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.

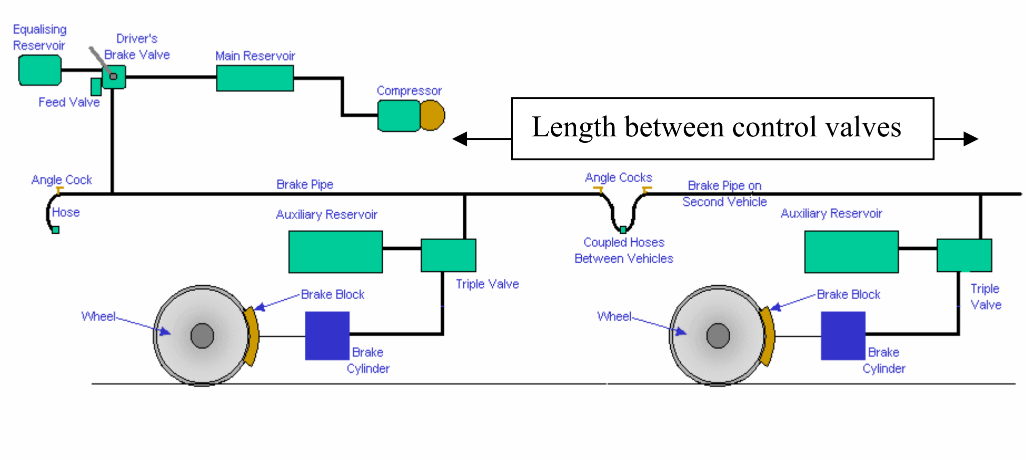


Figure 1: Schematic of a brake system.

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

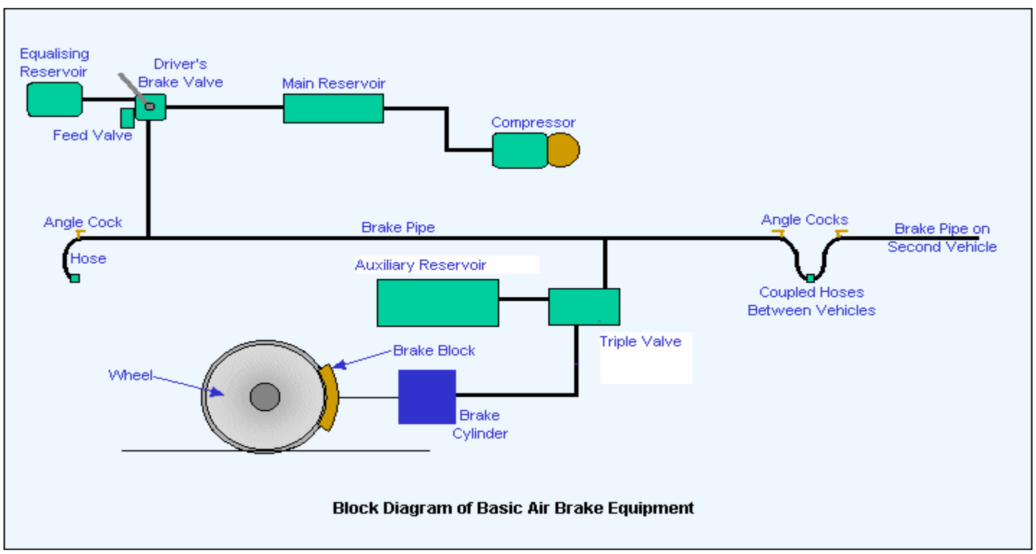


Figure 2: Automatic air brake direct release system.

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

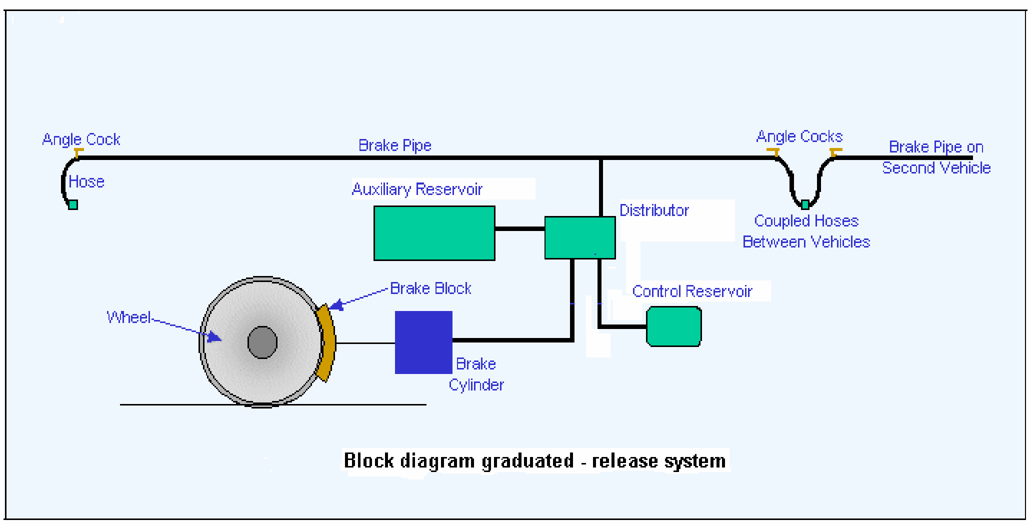


Figure 3: Automatic air brake graduated - release system.

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.

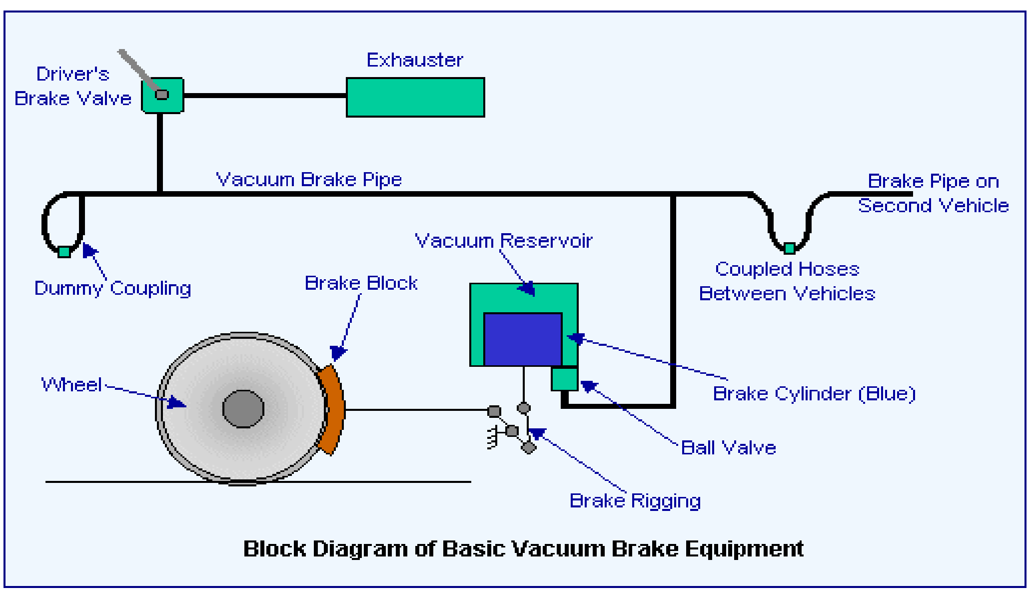


Figure 4: Vacuum automatic air brake system.

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).

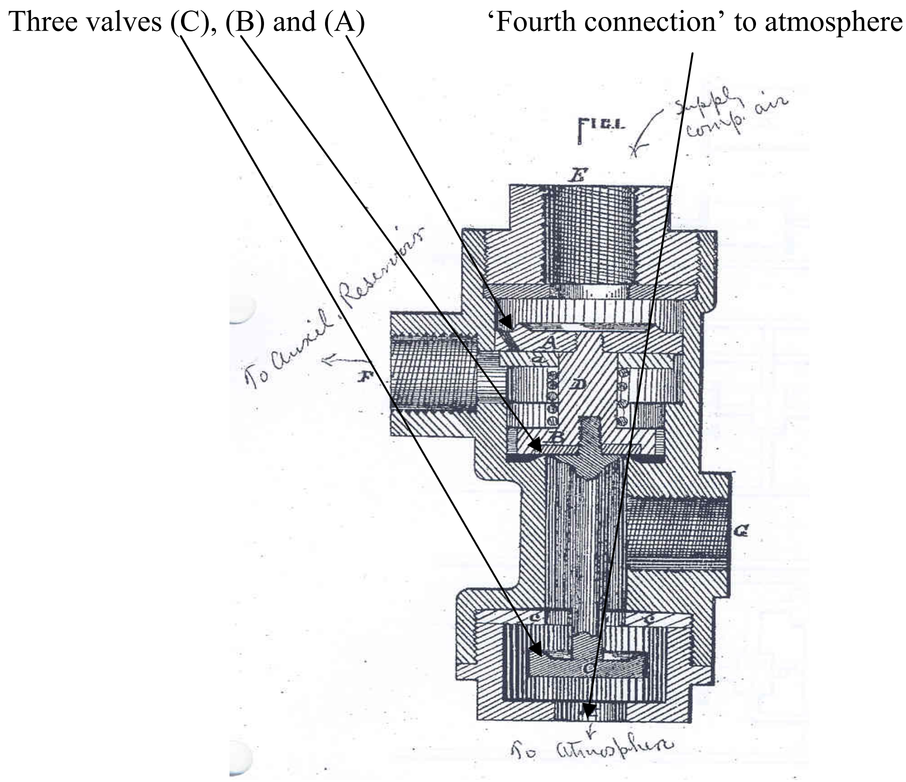


Figure 5: First triple-valve valve of Westinghouse. (Reproduced in whole from patent No. 144006 Westinghouse 1873)

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.

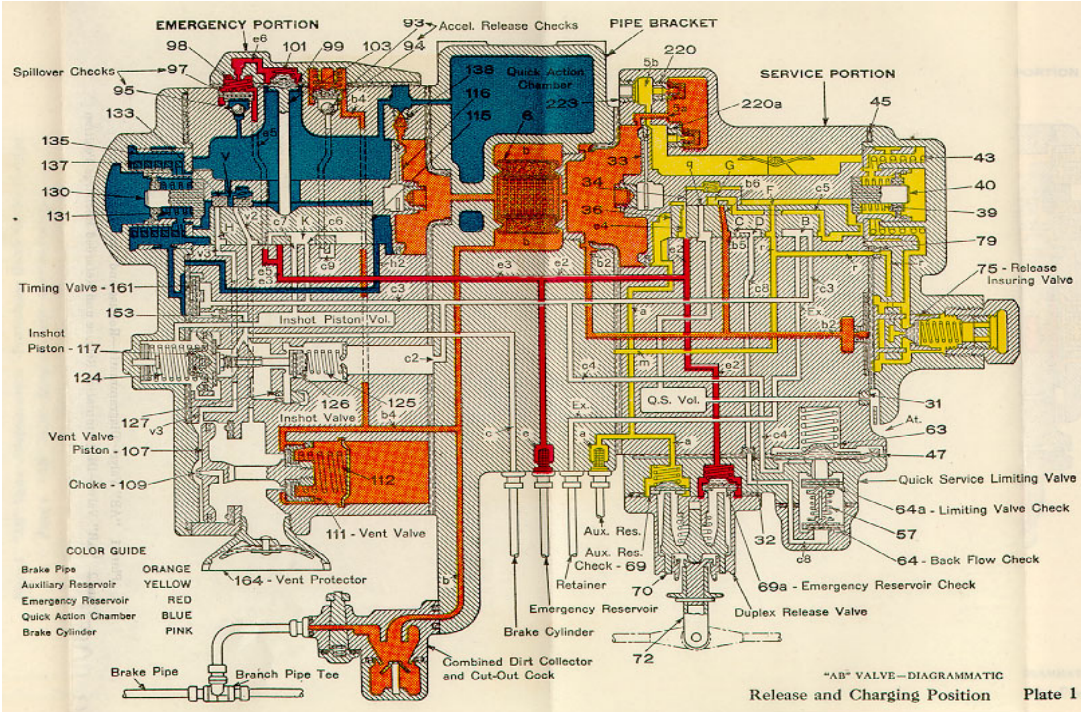


Figure 6: "AB" triple valve. (Reproduced in whole from Westinghouse Air Brake Company 1945)

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.

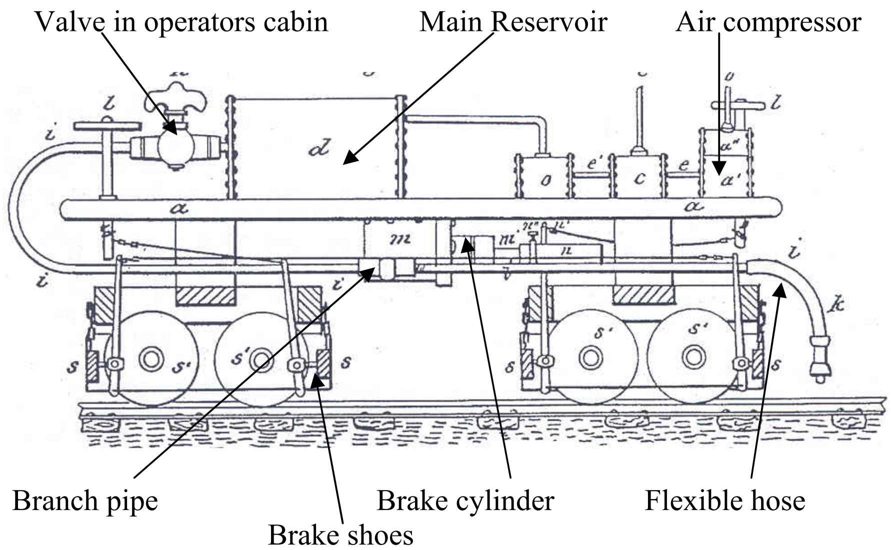


Figure 7: The Westinghouse straight air brake. (Westinghouse 1869)

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.

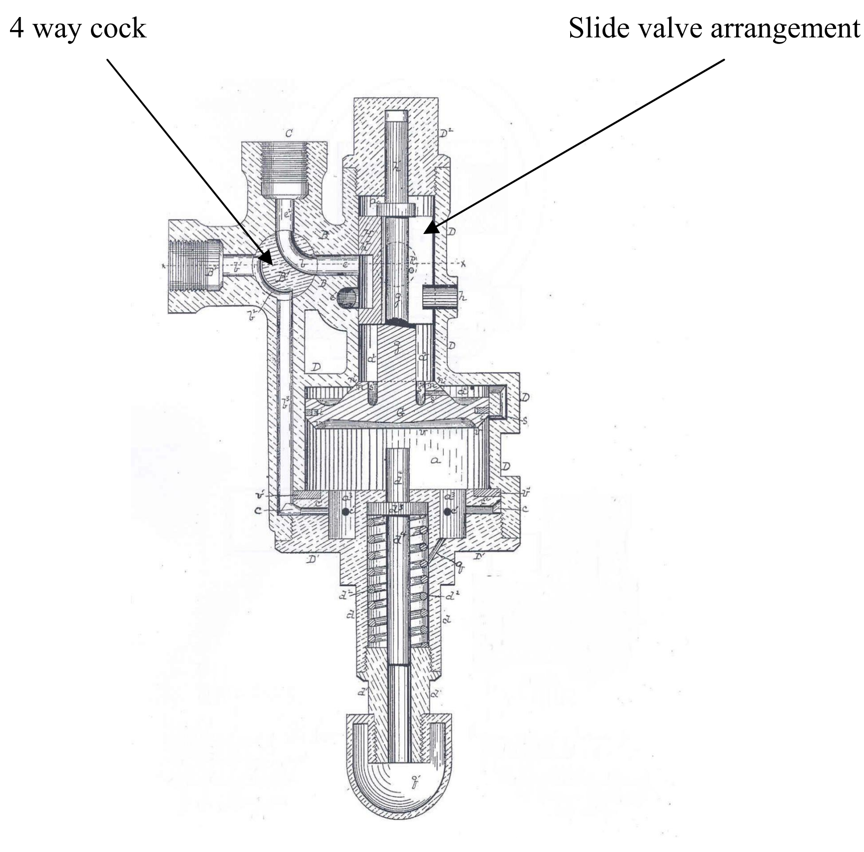


Figure 8: First triple valve with a slide valve. (Westinghouse 1875)

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.

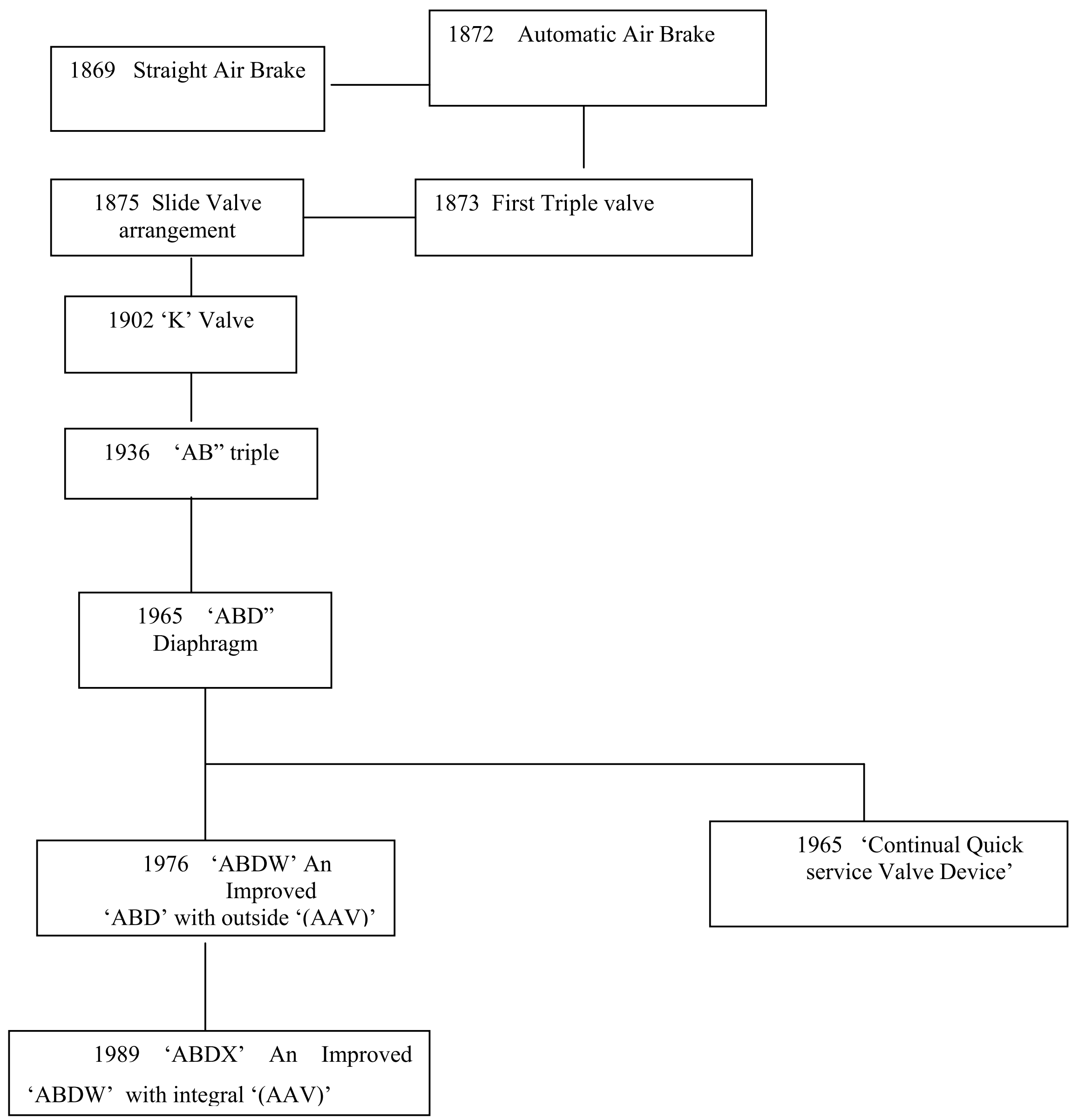


Figure 9: The Westinghouse American development tree.

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.

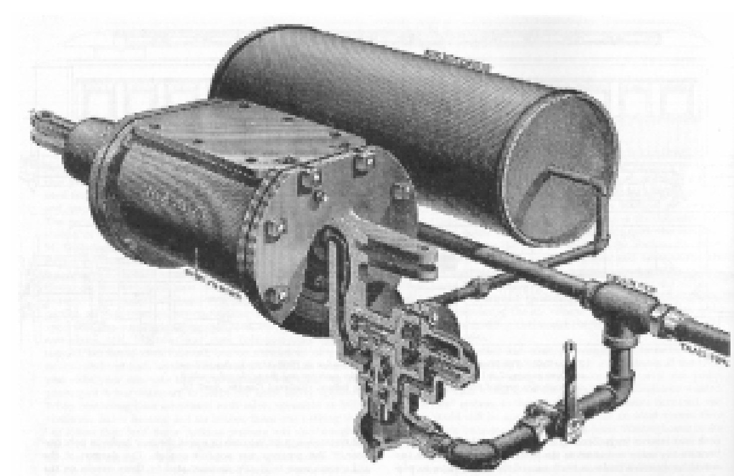


Figure 10: The ‘K’ triple valve. (Westinghouse 1902)

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:

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

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.

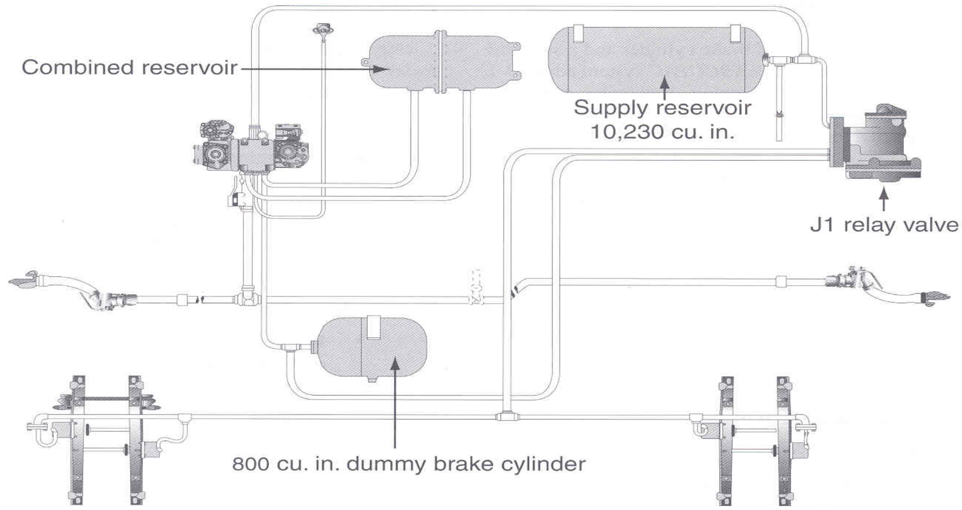


Figure 11: American relayed brake system. (Bureau 2002)

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.

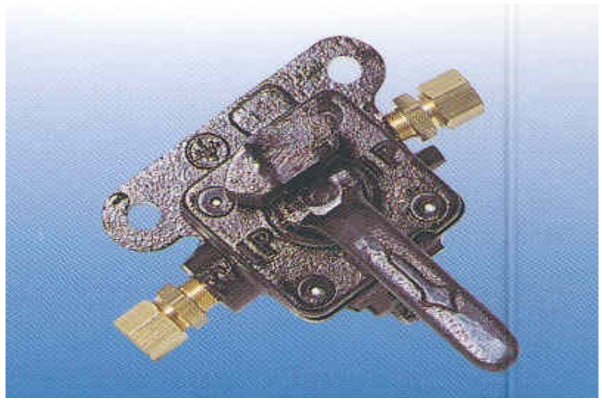


Figure 12: Grade control valve. (Westinghouse Brakes Australia)

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.

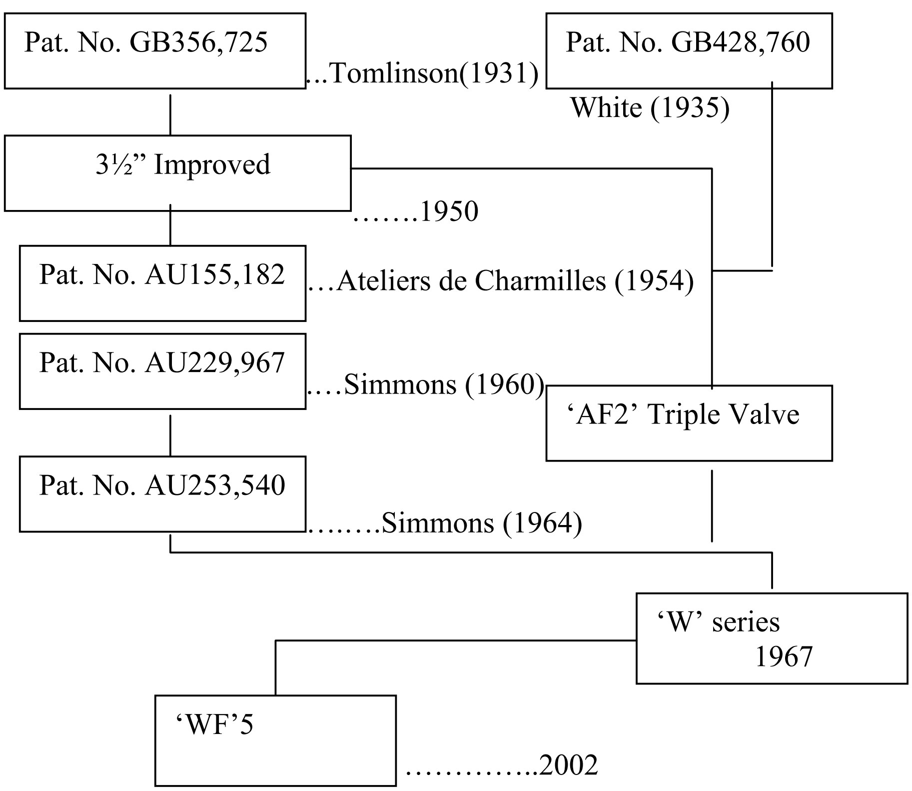


Figure 13: The Westinghouse Australian development tree.

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).

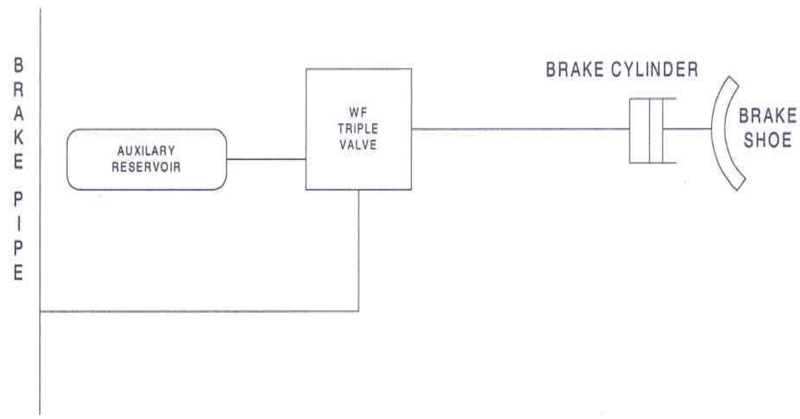


Figure 14: Australian basic control valve system

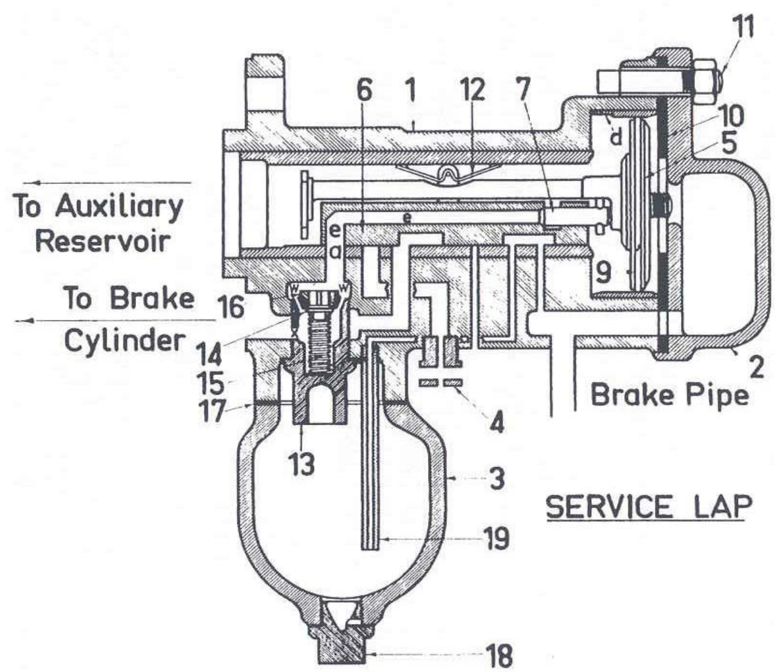


Figure 15: The 3 ½ " improved triple valve. (QR_Manual 1995)

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

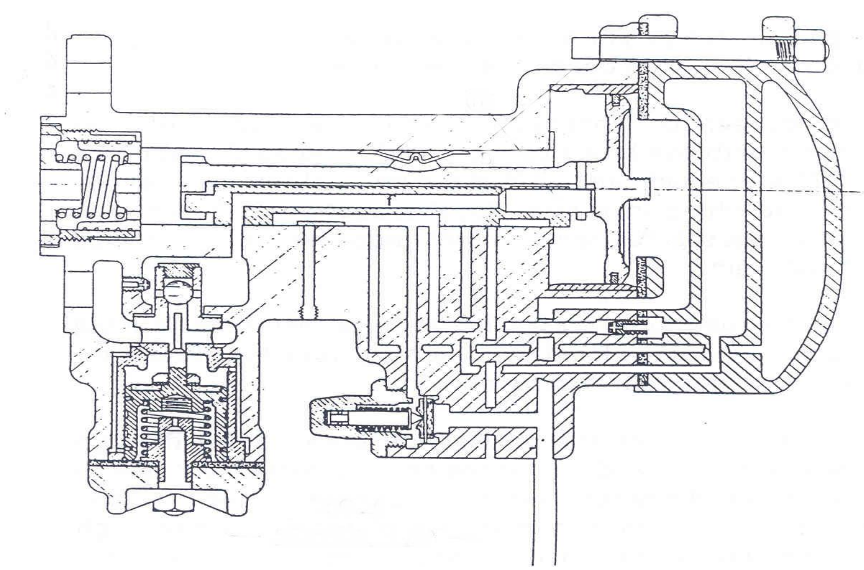


Figure 16: The AF 2 type triple valve. (QR-Manual 1995)

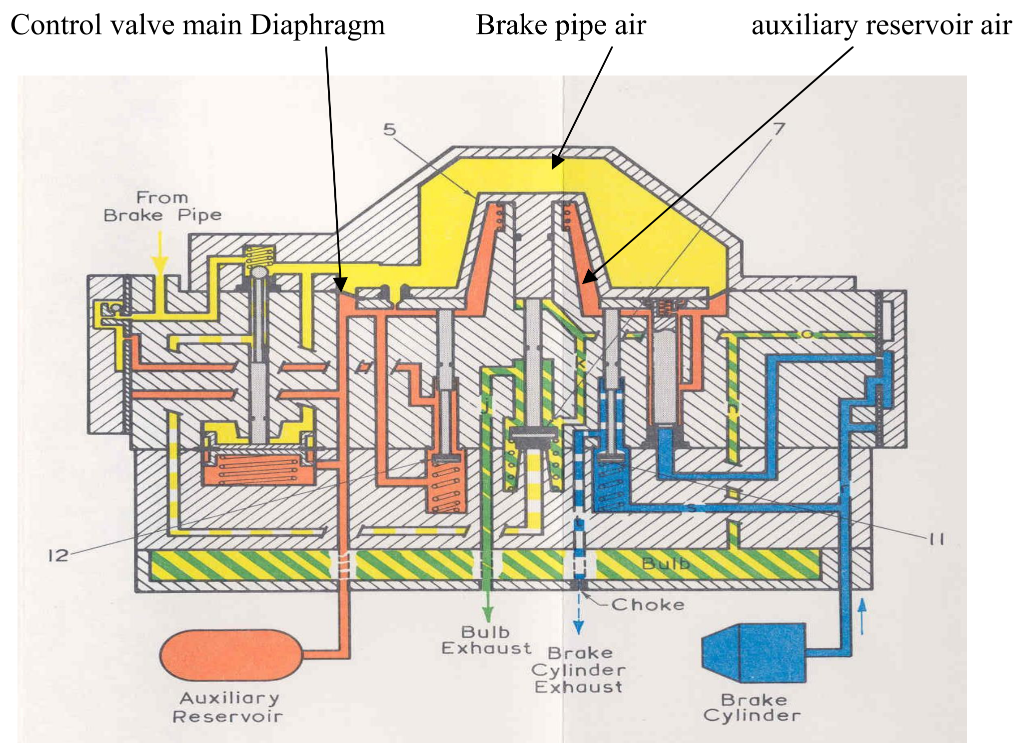


Figure 17: The basic 'W' type triple valve. (Westinghouse Brakes Australia)

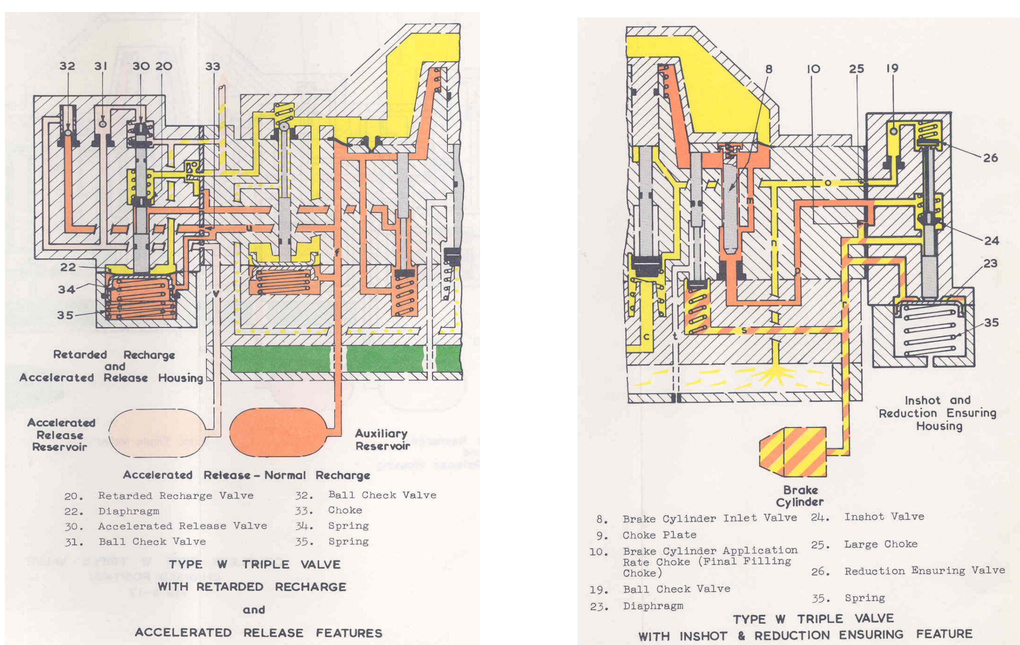


Figure 18: Added devices to the 'W' triple valve. (Westinghouse Brakes Australia)

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.

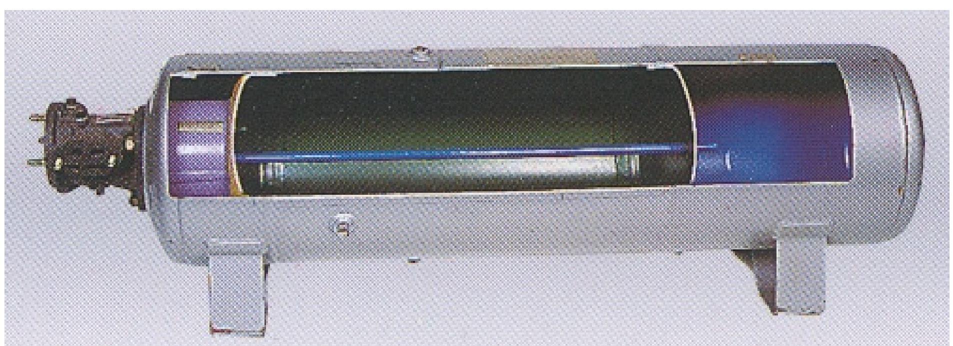


Figure 19: A combined Reservoir showing separate chambers.

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)

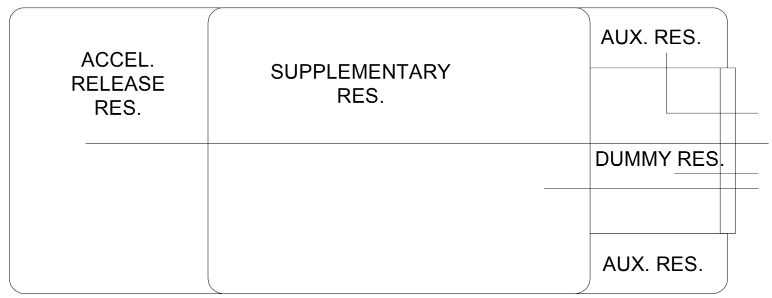


Figure 20: Multi-compartment reservoir.

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.

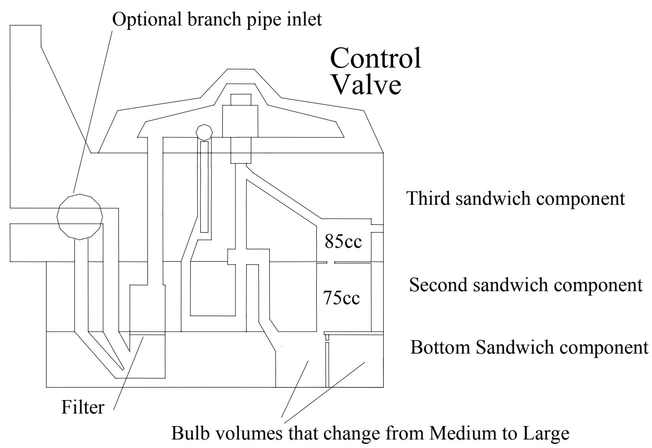


Figure 21: Control valve with bulb cavities placement

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).

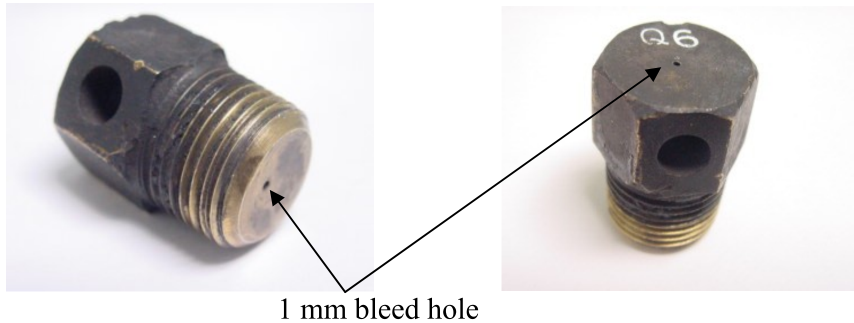


Figure 22: Q6 exhaust choke used on mineral freight wagons.

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.

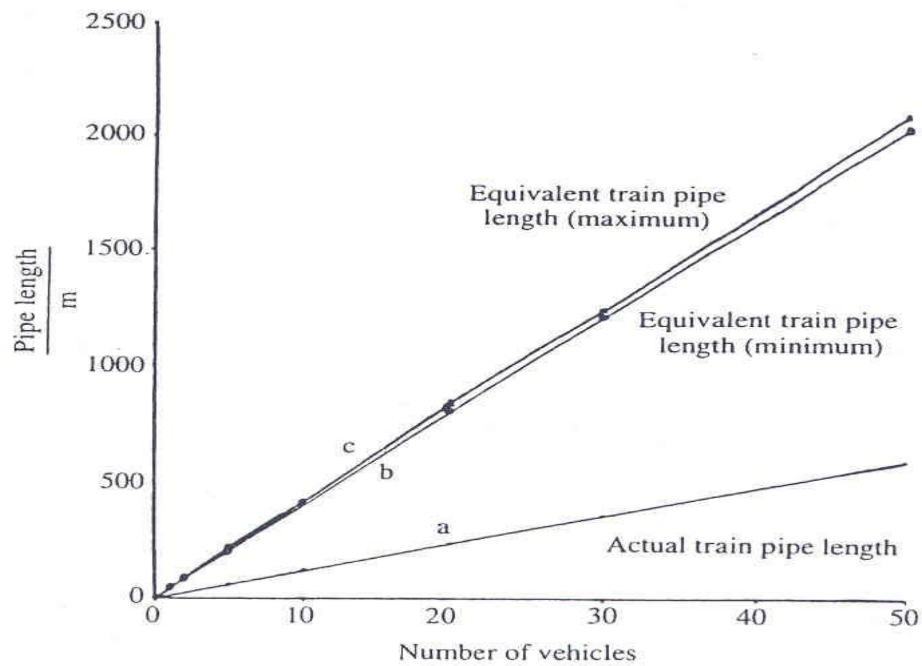
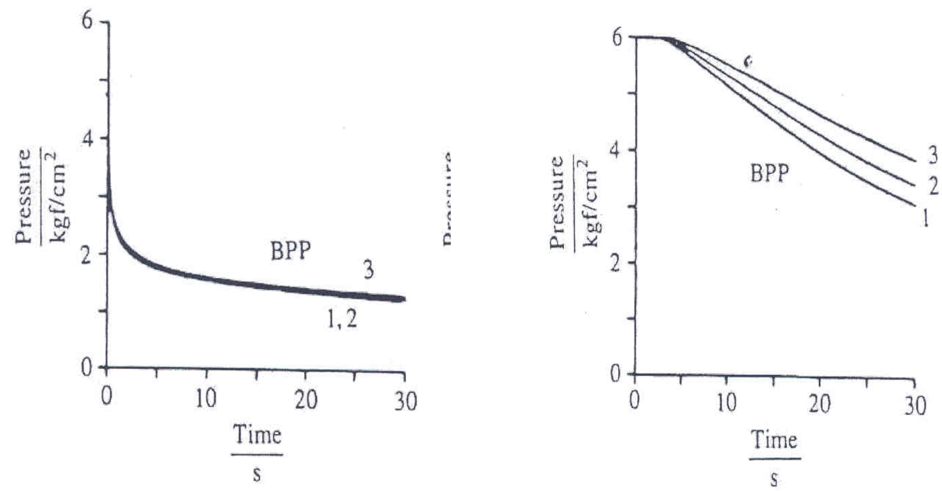


Figure 24: Brake pipe with equivalent and actual lengths. (Murtaza 1993)

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:

$$L_t = L_e + L_f \tag{2.4}$$

Using the governing equations of continuity and momentum are respectively

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

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

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)

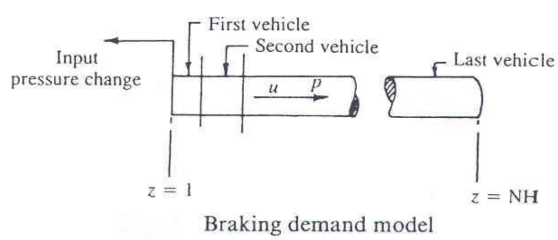


Figure 26: The braking demand model. (Murtaza 1989)

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)**.

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

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

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

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

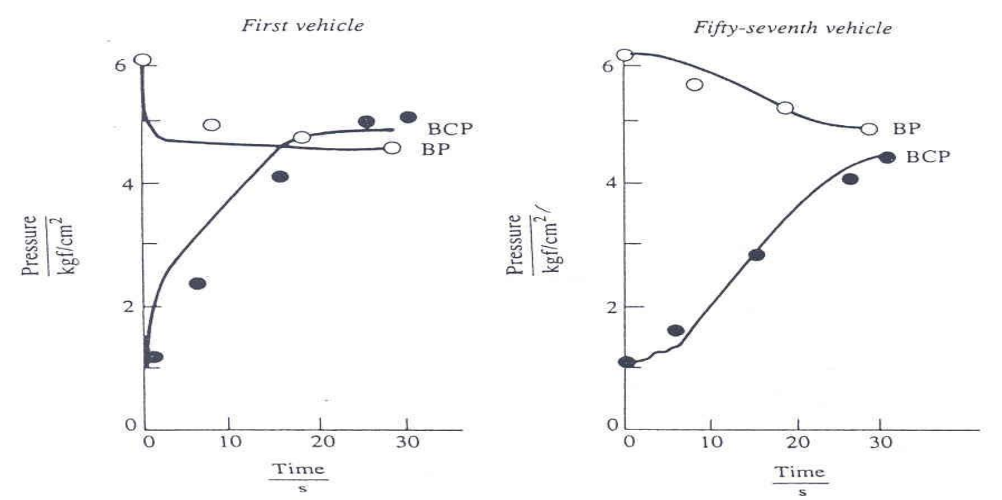


Figure 27: Results from a train pipe length of 690 m. (Murtaza 1989)

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.

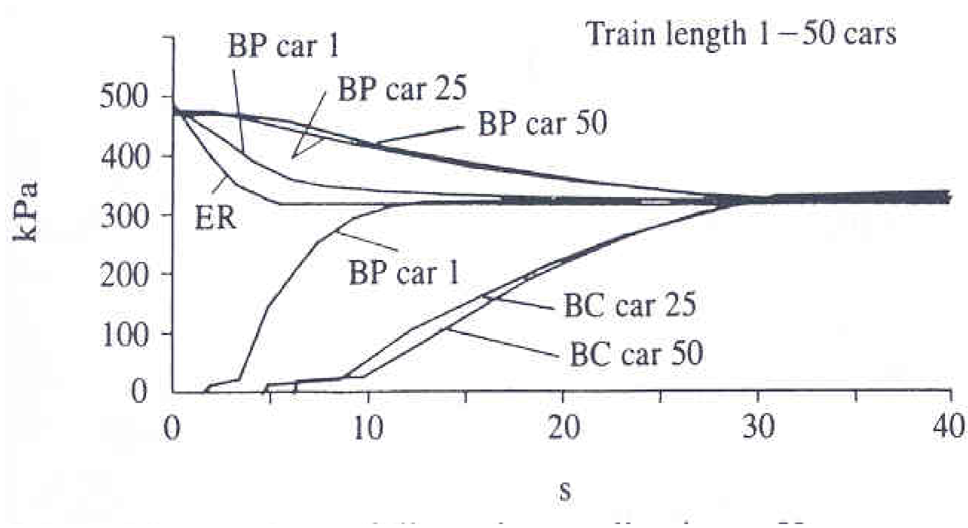


Figure 28: Full service application with an unrestricted exhaust choke. (Leigh 1990)

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).

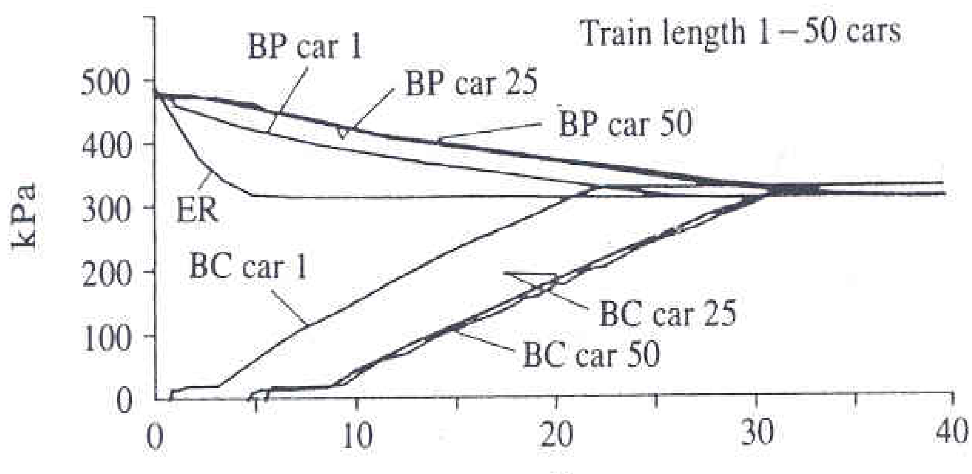


Figure 29: Full service application, with a restricted exhaust choke.(Leigh 1990)

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.

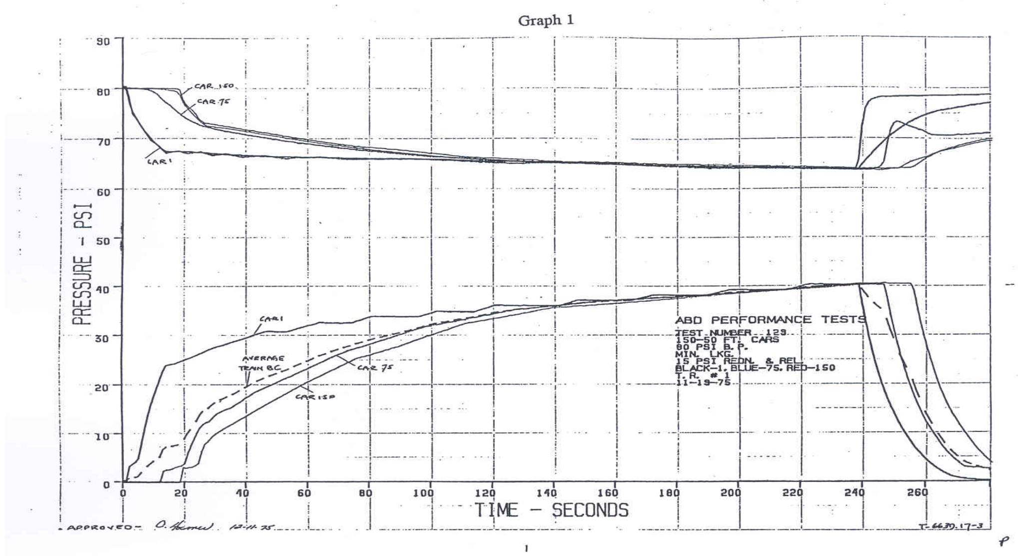


Figure 30: ABD tests for 150 cars of 50ft each in length. (WABCO 1996)

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.

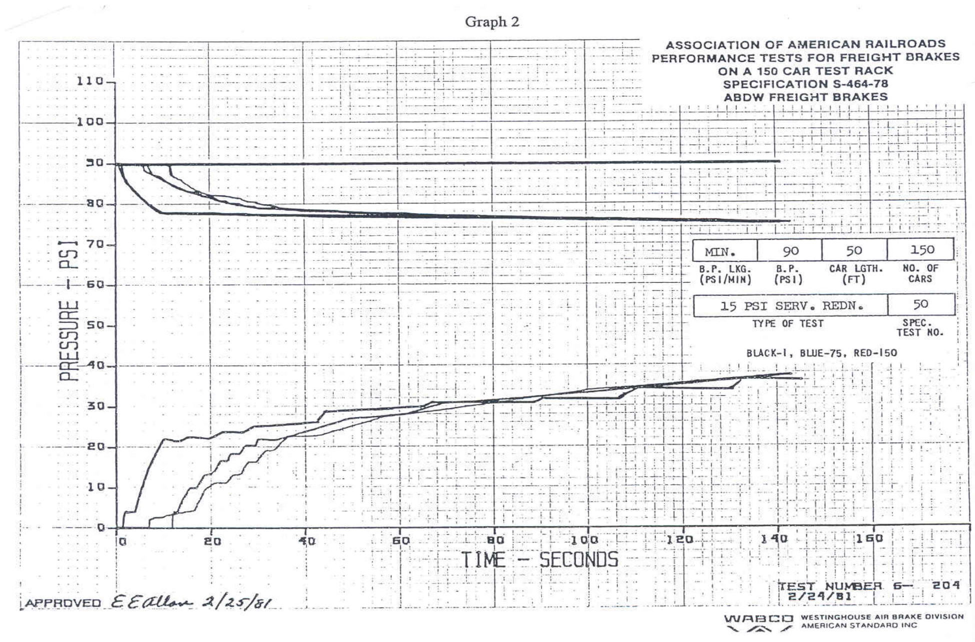


Figure 31: AAR tests of ABDW in 1981. (WABCO 1996)

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.

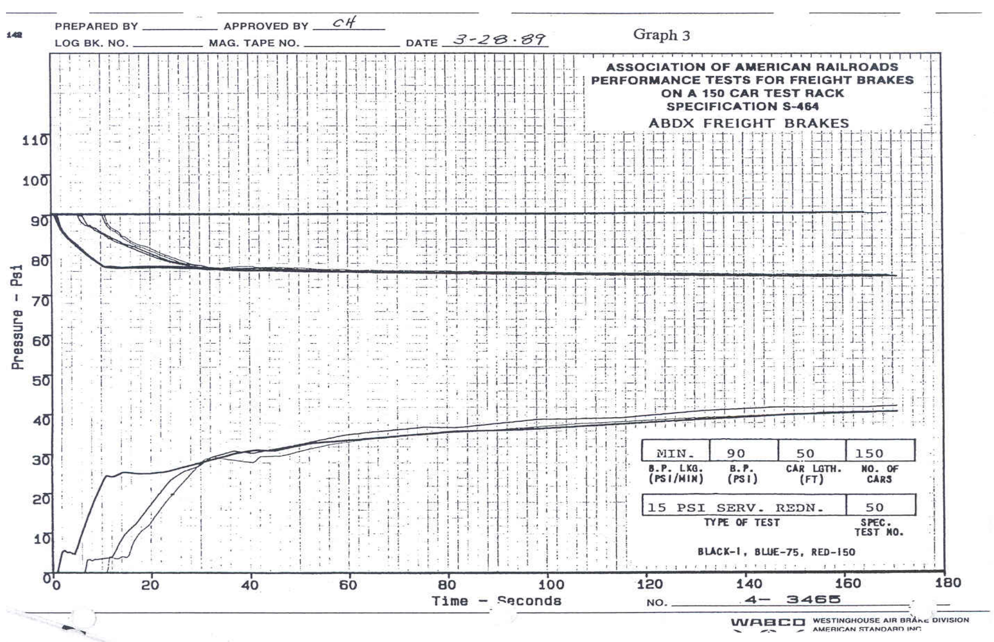


Figure 32: AAR tests of ABDX in 1989. (WABCO 1996)

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.

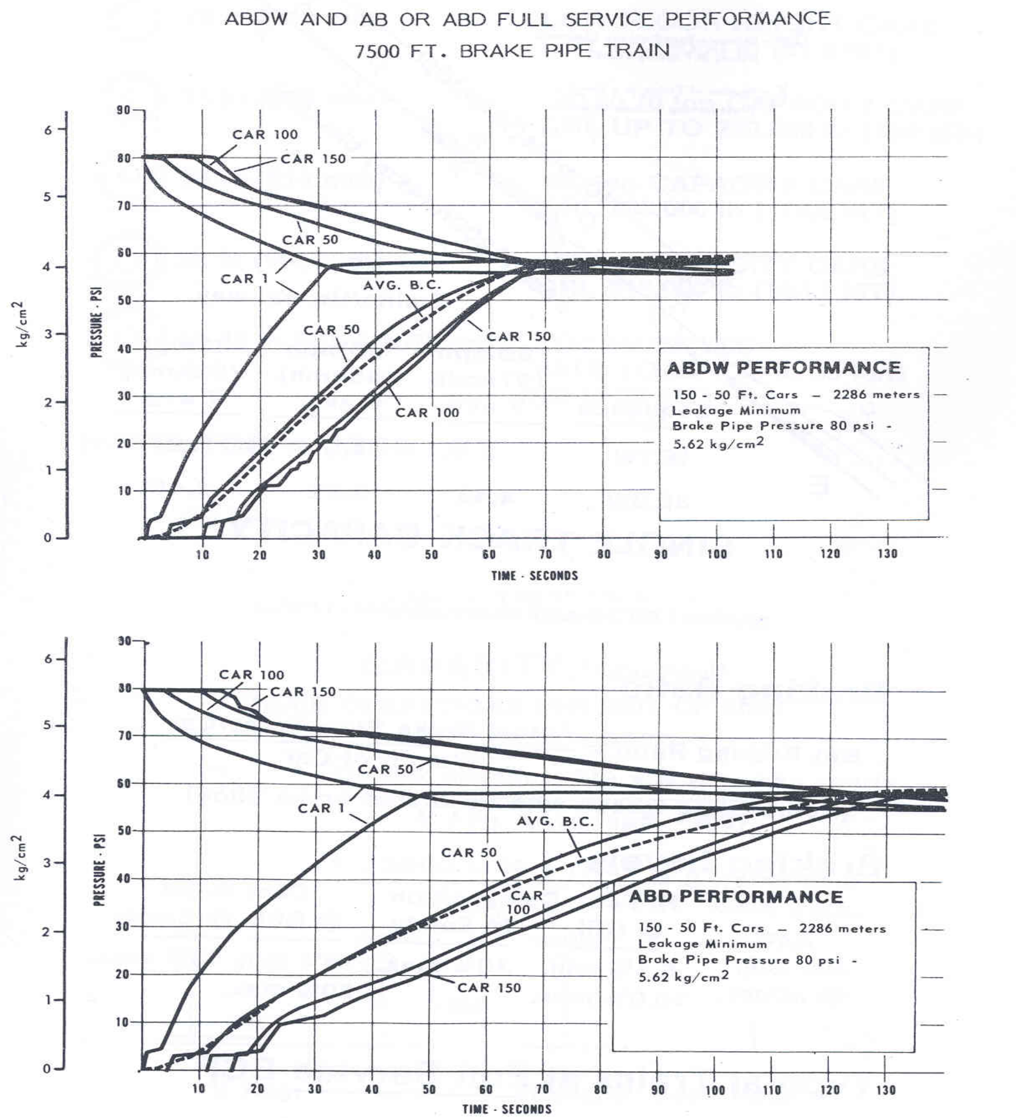


Figure 33: ABDW and ABD tests on a full service reduction.

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).

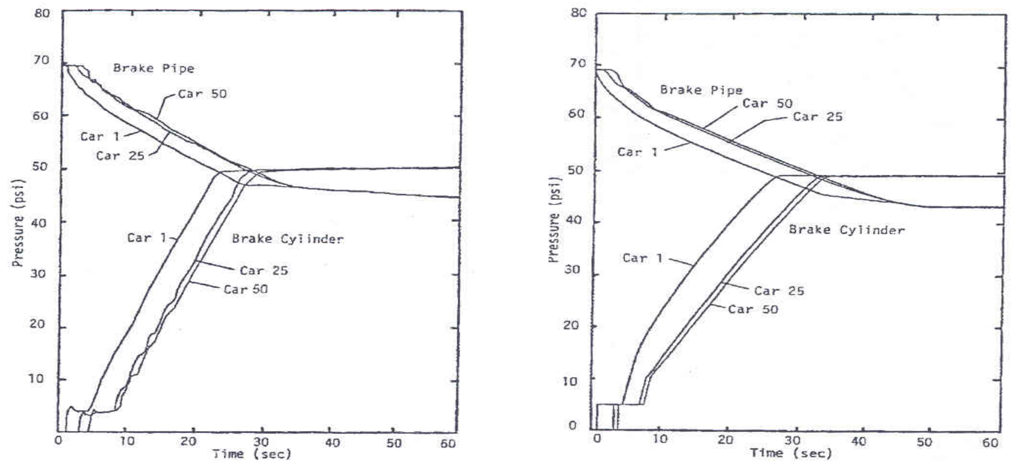


Figure 34: Results from a test rack and a simulated model. (Johnson 1986)

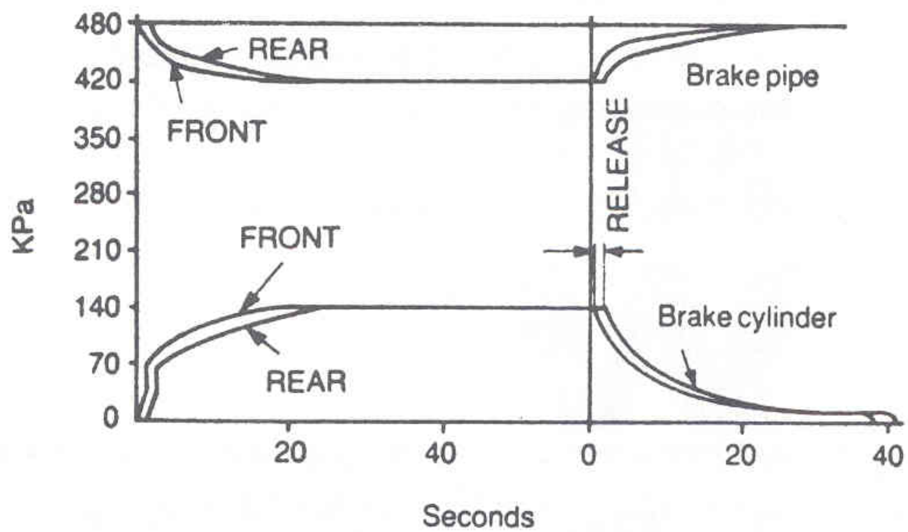


Figure 35: Service application on a 25 car train (Westinghouse Australia)

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.