathrm{RT} \quad \textbf{(3.7)}$$

Where *k* the adiabatic constant, characteristic of the specific gas =1.4, R = gas constant for air = 287.04 J/kg-K. T = absolute temperature in Kelvin.

The speed of sound in an ideal gas is calculated by:

$$c = \sqrt{k\frac{p}{\rho}} \quad \textbf{(3.8)}$$

or

$$c = \sqrt{k\mathrm{RT}} \quad \textbf{(3.9)}$$

3.1.1 Sonic Velocity Calculation

- When R = the gas constant for air = 287.04 J/kg-K.
- T = the absolute temperature in Kelvin.
- *k* = the adiabatic constant, characteristic of the specific gas =1.4

Using the above parameters, then the speed of sound when using equation **(3.9)** with dry air and at 32^0 C will be 350.18 m/s.

3.1.2 Calculation of pipe length for QR VSAL/S wagons.

From Figure 37 when a QR VSAL has a length of brake pipe with a measurement along the centre of the pipe, called a centreline method of length and has been calculated from the following:

VSAL Coal Wagon 106 Tonne is 14.936 m in length over headstocks, the dimensions in Figure 37 shows the pipe length in mm.

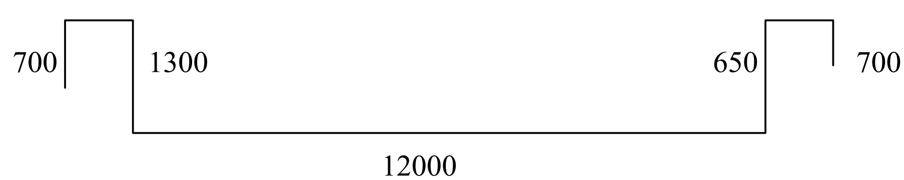


Figure 37: Centreline length of a QR VSAL/S 106t coal wagon.

Drawings supplied by QR for piping arrangement dwg. No. A0-33765

- For a VSAL when the length by centreline method is = 6 X 90^0 Pipe Bend (0.3 m radius) = 2.83 m and then +0.7 m (pipe) +1.3 m (pipe) +12 m (pipe) +0.65 m (pipe) +0.7 m (pipe) = **18.18 m.**

- For a VSAS when the length by centreline method is = 6 X 90^0 Pipe Bend (0.3 m radius) = 2.83 m and then + 0.75 m (pipe) +1.3 m (pipe) + 12 m (pipe) +0.45 m (pipe) +0.75 m (pipe) = **18.08 m.**
- The total centre line length of brake pipe for the two wagons is **36.26 m.**
- The inclusion of the hose coupling lengths between each wagon (4 x 0.7 m) will bring the overall length to **39.06 m.**

[Note] The overall centreline length of a train brake pipe is used in propagation rate calculations.

3.2 Equipment

To be able to investigate the brake application delays without a long train to use for experimental purposes, a laboratory setup of a train brake pipe was used. The test rig, because of project expenditure constraints, only had the use of 4 control valves and with appropriate pipe work was made in the same configuration as a coupled pair of wagons. The setup was to examine in detail the pressure in the brake pipe at different locations along the pipe while and after different brake reductions were made. Different sized brake pipe exhaust chokes were used to simulate a reduction rate in the experimental rig similar to results from train tests results.

For the experiments, complete sections of brake pipe from two coal wagons of the VSH type were utilised in a configuration as shown in Figure 39. The brake pipe also included nine (9) bends of 90^0 with radius of 200 mm and two flexible hose couplings, from the VSH installation. A further 83 meters of pipe with sixteen (16) bends of 90^0 with a radius of 100 mm and with four flexible hose couplings were utilised to give a combination long enough for the inclusion of four control valves. The total centreline length of the brake pipe was 120 meters. Pipe diameter was

0.0345 m (32 NB heavy gal. pipe) and the volume of the pipe was 112.17 litres. A 37 litre reservoir was fitted to the end of the pipe.

For the series of experiments completed, the 1st control valve is connected to the brake pipe via a tee junction. Positioned on one side of the tee junction is an exhaust, which has an exhaust choke for controlling the brake pipe flow and hence the rate of brake pipe pressure reduction. The other side of the tee is connected to the remainder of the brake pipe. At a distance of 37 meters from the first control valve a tee to the pipe connects a second control valve. A third control valve 37 meters from the second control valve is connected by using another tee to the pipe and a fourth tee connects the fourth control valve at a further distance of 43 meters. The pipe extends for a further 3 meters and is connected to a reservoir of 37 litres capacity. The 37 litre reservoir was added to the brake pipe to reduce reflection of pressure waves. An extra 6 meters of pipe was fitted between the third and fourth control valve because of space limitations within the lab area for positioning of control valves.

Positions of the pressure transducers on the brake pipe as shown in Figure 39 and on the brake equipment were as follows:

- Transducer P1 fitted at the brake pipe side of the main diaphragm on 1st control valve
- Transducer P2 fitted at the brake pipe side of the main diaphragm on 2nd control valve
- Transducer P3 fitted at the brake pipe side of the main diaphragm on 3rd control valve
- Transducer P4 fitted at the brake pipe side of the main diaphragm on 4th control valve
- Transducer P5 at the brake pipe tee to the 1st control valve
- Transducer P6 fitted after the tee to the 2nd control valve

- Transducer P7 fitted after the tee to the 3rd control valve
- Transducer P8 fitted 3 meters after the 4th control valve

A data acquisition system taking samples at 250 /s was used to record pressures.

From the data obtained after running the experiment, the velocity of the waves, the rate of reduction and the size of the reduction as it travels along the pipe could be calculated.

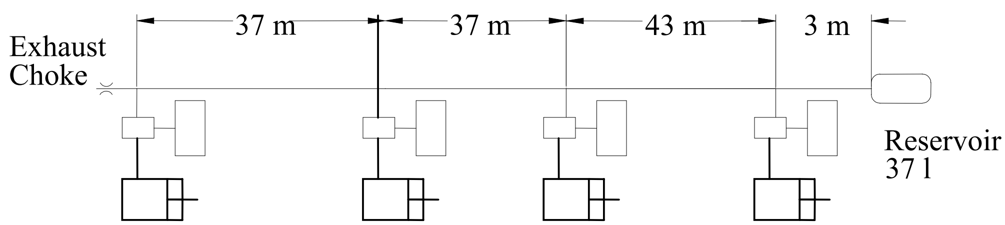


Figure 38: Schematic of brake pipe test rig setup.

Equipment used during testing:

- 120 meters of 32 mm NB heavy duty galvanised pipe
- Westinghouse WF4 Control valve # NWFF302
- Westinghouse WF4 Control valve # NWFF303
- Westinghouse WF4 Control Valve # NWFF304
- Westinghouse WF4 Control valve # NWFF305
- ‘K' type thermocouples.
- National Instruments DAQ comprising of PXI 1002 General purpose chassis.

  NI PXI-8175 Pentium III Embedded Controller for PXI.

  NI PXI-6040E 500 kS/s (1-Channel), 250 kS/s (Multichannel), 12-Bit, 16 analog input multifunction DAQ
- Sh 68 _ 68 EP 1m shielded cable.

- CB_68LP terminal block.
- WABCO (SETRA) transducer Model 207 pressure range 0-100 PSIG, Excitation 12-28 VDC, Output 0.1-5.1 VDC, Accuracy ±0.13% Full Scale (RSS method)
- Druck DPI 705 Digital Pressure Indicator
- Labwindows CVI 5.0 was used with ANSI C for the programming

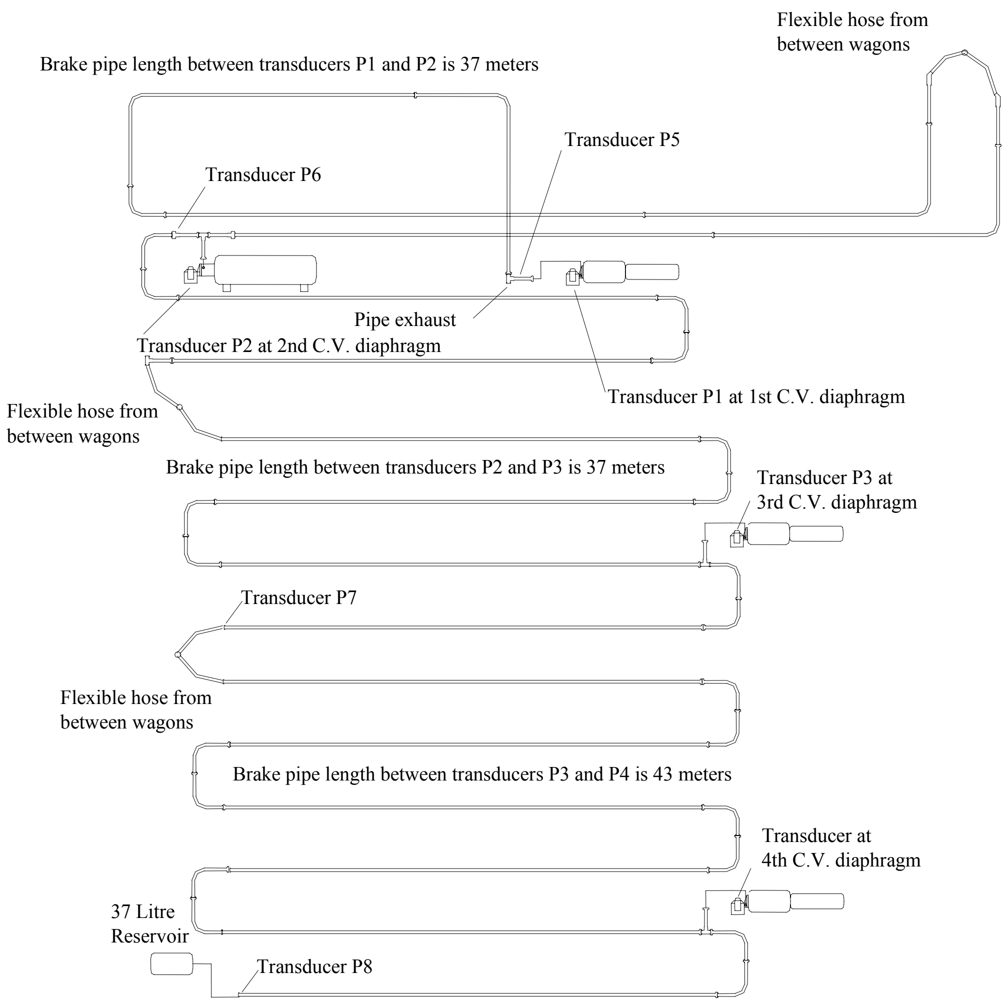


Figure 39: The experimental brake pipe rig and transducer locations.

The temperature of the air in the pipe at the time of testing was 32^0 C, humidity approximately 55%.

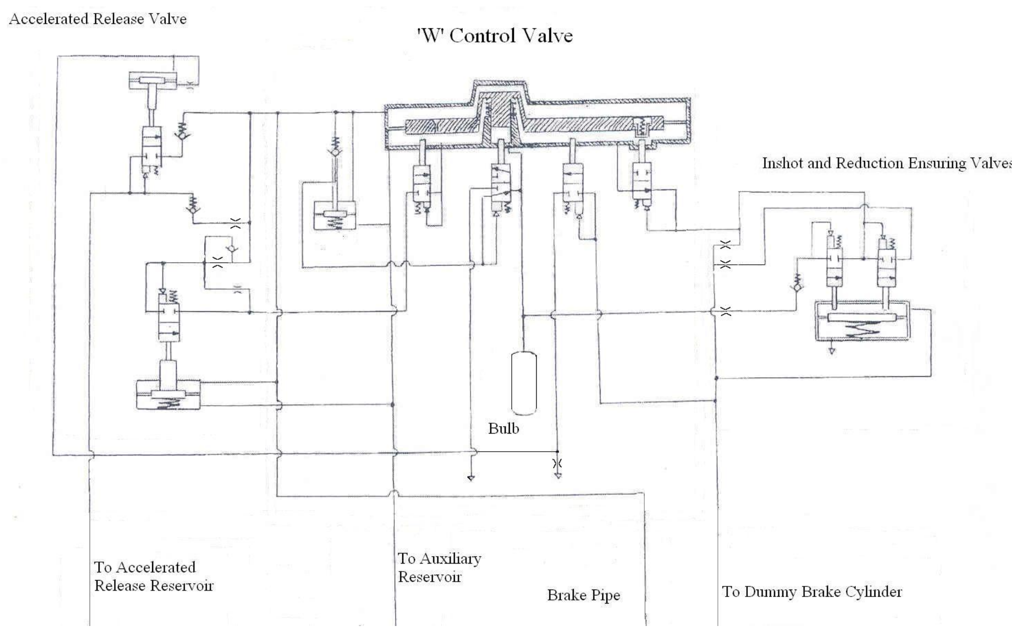


Figure 40: 'W' control valve schematic

The four 'W' style control valves used on the test rig are typical of the schematic shown in Figure 40 and of the diagrammatic view as shown in Figure 41. The both figures show the features and functions that are added to a standard valve shown in Figure 17.

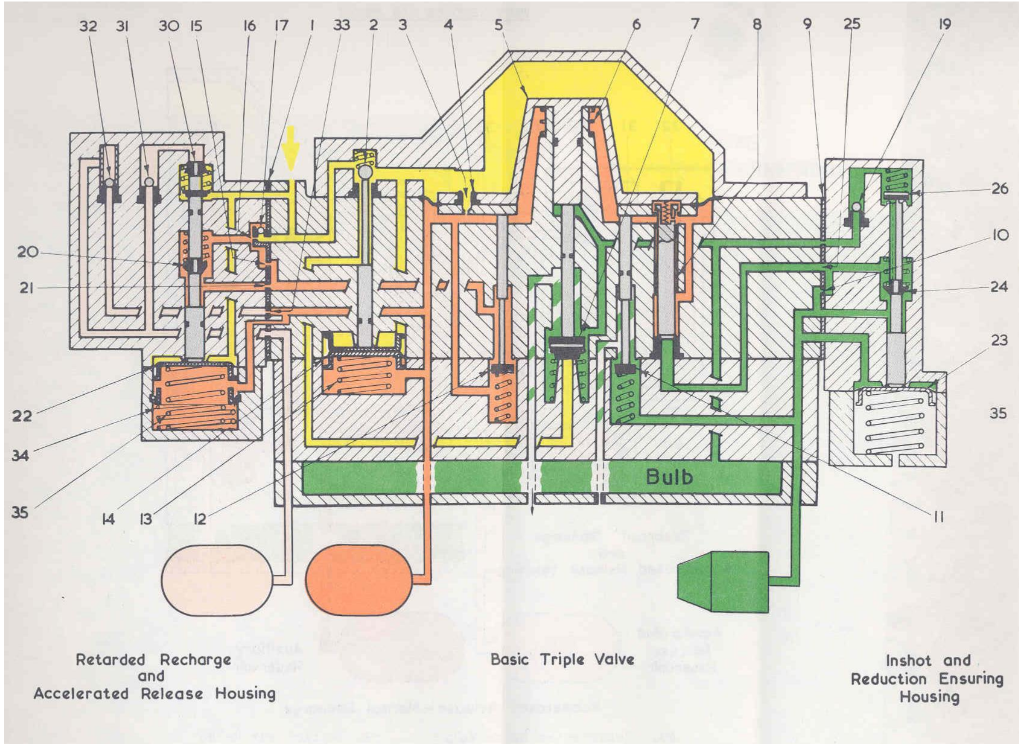


Figure 41: Diagrammatic view 'W' control valve

3.3 Data Acquisition and Processing

Transducers used had a rate of 137.89 kPa/volt. Using 12 bit resolution, pressures could be resolved to 0.6733 kPa. This gave a step like appearance to plotted data when small changes are highlighted in results.

The Wabco 207 pressure transducers were checked and calibrated by comparison with the Druck 705 pressure indicator.

While channels were scanned once every 0.004 seconds inter-channel time skew was minimised by a channel to channel time of 10^{-5} seconds.

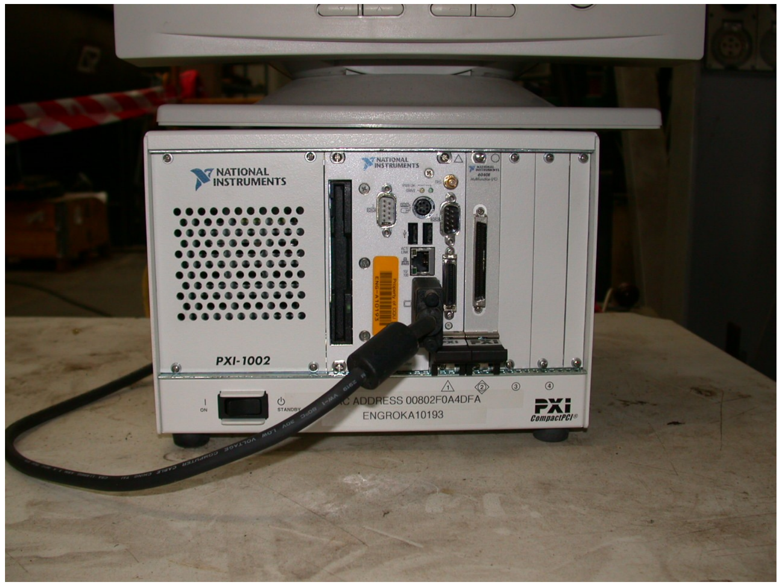


Figure 42: National Instruments DAQ

Figure 43: Sh 68 _ 68 EP shielded cable, CB_68LP pin terminal block.

Figure 44: Druck 705 pressure indicator.

4. Experiments and Results

4.1 The rate and size of reduction of the gulp in a brake pipe

4.1.1 Introduction

The purpose of this experiment is to measure the propagation speed compared to sonic velocity and size of a gulp in a short length of experimental brake pipe. The experiments are in two parts, the first part measures the effects of the bulb action in the brake pipe of 120 m in length from operation of one control valve. The second part measures the effects of the bulb action in the brake pipe when four control valves are operated within the same 120 m of brake pipe.

Control valves fitted to wagons are controlled by the pressure of the air in the train brake pipe. A reduction of pressure of the air in the brake pipe has the effect of activating a brake application. The reduction in pressure is initiated by exhausting air from the front end of the brake pipe, usually via a valve in the locomotive. The precise instant that a control valve is activated is determined by the instant when a specified pressure difference between the brake pipe pressure and the auxiliary reservoir pressure of that control valve is reached.

From gas dynamics theory it is known that a pressure disturbance wave from the sudden opening of a valve at one end of a pipe will be propagated at sonic velocity through the gas. While the pressure wave corresponding to the initiation of a pressure reduction in the brake pipe propagates at sonic velocity, the pressure reduction that is sufficient in size to activate the control valve is limited by the actual gas flow exhausting from the pipe. As the pressure reduction propagation distance becomes longer the pressure wave front diminishes in size and therefore the pressure reduction of adequate size to operate the control valve takes longer to occur.

In Australian designs, additional exhaust gas flows are initiated at each control valve by the use of a cavity or volume known as the “bulb” or the quick service volume. The opening of the brake pipe to connect to this volume creates a further pressure disturbance wave that assists the exhausting gas flow and locally increases the rate of pressure reduction available to the next valve. In the first of experiments in this chapter, the pressure wave developed by the quick service volume of a control valve in a wagon brake pipe is measured and analysed along a pipe by using pressure transducers and data acquisition.

4.1.2 Equipment

For measurements of the pressure reduction rate in a brake pipe, the equipment used is detailed in section 3.2 and the configuration is as shown in Figure 39.

4.1.3 Method

The air in the pipe was compressed to 500 kPa and allowed to stabilise. Air was then exhausted through the exhaust choke. Choke of diameters of 2 mm, 1.5 mm, 1.2 mm and 0.91 mm were selected by testing various sizes of chokes then measuring the reduction drop rate of each choke. The choke diameters were then enlarged or new smaller ones made so the results would compare favourably with drop rates observed in train line data. The selection of chokes diameters allowed the discharge from 120 meters of test rig brake pipe to give a pressure drop similar to that seen at either:

- The 21st wagon with head and mid locomotives shown in Figure 45
- The 41st wagon with head and mid locomotives shown in Figure 45.
- The 92nd wagon with only head end locomotive power as shown in Figure 46

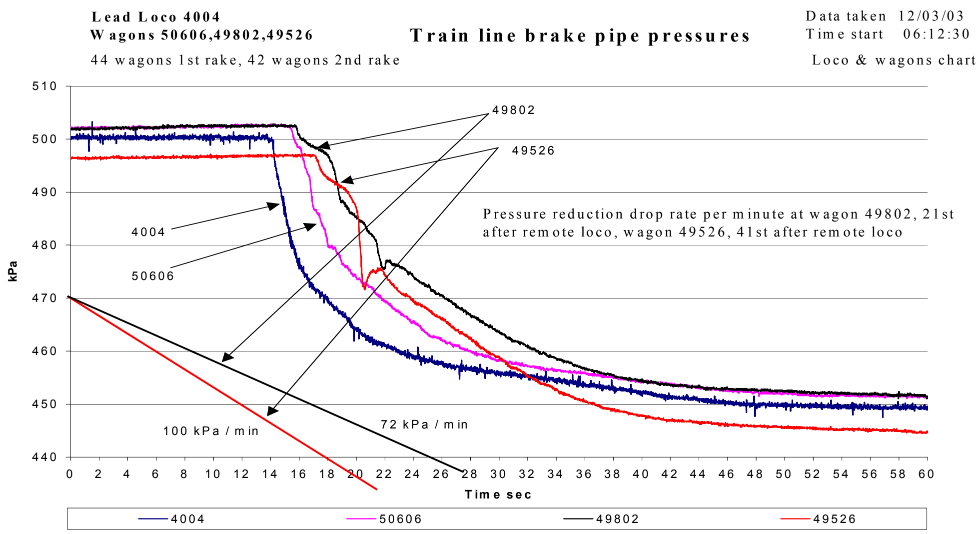


Figure 45: Brake pipe reduction. (CQU RA1.1 tests 2003)

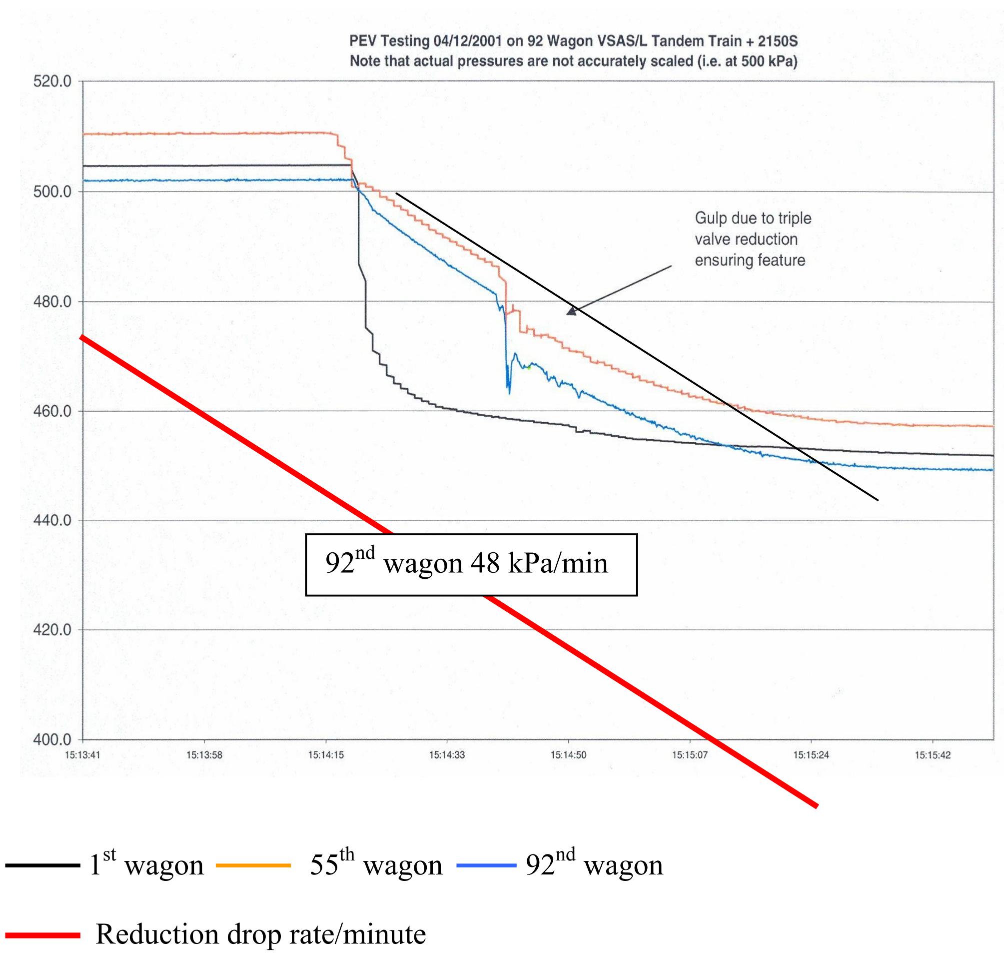


Figure 46: Reduction rate of a 92 wagon train. (Queensland Rail, 2001)

The uncertainty in measurements and calculations using the equipment and transducers is quantified as follows:

Using 250 samples per second this gives one sample every 0.004 seconds. (1/250)

With a pipe length of 120 meters then this gives 0.00083 m. (0.1/120)

Then a variance of +/- 0.00083 meters and +/- .004 seconds can be used.

A time period over the 120 m is 0.340 seconds as seen in Figure 48.

$$\Delta Z = Z\sqrt{\left(\frac{a}{A}\right)^2 + \left(\frac{b}{B}\right)^2} \qquad \textbf{(4.1)}$$

Where a = .0083 meters

A = 120 metres

b = 0.004 seconds

B = 0.340 seconds

Z = 352 m/s

$\Delta Z =$ 4.14 m/s

The velocity of 352 m/s from Figure 50 and an uncertainty of +/- 4.1 m/s therefore indicate sonic velocity in the range 347.9 m/s to 356.1 m/s.

4.1.4 Results

The results shown in Table 5 and in Figure 48 are from a brake pipe length of 120 meters and with 1 control valve. The brake pipe reduction rate was 54 kPa/min by using an exhaust choke of 1.2 mm diameter. The graphs show the brake pipe pressure being steadily reduced until the first control valve operates. After the valve operation the brake pipe air pressure (P1) measured at the control valve sharply reduces from 488.1 kPa at time of 13.9 seconds as the filling of the bulb from the brake pipe pressure in the cavity above the diaphragm has commenced. The pressure at P1 drops rapidly to 430.2 kPa at 14.1 seconds. When the bulb volume has begun to be filled, the brake pipe air pressure at P1 increases from a pressure of 430.2 kPa and is then stabilised at a pressure of 486.7 kPa at time of 14.7 seconds. The sharp reduction shown and the returning to a steady reducing pressure can be called a 'local gulp' and can best be seen in the brake pipe pressure of the control valve measured in the cavity above the diaphragm in the control valve. The gulp visible in the brake pipe is modified by the restrictions to flow in the branch pipe and passageways that connect to the cavity above the control valve diaphragm. The local pressure reduction in the brake pipe measured at P5 shown in Figure 48 is the final pressure drop produced in the pipe from the control valve bulb function. The 'local gulp' of brake pipe pressure seen at the control valve has been reduced to only a comparatively small reduction in pressure in the brake pipe. The smaller local reduction of the pressure in the brake pipe has been a result of the restricted mass flow through the branch pipe and passageways of the control valve from the larger volume of the brake pipe. It is this local reduction in the brake pipe pressure that propagates along the brake pipe and is shown at transducers P6, P7 and P8. The local reduction in pressure in the brake pipe is also known as the brake pipe gulp or more commonly the 'gulp' as shown in Figure 46.

Table 5: Brake pipe pressures from a single control valve operation.

Time (s)	P1 (kPa)	P5 (kPa)	P6 (kPa)	P7 (kPa)	P8 (kPa)	Comments
0	499.6	499.5	499.8	500.3	500.3	Pipe at stabilised pressure
2.8	498.9	497.7	499.5	498.9	498.9	Start of B.P. exhaust
13.9	488.1	489.7	489.4	489.5	489.5	Start of C.V. application
13.94	481.4	488.3	489.4	489.5	489.5	Start of pressure wave at P5
14.03	430.2	486.3	489.4	489.5	488.8	Bottom of 'gulp' at P1
14.06	436.9	484.9	488.7	489.5	489.5	Bottom of pressure at P5
14.28	478.7	486.9	487.4	486.1	488.8	Start of pressure wave at P8
14.41	481.4	486.9	488.1	486.8	485.5	Bottom of pressure at P8
14.68	486.7	486.9	488.1	486.8	486.8	P1 stabilising with P5

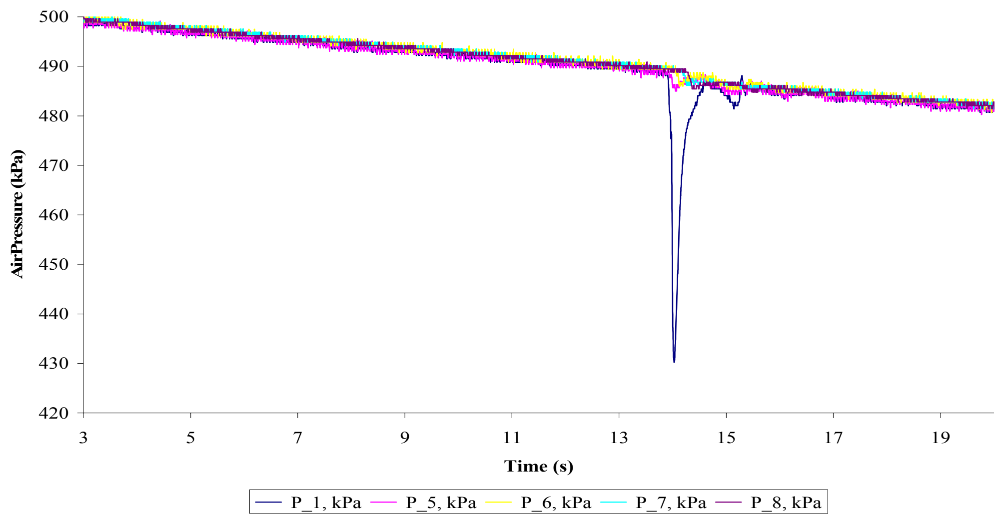


Figure 47: Pressure reduction rate of 54kPa/min and a single valve operation.

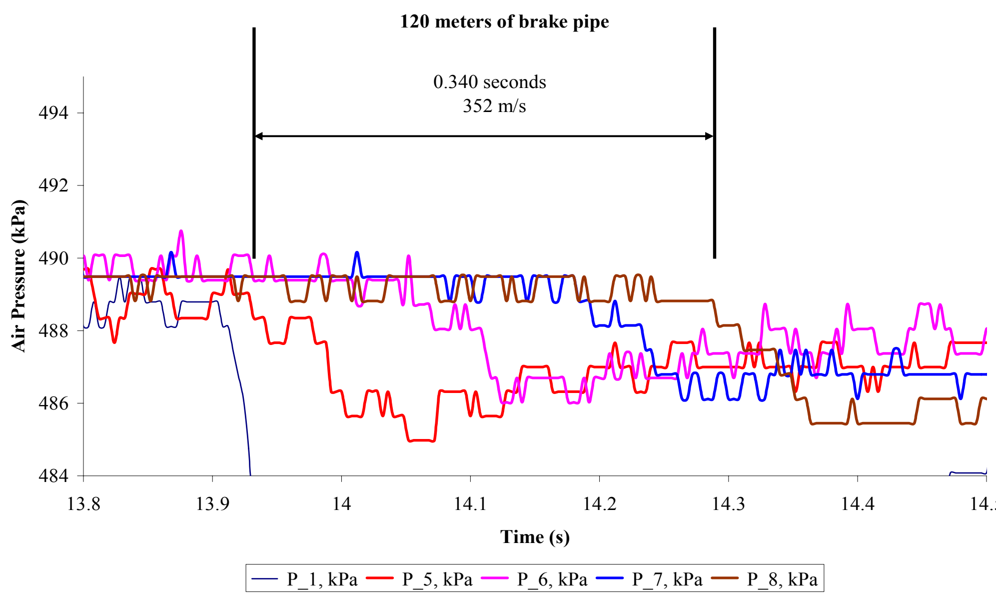


Figure 48: Gulp propagation resulting from the single valve operation.

The results shown in Table 6 and in Figure 49 and Figure 50 are from a brake pipe length of 120 meters and with 4 control valves. The brake pipe reduction rate was 48 kPa/min and was achieved by using an exhaust choke of 0.91 mm diameter. The graph in Figure 49 shows the brake pipe pressures being reduced after the exhaust choke is opened to atmosphere, leading to a steady reduction of 48 kPa / min. The four control valves each show a ‘local gulp’ measured at the cavity above the diaphragm of each valve. The graph in Figure 50 shows the local pressure reduction produced from each control valve bulb operation and the size of the local pressure reduction as it travels along the pipe. The results from Table 6 show the same characteristics of a 'local gulp' as shown in the previous table although with different results for the local pressure reduction in the brake pipe at each transducer location.

Table 6: Brake pipe pressures from 4 control valve operation.

Time (s)	P1 (kPa)	P5(kPa)	P6 (kPa)	P7 (kPa)	P8 (kPa)	Comments
0	500.2	500.5	500.2	500.3	500.3	Pipe at stabilised pressure
1.92	499.6	499.8	499.5	500.3	499.6	Start of B.P. exhaust
18.14	486.1	486.9	487.4	487.5	487.5	Start of C.V. application
18.17	478.7	486.9	487.4	487.5	486.8	Start of pressure wave at P5
18.31	445.1	483.6	486.7	488.1	487.5	Bottom of pressure at P5
18.51	476.7	485.7	484	484.1	486.9	Start of pressure wave at P8
18.75	480.1	481.6	483.3	483.7	478.7	Bottom of pressure at P8

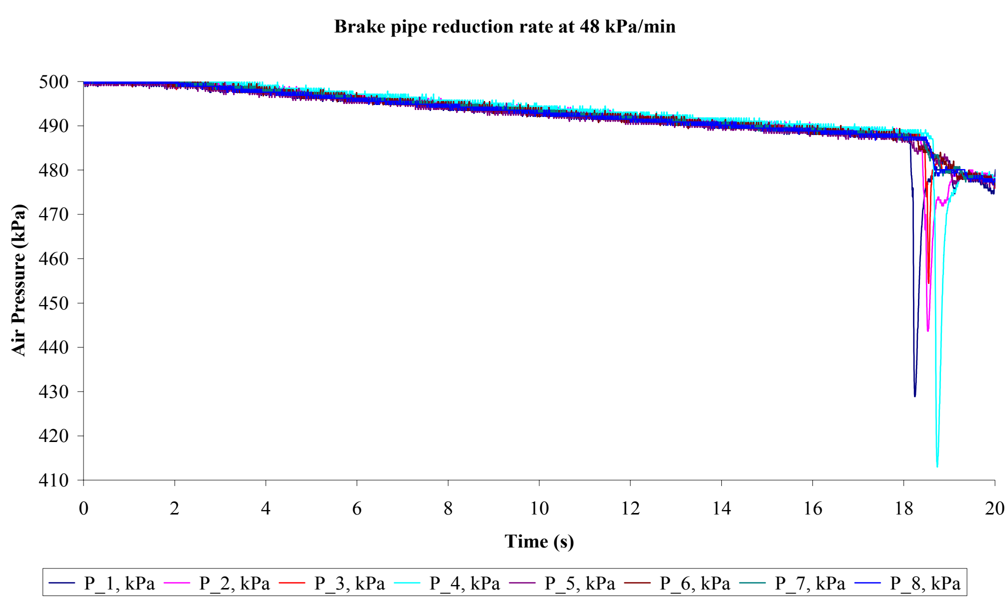


Figure 49: Pressure reduction rate and the operation of four valves.

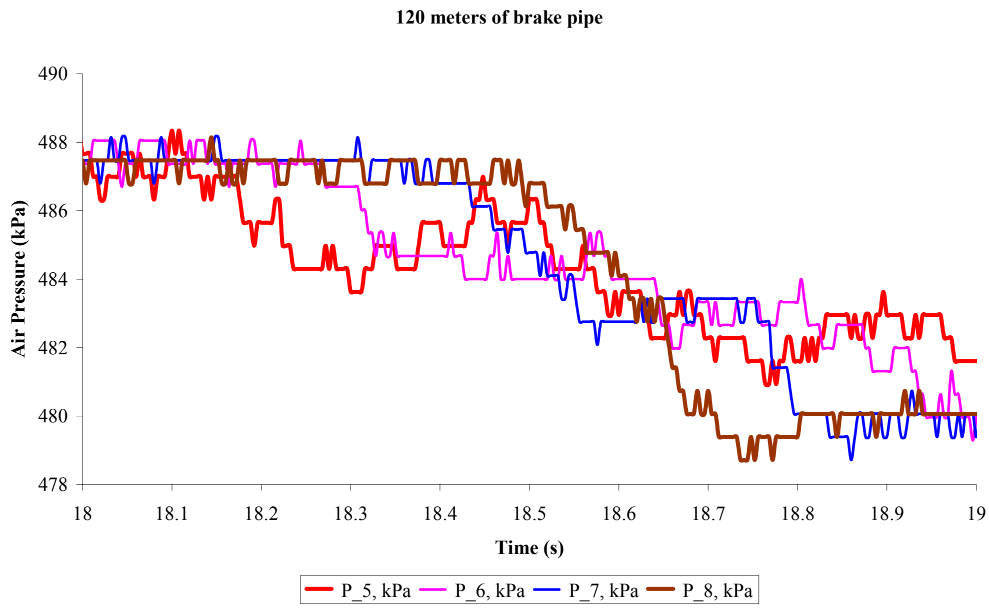


Figure 50: Gulp propagation with all 4 control valves operational.

4.1.5 Discussion

a) Rate of Reduction and Size of reduction from a single control valve operation

The minimum pressure reduction rate required to operate a control valve is 30 kPa/min (Westinghouse 1973). A reduction rate of ~ 100 kPa/min measured at the rear rake of 42 wagons has been shown for distributed power trains as shown in Figure 45. Further data from tests by Queensland Rail 04/12/2001 of longer trains have shown a reduction rate of 48 kPa/min at the rear of 92 wagons of a head end train, as shown in Figure 46. The pressure reduction rate of 54 kPa/min was chosen for the single control valve test to ensure that the valve would commence to operate. Previous testing with a pressure reduction rate of 30 kPa/min failed to operate the valve.

The results in Table 5 show that when a single control valve has been subjected to a brake pipe drop rate of 54 kPa/min, the valve has functioned as the pressure differential between the brake pipe pressure and the auxiliary pressure has approached 12 kPa. During the brake application, the start of the pressure wave in the brake pipe produced by the quick service volume of the control valve shows a sharp drop in brake pipe pressure of 3.4 kPa as measured at 13.94 seconds at transducer P5 shown in Figure 48. The pressure measured at 14.28 seconds at transducer P8 that is 120 meters from transducer P5, and shows the corresponding sharp drop of 3.3 kPa.

As shown in Figure 48 it takes 0.340 seconds for the pressure wave to travel from P5 to P8 a distance of 120 meters and with a velocity of 352 m/s. This compares well with the calculation of speed of sound in section 3.1.1 of 350.18 m/s of dry air and at 32^0 C with the variance from equation **(4.1)**.

b) Size of reduction from 4 control valve operation

The results from this test are shown in Table 6. Four control valves are operated in sequence by the propagation of the brake pipe pressure reduction, and local sharp pressure drops are developed by each quick service volume. The results have shown progressive increases in the local pressure reduction when four control valves are connected to the brake pipe.

The start of the pressure wave (a quick drop of 3.3 kPa) produced by the quick service volume of the first control valve was measured at 18.17 seconds at transducer P5 shown in Figure 50. The bottom of the pressure drop measured at 18.75 seconds at transducer P8 is a combination of the original pressure wave or gulp and the sum of the local reduction of pressure drop in the brake pipe produced by the quick service volumes of the control valves. The combined gulp has shown a drop in pressure of 8.2 kPa measured at the gulp at P8. The sum of the increase in the gulp had begun at P5 at 483.6 kPa and finished at P8 of 478.7 kPa and was an increase of 4.9 kPa in gulp in 120 meters with four control valves.

The tests as shown in Figure 50 show no increase in the combined local reduction in pressure after the 2nd control valve and it is this control valve that has a pipe bracket between the branch pipe and valve. The combined increase of the gulp pressure drop shown in this series of tests is seen after the 2nd control valve when the 3rd and 4th control valves have operated. Therefore the results of this test suggest that having an increasing number of control valves in a train line the pressure wave or gulp could be made to grow larger in size as propagation progresses. The gulp from the reduction ensuring feature and quick service volume is seen in Figure 46 where the gulp increases in size from wagon 55 to wagon 92.

4.2 Local Pressure reduction using different branch pipe sizes

4.2.1 Introduction

The objective of this experiment was to measure and compare the size of the gulp in the brake pipe due to the effects of different lengths and internal diameters of the branch pipe from the bulb action when a control valve operates.

Industry survey has shown that branch pipes connecting the control valve to the train brake pipe differ in both diameter and lengths see Figure 51. There are also two ways the branch pipe connects to the control valve. One system is to have the branch pipe connected to a port on the control valve. Another system uses a multi compartment reservoir which includes a mounting bracket. The control valve is mounted to this bracket and the branch pipe is connected to a port on this bracket. The mounting bracket is normally called a pipe bracket.

In these experiments the local pressure reductions developed when using different length and size branch pipes to a control valve pipe bracket were measured. The response of the system was recorded and analysed using pressure transducers and data acquisition.

Plastic branch pipe. Long steel branch pipe.

Figure 51: Branch pipes showing different material and lengths.

4.2.2 Equipment

For experimental measurements of the pressure reduction rate resulting from different length and size branch pipes, the pipe arrangement shown in Figure 39 was used. In this series of experiments, the 2nd control valve utilises a pipe bracket. A selected series of different branch pipes are connected between the tee at the brake pipe and pipe bracket. The transducers are fitted as shown in Figure 39 . From the data obtained after running the experiment, the rate of reduction and the size of the reduction can be established.

4.2.3 Method

Various branch pipes, of which two are shown in Figure 52, were connected between a tee in the brake pipe to the pipe bracket port. Each test was conducted with the air in the brake pipe being compressed to 500 kPa and allowed to stabilise. Air was then exhausted through an exhaust choke. The diameter of the exhaust choke used in these experiments was of 1.5 mm, this being to allow a comparison of the branch pipes with typical brake pipe train installations. These experiments were performed with all four valves connected to the brake pipe. The results were checked for consistency with three repeat tests. The pipe pressures were taken only from transducer P6 which is positioned after the 2nd control valve and as shown in Figure 39.

The test number along with the length and sizes of the branch pipe used are shown in Table 7.

Table 7: Branch pipe length and sizes.

Test No.	Branch pipe length (mm)	Branch pipe I.D. (mm)
608	150	20
700	800	20
718	800	12

Figure 52: 20 mm x 150 mm and 20 mm x 800 mm branch pipes.

4.2.4 Results

The results shown in Table 8 and Figure 53 are from brake pipe pressures from 120 meters of pipe with 4 control valves at spacings as shown in Figure 38 and with the 2nd control valve having the pipe bracket.

These results of the local reduction in brake pipe pressure are extracted from three experiments of test numbers #608, #700 and #716 with the pipe pressure measurements from transducer P6 at the location shown in Figure 39.

Table 8: brake pipe pressures from different branch pipes.

Time (s)	Test #608 (kPa)	Test #700 (kPa)	Test #718 (kPa)	Comments
0.412	488.4	488.4	488.4	All start to drop in pressure
0.436	487.1	486.4	487.1	#718 drops least
0.468	484.4	484.4	485.1	#700 drops below others
0.492	485.1	484.4	485.1	#700 bottom of trough
0.556	485.1	483.7	485.8	#718 starting to climb

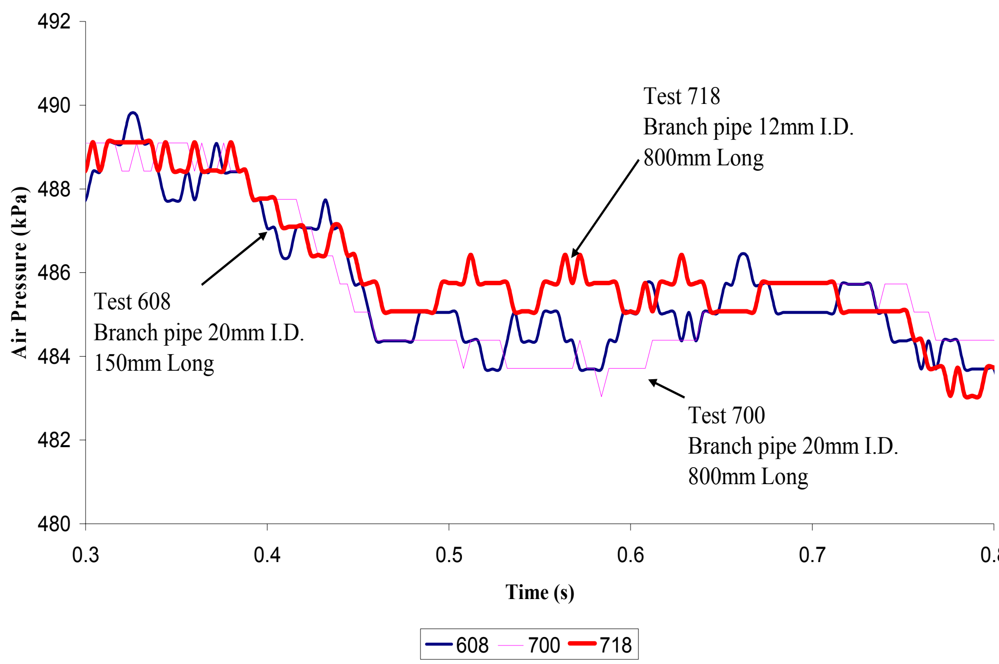


Figure 53: Gulps in the brake pipe with different size and length branch pipes.

4.2.5 Discussion

The results from Figure 53 show a difference between the greatest local pressure reduction reached in test #718 and test #700 was 1.3 kPa with #700 giving the lowest pressure at 483.7 kPa. Although measurable, this difference is almost too small to be significant when the measurement resolution is 0.67 kPa. The results show that the time that each local pressure reduction remains at the lowest pressure is also different for each sized branch pipe.

The time the pressure of test #608 remains at its lowest local pressure reduction of 484.8 kPa is 0.132 seconds. The time on test #700 is 0.076 seconds and the time on test #718 is 0.204 seconds. The results show that for test #608 the bulb function takes 0.072 seconds to induce the local pressure reduction of 4 kPa. The time for the brake pipe pressure in test #700 to reach its lowest point is 0.144 seconds with a local pressure reduction of 4.7 kPa. In test #718 the time taken to reach the lowest point is 0.076 seconds with a local pressure reduction of 3.3 kPa.

Results have shown that by using a 20 mm I.D. branch pipe 150 mm long and when compared to a branch pipe of the same diameter and 800 mm long, the local reduction in pressure drop reached can be the same. As $h_L \alpha L$ it can only be concluded that the difference is too small to measure with the equipment used. When the branch pipe is 12 mm I.D., both the reduced pressure in the brake pipe and the time the pressure is at the reduced level is less than for the larger diameter branch pipes as expected, (Darcy-Weisbach equation shows that $h_L \alpha \frac{1}{d^5}$). The three different branch pipes used in these tests have shown that the cross sectional area of the branch pipe has the major effect on the size and duration of the local pressure reduction in the brake pipe.

4.3 The Quick Service Volume

4.3.1 Introduction

The purpose of experiments with the quick service volume or 'bulb' was to compare the size of the gulp in the brake pipe from the two volumes that are used by the Australian rail industry. Another objective in this experiment was to observe the action of the bulb and the gulp in the brake pipe while performing a series of brake pipe reductions.

The quick service volume is a small reservoir built within the basic control valve body. Its purpose is to assist in the application propagation and therefore bring about a more instantaneous application of brakes throughout the train. Its capacity is designed to correspond to the volume of brake pipe on each vehicle and therefore there is a necessity for differing capacities for different wagon lengths. There are two capacities used with the current Australian control valve as explained in 2.4.3, namely 33 $x10^{-5}$ m^3 and 59 $x10^{-5}$ m^3. The train brake pipe lengths to which the aforementioned sizes are applied are up to 23 meters and from 21 to 37 meters respectively (Westinghouse 1989). The results from this series of tests will show the difference in size of the local reduction in the brake pipe for these two quick service volume sizes. The responses from bulb operations in the brake pipe of successive applications are also investigated in this series of tests.

4.3.2 Equipment

For experimental measurements of the local pressure reduction rate resulting from different size quick service volumes, the brake pipe rack and control valves as shown in Figure 39 was used. In this series of experiments, the 1st control valve is tested with the different quick service volume or bulbs. The tests also included the use of a volume connected directly to the brake pipe with the size of a large bulb. The transducers are fitted as shown in Figure 39. The container was connected to the brake pipe with a ball valve as shown in Figure 61.

4.3.3 Method

The different quick service volumes tested were of 33 $x10^{-5}$ m^3 and 59 $x10^{-5}$ m^3. These are referred to as medium and large bulbs (Westinghouse 1989).

Data was extracted from each file to produce a comparison graph showing the local pressure reduction in the brake pipe after the control valve. The 'local gulp' of air pressure was also measured at the control valve. Each test was conducted with the air in the brake pipe being compressed to 500 kPa and allowed to stabilise. Air was then exhausted through an exhaust choke. Testing that included a series of applications of the brakes from 500 kPa without making a release application was also included.

4.3.4 Results

The results shown in Figure 54 are from brake pipe pressures from 120 meters of pipe with a transducer connected to the brake pipe after the first control valve. These results of the local reduction in brake pipe pressure are extracted from two experiments of test numbers #073 and #074 with the pipe pressures from transducer P5 in each test with the location shown in Figure 39.

Table 9 is produced from the data in Figure 55.

Table 9: The local reduction in brake pipe pressures.

Time (s)	$MLPR_{(1)}$ (kPa)	$LLPR_{(2)}$ (kPa)	Comments
8.684	487.4	487.4	Pressures aligned with time
8.728	484.7	485.4	Start of corresponding gulp of the medium bulb
8.732	482.1	483.4	Start of corresponding gulp of the medium bulb
8.74	485.4	484.7	Bottom of corresponding gulp large bulb
8.748	484.1	485.4	Bottom of corresponding gulp large bulb
8.768	482.1	483.4	Bottom of local pressure reduction medium bulb
8.776	482.1	482.7	Bottom of local pressure reduction large bulb

Explanatory notes for the above table

(1) MLPR medium bulb local pressure reduction
(2) LLPR large bulb local pressure reduction

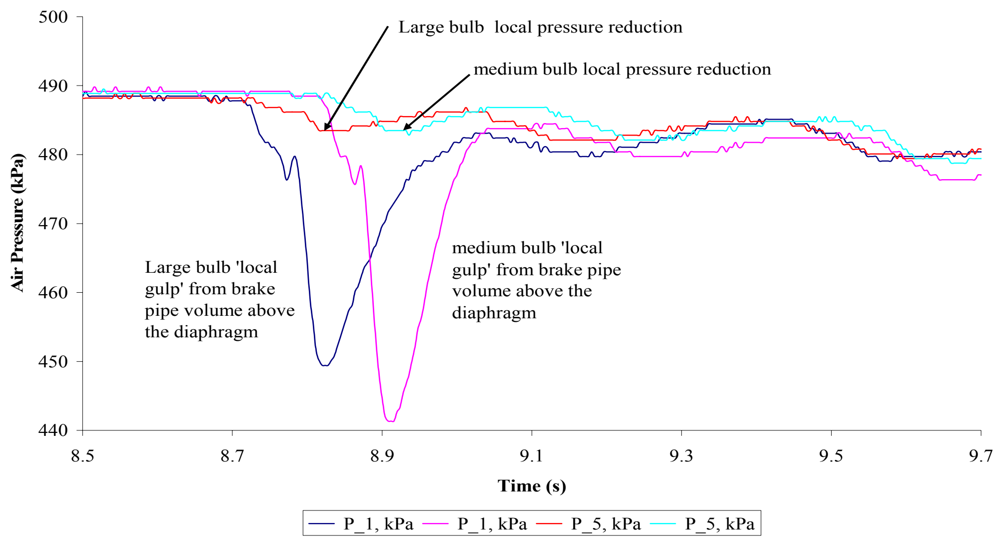


Figure 54: The 'local gulp' from a medium and a large bulb.

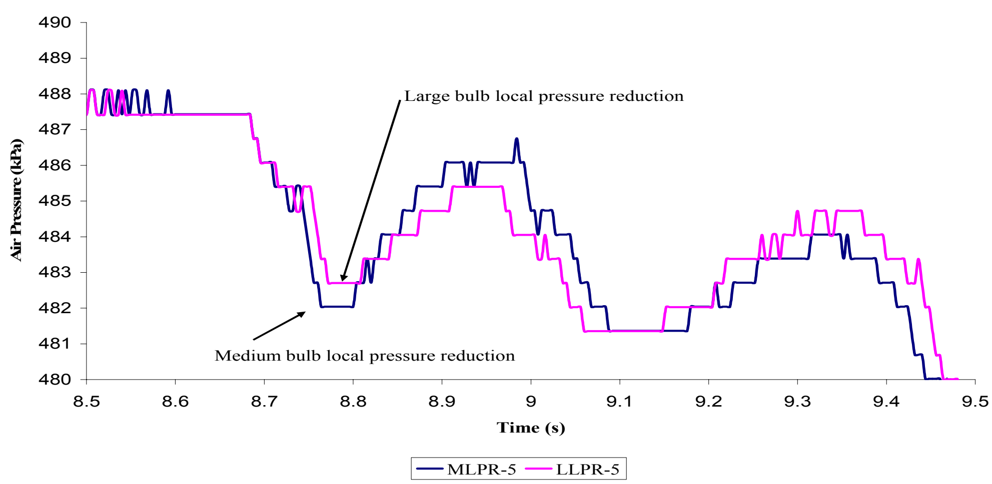


Figure 55: Gulps produced by a medium bulb and large bulb.

The results shown in Figure 56 are from transducers at the bulb and positioned after the control valve. The test shown in Figure 56 included successive brake applications without the brakes being released between the applications. These results show the bulb being filled at the first brake application and then leaking down before the next application.

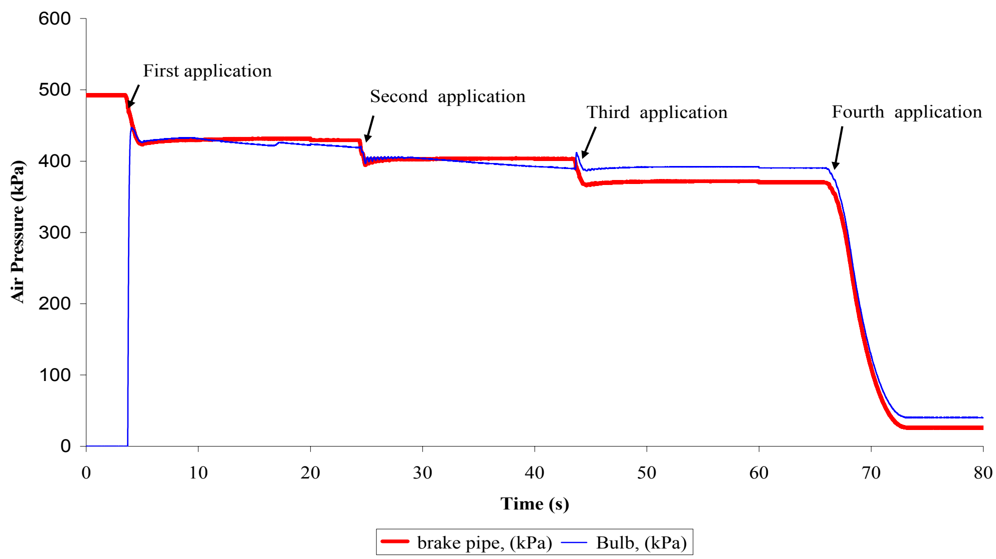


Figure 56 : Bulb pressures after four brake applications.

4.3.5 Discussion

The pressure measurements were taken from the location at transducers P1 and P5 as shown in Figure 39 of each test. The results in Figure 54 show the comparison of 'local gulp' reached between the medium bulb and the large bulb. The results also show a slight gulp within the larger 'local gulp' pressure line as the brake pipe pressure above the diaphragm is dropping as the bulb is being filled. This slight gulp corresponds to the instant when the control valve disconnects the bulb to the brake cylinder at the end of the inshot function. The local pressure reduction seen in the brake pipe as shown in Figure 55 corresponds to the larger 'local gulp' seen at the control valve and shown in Figure 54.

This experiment has shown a difference of 0.6 kPa between the local pressure reductions in the brake pipe after a control valve operation. This difference is the same as the bit resolution of 0.67 kPa of the data so the significance of the difference is debatable. The times that each gulp stays at the lowest pressure are also the same at 0.032 seconds.

The gulp or local reductions of pressure in the brake pipe as seen by these tests are between 5.3 kPa and 4.7 kPa. These reductions taken alone are not enough to produce a trigger effect on the next control valve as the valve needs a differential between the Auxiliary reservoir pressure and the brake pipe pressure of at least 10 kPa to 12 kPa. There is seen from the tests conducted of the different bulb sizes that little difference in the local pressure reduction has been achieved using the larger bulb. The use of multiple control valves may increase this little difference in the local reduction of the brake pipe pressure. The results shown in Figure 50 showing increases in the gulp size can give the impression that with further lengths of pipe and more control valves, the sum of the gulp or local pressure reduction from each valve would finally reach a stage to be large enough to trigger a control valve.

An observation from the multiple application tests performed, (when closing of the brake pipe exhaust after the control valve had operated and then opening the exhaust again) was that there was no second gulp function in the brake pipe shown in Figure 56. The pressurised quick service volume will stay pressurised until an increase occurs in brake pipe pressure. The control valve then releases the pressurised volume to atmosphere this is when the brakes are being released as explained in 2.4.1. The results from the repeat application tests show the bulb leaking down in pressure, but to below the brake pipe pressure between applications and this shows a faulty control valve. The bulb should follow the brake pipe pressure. At the third application the bulb pressure raised as the pressure from the brake pipe was connected but stabilised at 390 kPa which was surprising. Again because of the faulty control valve the bulb pressure should have followed the brake pipe pressure down to 370 kPa. Although at this time the application showed a small gulp in the brake pipe of ~ 4 kPa as the bulb was filled. The brake pipe pressure was 13 kPa above the bulb pressure, at the start of the third brake pipe reduction and would explain the small gulp seen in the brake pipe.

4.4 The Pipe Bracket and Isolation Cock

4.4.1 Introduction

The purpose of the pipe bracket and isolating cock experiments was to compare the size of the gulp in the brake pipe under two conditions, which are (1) when a pipe bracket is used and (2) when a pipe bracket is not used. A second objective of these experiments was to compare the size of the gulp in the brake pipe when different sized passageways are used in the isolating cock.

The pipe bracket used in these experiments is mounted between the control valve and the multi-compartment reservoir. Mounted to the pipe bracket and connected by internal air passages is the isolation cock. Air from the brake pipe via the branch pipe is routed with passageways through the pipe bracket into the isolation cock and then back into the pipe bracket where it is connected to the control valve. The air also flows through a filter attached to the isolation cock.

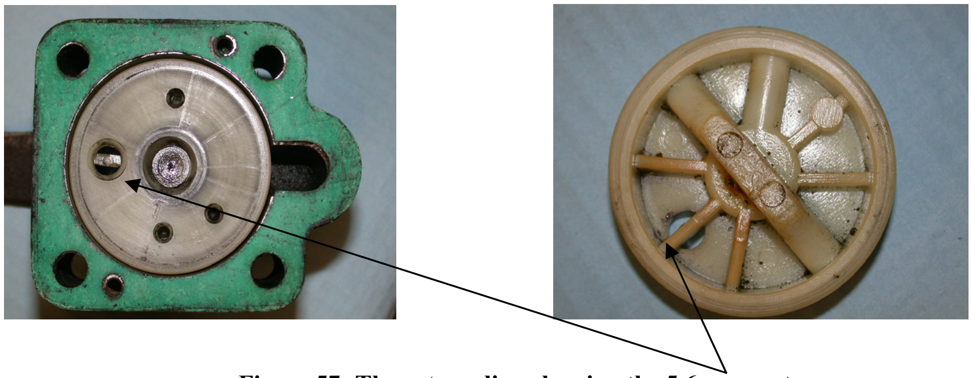


Figure 57: The rotary disc, showing the 5.6mm port.

These experiments are to investigate and compare the size of the local reduction pressure wave created in the brake pipe from the action of the bulb when using different sizes of the port in the rotary disc of the isolation cock. The comparison between using a pipe bracket and having the branch pipe connected directly to the control valve is assessed separately. The experiments in investigating the actions of the local pressure reduction in a brake pipe from (1) a pipe bracket with an isolation cock and (2) a control valve without a pipe bracket were completed as two separate sets of experiments.

4.4.2 Equipment

In the first series of experiments, the 2nd control valve within the brake pipe rack utilises a pipe bracket used in conjunction with a combined reservoir. The isolation cock that is connected to the pipe bracket is tested with two different port sizes. The pipe bracket was examined and found to have approximately 132 mm of passageways of which range in diameters from 11 mm to 8.7 mm. The brake pipe air was connected to the control valve via the pipe bracket. The combined isolation cock and filter utilises a rotary disc as a valve and has a filter of the paper element type of 25 micron attached. The passageways in the isolation cock have a combined length of approximately 150 mm and diameters range from 11 mm to 9.5 mm. The standard diameter of the port in the rotary disc is 7.65 mm with a bridge of 2 mm wide across it and therefore has an effective diameter of 5.6 mm and an area of 2.463×10^{-5} m^2.

Tests are also completed with a modified rotary disc. The size of the port in the modified rotary disc was 9.5mm with an area of 7.088×10^{-5} m^2 with the bridge removed. The 9.5 mm diameter matched the size of the immediate passageways either side of the rotary disc. The two diameters of the branch pipe that were used were 20 mm and 12 mm and were both of the same length. The inlet diameter of the control valve will not allow a larger diameter than 20 mm. For the second series of experiments a branch pipe of 20 mm and 800 mm in length is connected to a pipe bracket and a branch pipe with the same internal diameter and length is connected to a control valve which does not utilise a pipe bracket. The diameter of the exhaust choke used in these experiments was of 1.5 mm.

4.4.3 Method

Each test was conducted with the air in the brake pipe being compressed to 500 kPa and allowed to stabilise. Air was then exhausted through an exhaust choke.

4.4.4 Results

The results of the local reduction in brake pipe pressure created by different branch pipe diameters and rotary port diameters are presented from four experiments of test numbers 700, 705, 711 and 716. The branch pipe diameters and port sizes are shown in Table 10. The response of brake pipe pressure from transducer P6, which is located after the branch tee for the branch pipe connection to the pipe bracket, is shown in Figure 58.

Table 10: Branch sizes and port sizes.

Test No.	Branch pipe length (mm)	Branch pipe I.D. (mm)	Rotary disc port size (mm)
700	800	20	7.65
705	800	20	9.5
711	800	12	9.5
716	800	12	7.65

Table 11 is produced from the data in Figure 58 the transducer P6 is at the locations in Figure 39

Table 11: Pressures and times at transducer P6.

	P6 (kPa)	P6 (kPa)	P6 (kPa)	P6 (kPa)	
Time (s)	#700	#705	#711	#716	Comments
0.072	487.5	487.5	487.5	487.5	Pressures aligned with time
0.112	485.5	485.5	485.4	486.1	Small pipe shows less drop
0.152	483.4	484.1	483.4	484.1	Bottom of trough of 716
0.180	483.4	483.4	484.7	484.7	716 rising
.0.220	482.8	483.4	484.1	485.4	700 at bottom of trough
0.292	482.8	483.4	484.1	486.8	700 starting to rise
0.368	484.7	484.7	484.7	486.8	All at top of local gulp

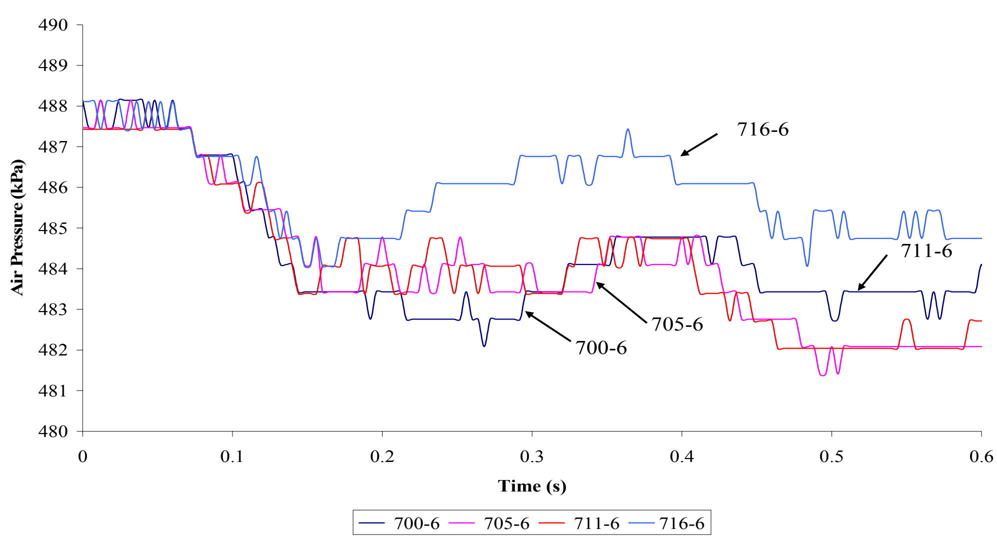


Figure 58: Gulps with different branch pipes and port diameters.

The results of the local reduction in the brake pipe pressure from a control valve connected to a pipe bracket and of a control valve without a pipe bracket are shown in Figure 59. The branch pipe for all four tests were of 20 mm I.D. and with a length of 800 mm, and are as shown in Table 12

Table 12: Test numbers with various pipe brackets.

Test No.	Transducer	Branch pipe length (mm)	Branch pipe I.D. (mm)	Comments
91	P3	800	20	No pipe bracket
91	P7	800	20	No pipe bracket
72	P2	800	20	Pipe bracket
72	P6	800	20	Pipe bracket

The results shown in Figure 59 and Table 13 are at the transducer locations as shown in Figure 39.

Table 13: Pressures from control valves with and without a pipe bracket.

	P3 (kPa)	P7 (kPa)	P2 (kPa)	P6 (kPa)	
Time (s)	# 91	# 91	# 72	# 72	Comments
8.732	486.1	486.8	486.1	486.8	Pressures aligned with time
8.788	484.8	486.8	484.8	486.8	Drop in pressure at above diaphragm
8.848	473.3	485.5	461.2	484.1	P7-91 shows corresponding 'small gulp'
8.88	448.4	483.4	420.8	482.1	P6-72 at lowest pressure
8.89	447.1	482.1	412.1	482.1	P7-91 still falling
8.988	459.2	480.1	433.6	482.7	P7-91 at lowest pressure P6-72 rising

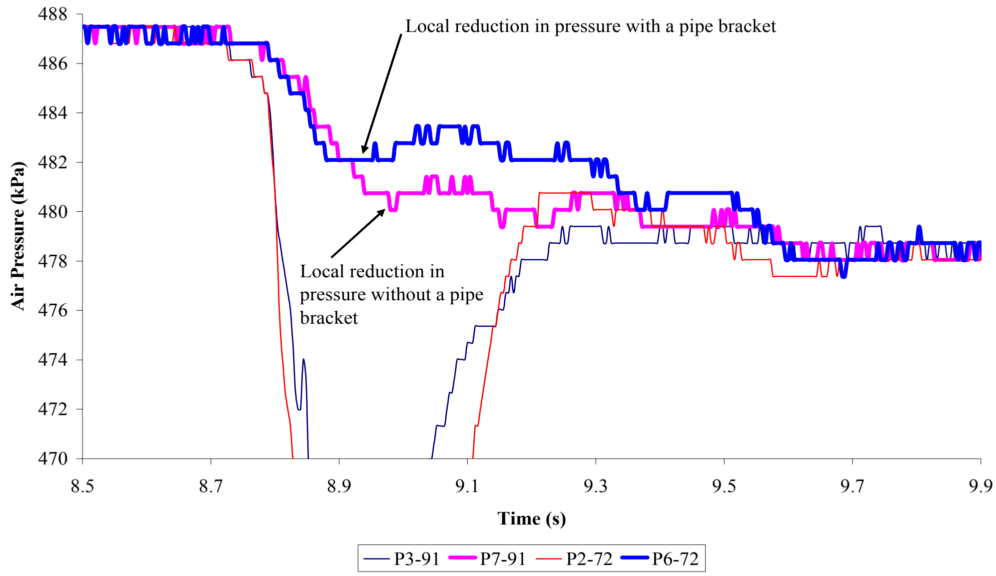


Figure 59: Gulps in the brake pipe with and without a pipe bracket

4.4.5 Discussion

The first series of experiments the brake pipe pressures were taken only from the transducer P6 which is positioned after the 2nd control valve as shown in Figure 39. The results from Figure 58 show a difference of the lowest local pressure reduction reached between test #716 and test #700 of 1.6 kPa with #700 being the lower at 482.8 kPa. The results show that the time that each local pressure reduction stays at the lowest pressure is different for each rotary port and branch pipe diameter. The time the pressure of #700 stays at its lowest local pressure reduction of 482.8 kPa is 0.076 seconds and could best be described as a ‘hold time’. The ‘hold time’ on test #705 is 0.172 seconds. Test #711 shows a ‘hold time’ at its lowest point is 0.172 seconds. The time period that the pressure of #716 stays at its lowest local pressure reduction of 484.3 kPa is 0.02 seconds.

The results show that for test #700 and test #711 it takes 0.072 seconds to reach the lowest local pressure reduction of 4 kPa and this could be called a ‘drop time’. The 'drop time' for test #705 to reach its lowest point is 0.092 seconds with a local pressure reduction of 4 kPa. In test #716 the 'drop time' taken to reach the lowest point is 0.08 seconds with a local pressure reduction of 3.3 kPa.

These results show that when a branch pipe is a smaller diameter, the enlarged port in the rotary valve has little effect on the local pressure reduction in the brake pipe.

These observations are of the pressure response of tests #705 and test #711 which show a similar response to each other.

The results from test #716 show that with a small diameter branch pipe and a standard port size in the rotary disc, the size of the local pressure reduction is the least in both pressure reduction and the time period which the lowest pressure is maintained. The results of test # 700 can be compared to a previous experiment seen in section 4.2 and Figure 53 of the local pressure reduction using different branch pipe sizes. In that section, test #608 was setup with a branch pipe of 20 mm I.D. and 150 mm long. This test # 608 showed a reduction of 4 kPa as seen in Table 8 and the branch pipe was connected via the pipe bracket with a standard rotary port. This compares well with the 'hold times' and 'drop times' with test #700 of which is using the same size diameter branch pipe but 800 mm long. The results from the tests completed in this series indicate that the smaller internal size of a branch pipe as shown with test #716 will give the smallest local pressure reduction, occurring in the least amount of time after a control valve operation. The enlarging of the rotary port size as shown with test #711 can increase the time that the local reduction pressure drop will stay at its lower pressure when the branch pipe internal diameter is smaller.

The second series of experiments give the result that a pipe bracket can have an influence on the size of the local pressure reduction drop in the brake pipe. The results from Figure 59 and shown in Table 13 show a difference of 2 kPa between the two lowest pressures obtained in these tests of the local reduction in brake pipe pressure or gulp. The local pressure drop in the brake pipe that has been shown is 6.7 kPa for a control valve without a pipe bracket. When a pipe bracket is used the local pressure drop is shown as 4.7 kPa. These pressures were obtained using a single control valve in separate tests.

4.5 Bulb filling times

4.5.1 Introduction

An evaluation of the effect of the restrictive passageways of the pipe bracket and control valve is the first objective of this group of experiments. The purpose of the second part of these experiments is to measure bulb filling times and to compare the time that was taken to fill two different sized bulbs with and without the restrictive passageways of the pipe bracket.

The quick service volume or bulb of a control valve is filled from the compressed air in the brake pipe at the beginning of a brake application. Between the bulb and the branch pipe are passageways of approximately 730 mm in length. The diameters of these passageways vary between 5.6 mm and 11.5 mm. The first of these tests is to compare the difference in brake pipe pressure response when a gulp is provided with and without these restrictive passageways.

A ‘container’ or ‘experimental dummy bulb’ as shown in Figure 61 with the same volume as a large bulb (59×10^{-5} m^3) was connected directly to the brake pipe to give a comparison of the bulb volume operation without the restrictive passageways. The second experiment a tube with a small diameter is connected between the 'container' and brake pipe. The length in the pipe is adjusted to give the same response in the brake pipe as would occur when a standard large bulb is connected via the pipe bracket and control valve.

The two different sizes of bulbs that are used in most present day Australian railway industry would be assumed to have different filling times and the passageways within the pipe bracket as seen from previous testing of the Quick service volume may have an influence on these times. These tests are to establish the time it takes to fill a quick service volume or bulb to its maximum pressure with two different configurations of branch pipe connections, where one connection of the branch pipe eliminates the pipe bracket.

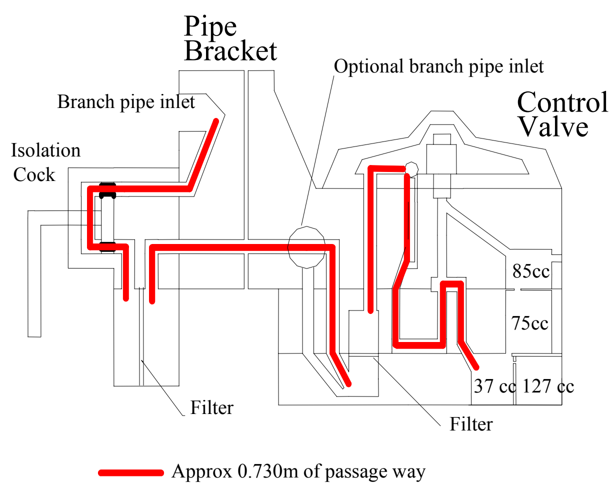


Figure 60: Passageways in the pipe bracket and control valve of 730 mm.

4.5.2 Equipment

The equipment used in the first of the tests was the brake pipe with transducers at the locations as shown in Figure 39. The 'experimental dummy bulb' or 'container' used was made to the size of a large bulb of a control valve and had a volume of 59 x 10^{-5} m^3. The length and diameter of the long length tube had been previously determined through experimental results comparing the 'local gulp' from valves with and without the pipe bracket and is of 2.1 m in length and 6 mm in diameter.

Figure 61: The 'container' connected to the brake pipe.

The second of these tests used two different control valves one having a medium bulb of 33 x 10^{-5} m^3, the other having a large bulb of 59 x 10^{-5} m^3. The branch pipe used was 20 mm I.D. and 800 mm in length.

4.5.3 Method

These tests are broken into two parts. For the first part of the tests, an 'experimental dummy bulb' or 'container' was connected to the brake pipe with a ball valve as shown in Figure 61. The brake pipe pressure, when at its stabilised pressure of 500 kPa was opened to atmosphere by a valve through an exhaust choke, as in a brake pipe application at a rate of ~ 270 kPa/min. Then after approximately 7 seconds the ball valve that connected the 'container' to the brake pipe was opened, the 'experimental dummy bulb' would then be pressurised and the brake pipe pressure would drop accordingly. This type of operation produces a gulp or local reduction of pressure in the brake pipe as would a control valve bulb operation.

The second part of these tests used the 2nd control valve in the brake pipe setup as shown in Figure 39. The branch pipe used was either connected to the pipe bracket or to the optional port on the control valve for comparison purposes of the fill times.

4.5.4 Results

For the first experiment, the 'container' was connected directly to the brake pipe and the results show a gulp or local reduction in the brake pipe of 11.5 kPa drop in pressure. The second part of the tests, the 'container' was then connected to the brake pipe with a long length tube. The results show a reduction or gulp in the brake pipe when the 'container' was filled was at a 5.4 kPa drop, both of these local pressure reductions or gulps are shown in Figure 62.

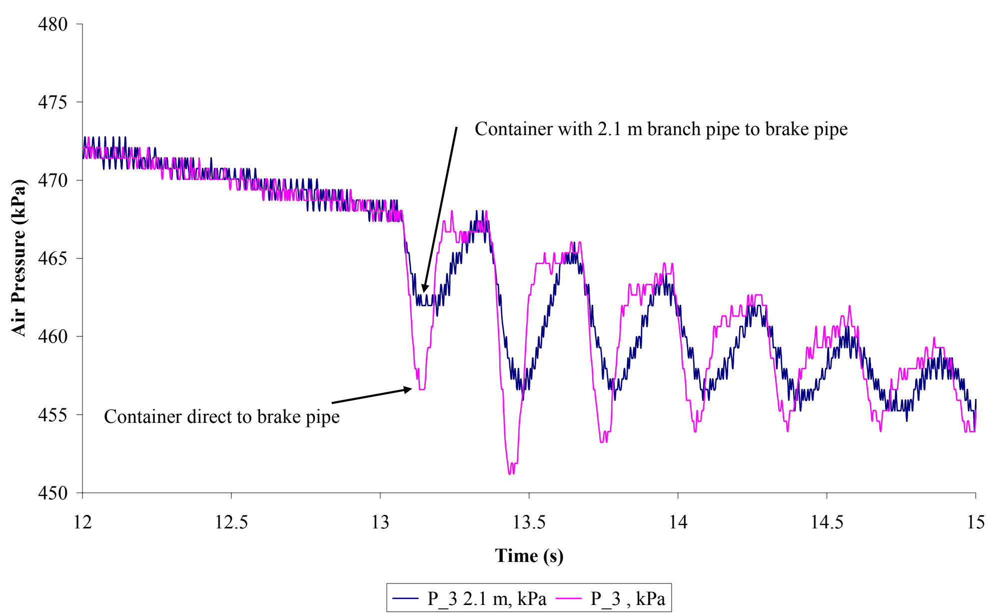


Figure 62: Comparison of bulb having long and short branch pipes.

The results from the 2nd part of these tests are shown in Figure 63, and show a decrease in the filling times of a medium and large bulb. The decrease shown is between 24.5% and 53% respectively. The decrease was achieved by changing the branch pipe connection from the inlet port on the pipe bracket to the alternative inlet port on the control valve. The filling times of the bulbs are shown in Table 14 and made from Figure 63.

Table 14:Times when medium and large bulbs have filled.

Test No.	Time when bulb has filled to maximum pressure (s)	Comments
620	0.208	Medium bulb, branch pipe direct to control valve
618	0.276	Medium bulb, branch pipe to pipe bracket
605	0.280	Large bulb, branch pipe direct to control valve
601	0.596	Large bulb, branch pipe to pipe bracket

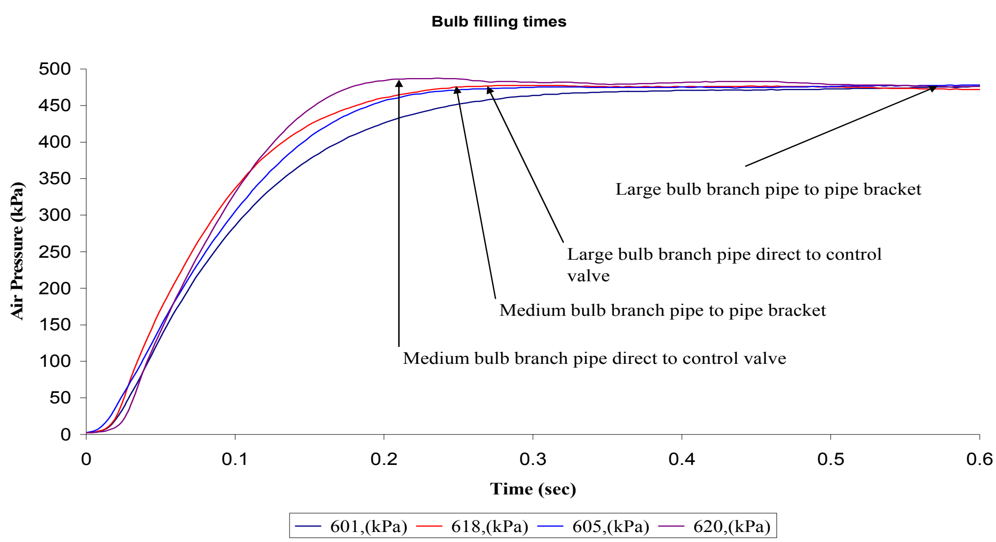


Figure 63: Bulb filling times for medium and large bulbs.

4.5.5 Discussion

The tests using an 'experimental dummy bulb' showed that the pipe bracket and control valve passageways have an effect on the local reduction pressure in the brake pipe. The crude, but directly connected volume as seen in Figure 61 gave an increase in pressure reduction or gulp of 6.1 kPa. The local pressure drop seen in the brake pipe when the container is connected to the brake pipe was 11.5 kPa. When the 'container' is used with a long length tube the local pressure drop seen in the brake pipe is reduced to 5.4 kPa. This pressure drop of 5.4 kPa is similar to the pressure drop seen in the previous tests in section 4.4 and Figure 59 when using a pipe bracket connected to a 20 mm diameter and 800 mm in length branch pipe using a large bulb taking into account the resolution of 0.67 kPa.

The second part of these tests has shown that the passageways in the pipe bracket have an effect in the filling of the bulb. The tests show that a large bulb with a pipe bracket takes the longest time to fill to its maximum pressure, as shown in Figure 63 and Table 14. The tests also show that when the pipe bracket is not used as a connection for the branch pipe, the time to fill a large bulb is decreased by 53%.

From the tests preformed, it can be seen that the pipe bracket and control valve that restrictions that can be grouped into individual classes. All the passageway restrictions in the pipe bracket could be classed overall as an orifice or a choke for that part of the control valve assembly. The restrictions of the passageways in the control valve that lead to the bulb could also be classed overall as an orifice or choke. For the purposes of discussions these 'orifice' or 'choke' will be called 'combined orifice'

By analysing the graphs in this section can give an indication of what variation on the size of the local pressure reduction seen in the brake pipe is made by each 'combined orifice'.

From the examples seen,

- When a control valve with a pipe bracket has shown a local reduction 4.7 kPa. It is noted that there are two 'combined orifice' between the bulb and the brake pipe.

- When the control valve is used without the pipe bracket the local reduction is at 6.7 kPa. Only one ‘combined orifice’ is between the bulb and the brake pipe.
- When the ‘experimental dummy bulb’ is connected to the brake pipe the local reduction in the brake pipe is seen as 11.5 kPa. There is no ‘combined orifice’ between the bulb and the brake pipe.(Although small restrictions will exist in pipe fittings)

Pressure drop variation due to different restrictions in the flow path:

- Pipe bracket and control valve……..4.7 kPa
- Control valve……………………….6.7 kPa
- ‘experimental dummy bulb’ ……….11.5 kPa

Against the 11.5 kPa value of the ‘experimental dummy bulb’ the above values indicate that the ‘combined orifice’ effect of the pipe bracket is of a value of 4.8 kPa. The value the ‘combined orifice’ effect of a pipe bracket and a control valve is at 6.8 kPa.

It must be noted that the start of local reduction in the brake pipe of the ‘experimental dummy bulb’ was when the brake pipe pressure was at a value of approximately 468 kPa. It would have been expected that if the brake pipe pressure was at the normally higher value of 10-12 kPa below the nominal 500 kPa when a control valve operates, then the local pressure reduction of the ‘experimental dummy bulb’ would be more than 11.5 kPa due to greater branch pipe flow.

If this could be the case, then the value of the ‘combined orifice’ effect of a pipe bracket and a control valve may be greater than 6.8 kPa.

5. General Discussion

The first of the tests using the four control valve test rig showed that the magnitude and duration of the pressure drop (in the brake pipe) associated with the gulp increases as more control valves in a brake line are triggered. The pressure gulps in the experimental rig propagated with a speed equal to sonic velocity. Note also that the experimental setup included one control valve with a pipe bracket, the other three valves were not fitted with pipe brackets. Testing showed that valves without pipe brackets induced larger gulps. The increase in the gulp is interesting, as the pipe lengths on the test rig were matched to the bulb volumes used with spacings of 37 m, 37 m and 43 m

Figure 38. The speed of propagation observed, over four control valves is faster (352 m/s) than was expected as the pipe lengths of the VSAL/S type wagons (39.06m) are above the upper extreme of the recommended operational range for a long bulb (23-37 m). Results from a QR train test, Figure 46, do not show this type of propagation speed up to wagon #52 (161 m/s). The possible reasons for this anomaly will be discussed along with other issues with pressure profiles in page 94.

The testing of medium (33 x10^{-5} m^3) and large (59 x10^{-5} m^3) bulbs using a pipe bracket for the branch pipe connection, did not show any measurable differences in the size of the gulp in the brake pipe on the experimental rig. It was also noted there has been no information from industry or valve manufactures in the literature review as to what size gulp would be obtained from bulbs to the equivalent pipe lengths they were assigned to. Manufactures only provide recommendations for pipe length for the two different bulb sizes (Westinghouse 1989).

Results from a ‘experimental dummy bulb’ of 59 x10^{-5} m^3 (manufactured from pipe fittings) that was directly connected to a brake pipe without the restrictive passageways of the pipe bracket and control valve gave large brake pipe gulps (~11.5 kPa). The only difference between this bulb and the standard large bulb is the restrictions provided by different size, lengths and sharp angles of the many small passageways within the pipe bracket and control valve.

The approximated restrictions were tested from comparing valve response by using the ‘experimental dummy bulb’ or ‘container’. A long tube from the ‘container’ to the brake pipe was used to represent the restrictions of a pipe bracket and control valve. The result of this comparison by using the long tube showed that the restrictions equated to a branch pipe of 2.1 m long and with a diameter of 6 mm. In a previous test it is noted that in Figure 59 section 4.4 in a comparison where a pipe bracket is used and also bypassed when using a large bulb, gave results of the gulp size at 4.7 kPa with a pipe bracket Figure 64 and 6.7 kPa without the bracket Figure 65, these results again showed the effects of restrictions of a pipe bracket. These effects of the restrictions of the pipe bracket are also seen in the bulb filling times of a large bulb when the branch pipe was used to bypass the pipe bracket as shown in the results from section 4.5 and Figure 63 when the times for filling a large bulb was at 0.280 seconds when bypassing the pipe bracket and 0.596 seconds when using a pipe bracket.

The possible benefits of the large bulb therefore appear to be lost due to the restrictions in the pipe bracket passageways. The possibility is thus raised that the resizing of bulbs to the large volume of brake pipe for VSAS/L wagon pairs was ineffective in increasing the gulp size. This argument is given more weight when it is realised that the large bulb development was only supported experimentally by the Westinghouse testing facility which was not equipped with pipe brackets with combined reservoir and relay systems. Overall results from all tests show that the pipe bracket or more exactly the passageways within the pipe bracket restrict the size and duration of the gulp in the brake pipe local pressure.

The pipe bracket restrictions Figure 64 also appear to nullify effects of different branch pipe lengths. It was noted that when the branch pipe has shorter or longer lengths in the range of 150 mm and 800 mm and having the same internal diameter there is no measurable difference seen in the size of the gulp when tested with a connection between the brake pipe and pipe bracket as in section 4.2, Figure 53. Conversely, gulp behaviour was observed to change when the branch pipe was connected to the control valve directly as in section 4.4, Figure 59.

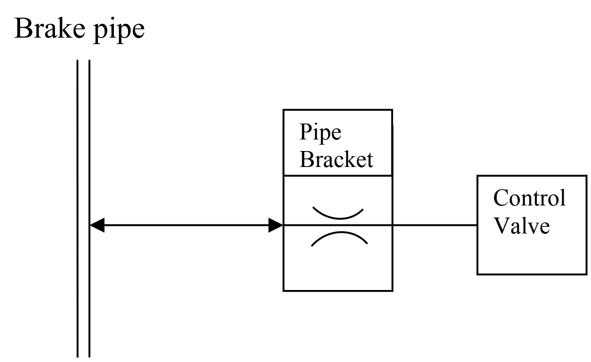


Figure 64: Branch to pipe bracket.

The tests of different branch pipe dimensions also showed that when the branch pipe was connected directly to the control valve, the bulb filling times were reduced as shown in Figure 63, section 4.5.

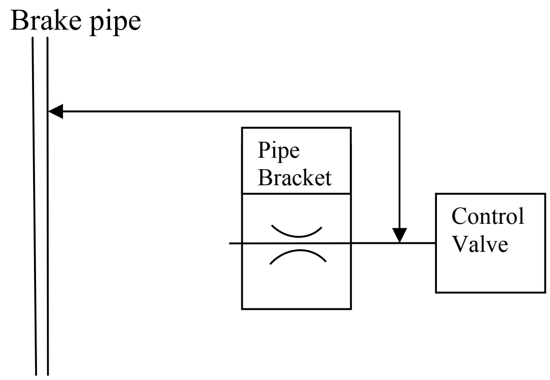


Figure 65: Branch pipe to control valve.

In the tests conducted two sizes of branch pipe used and were at 20 mm and 12 mm internal diameter. The branch pipe when connected to the pipe bracket has shown that the gulp was basically unmodified with different lengths of a branch pipe of 20 mm internal diameter, but when the branch pipe was changed to 12 mm internal diameter the gulp changed. At this point the restriction in the branch pipe became as significant to the gulp as the restrictions in the pipe bracket passageways.

Testing has confirmed that no benefit from the quick service volumes can be expected in the next successive brake applications. The design of the Australian brake system is such that the bulb pressure is held until the brakes are released (2.4.1).

A further issue to be considered is that the reduction ensuring valve only gives an initial charge of bulb air to the brake cylinder or dummy. It is to be noted that a change to the Australian control valve (WF5) now has the reduction ensuring valve being pilot operated by the bulb pressure (Westinghouse 2001). In this type of operation the brake pipe air above the diaphragm is connected to the dummy brake cylinder volume via the pilot operated reduction ensuring valve instead of using the bulb volume to a pressure of 70 kPa. The reduction ensuring valve of the WF5 valve, as with earlier valves when using relayed type equipment will connect to a fixed volume, which is the dummy brake cylinder and therefore adds a fixed volume to the bulb and therefore size of the gulp in the brake pipe by the bulb action. When the control valves are used with non-relayed equipment the reduction ensuring valve will be connected directly to the brake cylinders. It is noted that the volume of the cylinders can change to a larger volume than the fixed dummy cylinder volume, due to the number of cylinders and the stroke of the cylinders (if adjustment is not maintained). The gulp in the brake pipe will then change due to the larger volumes of the cylinders.

The interesting results from tests of quick service volume (see section 4.3.4), with a leaking bulb has demonstrated the possible benefits of venting bulbs. The effects of the leaking bulb has shown results that could be seen as similar to the American control valves in sections 2.3.2 and 2.3.3, where the Accelerated Application Valve (AAV) and the preliminary Quick Service Valve (Q.S.1) each vent a small part of the brake pipe air to atmosphere as a measure to increase the pressure reduction propagation rate. The AAV of the American system will vent a small portion of brake pipe air also on the next lower brake pipe reduction and therefore help to increase the pressure reduction propagation rate for the next brake application.

The construction of the bulb in the Australian valve as mentioned, if vented before a brake release is made, would cause the diaphragm to move to the release position and apply the brakes. It remains that the separate use of an Accelerated Application Valve or similar style valve may improve the lost propagation performance of successive applications on Australian systems.

The remaining discussion attempts to relate these experimental results to the brake pipe operation in long pipes. The exhaust of a brake pipe at a head end locomotive to give a reduction of pressure would make a pressure profile along the train as shown below in Figure 66. Superimposed on the profile will be the gulps as they are generated. Based in the experimental and industry data four different regions of operation can be suggested.

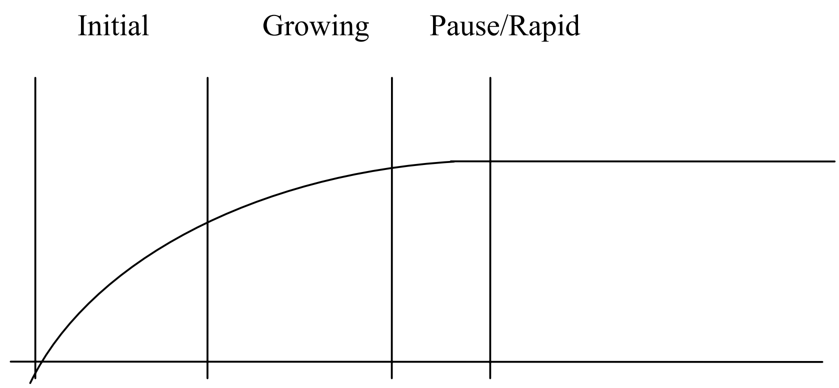


Figure 66: Brake pressure profile along the train.

1. In the initial section as the pressure is falling at the beginning of a long rake. The behaviour is dominated by the brake pipe exhaust. The first of the control valves are triggered by the high brake pipe pressure reduction rate and the high differential between the brake pipe and auxiliary reservoir pressures. The bulb action in this region is difficult to observe experimentally as air is exhausting rapidly via the exhaust in the locomotive. This was also demonstrated on the test rig with large exhaust chokes. With results from the QR graph (Figure 46) the propagation rate up to wagon #28 of a long rake where one valve operates over 39.06 meters of brake pipe has shown to be 123 m/s but begins to increase as the pressure profile tends to flatten, wagon # 52 shows a rate of 161 m/s.

2. A second issue is that a very long train stores a large volume of air in the pipe. With larger pipe volumes, bulbs will be less effective at dropping pressure. This effect is nullified at the front of the train via the rapid exhaust at the locomotive. So, as the pressure reductions due to pipe exhaust, at a given time can be smaller when located further from the locomotive exhaust, and the accumulated air volume in the rear of the pipe can still be large, bulb operation can become more marginal. Propagation rates of ~120 m/s are measured compared to full sonic in the test rig. So in mid-train pipe regions propagation slows. Modifications to either exhaust choke or bulb sizes made to suit the train length could be considered.
3. The pressure profile is usually fairly flat towards the rear of the brake pipe. When the brake pipe pressure in this region drops sufficiently to trigger the control valve, sonic propagation can occur as successive valves are triggered by bulb action alone. The results shown in Figure 46 from wagon #55 to wagon #92 (700+ m and within ~2 seconds) are indicative of sonic velocity. This is supported by the sonic velocities measured on the testing as shown in Figure 48, as the pressure drop rate used in the test rig approximated the rear of train behaviour.

6. Conclusions

Information arising from the literature review has shown a comparative difference in the Australian and American wagon brake systems in the way both assist the propagation of the pressure reduction rate. The American system has shown the development of the control valve evolved from using wagon lengths of 12 meters with 20 wagons to lengths of 9 to 27 meters and up to 150 wagons by the 1970's. Later control valve changes included the use of accelerated application valves and initial quick service valves both expelling portions of brake pipe air to the atmosphere.

The Australian wagon brake system by the 1970's went a different direction by holding a charge of brake pipe air in a quick service 'bulb' and then expelling this charge of air to the atmosphere when a brake release was made. The length of the Australian freight and mineral wagons has grown to similar lengths, with freight or mineral wagons having up to 19.5 meters of brake pipe. Developments in Australia have seen the number of wagons increase per train, and also the use of multi-coupled wagons with one control valve to operate brake pipe lengths of up to 39 meters.

The conclusion from the literature review is that the increase in the length between control valves and number of wagons in a rake adapted by Australian operators has not been fully supported with research and development of the control valve and/or other apparatus in the assistance of the propagation of the brake pipe reduction rate.

Results from the test rig show that when the brake pipe pressure drop rate is set close to the recommended minimum to operate a control valve, the size of the gulp progressively enlarges during propagation in a short pipe. Testing with more control valves is required to give an indication as to the size that gulps can progressively reach and when the air stored in a long pipe nullifies this effect. Further tests are also required using higher brake pipe pressure drop rates to fully investigate the mechanism of valve triggering from the local gulp verses the pressure drop from the brake pipe exhaust.

Branch pipes with larger internal diameters produce larger sized pressure reductions or gulps when connected to control valves via the pipe bracket. This conclusion is valid for comparing pipe diameters of 12 mm and 20 mm.
Branch pipes with different lengths but with the same internal diameter of 20 mm gave no measurable increase or decrease in the brake pipe local pressure reduction or gulp size when used with a pipe bracket connection. Branch pipes with shorter lengths but with the same internal diameter gave an increase in the brake pipe local pressure reduction or gulp size when used without a pipe bracket connection.

The quick service volume was shown to give a larger size gulp in the brake pipe when a control valve was connected without a pipe bracket. It is concluded that the restrictions to flow in the pipe bracket passageways reduce the size of the brake pipe gulp that can be achieved. This conclusion is further confirmed by the improvement gained by enlarging of the port in the rotary port plate in the isolation cock located in the pipe bracket.

Larger gulps were proved to be possible by testing an ‘experimental dummy bulb’ or ‘container’ as a bulb directly connected to the brake pipe. By removing the restrictive passageways caused by the pipe bracket and control valve, and connecting the ‘container’ direct to the brake pipe gave pressure gulps 113% larger than the 4.7 kPa reached when using the pipe bracket.

The investigations into the bulb fill times on both large and medium bulbs have shown that the passageways restrictions within the pipe bracket increase the time required to fill the bulb volume. In so doing these restrictions nullify any increase that could be obtained from large bulbs. The restrictions of the passageways in the pipe bracket can be therefore seen to be the major contribution to the gulp size. Improvements to propagation rate cannot be achieved with large bulbs combined with pipe brackets unless the pipe bracket passageways are modified.

From the above conclusions, the aims of this project has been met with conclusion of the literature review that a review and further research into the assistance of the propagation of the brake pipe reduction rate on Australian brake systems is needed.
The experimental work of this project has shown that with modifications to the brake system, improvements in the performance of the application delays can be achievable. The very simple modification of relocation the branch pipe connection from the pipe bracket to the control valve port will improve brake pipe propagation speeds.

7. Recommendations

Throughout the testing on the rig it has been shown that the restricted passageways in the pipe bracket has been responsible for a reduced gulp in the brake pipe. It has also been shown that larger branch pipe diameters can give a larger gulp in the brake pipe when connected to the pipe bracket. From the conclusions of the experiments performed on the test rig the following recommendations of focusing are suggested for further work.

1. Further investigations of branch pipes including:

    - The examination of results of gulp size in the brake pipe by having branch pipes connected directly to the control valve that eliminate or by pass the pipe bracket passageways.
    - Examine results of the gulp size and propagation rates from the effect of having a branch pipe larger than 20 mm internal diameter.

2. Further investigations of the pipe bracket including:

    - Improve the restriction of flow through the pipe bracket by enlarging the passageways.

3. Further investigations of the bulb including:

    - Investigate the possibility of using a choked exhaust of the bulb

- Investigate sizes of the bulb for bar coupled wagons controlled by one valve and multi-car packs with reduced numbers of control valves.
- Investigate the use of the Australian style Accelerated Application Valves (Westinghouse 1989) combined with the Australian style control valves.

8. Reference List

Abdol-Hamid (1986). "Simulation of a Freight Train Air Brake System." ASME Paper No. 86-WA/RT-15 (Anaheim, California U.S.A).

Ateliers Des Charmilles, S. A. (1954). Improvements in or relating to control devices for compressed air brakes. AU Pat. No. 155,182 Commonwealth of Australia.

Australia,. (2002). Code of Practice for the Defined Interstate Rail Network.

Blaine, D. G. (1979). Freight car air brakes, Simmons-Boardman Books Inc.

Bureau, R. E. (2002). Guide to Freight Car Brakes, Simmons- Boardman Books Inc.

Eder, H. (1984). Service Brake Accelerator for Rail Vehicle Air Brake Systems. US Pat. No. 4,536,040 . Fed. Rep. of Germany, Knorr-Bremse GmbH, Munich.

Farmer, C. C. (1936). Fluid Pressure Brake. US Pat. No. 2,031,213. America, Westinghouse Air Brake Company.

Gerhart, P. M., Gross,R.J. (1985). Fundamentals of Fluid Mechanics, Addison-Wesley Publishing Co.

Hart, J. E. (1987). Freight Brake Control Valve having an Emergency Piston Slide valve Arranged to Provide an Accelerated Brake Application Function. US Pat. No. 4,690,463, American Standard Inc.

Hintner, J. (1978). Brake Accelerator for Air Brake system of a railway vehicle .US Pat. No.4,108,501, Knorr-Bremse GmbH.

Johnson, M. R., Booth, G. F. and Mattoon, D., W. (1986). "Development of practical Techniques for the simulation of train brake operation." ASME Paper No. 86-WA/RT-4.

Jones, H. H. (1979). "Railway Freight Braking, North American Style." Int. J. Mech. Sci. C165/79: pp. 283-297.

Kirk, W. B. (1965). Fluid Pressure Brake Control Apparatus with Accelerated Release after Service Application. US Pat. No. . U.S.A., Westinghouse Air Brake Company.

Leigh, M. J. (1990). "Development and Implementation of a modular Locomotive brake system." Journal of Rail and Rapid Transit Part F (204): pp. 113-121.

Leigh, M. J. (1992). "Brakes along the line." Part F: Journal of Rail and Rapid Transit 206: pp. 79 - 91.

Macfarlane, I. (2000). Railway Safety, Brakes. Crows Nest N.S.W. Australia, Engineers Australia.

Murtaza, M. A. (1993). "Railway Air Brake Simulation, an Empirical Approach." Proc Inst. Mech Engrs. 207: pp. 51-56.

Murtaza, M. A. a. G., S. B. L. (1989). "Brake modelling in train simulation studies." Journal of Rail and Rapid Transit Part F(203): pp. 88-95.

Murtaza, M. A. a. G., S. B. L. (1990). "Transients during a railway air brake release demand." Journal of Rail and Rapid Transit Part F(204): pp. 31-38.

Murtaza, M. A. a. G., S.B.L. (1989). "Brake modelling in train simulations studies." Journal of Rail and Rapid Transit Part F 203: pp. 88-95.

Murtaza, M. A. a. G., S.B.L. (1990). "Transients during a railway air brake release demand." Journal of Rail and Rapid Transit Part F 206: pp. 21-36.

Murtaza, M. A. a. G., S.B.L. (1993). "Railway Air brake Simulation." Proc Instn Mech Engrs 207: pp. 51-56.

QR_Manual (1995). Wagon & Carriage Brake Manual (unpublished).

Roberts, H. P. (1979). "Railway Freight Braking on the Continent.(UIC)." Int. J. Mech. Sci. C159/79: pp. 299-308.

Simmons, A. W., and Washbourn. (1964). Improvements relating to Triple Valve Devices. Commonwealth of Australia Au Pat. No. 253,540, Westinghouse Brakes (Australasia) Pty. Ltd.

Simmons, A. W., Washbourn, J. (1960). Improvements Relating to Compressed Fluid Braking Apparatus. Commonwealth of Australia Au Pat. No. 229,967, Westinghouse Brakes (Australasia) Pty. Ltd.

Sismey, B. a. S., W. (2000). Keeping the lights on in South Australia. Conference on Railway Engineering, Adelaide, South Australia.

Tomlinson, A. V. (1931). Improvements relating to Fluid Pressure Braking Apparatus. AU Pat. No. 356,725. United Kingdom, Westinghouse Air Brake Company.

WABCO (1996). Advanced Train Braking Technology from Wabco. International Heavy Haul Conference, Montreal Canada. June 1996

Westinghouse (1973). "Service Sheet."

Westinghouse (1989). "C187 Accelerated Application Valve." Product Information. Westinghouse Brakes Australia

Westinghouse (1989). "Service bulletin #69." Westinghouse Brakes Australia

Westinghouse (2001). "Service bulletin # 86." Westinghouse Brakes Australia

Westinghouse Air Brake Company (1945). The 'AB' Freight Brake Equipment.

Westinghouse Brakes Australia Air Brake Training.

Westinghouse, G. (1869). Improvement in Steam-Power-brake Devices .US Pat. No. 88,929. U.S.A. George Westinghouse

Westinghouse, G. (1873). Steam and Air-Brakes .US Pat. No. 144,006. U.S.A, George Westinghouse.

Westinghouse, G. (1875). Air Valve for Power Brakes. US Pat. No.168,359 U.S.A., George Westinghouse.

Westinghouse, H. H. (1902). Fluid Pressure Brake. US Pat. No. 707,473. U.S.A., George Westinghouse.

White, J. (1935). Improvements in and Relating to Fluid Pressure Braking Apparatus.. AU Pat. No. 428,760. United Kingdom, Westinghouse Air Brake Company of Australia.

Wilson, R. L. (1965). Continual Quick Service Valve Device .US Pat. No.3,175,871 . U.S.A., Westinghouse Air Brake Company.

Wilson, R. L. (1973). Quick Service Valve Device. US Pat.No.3,716,276. U.S.A., Westinghouse Air Brake Company:.

9. Bibliography

Anscombe, R. L. (1979) *Int. J. Mech. Sci,* Specification and Design of Braking Systems C170/79, pp. 263-276.

Bharath, S. (1989) *Journal of the Institution of Engineers (India) Part ME,* modeling and analysis of railway brake systems.

Bharath, S., Nakra.C and Garg, S. B. L. (1990) *Int. J. Mech. Sci,* A Distributed Mathematical Model for Pressure Transient Analysis in Railway Pneumatic Brake System, pp. 133 - 145.

Bharath, S., Nakra.C and Gupta, K. N. (1990) *App. Math. Modeling,* Modeling and analysis of pneumatic railway brake system, pp. 58 - 66.

Crane Co. (1995) *Flow of fluids through valves, fittings and pipe,* Crane Australia Pty Ltd, St Mary's, NSW.

Dalton, J., Eichinge, M. and Nemtas, M. (1998) Conference on Railway Engineering (Adelaide) performance Prediction and Operational Investigations of WF4 Triple Valves Using High Fidelity Simulations

Duncan, W., Nemtsas, M., Eichinger, M. and Dalton, J. (1998) Conference on Railway Engineering (Adelaide) A Cost Effective Method Of Brake Testing On Long Trains

Funk, J. E. and Robe, T. E. (1970) *Int. J. Mech. Sci.,* Transients in pneumatic lines subjected to large pressure changes, pp. 245 - 257.

Hooper, W. B. (1981) *Chemical Engineering,* The two-K method predicts head losses in pipe fittings, pp. 96-100.

Leigh, M. J. (1994) *Part F 208 : Journal of Rail and Rapid Transit,* Brake Blending, pp. 41 - 49.

Makinen, J., and Piche, R, and Ellman, A. (2000) *Dynamic Systems, Measurements and Control,* Fluid Transmission line Modeling Using a Variation Method**,** pp. 153-162.

Murtaza, M. A. and Garg, S. B. L. (1989) *Int. J. of Vehicle Design,* Dynamic response of a railway vehicle air brake system**,** pp. 481 -496.