INVESTIGATION OF APPLICATION DELAYS IN AUSTRALIAN TRAIN BRAKE SYSTEMS
Colin Cole* (PhD, MEng, BEng), Ian Ripley** (MEng, BET)
*Centre for Railway Engineering, Central Queensland University
**Rail CRC, Central Queensland University

SUMMARY
A brake pipe system consisting of 120m of pipe and four triple valves was assembled to test the response of the pipe and triple valves with variations on branch pipe, bulb size and the triple valve design. The rig included one combined reservoir relay type and three triple valves directly connected to auxiliary reservoirs and cylinders. The research work established that it was possible to propagate valve operations in a pipe at sonic velocity if pressure profiles and bulb volumes were optimal. The reasons why propagations are limited to 100-150m/s for minimum and service applications and 250 m/s for emergency applications are discussed. It was also established that larger 'gulps' in brake pipe pressure can be achieved with either larger bulbs and/or less restrictions to bulb mass flow. Branch pipe diameter was also found to be significant to bulb mass flows. The change to the use of combined reservoir relay type systems was identified as a significant source of reduced performance of the bulb resulting in reduced ‘gulp’ in the brake pipe. Recommendations are made for the improvement of existing systems and for directions of future research.

1 INTRODUCTION
The scope of work in the Rail CRC: Theme 1 "Smart Train - Intelligent systems" extended to the examination of train braking systems with a view to optimise use of existing capabilities as well as investigate opportunities presented by smart train systems such as 'Health card'. A Rail CRC project was approved and completed with the scope to investigate application delays in Australian Train Brake Systems. It was envisaged that this research may offer opportunity to further exploit and improve existing designs and add the possibility of saving costs in equipment upgrades. It was also envisaged that the work would increase understanding of the limitations of traditional train brake systems. Literature review, consultation with industry and examination of data led to the focus of research being directed toward improving the propagation of minimum brake applications.

2 NOTATION
f Friction factor
$F_b$ Branch Pipe Friction factor
k Isentropic index
v Valve subscript
A Cross section area, m²
$C_p$ Specific heat of air, J/(kg.K)
D Diameter, m
$D_b$ Branch Pipe Diameter, m
L Pipe Length, m
$L_b$ Branch Pipe Length, m
M Mach number
R Gas constant for air, 287 J/(kg.K)

P Pressure, Pa
$P_2$ Pressure at branch pipe connection to triple valve, Pa
$P_3$ Pressure measured above triple valve diaphragm, Pa
$P_{Bulb}$ Pressure in the quick service volume, (Bulb), Pa
$P_{BP}$ Pressure, brake pipe, Pa
$P_o$ Stagnation Pressure
T Absolute Temperature, K
$T_o$ Stagnation Temperature, K

3 BACKGROUND
It is known that the existing brake systems impose certain limitations on the maximum train lengths that can be operated. The case of the very long wagon rack operating to supply the Flinders power station in South Australia [1] is a case where these limitations have been exceeded, but operation has been achieved with reduced functionality and changed driving practice. There also seemed to be a consensus that problems existed in some installations despite a lack of documented information. It is believed that the lack of information is due to confidentiality agreements.

Discussions with industry partners initially indicated that key areas of focus for understanding and possibly improving the rate of propagations of brake pipe reduction were:
- The size of the quick service volume ('bulb')
- The dimensions of branch pipes
- The number of triple valves per unit length of pipe

Conference On Railway Engineering
Melbourne 30th April – 3rd May 2006

153
The ratio of bulb volume to pipe length between valves.

Figure 1: Short Plastic Branch Pipe

Figure 2: Long Steel Branch Pipe

Literature was reviewed for details of experimental work relevant to Australian valve designs and in the process details of North American systems were also reviewed.

An experimental program was developed based on the QR VSH wagons which utilise Westinghouse triple valves and large bulbs. As these are bar coupled wagons the bulb is at the upper extreme of its operational range, 37m of pipe. The above issues were explored.

4 LITERATURE RESEARCH

A full and comprehensive literature review was completed by the Masters candidate. Information of particular relevance to future research is reproduced here. It is instructive to compare the differing developments in the Australian and North American systems in regard to propagation of brake pipe reductions.

4.1 The Australian System

The first brake pipe reduction in an Australia system is assisted by the opening of bulbs as each triple valve operates. Air flows into the bulb until it equals pipe pressure. Further air is also extracted at this time using the reduction ensuring feature that exhausts brake pipe air directly to cylinder until the cylinder reaches 70 kPa, (cylinder in-shot).

The resulting sudden drop in the brake pipe increases the speed of propagation by assisting in the triggering of the next valve. This system would, theoretically allow propagation at sonic velocities for the first propagation, if bulbs were of sufficient size. Importantly, there is no bulb volume or other reduction ensuring functions available on a second pipe reduction.

4.2 The North American System

The North American System differs from the Australian system in two important respects. It utilises a small bulb and the reduction ensuring feature is achieved by an Accelerated Application Valve (AAV). The AAV allows air to be vented to atmosphere so that the reduction ensuring feature function is available on a second application.

While both systems have achieved similar published propagations rates [2] the North American system has the advantage of faster propagation rates on second and subsequent applications.

5 THEORY

Analysis of compressible flows is complicated by the various phenomena associated with sonic velocities. Small pressure waves in a duct propagate through the duct at sonic speed. From continuity it can be shown that a velocity (celerity) c is given by $c = \sqrt{\frac{\gamma P}{\rho}}$ for a perfect gas is $c = \sqrt{kRT}$ , [3].

Steady flows in a duct, either adiabatic or isothermal will not exceed sonic velocity. This limitation is often referred to as choking. Choking limits the maximum flow through a given area. The relationship that describes this limit is, [3], see also Figure 3:

$$\dot{m}_{\text{max}} = \frac{A_p P_o}{\sqrt{T_o}} \left( \frac{2}{k+1} \right)^{\frac{k+1}{k-1}}$$

where $P_o$ and $T_o$ are stagnation quantities,

$$T_o = T + \frac{V^2}{2C_p} \quad \text{and} \quad \frac{P_o}{P} = \left( \frac{T_o}{T} \right)^{\frac{k}{k-1}}$$

Devices added to pneumatic systems, and in particular train brake systems, which have the function of providing flow limitation are called chokes.
Figure 3: Mass Flow vs Choke Diameter

Choking is, however, not just limited to flow through contracted areas. Choking also occurs in pipes as entropy increases in the direction of pipe flow. At subsonic regimes, velocity must increase in the direction of flow. This is true of both adiabatic and isothermal flows in ducts and pipes with friction. Therefore, a length at which the flow velocity at the end of the pipe will be sonic (adiabatic case, Fanno Flow) and 0.845M (isothermal case). For ducts or pipes longer than this length, flow will be reduced as the exit conditions do not exceed sonic (M) or 0.845M, hence choking due to pipe length.

The governing equation describing Adiabatic Flow with Friction (Fanno Flow) from [3] can be expressed as:

$$f \frac{L}{D} \frac{1}{2} \frac{m^2}{A^2} = \frac{1}{2} \rho_1 dp + \frac{2}{A^2} \ln \frac{p_2}{p_1} \quad (2)$$

Calculation of critical lengths,[4] can be obtained from

$$f \frac{L}{D} = \frac{1-M_1^2}{kM_1^2} + \frac{k+1}{2k} \ln \left( \frac{(k+1)M_1^2}{2(1+ \frac{k-1}{2} M_1^2)} \right) \quad (3)$$

Isothermal Flow with Friction (quoting [3])

$$p_1^2 - p_2^2 = \frac{m^2 RT}{A^2} \left( f \frac{L}{D} + 2 \ln \frac{p_1}{p_2} \right) \quad (4)$$

And the same equation re-arranged allows calculation of choking lengths

$$\frac{M_1^2}{M_2^2} = 1 - kM_1^2 \left( f \frac{L}{D} + 2 \ln \frac{M_2}{M_1} \right) \quad (5)$$

Using the isothermal approximation offered by equation 5, choking flows for pipe lengths were checked for 12 and 20 mm pipes, these being relevant to results obtained for branch pipes.

Figure 4: Choking Flows and Lengths, φ12, φ20.

6 EXPERIMENTAL RESEARCH

6.1 Equipment Details

Due to difficulties in gaining access to rollingstock, an experimental program was developed using four control valves and brake pipe lengths approximating VSH wagon pairs. Data was collected using data acquisition systems so that both timing and magnitudes of pressure levels could be studied.

The laboratory set up included a single combined reservoir relay type system and three direct acting triple cylinder systems as shown in the schematic in Figure 5.

Figure 5: Schematic of Brake Valve Test Rig

6.2 Experimental Results

Experiments in the areas of interest were completed with results presented in the following sections. A summary of results the entire test program is presented in Table 1.

Plots of data from tests of medium bulbs and large bulbs are shown in Figure 6. Further tests were completed using a manually operated 'bulb' of the same volume as the large bulb, 59x10⁻⁵m³, (or 590 cm³), Figure 7 and Figure 8. Investigation of branch pipe dimensions is shown in Figure 9. An estimate of the length of the passageways in the valve and pipe bracket is presented in Figure 10. The action of the quick service valve for different branch pipe connections is compared in Figure 11. The effects of modifications to the isolation cock disc are compared in Figure 12. The investigation
of the pipe bracket included modification of the rotary valve. The reasons for this were:

- The area available in the disc was obviously smaller than the porting in the casting.
- It would be easy to modify.

The steps in the data are due to bitwise limitations on resolution in the data acquisition process. Data was not filtered to ensure that no phase shifts were introduced to results.

### Table 1 Summarised Results from the Experimental Program

<table>
<thead>
<tr>
<th>Experiment Type</th>
<th>Experiment Detail</th>
<th>Result</th>
</tr>
</thead>
<tbody>
<tr>
<td>Measure Propagation Rates</td>
<td>Single Triple Valve</td>
<td>Gulp propagated at sonic speed</td>
</tr>
<tr>
<td></td>
<td>Pipe Exhaust at 54 kPa/min</td>
<td></td>
</tr>
<tr>
<td>Measure Propagation Rates</td>
<td>Four Triple Valve</td>
<td>Gulp propagated at sonic speed</td>
</tr>
<tr>
<td></td>
<td>Pipe Exhaust at 48 kPa/min</td>
<td>Gulp magnitudes increasing</td>
</tr>
<tr>
<td>Compare Bulb Sizes</td>
<td>Medium and Large Bulbs and Valve with Pipe Bracket</td>
<td>No measurable increase in ‘Gulp’</td>
</tr>
<tr>
<td>Compare Bulb Flow</td>
<td>1. Large Bulb Volume with Pipe Bracket</td>
<td>Gulp ~4.7 kPa</td>
</tr>
<tr>
<td>Restrictions</td>
<td>2. Large Bulb Volume without Pipe Bracket</td>
<td>Gulp ~6.7 kPa</td>
</tr>
<tr>
<td></td>
<td>3. Equivalent large bulb volume direct connected to</td>
<td>Gulp ~11.5 kPa*</td>
</tr>
<tr>
<td></td>
<td>Brake Pipe.</td>
<td></td>
</tr>
<tr>
<td>Compare Branch Pipe</td>
<td>1. 12 mm ID, 800 mm long</td>
<td>Gulp reduced for 12mm ID pipe</td>
</tr>
<tr>
<td>Lengths</td>
<td>2. 20 mm ID, 150 mm long</td>
<td>No Measurable difference for longer pipes</td>
</tr>
<tr>
<td></td>
<td>3. 20 mm ID, 800 mm long</td>
<td></td>
</tr>
<tr>
<td>Pipe Bracket</td>
<td>1. Test without Pipe Bracket</td>
<td>Gulp ~ 2 kPa larger without pipe bracket.</td>
</tr>
<tr>
<td></td>
<td>2. Test with Pipe Bracket</td>
<td></td>
</tr>
<tr>
<td>Pipe Bracket Rotary Valve</td>
<td>1. Test with 20 mm ID branch pipe</td>
<td>The enlarged valve port increased the</td>
</tr>
<tr>
<td>Valve enlargement</td>
<td>2. Test with 12 mm ID branch pipe</td>
<td>duration of gulp for the smaller diameter</td>
</tr>
<tr>
<td></td>
<td></td>
<td>branch pipe.</td>
</tr>
</tbody>
</table>

* Value conservative as bulb was manually activated at 468 kPa, 22 kPa lower than normal activation time.

![Figure 6: The 'gulp' and local pressure reduction from a medium and a large bulb control Valve](image)

The final tests presented are bulb filling times, Figure 13. A sample of some field data, supplied by QR is shown in Figure 14.
Figure 7: Manually operated bulb.

Figure 8: Results of Manually Operated bulb.

Figure 9: Local pressure reduction for different size and length branch pipes

Figure 10: Passageways in the pipe bracket and control valve of 730 mm.
Figure 11: Gulps in the brake pipe with and without a pipe bracket

Table 2 Branch Pipe and Rotary Port Details

<table>
<thead>
<tr>
<th>Test</th>
<th>Branch Pipe Length, mm</th>
<th>Branch Pipe ID, mm</th>
<th>Rotary Disc Size, mm</th>
</tr>
</thead>
<tbody>
<tr>
<td>700</td>
<td>800</td>
<td>20</td>
<td>7.65</td>
</tr>
<tr>
<td>705</td>
<td>800</td>
<td>20</td>
<td>9.5</td>
</tr>
<tr>
<td>711</td>
<td>800</td>
<td>12</td>
<td>9.5</td>
</tr>
<tr>
<td>716</td>
<td>800</td>
<td>12</td>
<td>7.65</td>
</tr>
</tbody>
</table>

Figure 12: Local reduction pipe pressure with different branch pipes and a change in the port diameter of the isolation cock rotary disc.
**Field Data**

*Figure 13: Bulb filling times for medium and large bulbs*

*Figure 14: Brake pipe reduction rate of a head end loco and a 92 wagon train (Queensland Rail 2001)*
7 MODELLING AND ANALYSIS

Using measured data from Figure 6 for the large bulb, further analysis was carried out using equation 4 to model the flow in the branch pipe and flow through the valve to the volume above the diaphragm. The version of the triple valve used had the feature that a choked flow of air direct from the diaphragm cavity is made available to the cylinder or dummy cylinder. The first stage of valve operation is therefore as shown in Figure 15 the second stage is shown in Figure 16.

![Figure 15: Initial Flow Paths To Bulb and Inshot](image)

![Figure 16: Second Stage – Connection to Bulb](image)

The measured data from Figure 6 gives values for $P_{BP}$ and $P_3$ allows the mass flow rate to be calculated. As it is desired to understand the effect of the branch pipe and valve restrictions separately, in iterative algorithm was used to guess $P_2$ and then calculate mass flows for the two segments. This is done by simultaneously solving equation 4, eg

$$p_{BP}^2 - p_2^2 = \frac{m^2RT}{A_b^2} \left(f_b \frac{L_b}{D_b} + 2 \ln \frac{P_{PB}}{P_2} \right)$$

$$p_2^2 - p_3^2 = \frac{m^2RT}{A_v^2} \left((f \frac{L}{D})_v + 2 \ln \frac{P_2}{P_3} \right)$$

While the pipe friction in the branch pipe could be approximated by an empirical formula and the length and diameter known, this could not be estimated for the valve restrictions. As isothermal analysis was used, the mass flow to the bulb could be calculated directly from the gas law, i.e. $P = \frac{mRT}{V}$ gives $dP = dmRT/V$. The mass flow through the branch pipe is however the sum of the inshot flow and the flow to the bulb therefore to determine the mass flow in the branch pipe the flow from the auxiliary reservoir to dummy cylinder also needs to be modelled. As the precise manufacturer’s information was not known, chokes were selected to give a cylinder fill time of 13 seconds. Using the selected cylinder fill rate inshot flow was calculated allowing the branch pipe flow to be equated to the sum of inshot flow and bulb flow. The friction factor in the branch pipe was approximated to: $f = 50$ ($Re < 1$), $f = 64/Re$ ($1 < Re < 2000$), $f = 0.0426$ ($2000 < Re < 3000$), $f = 0.316/Re^{0.25}$ ($Re > 3000$). The $f LID$ characteristic of the valve restrictions were selected iteratively in the algorithm until the mass flow in the branch equalled the mass flow to the inshot and bulb.

The resulting response is shown in Figure 17 and Figure 18.

![Figure 17: Modelled Branch Pipe Response](image)

![Figure 18: Modelled Branch Pipe Response - Detailed](image)
Investigation of application delays in Australian train brake systems

installation has only 120 m of pipe and is fitted with a 37 litre reservoir. The added end volume approximates to the volume of 39.6 m of pipe. Two issues are raised. The first is that pipe and added volume used for the laboratory tests approximates to a very short train. The second is that adding a volume to simulate extra pipe is not exactly the same as adding extra pipe.

As the laboratory equipment approximates a short train the mass of air stored in the pipe is relatively small. The action of the bulb is to remove air from the system as air is also being exhausted from the exhaust choke at the end of the brake pipe, Figure 21.

If a large reservoir was added at the brake pipe at the same point as the triple valve and bulb was connected it would have the effect of replenishing the brake pipe with a mass flow of air, Figure 22. If the reservoir was large, its pressure would remain high longer, and unless restricted could cancel the effect of the bulb. In the case of the laboratory equipment the reservoir exists but is relatively small and so sonic propagation is possible.

The second thing to note is that a large reservoir added to the pipe is not the same as a long narrow reservoir (or pipe), Figure 23. In the case of a long tail pipe, flow has to occur along the pipe and the pressure delivered to the valve is less than the maximum pressure in the pipe. As this pressure is less than that in Figure 22, the difference between the local pipe pressure and bulb pressure is also reduced and so less mass flow from the long pipe to the bulb can occur. So in the case of the laboratory rig, the added volume would have delivered more flow than its equivalent volume in pipe, (eg 39.6 m of pipe), but it was not sufficient to limit propagation. As shown from the data, propagation was limited to the speed that the pressure wave travelled, which is limited to sonic velocity due to the properties of Fanno flow.

8 DISCUSSION

In considering the first results in Table 1, it is immediately apparent that propagation rates reach sonic velocities. It is known from literature and industry anecdotes that this is not achieved in practice, Review of Australian and North American systems place propagation rates in the region of 100-150 m/s for minimum and service applications while emergency applications reach ~ 250 m/s. [2]. Clearly, the laboratory installation does not reflect these conditions. It is noted that the laboratory
Investigation of application delays in Australian train brake systems

C Cole, I. Ripley
CRE, CQU; Rail CRC, CQU

Figure 23: Schematic of Quick Service Valve or Bulb in Operation (With long Brake Pipe)

From these observations it can be deduced that sonic velocity of propagation of pressure waves can be used to achieve the sonic propagation of valve operations if conditions are correct. To achieve valve operation, the pressure of the brake pipe must be about 12 kPa below the auxiliary pressure. To achieve sonic operation of valves after the first valve and bulb operate requires that the pressure wave (or 'gulp') induced in the brake pipe is sufficient to lower the brake pipe pressure at the next valve from its existing pressure to a pressure 12 kPa below auxiliary pressure. From the data collected, brake pipe gulp can supply about 3 to 5 kPa, so for sonic propagation the brake pipe pressure at the next valve must not be greater than about 3 kPa above the pressure at the valve just triggered. If the pipe pressure at the next valve is not within the range where the gulp can trigger the valve, (i.e. a steeper pressure profile), then the gulp will travel past the valve and not assist propagation. In such a case propagation must slow until it is reinitiated at the next valve by the steady drop of the pipe pressure due to the end exhaust. Several things are now evident:

- The pressure wave or gulp will not be visible in the lead area of the train where exhaust flow is large anyway, see Figure 14.
- The propagation will slow or pause and drop below sonic if the pressure profile conditions are steeper than the capabilities of the bulb.
- There will often exist a point in a train where propagation will pause due to insufficient bulb size and 'gulp' to operate the next valve due to either or both:
  - Limited pipe exhaust flow due to choking
  - Too much mass flow from air stored in the tail of the pipe.
- The pause will logically occur somewhere mid wagon rack.
- As the pressure profile in the tail of the train can be quite flat, resumed propagation can be quite rapid, even sonic, with well defined guls, see Figure 14.

Of interest in the experimental data is the unexpected lack of effect of the increased bulb size as shown in Figure 6. The difference is so small that it is not significant as it corresponds to only one bit resolution in the data acquisition system. The work completed by the Masters student Ian Ripley [2] in using a manufactured volume equivalent to the large bulb showed conclusively that the performance of bulbs is limited or 'tuned' by the restrictions, filters and chokes in the brake pipe bracket and triple valve, Figure 8.

What is significant is that installations with pipe brackets will have significantly more restrictions than installations where the triple valve is used alone, Table 1, Figure 10, Figure 11 and Figure 13.

There exists inconsistency in branch pipe lengths and diameters on wagon installations as noted in Figure 1 and Figure 2. The experimental program was limited to only two branch pipe sizes, 12mm and 20 mm. In agreement with the Darcy's law $h_L = \frac{F \cdot v^2}{2gd}$, or $h_L = \frac{8\pi Q^2}{(\pi d^5)}$, changes in diameter gave larger changes in gulp performance. Changes due to changes in length were also expected, based on Darcy's law but were not measurable, see Figure 9.

Some research was completed exploring the flow restrictions added by the pipe bracket. It will be noted in, Table 2 and Figure 12 that the enlarging of the hole in the pipe bracket rotary valve gave improvement for the 12mm branch pipe.

As the branch pipe comparison data did not conclusively demonstrate the advantage of larger diameter shorter branch pipes, the experimental data was used to develop models that quantified the pressure drops in branch pipes, pipe bracket and valve as shown in Figure 15 through to Figure 19. The modelling confirmed two things. That a smaller branch pipe, eg 12mm diameter would significantly reduce mass flows, as noted in Section 7 and that pressure drop within the pipe bracket and valve combined is quite large, confirmed by minimal pressure drop in the 20 mm diameter branch pipe, Figure 19.

It seems fairly clear that there is scope to further optimise train brake reduction propagation for long trains. Recent innovations using combined reservoirs and pipe brackets appear to have reduced the effectiveness of the bulb and reduced access to the volume available for the reduction ensuring function.

9 CONCLUSIONS

Improvements to brake propagation can be achieved with the relatively minor modifications of:

- Connecting the branch pipe to the alternate port on the triple valve instead of the connection on the pipe bracket.
- Ensuring branch pipes are of 20mm ID and minimum length.

It would appear that larger bulbs will have little effect as flow to the bulb and the resultant gulp appears to be controlled by internal valve restrictions, even when the pipe bracket is bypassed. A review of these restrictions for operations where large bulbs were specified, (i.e.
C Cole, I.Ripley
CRE, CQU; Rail CRC, CQU

bar coupled wagons with longer brake pipe lengths per valve) is recommended.

If valve restrictions can be reduced it is believed that larger bulbs could improve and allow effective propagation in longer trains. If achieved, this will contribute to rail achieving growth by extending capability and useful life of existing rollingstock assets.

10 FUTURE WORK

It is planned to continue investigation using the modelling of the branch pipe, pipe bracket and valve as developed in Section 7 to determine what combination of design and conditions are required to achieve larger branch mass flows. From this work it is hoped to produce minor modifications that can be applied to pipe bracket and valve combinations to achieve improved propagation rates.

11 ACKNOWLEDGEMENTS

This project was funded by the Cooperative Research Centre for Railway Engineering and Technologies of Australia (Rail CRC) Theme1: Smart Train System. The work has also been supported by direct contributions from other Rail CRC Partners, QR and RIC. Experimental work was completed at the Centre for Railway Engineering (CRE) at Central Queensland University.

12 REFERENCES

2. Ripley, B, An Investigation of Brake Application Delays in Australian Train Brake Systems