QUEENSLAND RAILWAYS STEAM LOCOMOTIVES 1900 - 1969

DESIGN AND OPERATION

CORRIGENDA

(The book is available from ANGRMS Sales, PO Box 1135, Woodford, Queensland 4514, tel (evgs) (07)3273 2014.)

This corrects various statements in the book, and expands on others to avoid misunderstanding. It also corrects errors in a review of the book. This is the October 2008 version, containing additions to earlier versions. Corrections of further errors which come to light will be added to this page.

Various expansions, explanations and further examples have occurred or been told to the author since the book was published. They are being progressively added to the text of the book on his computer, along with these corrections. They are available to anyone interested. Contact the author at the email address given on the home page. Two of these expansions etc are so close to corrections that they have been added to this page. These are an important addition to 10.12, and a whole section on Conclusions from the Perform Program.

1.02 Construction Standards

p 10 RH, para 1 - There were indeed vertical curves between changes in gradient, even if not shown in the Working Plans and Sections. The Standard Specifications for construction of 1896 required that a vertical curve of two miles radius be introduced between changes in gradient during construction. Such curves were important for preventing coupled axleboxes of locomotives from striking the tops of cut outs or horn blocks above and keeps below (as sometimes happened where a siding on a steep gradient joined another line without such a vertical curve).

1.022 Main Line Track Standard

p 15 LH - The ten per cent addition to the speed limit for emergency did not apply to the 50 mph limit, and that addition was dropped from all speed limits in the 1962 GA.

1.043 Drawgear

p 30, LH para 2 - Only on the PB15 and B15Con was the dragbox a casting. On other engines, it was fabricated.

1.055 Breaking out of the Technology

p 40 A draft letter for the then Minister to send to Wilmot of Beyer, Peacock in August 1951 appears on several QR files (eg Secretary's 69.753, Batch 3, PB15, General). On account of the drawgear limiting loads to 650 tons and 60F wagon units, with loops and yards to suit, it was considered that strengthening to allow the heaviest current engines to run, and reduction in gradients, were the best policies to follow. The intended 15 tal from Ipswich to Rockhampton was for the longer term (see 1.06 below on the locomotive policy put forward in this letter).

1.06 War and Post-War

p 43 The draft letter to Wilmot of Beyer, Peacock in August 1951 had this to say about policy on locomotive construction and use. The C16, C17, C19 and CC19 were not satisfactory designs, and were to be discarded. The AC16 had an axle load too heavy for its tractive effort. The following were to be retained: the PB15 and B15 for the lightest lines, the BB18¼, CC17, DD17, the Garratts for heavy passenger and goods, heavy diesel and light diesel. Specific shunting locomotives were not needed, the B15 classes proving excellent. More Garratts could not then be recommended.

3.02 Influences on Design, Standardisation

p 52, RH, para 1 - The DD17 superheater header was not the same as on the C17 and D17. Although of similar size, it differed in having the Melesco air snifter discussed in part 5.

3.03 Frames

p 53 LH para 2 and Fig 3.01 - On the B18¼, the main frames passed through the transverse stretcher mentioned, and at that point were much reduced in height.

p 53 RH para 2 - On the DD17, the saddle, and carriers for the bogie pivots front and rear, were welded, fabricated, steel plate.

p 54 LH para 1 - The carrier for the rear bogie on the suburban tank engines was fabricated.

p 55 RH para 3 - The cylinders were affixed to the frames by bolts, but the strain on the bolts was reduced by a rectangular spigot attached to the cylinder casting, which penetrated the frames and saddle. The dimensions of the spigot varied from 6¼ by 6½ inches on the D16 to 19 3/8 by 10 inches on the B17. Exhaust steam flowed from the cylinders through a cavity in this spigot then through the saddle to the blast pipe (see Figs 3.04 and 6.02). On the engines with slide valves, so too did the admission steam to the cylinders, which explains why the spigots on those engines were longer than on the piston valve engines, which had direct pipes for the admission steam.

p 58 RH para 5 - the distance at the lower level was 8ft 4ins, not 8ft 10ins.

p 58 LH paras 2 and 4 - The frames of the AC16 were cast steel, one casting each side.

p 58 RH para 4 - Only the leading coupled axle block of the DD17 was fitted with a horn block, the same as on the D16 and D17.

3.04 Axleboxes

p 62 LH para 1 - Where the spring was under the axlebox, the hanger was an upright T. Where the spring was above the box, the oil box was supported by two pins.

p 62 LH para 2 - During most of the period under review, there was no keying of coupled wheels to axles on QR locomotives (a key was a piece of metal pressed into the end of an axle and a part of the wheel on each side to prevent movement of the wheel on the axle).

p 62 RH final para - As reciprocating motion was converted to rotary, the maximum piston thrusts occurred when the crank pin was directly in line with the piston rod.

p 63 LH paras 3 and 4 - All QR axleboxes except those on the BG had wicks passing from the reservoir in the top of the boxes through passages in the body of the boxes to lubricate the front and rear faces. The brass boxes had wicks passing through similar passages to lubricate the slog faces. See Fig 3.07 for the six passages through the body of these boxes for lubricating the bearing, front and rear and side (slog) faces. On the BGs, the oil for the front and rear of the box was supplied from an oil box on the side of each engine unit. The loop in the top of PB15 axleboxes had been removed by the 1950s. Some of the brass boxes had threaded holes into which lugs could be screwed for lifting.

p 65 Fig 3.07 - The pipe entering the top left of the side elevation existed only while the oil supply for each axlebox was a oil box on the frames or in the cab, as described in 3.11.

p 67 Fig 3.09 - There was no oil reservoir at the top of the (BG) box. The bearing was originally lubricated by oil from the mechanical lubricator, later by grease. The oil was supplied for the front and back of the box (the X grooves) from a box on the side of each engine unit.

3.06 Axles etc

p 68 LH para 5 - The BB18¼, DD17 and roller bearing versions of the C17 and D17 did not have collars, the taper on the rollers providing side control. PB15s were built with collars, but collars were removed from both coupled and bogie axles on these engines, probably when replacement axles were fitted, perhaps following the design of the 1924 engines (engine axles were renewed after 25 years). No hub liners were fitted to PB15s thereafter to allow easy compensation for wear at the slog face.

3.08 Drive

p 80, LH para 1 - Although it was stated at the time that cylinder blocks were to become cast steel they remained cast iron except for some C17s. The cylinders of saturated engines wore little and were rarely replaced. Hence, liners were fitted only to the cylinders of superheated engines, except the BG, which had none. The DD17 had welded fabricated cylinders with cast iron liners.

P 80 LH para 4 - The vertical component of the thrust on each side of a C19 is about 1900 lbs.

p 80 RH para 5 - When new, the BGs, at least Nos 1001-10, had pipes taking the cylinder cock drains to the outer ends of the locomotive, but they were soon removed. It was to this arrangement that 1009 was restored in 1994. The automatic opening of the cylinder cocks when no steam was in the cylinders did not apply to the saturated engines.

p 80 RH para 5 What seemed to be open cylinder cocks on engines running fast was in some cases at least the pressure relief valves blowing on every stroke, the result of compression in the cylinders exceeding boiler pressure on engines with short travel valves, as explained in amendments to Section 5.03 below.

p 81 LH para 4 - The adjustments to the guide bars were made by fine shims, and not wedges.

3.09 Cabs and Controls

p 85 LH para 2 - The sliding glazed sashes in the flat side sheets of the B18¼s built up to 1930 were fitted when the engines were built.

p 85 RH final para - The windows on each side of the rear wall of tank engine cabs could be opened inwards, but on account of the coal dust problem, they were kept closed with the engine running in reverse. The lower part of these windows was protected from damage from coal by horizontal metal bars.

p 86 LH re cab roofs - The steel roofs of the DD17, BB18¼ and BG were indeed double with a small gap between, and a layer of marine ply or similar beneath the lower steel roof. The Maloney ventilator was about six inches high in the centre and eight inches at the sides. The DD17 had no Maloney ventilator at all.

p 86 RH para 3 - The only B18¼s with the main mount outside the cab were those fitted with the standard 18¼ boiler from the 1950s.

p 86 RH para 4 - In all cases, there was a cock in the steam pipe to the steam pressure gauge, to allow it to be removed for checking or substitution.

p 87 RH para 1 - The electric staff picked up at Grandchester was small size, attached to a hoop.

P 87 RH para 3 - The pieces to place above axleboxes if springs broke were mild steel (see also p 181 LH para 3).

3.10 Tenders

p 90 LH, para 2, slope of the coal space in tenders - On the B13, there was no slope at all. Beyond the shovelling area, the coal sat on a level space above the tender tank. On the B15Con, the slope of the coal space was 1 in 18, and on the original PB15 1 in 14. On the 1924 PB15, the shovelling plate was lower, and the slope was 1 in 8.3. On the BB18¼, the coal space was level for some eight feet beyond the shovelling plate, then rose at 1 in 1.43 over the remaining four feet of the coal space.

p 90, LH, para 3 - The plates mentioned fitted above the coal space to 231 and 907 were also fitted to B18¼ 908 and C17 804. They were intended to prevent overloading of tenders and ensure that spillage occurred at the coal stage and not on the road. They were removed from 1962 (Sec file 73-432). There were no plates directing coal towards the shovelling plate.

p 91 LH para 3 - The side tanks on the D16 were entirely above the low running board, and were scalloped inwards to be slightly wider below the boiler centre line. On the D17 and DD17, the rear part of the tanks was some seven inches higher than the cab floor, but the forward portion was about a foot higher again, to clear the weigh shaft and give space for the link in the valve gear. The lower portion was scalloped on the inside as on the D16. In all cases, the lower portion of the tanks was recessed to clear the coupled wheels, with a further recess towards the outside to accommodate parts of the drive.

p 91 LH para 4 - The reach rod and sand pipes did not pass between the tanks and boiler. There was a 5 ins diameter horizontal cylinder through the tank on the driver's side of all these tanks engines to accommodate the reach rod, and on the D17, vertical cylinders through the tanks on each side for the pipes from the rear sandbox.

p 91 LH para 5 - The lids on the side tanks of the tank engines were not clamped. Their control on spillage came from their weight.

3.11 Lubrication

p 92 LH para 6 - The lubrication of axleboxes is more fully and accurately described in 3.04, as corrected above.

p 92 RH para 3 - The BB18¼ and DD17 were built with oil lubrication of the items mentioned and later converted to grease. Grease lubrication was applied to most aspects of the drive and motion of many engines in the 1960s

p 94 RH para 4 - On the AC16, the atomisers for the steam pipes were at running board level. Those within the cylinder block were for the cylinders only.

3.12 Braking

p 95 RH - The Westinghouse brake required the fitting of a main reservoir on the engine, sometimes as a single cylindrical reservoir, mostly between the frames, as on the C16 and B17 when built. Generally, however, the main reservoir was split into two cylinders, slung beneath the running boards on each side, or beneath the bunker on each side on the suburban tank engines. There was also an equalising reservoir, mostly on the fireman's side, sometimes in the cab, sometimes on the running board in front of the cab. Somewhere about the engine was another reservoir, the auxiliary reservoir providing the air used to apply the brakes on the engine itself (on the driver's side under the cab on the B17 for example), and the brake cylinder, in which compressed air pushed on a piston to apply the brakes.

Many engines ran until the end of steam with the No. 4 brake valve, with which locomotive brakes were applied along with the train brakes, unless the driver intervened. Unless he did, the locomotive itself was immediately heavily braked. As the train itself was less heavily braked, and its braking was slightly delayed, it tended to run in or bunch up on the locomotive. The driver often intervened to release air from the locomotive and tender brake cylinders to reduce the braking force there, even cut out the brake on engine and tender. As well as providing better control of the train, this reduced heat on the coupled wheels of the locomotive, and the possibility of loosening tyres thereon. There was nevertheless considerable braking power on the engine for use when needed. To prevent the engine surging away from the train on release of the brakes, the brakes on the engine released slowly, unless the driver intervened to make them release quickly, as when shunting or running light engine. Gradual release of the engine brake was practised as engines backed on to trains to couple up ("easing up").

Many engines, but not the B15 types, were fitted with the **independent engine and tender brake** allowing engine and tender to be braked separately from the train**.** Three engines had this brake in 1926, and it was fitted to the B18¼s built in 1929-30. Retrospective fitting continued to selected classes, old C17, and the D17 from 1954. It was direct acting, using air at 45 lbs pressure. It could be applied and released rapidly and to varying degrees, and the necessary pressure was automatically maintained in the brake cylinders notwithstanding leakage. It was also possible to use it to override application of the automatic brake on engine and tender. When the independent brake was fitted, it was used when shunting or running light engine. It was used to hold the train stationary to recharge the automatic brake on long descents, and to hold the train on an up gradient prior to a start when the train brakes had been released. Where it was fitted, only the train was usually braked in running. Engines so fitted had an additional brake valve beside the driver's main automatic brake valve, and an additional gauge to show its operation, a reducing valve to give the desired pressure in the system, and a check valve between the auxiliary reservoir and the brake cylinder (the last on the tender as well).

From November 1939, a more sophisticated form of brake was fitted to various engines, the **A6-ET** version of the Westinghouse brake, first to B18¼ (873 the first), then from March 1945 to new C17 (923 the first), all BGs and BB18¼s (but not the DD17s). The AC16s came with the similar 6-ET. Although ET stood for engine and tender, the brake contained other improvements. Again there were two separate brake valves beside the driver, the larger for the automatic and the smaller for the direct air engine and tender, and two Duplex gauges showing the operation. So far as the independent brakes are concerned, the engine and tender brakes could be applied or released, in whole or in part, slowly or rapidly, with or independently of the train brakes, to any desired pressure, with that pressure maintained regardless of any leakage and of the travel of the pistons in the brake cylinders. In addition, there was provision for rapid charging or release of the train brakes with the governor controlling the compressor altered to have it compress to a pressure of 120 lbs solely during the release (the usual 90 lbs otherwise), this with an air blow to ensure the driver was aware of the location of the control to avoid overcharging. While the train brakes were thus rapidly released, the release on the engine and tender was retarded to avoid the engine surging away from the train. The high pressure governor was also brought into operation with the brake on lap, release or emergency. There was provision for making a minimum reduction of pressure of 7 lbs in the train pipe to set the brakes on the train for a further application, during which the application on engine and tender was retarded. There was also high pressure braking on engine and tender in emergency (unless overridden). The ability to charge or release the brake more rapidly also allowed more frequent applications.

The A6-ET required a distributing valve, relay chamber and body for a relay piston to allow these additional functions. On the B18¼ and C17, these and the associated piping were located on the running board on the fireman's side, on the BB18¼ on the driver's side (in each case on the opposite side to the air compressors).

3.13 Whistles

p 97 LH - The PB15 was introduced with the whistle above the cab roof. When in later construction, the safety valves were moved from above the firebox to the top of the dome, the whistle was placed above the firebox. On the B13, 6D13½ and B13½, it was above the cab roof. In both locations, whistles took their steam from directly beneath, above the firebox.

3.14 Headlights

p 97 RH para 3 - The rear electric headlight on the D17 was made at Ipswich, was of smaller diameter than the standard Pyle, and never had side number panels. The electric headlights on both D17 and DD17 had main beams.

3.15 Liveries

p 98 - The BG livery was decided by the builders after the QR advised them that it would not want a conflict with the green, brown and blue chosen for the classes above (CME file 47/1335/A1). The paint came with the engines, which arrived in grey undercoat, and the top coats were applied at Ipswich.

4.03 Coal

p 114 LH para 4 - Mt Mulligan coal was normally used only on the Tablelands lines, and was sent to Innisfail and Cairns only when Collinsville could not supply those depots. The haul from Collinsville to Innisfail was 319 miles and to Cairns 373.

4.04 Firebox

p 117 RH last para - The AC16 had no combustion chamber.

4.06 Stays

p 121 - With copper fireboxes, the inner or firebox end of the roof or crown stays were threaded, but passed through an unthreaded clearance hole in the firebox roof there. The upper end of rigid stays and the outer wrapper were both threaded, and the stay was screwed into the wrapper there, with some protruding outside. The end of the stay in the firebox had a squared section on the end to allow tightening of the stay, with a cup head immediately on top of that. This cup head was slightly dished underneath by 1/32 inch, to bite into the metal of the firebox roof. Above the roof, in the water space, a nut was tightened on to the roof. This nut had a ridge at its outer edge to bite into the roof of the box as it was tightened. The head of the stay and this nut both biting into the metal on each side of the roof made the hole water and pressure tight. When the stay had been made tight, the end above the outer wrapper was rivetted over, while the squared portion on the end in the firebox was chipped off. This system applied to all the roof stays of copper fireboxes, rigid, slung and flexible. This method of sealing the passage of the stay through the roof of the firebox seems to have been a QR peculiarity.

From early drawings, it would seem that the cup head inside the box was originally a nut also, probably with a similar ridge on its edge to the one in the water space. Such nuts would have burnt away more than cup heads.

When roof stays to copper fireboxes broke, their replacement was a major task. The dome, regulator valve, superheater elements flues and tubes had to be removed to allow access to the boiler through the hole for the dome with the man concerned standing in the boiler shell in front of the firebox. Then a wedge of sound stays had to be removed to give access to the broken stay, then all replaced. All this was necessary to obtain access to roof of the firebox in the water space to tighten the nut on the stay there.

Roof stays for steel fireboxes had a parallel thread at the top to screw through the outer wrapper, and a tapered thread of the same pitch to screw into the inner wrapper, with a squared section on the end to allow the stay to be screwed in and tightened. On completion, the portion of the stay above the outer wrapper was cut to a specified length and rivetted, while the squared portion inside the box was cut off. If these stays broke, they were drilled out, the threads renewed, and a replacement stay fitted as for a new stay.

Wall stays were screwed into both wrappers, even where the outer wrapper was curved. On new B18¼ boilers, the threaded portions of the wall stays were the same 13/16 ins diameter on each side and had twelve threads to the inch. More usually, however, the thread diameter on the end which screwed into the outer wrapper was of smaller diameter than that on the firebox end so they could be screwed from the inside, which allowed renewal of these stays between overhauls. Even if not so designed into new boilers, when replacements occurred, the holes were retapped and replacement stays manufactured to that arrangement. (New holes are needed because after the removal of a stay the thread in the shell is fatigued and brittle.)

Wall stays were drilled hollow for some distance (1 3/16 ins on the B18¼) on both the inside and outside of the sides of the box to provide "tell tales" as described.

p 122 LH para 4 - The AC16 had no longitudinal stays. Instead there were stays from central parts of the smokebox tube plate above the tubes, and from the backplate above the inner firebox at an angle to brackets on the sides of the barrel.

4.09 Ashpans

p 124 RH para 2 - The quenching water, provided by the injector, was hot.

4.10 Dampers and Ashpan Air Space

p 126 LH para 2 - The C17 also had gauze covered slots in the sloping section of the bottom of the ashpan, and these slots came to the C16 when C17 type ashpans were fitted to them.

4.14 Domes and Safety Valves

p 128 LH para 2 - It was the cast iron dome cover (part of the boiler clothing) which was difficult to cast, and led to the adoption of low domes. There was no difficulty with the high cast steel dome itself.

p 128 Fig 4.04 - The smaller of the two pipes provided steam for the blower. On other classes with the steam circulating and snifting system, this pipe provided the steam for that (see part 5.06). On the C17 that system was replaced by air snifters, but the steam supply from the dome and the wheel control for the blower were retained. The pressure gauge always had a separate pipe from the boiler (see p 86 correction above and Fig 3.05 p 61).

4.16 Lagging

p 130, RH, end of para 2 - Cylinder ends were lagged, by an asbestos solution, covered by a light metal plate. That plate was different from dummy cylinder covers, discussed in 3.16, p 99. The plate disguised the irregular shape of the front cylinder covers, which had protrubences to accommodate the fixing of the piston to its rod. Such can be seen in Fig 5.06, p 152.

4.17 Superheating

p 131 RH para 3 - Most superheater headers fitted within the smokebox and were not covered by a flap. They were removed through the front of the smokebox by a gooseneck device fitted to an overhead crane. The exceptions were the headers fitted with multiple regulator valves - see remarks on part 4.23, p 140, below.

p 132 RH last para - See correction above to 3.02 p 52 for the way in which the DD17 header differed.

p 133 RH para 3 - All the South Australian 3ft 6ins gauge T class 4-8-0s were eventually superheated.

p 136 LH para 3 - When the clacks were forward of the dome, the spray pipes went forward.

p 137 caption to Fig 4.10 - The same correction as made to p 136 applies here.

4.18 Injectors

p 134 LH para 3 - The BG were also delivered with vertical injectors below the cab floor, and the ASG had a horizontal injector below the cab floor on the driver's side. Fitting vertical injectors below the cab floor to C17s was a retrofitting.

4.23 Regulators

pp 138-9 - From the late 1940s, new D17 boilers were fitted with the lower height version of the C17 dome and C17 upright regulator valves. Nos 882 to 886 might well have been built with such.

p 139 LH - The regulator valve of the ASG is described twice. It was of the single seat type, and had a pilot valve.

p 140 LH para 3 - The superheater headers fitted with multiple-valve regulators were just outside the smokebox dimensions, and were covered by a flap in the smokebox. When multiple-valve regulators were discontinued, these headers remained in use as headers for some years afterwards without the regulator valves.

4.24 Steam Pipes

p 141 RH - The branch steam pipes were finally steel except on the saturated engines, on which they were copper.

5.01 Valve Gear

pp 142-3 - The link trunnion on the D16 was attached to a bracket similar to that on the locomotives mentioned earlier in the paragraph, itself attached to the motion bracket which held the outer ends of the crosshead guide bars, but the trunnion was high relative to the low running board, requiring the raised section in the running board mentioned (see Fig 5.01).

5.03 Valve Dimensions and Events

p 146 Much is clarified in the section below on Conclusions from the Perform Program. The blast nozzle diameter is of considerable importance for power and efficiency.

p 148 The principal problems with the use of full regulator and short cut offs with the short travel valves (STV) compared with long travel valves (LTV) have been analysed in detail with the Perform program. The results appear in a separate section below. So far as valve travel is concerned, the content of this section can as a result be better summarised as follows. At short cut offs, less steam enters the cylinders than with LTV, the result of the smaller port openings. To obtain the same work as can be obtained on an engine with LTV, longer cut offs have to be used. The pressure at the end of the expansion phase is therefore higher, for the same work, than with LTV. On the exhaust side, the port openings are again smaller than with LTV, in the QR examples given, at any cut off. Notwithstanding the longer exhaust phase and shorter compression phase resulting from the longer cut offs used with the STV, steam has to be pushed out the exhaust system by the power stroke on the other side of the piston, and more steam has to be compressed during the compression phase. The pressure during the compression phase will tend to exceed the pressure in the steam chest. This causes the pressure relief valves to blow on every stroke (as could sometimes be heard, a sure sign of that form of working). As there is a sudden increase in pressure in a fifth or so of the stroke, the elements of the drive are jolted, big end against pin, axle against bearing, and axleboxes against their guides. This results in vibration felt on the engine, and increased tendency for the big end to heat.

p 151 Fig 5.04, caption - The sheath for the piston tail rod is full length; the forward part is difficult to discern in the shadow.

5.04 Drafting

p 154f - The constriction of the nozzle to provide a self cleaning smokebox on the AC16 and DD17 reduced power potential, as did the samll nozzle on the C19. The smaller nozzle on the B18¼ reduced power compared with that on the BB18¼. See the section below on Conclusions from the Perform Program. Compare p 157 where restricted ports were judged more important in restricting power.

5.05 Spark Arresting

p 163 RH para 4 - There were receptacles or sheds for ash blown out of smokeboxes in some places, at Mayne and Toowoomba at least.

5.06 Snifting

p 164f - See the section Conclusions from the Perform Program below. With air, and steam at atmospheric pressure a few pounds above as the snifting medium, a partial vacuum was created during the power stroke, which was filled by a flow of gas with hot char from the smokebox as the valves opened to exhaust. It needed a considerable pressure in the steam chest to avoid that vacuum forming, and no single pressure was satisfactory for all speeds. IN all cases, a consdierable quantity of steam was used. The most favourable situation was to run the engine in mid gear with full regulator when coasting.

p 165 RH para 5. The snifting valves fitted on the valve chest cover above the balancing plate to saturated engines (and on the piston valve AC16) were horizontal spring loaded valves, which opened in the absence of steam pressure and allowed air to enter the steam side of the chest as soon as steam was shut off.

pp 165f - Steam conveying oil from the hydrostatic lubricator was admitted to the valve chest and cylinders while the engine was coasting along with the air or steam admitted by the various types of snifter. The effect on snifting was the same as that of the atomising steam admitted with mechanical lubricators (p 172).

p 166 LH para 3 - The automatic opening of the cylinder cocks when no steam was in the cylinder did not apply to the saturated engines (p 80 also).

p 169 RH para 2f - The valves on the sides of the cylinders at the ends (old C17s at times and D17s to the end) (see Figs 5.15 on p 171 and 10.02 on p 257) were pressure relief valves. There is therefore no Type 1½ snifting. On the D17s, air snifting valves were indeed fitted to the cylinders, but on the cylinder covers. These are not visible in Fig 10.02, where they coincide with the piston rod and tail rod. The air snifting valves can be seen in photos in "Locomotives in the Tropics", pp 72 to 76. It would seem these snifters were fitted from the mid 1940s. They comprised a horizontal cylinder about 3½ ins diameter and two ins deep, joined to the cylinder cover by a pipe. There was a button on the top held up by light spring pressure. As a vacuum formed in the cylinder while the engine was coasting, air was drawn through the valve, performing the same function as other air snifters. Drivers sometimes pressed the button in and inserted cylinder oil there. As these snifters operated while the engines were fitted with Type 3 snifting, thus to the end, this is best described as Type 3½. As the valve had to open and shut with each piston stroke, it can be expected to have been subject to considerable wear, in contrast to the other forms of air snifting, which remained open while there was no steam pressure in the steam chest.

p 171 Fig 5.15 - The reference to snifter valves on the side of the cylinders should be replaced by one to pressure relief valves. See the correction above concerning p 169.

p 171 LH para 2 - When the engine was stationary, admitting steam to the blower to enliven the fire also admitted it to the snifting valve, header, elements, valves and cylinders. The snifting valve cut off the passage of steam when pressure was about one pound above atmospheric, and if the pressure was about one pound higher again, the discharge valve (see below) opened. The circulating steam was intended to keep the elements cool as hot gases were drawn past them. When the engine was coasting, turning on the blower added to the circulating and snifting steam.

p 171 RH para 3 - the discharge valve also protected against build up of pressure from the hydrostatic or mechanical lubricators.

p 172 - the effect of carbon build up is demonstrated by the Perform program. See the section below.

p 173 LH at end of para 2 - The fitting of air snifter valves to D17 cylinder covers (as above) indicates that the Type 3 snifting was not a success, especially with respect to the loadings on the big end while coasting.

p 173 RH para 3 - There was no system of pressure relief in the AC16 valves. There were grooves in the cylinder covers. The intention in the design was that if there was excess pressure, the cylinder covers would break, preventing further damage to the mechanism. This seems a drastic solution in comparison with the QR relief valves. The AC16 were fitted soon after arrival with the same air snifters on the cylinder covers as on the D17s - see the correction above for p 169. At their overhauls and restoration to service in the early 1950s, those snifters on the cylinder covers were removed and the Type 4 snifting discussed on p 173 fitted instead.

p 175 LH para 2 - Pressure relief valves were fitted to the cylinder covers of the BGs at their introduction, and not when the bypass valves were removed.

6.04 Shed Maintenance

p 178 RH para 3 - Drop pits were hydraulically operated. Once wheelsets with axleboxes were lowered, they were slid into a chamber to the side, from where they were raised by chain block.

6.05 Workshops Overhauls

p 182 RH para 4 - The clearance at crosshead guide bars ex overhaul was .004 inch.

8.08 Passenger Train Loads

p 203 RH para 5 - The additional carriage attached to the "Sunshine Express" between Rockhampton and Mackay was not attached behind the brake van.

8.10 Banking

p 207 LH para 4 - Up goods trains from Brisbane reversing into Toowoomba Yard were sent on to the Western Line, within the down outer home for that line, and not the line to the locomotive depot.

9.05 Driving

p 227 LH paras 4 and 5 - There were vertical curves between these changes in gradient (see correction to 1.02 above) but these made no difference to the problems of train handling on the extreme changes of gradient at the places mentioned.

9.07 Full Goods Loads, Work and Power

p 232f The Perform program (see section below) shows that it was impossible for the C19 and BG to have worked their full goods loads at 10 mph. They must have worked these full loads on the sections which controlled the load at about 9 mph.

Section 10 - The Various Classes

The various conclusions drawn on each class following analyses by the Perform program are given in the section on that program below.

Wind Effects on Calculated Outputs

When the horsepowers given in this Part were calculated, no regard was paid to wind conditions. Wind can add considerably to the resistance of a train. This is shown in:

S O Ell: Developments in Locomotive Testing, JILE, 1954 p 561, Appendix III, The resistance of Coaching Stock, and

H I Andrews: The Measurement of Train Resistance, JILE, 1954, p 91

and elsewhere in the literature.

Both contain results for British trains under various wind conditions. In cross section, these trains are much the same as the larger QR locomotives, and carriages. The Ell figures can be modified for the shorter length of most QR carriages, and where necessary for smaller cross-section.

The Bureau of Meteorology retains records of wind speeds and directions at many places during the period covered by many of these efforts, and some of these places (Brisbane, Amberley, Charleville, Longreach, Charters Towers) are close to sites where high outputs were developed.

As it happens, allowing for wind makes very little difference to the calculated outputs. The wind was in some cases zero - that applies to the effort of engine 1015 in the LH column on p 269. In others, it was so slight that the effect cannot be considered material. Corrections were made for some effects not considered in the original calculations, such as effect of carriages with end platforms, of the leading end of the first vehicle after a bogie water truck, and removal of the head end effect from the second of two locomotives.

There were modest effects in two cases. PB15 732 On the "Westlander" north of Mangalore (p 249) was working against a 10 mph side wind at 82½° to the train. That added about 6% to the ihp. C17 982 on the SWL from Greysholme encountered a 7 mph wind at 90° to the train. That added about 2½% to the ihp, which is really within the range of error of the resistance data used in any case.

In some cases, gaps in the data made correction for wind impossible. On the basis of zero wind, the BB18¼ hauling the special train of the Victorian Division of the ARHS on 6th September 1964 developed some 1080 ihp at 53 mph between Balfes Creeek and Southern Cross, just west of Charters Towers just after 3 pm. That was the very hour when winds were usually recorded in Charters Towers. On that day, no record was made, which is a pity, for the next day the speed and direction were such as would have added about 50 ihp for the same speeds. The effort of 1078 with 400 tons north of Kunwarara (p 269) might have been slightly greater if wind conditions at St Lawrence applied there, but the location is too far away and too far inland for it to be possible to claim that St Lawrence conditions applied there.

In many cases, the locations were too far away from any location of wind data, or the conditions were such that wind conditions at a nearby location could not be assumed to apply to location of the performance. The western side of the Little Liverpool Range is not far from Amberley, but the existence of that range would be expected to have changed the wind. The changes in the direction of the track on that climb would have rendered the calculations difficult in any case.

10.02 PB15

p 249 LH 2nd para - The coal space of the 1924 PB15 tender was wider along with the rest of the tender, but was also deeper, with a 1 in 8.3 slope, compared with 1 in 14 on the earlier. The earliest rebuilds of 1899 tenders to have a wider body are thought to have had the same depth of coal space as on the original tender.

10.03 C16

p 251, conversion to C17 - Further to rebuilding the C16s to C17s, considered in the mid 1930s (see Part 4), CME file 34/1564A (QSA item 988755) survived and reveals that the proposal started with a conversation between the CME and the Workshops Superintendent. In October 1933, the Secretary asked the CME to prepare a programme for conversion of twelve C16s to C17s per year. The CME thought the proposal worthwhile and that the conversion would present no difficulty. The conversion would entail new cylinders, a new saddle, about 80% new parts for the motion, raising the footplate (running board) for about half its length, and the superheater and elements, new tubeplates, drilling and filling holes in the frame, and making a small alteration in brake cylinders. The cost was estimated to be £500 per engine. The displaced parts could be used on the remaining C16s, and the C16s already superheated could be done first (they had the necessary supheater, tube plates, flues and elements). The headers, flues snd elements for 20 locomotives were ordered.

At CC April 1934, the Workshops Superintendent said that tests (not yet completed) did not substantiate that the C17 was much more economical than the C16. No explanation was given, then or later, but the WS is noted on the file as doubting the advantages of the conversion. Just after this, C16s and C17s were tested, with the cylinders indicated, on ranges and on easy sections, to calculate the fuel savings. The file contains estimates of the coal savings and load advantages. A handsome economic case was made.

But the conversion did not occur. In April 1934, in a letter to the Superheater Company, the CME said that the conversion was not proceeding at present. In August 1934, the CME advised the Secretary that the Drawing Office was too busy with construction work to carry out the design, although he had no doubt that superheating was best from an economic standpoint. The Secretary (ie the Commissioner) did not ask subsequently whether there were sufficient resources for the design to go ahead.

The design work would not have been great. It would seem that the Commissioner had differing advice from his two mechanical advisers and chose to do nothing, and that the CME lost heart.

10.07 D17

p 257 Fig 10.02 - See corrections above to 5.06 p 169 re pressure relief and snifting valves.

10.11 DD17

p 262 RH para 5 - The running of steam suburban services in Paris with shorter distances between stops was also equalled in Brisbane. Large 2-8-2Ts of 122 tons gross, 88 tons adhesive weight, hauling trains of 250 tons, covered 18.2 miles with 14 intermediate stops (most of 20 seconds) near Paris, reaching 37 mph in 60 seconds on the level (see Vuillet: Railway Reminiscences of Three Continents). The average speed was 23.2 mph, with an average of 1.21 miles between stops. Between Wacol and East Ipswich, 18¼s with 6, 7 or 8 cars ran 10.9 miles with 9 intermediate stops in 29.67 minutes (allowing for a 20 second stop), an average of 1.21 miles between stops at an average of 22.0 mph. Between Northgate and Sandgate, 5.94 miles with four intermediate stations (Bindha then open for only a few trains), all engines hauled trains of six to eight cars in 15.5 minutes, average distance between stops 1.19 miles, average speed 23.0 mph. On the Petit Ceinture circular line in Paris, there was an average of 0.653 miles between stops, and the average speed was 16.9 mph, about the same as Central to Nundah in Brisbane (0.677 miles, 16.2 mph), but the Paris trains were lighter. On the Paris line, starts were helped by a starting valve on the tandem compound 4-6-0Ts (ie starts were made with four cylinders working at a long cut-off (see Railway Magazine, June 1931, p 427).

10.12 Beyer Garratt

p 263f The order was not dealt with by the Secretary's Office with no involvement by the CME.

It is possible to trace from CME files (47/1335A and 47/1335A, QSA ID 988805 and 988806 how the design evolved. A BP Sales Director visited Australia in the immediate postwar years, and met QR officers. Once the initial order had been made, the CME said several times, in various ways, that while the engines were to be provided to broad QR specifications, they were to be designed in detail by BP. The specifications were to be haulage ability, weight and the clearance diagram. Eventually he considered that BP should be advised about minimum curve radii, gauge widening, gradients, height of firedoor and shovelling plate, providing tuckerbox lockers, well ventilated cab, use of Westinghouse brakes, drawgear, buffers, and points of standardisation such as flues and tubes, lubrication, tyre profile and class of coal. The rest was to be for the BP designers. This comment was made to various suggestions and comments from the Divisional Locomotive Engineers.

Nevertheless, BP asked for QR advice about the acceptability of various items, and the QR continued to make further suggestions of its own. BP put forward proposals about drawgear, the speed recorder, the regulator, a snifing valve, and altering cylinder lubrication from grease to oil (on account of unhappy experience with grease on some BGs elsewhere), all of which were accepted. The CME did not want a proposed drifting valve, ferrules to tubes at the firebox end, wanted the tubes to be steel, and the inside of the boiler to be painted with an anti-corrosion compound. At a late stage, the CME decided to require certain extra spares. When asked about the livery, he said that it should not conflict with the green, blue and brown already chosen for various classes. The red scheme was therefore decided by BP, although the paint came with the engines and the painting was done at Ipswich after erection.

BP determined on the design by June 1947, the wheel arrangement, maximum weight and tal, the cylinder dimensions, coupled wheel diameter, boiler pressure, grate area, water and coal capacities. These arrangements were to suit the bridge (and presumably track) and the 5 chains curves limitations,. And a speed of 45 mph was mentioned as a design desideratum. The firebox was to be welded. Only the idler axles were to be fitted with roller bearings on account of the weight restrictions. The number of flues and elements was 32, but the number of tubes was given as 151, which number was reduced to 143 during construction. Reversing was to be steam powered, while lubrication of cylinders and ball joints was to be mechanical, while coupled wheel axle lubrication was to be grease (see above for alteration to oil). On accountof the design input, the minimum order would be for ten, but complete drawings would be provided so that more could be built in Australia. The main mount was to be located outside the cab. The cold water for crew use was to come from the levelling pipe between the front and rear tanks, under the cab.

The CME would have preferred a higher coal capacity, but had "no hesitation in recommending" the design and price. If the air conditioned train on the NCL was to run six days per week, eight would be needed, so ten was a reasonable purchase. He proposed changing en route at Gladstone, Mackay and Townsville, coaling en route at Bundaberg, Rockhampton and Bowen. Further alterations were made. The steam sanding was later altered to air, both being BP proposals. BP recommended against overhead springs. To maximise coal capacity, the back plate of the bunker was to be made moveable on the first engine completed. The firedoor was only 19 inches above the working floor of the cab (the BP letter mentions 22 inches), but the arch tubes made a greater height impossible. In December 1947, the QR agreed to 10.15 tal on the outer bogies and an all up 137 tons, but these were not to be exceeded; if coal capacity was to be seven tons, that must be accommodated in those weights.

The QR had an inspector in the UK at the time, inspecting wagons and other locomotives, J A Baxter. He was to report anything unsatisfactory, draw the attention of BP to the items, and witness the test run and final weighing. The CME visited BP during the construction, but not to inspect. Baxter observed the testing of much material. He found the cylinder covers were too close to the cylinders. All were removed and scrapped and a modification made. He observed the test, on 100 yards of 3 ft 6 ins gauge track, which included a five chains radius curve. He also observed the weighing, with boiler and tanks full and six tons of pig iron in the bunker. This gave 136.74 tons. The bogies were both of the 20.3 tons given on the diagram, all the other axles very slightly lower than the diagram weights. The Beyer, Peacock General Dimensions Book 1A (NW Museum of Science and Industry, Manchester) shows the weights which appear on the drawings. (See below on overweight).

From Baxter’s reports, it is known that BP in Manchester supplied many items to Franco Belge, who constructed the final twenty engines. These included cylinders, reversers, piston rods, boiler back plates, boiler flangings, throat plates, but there were more.

Some erecting staff from Ipswich went to Pinkenba wharf, where the boiler units were placed on the power units. In that form, the engines were towed to Ipswich for completion of the assembly and for painting. It was instructed that five of the first ten were to go to Rockhampton to work to Emerald, and five to Bundaberg for working to Rockhampton. They were to be given good coal, and spend the maximum time in traffic.

p 267 end of LH column - In months of 1955 and 1956 for which the locomotive statistics are available, the casualty rate for BGs was five to six times that of all other QR steam locomotives, and the failure rate from 1.2 to 2.9 times. The other steam locomotives were on average over 30 years old.

10.13 BB18¼

p 269, Table - the distance at Grandchester should read 43ml 45ch.

11.05 General Improvements

p 285f See the Conclusions from the Perform Program, below, especially about the effect of higher superheat temperatures and the size of blast nozzles.

Appendix 1 Largest Steam Locomotives on Selected Medium Gauge Railways

p 295 Fig A1.06 - The engine is a class 23 4-8-2 of 1938. The Class 25NC 4-8-4 of 1953 was slightly heavier, locomotive only, but the weight comparisons hold.

Appendix 5 Workshops, Depots and Engine Workings

p 313 RH para 1 - "goods and goods" north of Townsville should read "passenger and goods".

Appendix 8 Calculation of Locomotive Output

p 323 Table A8.01 - Under Equivalent Gradient, the figures in the RH column should be applied to the other line in each case, ie 1592 for locomotive and 6654 for tender and train.

APPLICATION OF THE PERFORM PROGRAM TO QR LOCOMOTIVES

The program Perform2, developed by the late Professor W B Hall, has allowed a much fuller analysis of the operation of the valves and cylinders of QR steam locomotives than was possible when the book was written. The basis of the program is given in articles in the *Stephenson Locomotive Society Journal* for May and June, and July and August 1999 pp 88 and 134, and May and June 2003 p 100. There are further articles on the effect of various aspects of locomotive design using the program in articles by David Pawson in that Journal in 2003 (July/August p 156, and September/October p 200).

The program is based on the action of the valves and the behaviour of steam as a gas through the valves, cylinders and the blast nozzle. Essential inputs are valve gear and valve dimensions, initial steam pressure and temperature, cylinder temperature, speed, cut off, and coefficients for the behaviour of steam as it passes through constrictions, expands, condenses on the cylinder walls (if that happens) and re-evaporates, is compressed and is discharged through the blast nozzle.

The documentation of Perform2 in Prof Hall's article in the SLS Journal May and June 2003 shows how condensation affects the passage of steam through the cylinder. For highly superheated steam, the process is fairly simple, and the assumption of a single cylinder temperature is adequate. For saturated steam, however, the assumption of a single cylinder temperature is a considerable simplification, as Prof Hall stated. At the steam chest, the steam might be say 190 ^{0}C. As the steam meets the cooler metal of the ports and passages, and of the cylinder, it is immediately cooled. This metal is cooler than the inlet steam because it last encountered the cooler exhaust steam from the previous stroke. The temperature of the exhaust steam on some C16 tests was below 100^{0}C, and it can be shown that the heat drop required to develop the horsepowers recorded in most tests will have brought the steam at the end of the power stroke to close to 100ºC, if not below it.

Some of the steam condenses to a thin film of water on the cooler metal. That film insulates the remaining steam from the cool metal, a film which can re-evaporate as temperature and pressure fall. It is nevertheless possible for a higher degree of condensation to occur, and water to form, to pass from the cylinder in the exhaust. Perform2 models the condensation of the first steam to enter, so that the pressure at the beginning of the stroke is often a good 10 lbs below chest. Typically, there is a value of the cylinder temperature, in the 130 - 145 ^{0}C range, at which the specific steam consumption (lbs of steam per ihp/hour, SSC) is a maximum and often over 30. As this temperature is encountered, the higher degree of condensation will occur. This temperature is always in the range between inlet and exhaust temperatures, ie in expanding and giving up heat, the steam will always pass through this temperature. Whether the thin film of water will suffice to prevent the higher degree of condensation is a matter of particular conditions, however, influenced by cut-off and speed. It certainly seems that in some cases the higher degree of condensation allows the higher SSCs in the low 30s to be avoided, and for SSCs to be kept to the 27 - 28 lbs range.

Steam which is modestly superheated avoids some of the condensation, but can still encounter the film of water.

The program gives indicated tractive effort and resulting steam consumption for the inputs mentioned. The consumption is for cylinder steam fully utilised. Leaks and losses are excluded, as is the use of steam by auxiliaries (one auxiliary, the injector, does not add to water consumption, because the steam it takes is returned to the boiler as water). Thus steam used by the lubricator, the air compressor for brakes and lifting water in carriages, and for lighting on the locomotive, steam lost at safety valves and through the cylinder cocks, water and steam lost at the injector overflow, and water used by the ashpan quencher, for laying coal dust, and for washing, are all excluded.

The program has been used in four ways. First, using the principles of physics and mechanics and the mathematical processes built into the program, to check how sensibly it operates. This includes checking whether the calculated efforts given in the book require credible regulator and cut off settings and steam consumption; this has been varied to the check of whether the driving practice observed, in cut-off and regulator settings, gives credible outputs. Second, to compare the results with the small amount of QR steam locomotive test data, indicator diagrams and water consumption (these are outlined in the book). Third, to consider aspects of the design of QR locomotives, how various characteristics affected outputs, and the effect of various changes in those characteristics. Fourth, to examine certain miscellaneous questions such as the effects of snifting, coasting and carbonised ports.

These exercises have provided a great deal of interesting information and many insights, but some mysteries about QR steam locomotive performance remain. The major insight is that the valves of the C17, D17, B18¼ and C19, once enlarged to 9½ ins diameter, were not as limiting as I thought. The program confirms that the AC16 was the most constricted engine to run on the QR.

Back pressure predicted by the program is generally considerably less than measured in tests, however (in one B17 test, 14.5 lbs declared by the indicator, 6.4 lbs from Perform). This has to be a problem of the zeroing of the indicator. The ihps declared for the B17 on the Toowoomba Range and those given by Perform are in close agreement. The same is true for the B18¼. The resistances given by gradient and the constant sharp curvature on the Toowoomba Range represent some 75% of the calculated ite. If the ite were to be increased to allow for the difference between the back pressures given by Perform and those given by the indicator being entirely at the exhaust line, ites would have to be 10 per cent higher. If they were in fact 10% higher, then the vehicle and locomotive resistances used in the calculations would have to be 43% higher. It is to be concluded that the indicator used from at least 1913 to 1935 over-registered pressures in absolute terms, but gave a satisfactory area, ie the top and bottom readings were too high by a similar amount. (see also below about the ability of the indicators to record initial inlet pressures).

The cut offs and inlet pressures needed to obtain calculated ihps according to Perform are largely what would be expected from observation. The vibration of the B18¼s in the 40s mph is explained by compression loops at short cut-offs, the result of some or all of inadequate port sizes, inadequate clearance volume and small blast nozzle.. The matter not explained is why higher outputs were not observed from those engines and the C17s.

The valve events built into Perform give points of release and compression consistent with those shown on QR diagrams of valve events.

QR engines did not have steam chest pressure gauges. From the mep of each diagram, the height of the inlet above the atmospheric line was obtained, thus the inlet pressure (gauge). Except at low rpms and long cut-offs, these read off inlet pressures give a lower mep than given by Perform for the cut-off and speed concerned, ie the true inlet pressure was higher than given by the indicator. The reason has to be the inertia in the indicator in the short time available. This gap could have been researched and allowances made in the indicator readings, but this was not usually done. On the other hand, the bottom left hand corner of the diagrams given by the indicator are less reduced from a right angle than those given by Perform, again on account of inertia in the indicator, which could similarly have been allowed for, but was not. It would seem, however, from the broad agreement between meps given by the indicator diagram and by Perform that these two sources of error largely compensated each other.

For the saturated engines working at full boiler pressure, inlet steam temperature (Ts) was obtained from steam tables.

B13, saturated slide valve

Nozzle tests were conducted on the Toowoomba Range in 1903 with the aim of increasing the loads of these engines (CME file 03/3399). The load, time from Murphys Creek to Harlaxton (between 87 and 103 minutes), nozzle diameter and water consumption were recorded. No indicating was done, but I have calculated ihp. Perform allows for potentially high condensation losses from the small (13 by 20 ins) cylinders, and has it that the climbs could be made at an average 41 to 51% cut-off, depending on the load, at a SSC of 28 to 31, in all cases taking the cylinder temperature which maximises SSC (135 to 139^{0}C). Using the same calculated ihps and the water consumption given on the files, the SSC based on the water consumption is in all cases very close to the Perform figure, 4, 6, 7 and 10% below.

Moving the full goods load of these engines on any gradient at 10 mph required a calculated ihp of 260. According to Perform, this required a 71% cut off and an SSC of 35 lbs. This is higher than the cut-off observed used on other saturated engines moving their full goods loads at 10 mph in the 1950s and 1960s. The full loads were probably moved at 9 mph instead, at which the cut off had to be 67%, that observed used on saturated locomotives on the QR in the 1950s moving a full load.

B15 Converted, saturated slide valve

Moving the full goods load on any gradient at 10 mph required a calculated ihp of 350. According to Perform, this required a 67% cut off and an SSC of 27.6 lbs if the cylinder temperature was 140^{0}C. The cut-off is that observed on these and other saturated engines moving their full goods loads at 10 mph in the 1950s and 1960s. The SSC was maximised in Perform at 33.7 at a cylinder temperature (Tc) of 150^{0}C at which the cut-off needed to be 71%, longer than observed. As with the PB15 (with the same 15 by 20 ins cylinders), the actual SSC seems to have been lower than the maximum. This means that the actual Tc was lower.

PB15, saturated slide valve

The lead of the original locomotives of this class with Stephenson valve gear is not known. [While the lead at full cut off of the B15Cons is known to have been 3/32 ins, the layout of the valve gear on those engines is considerably different from that on the Stephenson PB15s. The latter had much longer blades (the connections from the eccentrics to the link) than on the B15Con.] Lead is variable with the cut off with Stephenson valve gear, and increases as cut off is shortened. The lead on the Walschaerts engines, 3/32 inch, is constant at all cut offs, and was probably chosen to be the same as on the original engines at the working cut off of about 30 to 40 %. In all the simulations of the running of the original engines using Perform, the same 3/32 inch lead was used, but the effect of different leads tested. If the lead was shorter, 5/64 inch, achieving a certain ihp requires a slightly longer cut off and higher SSC. With longer leads, of 1/8 and 5/32 inch, the engines require a slightly shorter cut off and have slightly better SSCs. Even if it is slightly wrong, the use of 3/32 inch as the lead in all cases does not have any important effect on the conclusions below.

Trials were conducted on both passenger and goods trains on the Toowoomba Range in 1902 with different nozzles and chimneys (CME file 01/3391). The same data is available as for the B13s above, but on the passenger train the time was not given; it was assumed the trains concerned ran to time - had it been lost, that would surely have been remarked on. On the passenger train, an SSC of 27 is given by Perform at the cylinder temperature at which SSC is maximised, where my calculated ihp and the water consumption give 26.2. On one goods train, these figure are 28 and 33, and on the other 27.4 and 27.5. The 28/33 case is the only one for saturated engines for which there is not close agreement between SSC from Perform and the actual. (My rolling resistances cannot be the explanation, given the gradients and curves; my vehicle and locomotive resistances would have to be reduced over 60% to bring about agreement).

I have records of numerous runs from these engines, recorded in the 1950s and 1960s. On the "Westlander", No 732 gave a calculated ihp of 350 at 29 mph, working with 150 lbs pressure and about half regulator. According to Perform, if this required about 128 lbs in the chest (it was not making steam especially well), a cut-off of about 37% was required, which is about that observed. By Perform, SSC was about 26.6

lbs.

On a dash at the end of an ARHS tour in 1964, No 488 averaged 39 mph from before Fairfield to Park Road, 460 calculated ihp. With full regulator, that would have required 36% cut-off, this being close to one of the cut-off settings given by the nicks in the quadrant, an SSC of 25.4.

On an inexplicably vigorous climb from Cannon Hill to Morningside on a down Manly suburban, a calculated 450 ihp was developed at 25 mph, requiring 41.5% cut-off with full regulator, SSC 24.4.

The best climbs by Southport Expresses from near Salisbury to the summit at Altandi on the old SCL, on which the gradient gradually steepened, seem to have been accomplished by a mix of these two efforts, starting at about 35% cut-off, and advancing cut-off to about 40 near Sunnybank, with full regulator throughout, giving ihps in the same range.

Moving a full goods load at 10 mph required 330 ihp. According to Perform, if the cylinder temperature was 150^0C, at which SSC was maximised at 35, a 72% cut-off was required. At 140 - 142^0C, the observed cut off of about 67 could be used, at which SSC was about 28. This is another case where the cylinder temperature which maximised SSC seems to have been avoided in practice.

C16, saturated slide valve

C16 655 was indicated on two climbs of the Toowoomba Range on goods trains in 1934 with the original 4.125 ins nozzle. The data sheet accompanying the diagrams claims chest temperature of 177 to 183^0C, cylinder temperature 173 to 176, and exhaust temperature 87 to 92 (below boiling point) at the various locations where the recordings were made. There is no information on how or where these temperatures were measured. The inlet pressure obtained from the indicator diagrams fed to Perform gives meps the same as or a little less than shown in the QR diagrams. The QR diagrams and Perform diagrams are similar in shape, although Perform generally has a faster decline from inlet than on the QR diagrams, even while the valve was still open and the speeds were low, while the bottom left hand triangle from compression given by Perform is generally larger than on the QR diagrams. The QR diagrams have higher back pressures than Perform, reconcilable in the sense mentioned above.

In one location, at which speed was only 7 mph, the cut off given with the QR diagrams of 50% is obtainable at the ihp given only at 130^0C cylinder temperature, at which the SSC is 24. At this speed and for the ihp recorded, the maximum SSC is obtained at 146 ^{0}C cylinder temperature. which requires a longer cut-off than that observed, of 54%, at which the SSC is 30. The point of maximum SSC seems to have been avoided in this case.

The pattern engine with various modifications designed to improve steaming on low quality coal (No 177) was indicated in 1945. As a data set, these diagrams are unreliable - after removing the gradient effect, the ites at various speed/cut off combinations vary widely.

A C16 moving a full goods load at 10 mph required 467 ihp. According to Perform, with the 3¾ ins nozzle used on the modified engines postwar, this required 67% cut-off, which gave an SSC of about 27. Both figures are stable over a wide range of cylinder temperature, from 90 to 140^0C. This was the only instance of variation in cylinder temperature over such a range having no impact on the cut-off needed, nor the SSC.

B17, saturated slide valve

These engines were extensively tested on the Toowoomba Range in 1913 to optimise the nozzle size and type. Indicator diagrams were taken. These have not survived, but their ihp values are reported in the report in places (CME file 1913/6937). They seem to have been taken only at one mile intervals, at full mileposts, not especially satisfactory on an incline with such variation in gradient and curvature. The times from Murphys Creek to Harlaxton, the water consumption, coal consumption, and load are also available. I have calculated ihp. The question of back pressures is considered above.

Very close correspondence in SSCs again occurs. The SSCs by Perform are 3% and 9% below those of the calculated ihps and recorded water consumption, and that might be explained by a little consumed from the top of the Range down to Toowoomba and the shed there, and by consumption from the air compressor.

Moving the full goods load required 525 ihp at 10 mph. According to Perform, this required a 67% cut-off and an SSC at the cylinder temperature which maximised SSC (145) of of 26.5. The cut off is that observed.

Superheated Engines

From British Railways test data, it is clear that Ts varies with the intensity of the fire (heat per sq ft of grate per hour), and the proportion of the combustion gases passing the elements. The BR test data refer to locomotives brought to a certain fire condition and operated continuously at that state. On the road on the QR, QR test data shows that Ts takes time to build up.

Some Ts data are available for C17s and B18¼s. The C17 Ts data agree with test data for other locomotives with a similar percentage of the FGA passing the elements (PFGA), and are consistent with ihps observed in tests and calculated from Perform. The B18¼ data on Ts are not credible. During the climb from Lockyer to Murphys Ck (800 ihp, 30 mph), the Ts was only 60ºF above saturation, and reached an average maximum of only 120ºF above saturation at 670 ihp (both measured and calculated) at 19 mph on the Toowoomba Range. Using such temperatures in analysis of day-to-day working results in the B18¼ using a lot more coal and water hauling the same load at the same speed than a C17 (the B18¼ ihp is slightly higher in such a case because it is heavier). Furthermore the B18¼ degree of superheat is much lower than on the BR Standard 4 4-6-0 which has the same 46% PFGA as the B18¼ when working at the same rate of heat production in the firebox, after correcting for the small difference in grate areas (the higher boiler pressure of the 4 is responsible for both higher temperature (12ºC) and higher heat content (5 more BTU per lb) before the steam passes to the superheater. In addition, the experience of enginemen was that although there was little difference in coal consumption between the C17 and the B18¼ when hauling the same load at the same speed, the B18¼ must have used a little less coal because the C17 was slightly the more likely to run out of coal. (The B18¼ loses compared with a C17 by having longer tubes (which reduces the intensity of a given draft on the fire), a lower PFGA and being heavier, but gains from the higher boiler efficiency on account of the considerably larger grate, and the greater scope for expansive working on account of the considerably larger cylinders). By trial and error, it was found that when a C17 was producing 500 ihp at 20 mph with 320ºC Ts, a B18¼ could develop the equivalent (same load, same speed) 506 ihp with 312ºC Ts, and with a coal consumption about 98% that of the C17.

The maximum Ts to use in simulating performance of the superheated engines using Perform depends on data from elsewhere. For the 18¼s the maximum obtained by the BR 4 4-6-0 of 655ºF, 346ºC, has to be reduced by 12ºC to allow for the higher pressure pf the BR 4 before the steam passed the elements.. For the C17 up to 715ºF, say 380ºC, is justified, based on the average results for the LM Duchess and the BR 9F. These maxima can be considered to apply to outputs of 50ihp per sq ft of grate per hour. For the C17 the Ts therefore increases from 320ºC at 27 ihp per sq ft of grate per hour (500 ihp) to 380ºC at 50 (925 ihp), not that I have known the latter figure to have been developed. For the B18¼, the Ts increases from 312ºC at 20 ihp per SFGH (506 ihp) to 334ºC at 50 (1250 ihp)(again higher than I knew from those engines).

Strictly, allowance should be made for the different blast nozzles on the two types of 18¼, in that the smaller nozzle on the B18¼ should lead to a more intense fire and higher Ts than the larger nozzle of the BB18¼, but there is no realistic way that can be done, especially when the valve events of the BB18¼ were such that the exhaust was sharper, something which crews considered led to a more responsive fire. The Ts of the BB18¼ was never recorded. At any given ihp, it should have been lower than on the B18¼ for the following reasons. Producing a given ihp required less steam on the BB18¼, on account of the better valves and larger blast nozzle, hence less draft and a less intense, cooler fire. The larger blast nozzle reduced the draft at any ihp, a further reason for a cooler fire. At a later stage, it might be possible to reduce the B18¼ ihp figures to steam rates, to give a relationship between steam rate (Q) and Ts. In the meantime, the Ts values used in all BB18¼ calculations are the same as those used for the B18¼.

For other classes, the maximum and minimum Ts can be proportioned in relation to PFGA, boiler pressure and ihp/SFGH. In summary the Ts figures used for the superheated engines were:

C17 320ºC at 500 ihp, 380ºC at 925 ihp

18¼s 312ºC at 506 ihp to 334ºC at 1250 ihp

C19 340ºC at 500 ihp to 400ºC at 1000 ihp

DD17 316ºC at 500 ihp to 360ºC at 1000 ihp

BG 295ºC at 500 ihp to 372ºC at 1600 ihp

AC16 308ºC at 500 ihp to 363ºC at 1000 ihp.

In view of the time taken for Ts to build up after a high rate of output commences, in some cases, the Ts has been reduced when high outputs came suddenly, after only moderate rates of working. This has been done judgementally.

David Pawson found that BR test results could be obtained in Perform2 if the following cylinder temperature (Tc) figures were used in conjunction with the steam temperatures (Ts) given: for Ts Tc of 140^{0}C for Ts of 220 and 220^{0}C, then the following combinations 144/250, 163/300, 183/350, 189/400 and 184/420.

The tests of the C17 in 1934 show that at least for the range of Ts between 290 and 340 ^{0}C, the Tc given in the table and interpolated values (160 to 179 ^{0}C) are exactly what is needed to reconcile all the data, ie for the speed, cut off, Ts, the Tc above gives the ihp recorded by the QR. (Chest pressure is in some cases obvious, in that the regulator setting is clearly full. Where the regulator was not fully open, the chest pressure needed to bring about reconciliation is reasonable.)

C17

Considerable data is available for No 785, tested hauling goods trains on the Toowoomba Range in 1934. It then had 8 ins diameter valves. Interpreting the test data is not straightforward. The ihp was clearly registered by the indicator at a point, a half milepost. Is the speed that at the point or timed over the previous half mile? It is not possible to say whether the train was accelerating or decelerating at the point. The number of vehicles, hence axle load and resistance and length (important for knowing whether the train was on a curve at the time, and whether the vehicles concerned were four wheelers or bogie) are not known. The C17 data was therefore taken over two mile ranges. If the speed is point recorded, the average speed for the section cannot be perfectly known. Compared with calculated ihps, the recorded ihps seem a little on the high side, although not improbable.

Perform reproduces the ihps given by the QR indicator, at least so long as the chest pressures from partly opened regulators have been satisfactorily estimated. After the stop at Murphys Ck, Ts reached 320ºC after about nine minutes. The SSCs on the climb were highly satisfactory for the low speeds, as low as 18 lbs per ihp per hour, the result of the high steam temperature.

Hauling its full goods load, producing 500 ihp at 10 mph, starting with zero superheat, SSC was 29 lbs. With superheat progressively added until the Ts was the maximum for 500 ihp, 320ºC, SSC progressively falls to 21.7 lbs, in all cases working at 60% cut off.

The outputs noted on passenger trains on engines with 9.5 inches valves, 675 ihp, at 27 mph or so, can be developed at with 29% cut off with 345ºC Ts using only 9570 lbs of steam (SSC 14.2) At 40 mph, the same 675 ihp can be developed with the same Ts and 24% cutoff, using even less steam, 9210 lbs an SSC of 13.6. With the original 8 ins valves, slightly longer cut offs were needed and steam consumption was a little higher. The 8 ins valves were not the hindrance to the operation of these engines in the way they were to the B18¼ and C19, because less steam had to pass them, on account of their smaller cylinders. Other possible aspects of redesign made little difference to these engines, even increasing the lap and pressure. A larger nozzle, if draft could be maintained, would confer steam economy.

Apart from starting its full goods load in saturated form, the C17 was able to develop almost all these horsepowers at steam rates in the 9000 to 10,500 lbs range, the latter only 568 lbs of steam per sq ft of grate per hour, not requiring a particularly intense fire.

BB18¼

The best effort I recorded with these engines, calculated 1175 ihp at 51 mph, with 332ºC Ts and 177ºC Tc would require full regulator and a cut-off of 26% according to Perform, which is about that used. With these inputs, Q was 18,100 lbs SSC was 15.3, and the blast nozzle pressure 6.5.

If the pressure were higher, 200 lbs compared with 170, developing the same 1175 ihp at 51 mph, but with the same temperature and 18¼ inches cylinders, the cut-off would be 27% and the SSC some 3% lower. With this pressure, the engines would not be able to start and move their loads at low speeds on account of poor adhesion. With cylinder diameters reduced in proportion, the cut off would be 29% SSC would be 14.9. A proportion of the heat of the fire will go to producing the higher pressure.

If the stroke were longer than the 24 ins employed, 26 or 28 ins, with the usual boiler pressure, but cylinder diameter reduced to compensate, steam consumption would be increased a little.

Although 50 mph was the speed limit for these engines, and about 60 about the highest drivers dared, higher speeds were looked at, not for practicable running, but to see how good the valve events were. At higher speeds than 51 mph, using the same 26% cut-off, the ihp rises a little to 1230 at 60, and 1260 at 70 mph. but falls gradually to 1030 at 100 mph, with the SSC increasing by 25% to 25.6 at 100 mph. At 80 mph and above considerable compression loops develop, even with reduced cut offs. These increase back pressure and steam consumption, and through uneven effects on the mechanism, would cause bearings to run hot. Up to 90 mph, these can be eliminated by increasing the clearance volume above the designed 10% to 12% (Increased lap and port opening have no effect, in view of the small cut offs employed.) Such an increase in clearance volume would increase the steam consumption at the lower speeds at which the engines normally worked. 75 mph or so can be regarded as the desirable top speed for the mechanism of these engines. Even at these higher speeds steam consumption is still well within the steaming capacity of the boiler.

If the coupled wheels were larger, producing the same 1175 ihp at 51 mph, with the same Ts and Tc, and boiler pressure increased to compensate, to 180 lbs with 54 ins CWs, 190 with 57 ins and 200 lbs with 60 ins, very small savings in steam consumption would result. Compared with the 51 ins CWs on the engines as built, savings could be made of 0.7, 1.3 and 2% respectively. These result mostly from the slightly shorter cut-offs possible.

With the 4.8 ins diameter blast nozzle, there is 6.5 lbs back pressure at the blast nozzle when developing 1175 ihp at 51 mph and SSC is 15.3. With a 5.2 ins nozzle that pressure would be 4.2 lbs and SSC 14.7, one of 5½ ins diameter would have reduced nozzle pressure to 3.2 and SSC to 14.5, one of 6 ins to 2.1 lbs and 14.3, and one of 7 ins to 1.1 lbs and SSC to 14.1 lbs. One of 20 ins diameter (actually larger than the chimney) would have reduced nozzle pressure to a figure imperceptible from atmospheric and SSC to 13.3. These larger diameter nozzles would have required some more sophisticated drafting system then the plain nozzle and bell to preserve the draft and steam production. They are given solely to illustrate the effect of back pressure from the blast nozzle on SSC.

The biggest improvement to the efficiency of these engines (and steam engines generally) would have resulted from an increase in the Ts. Developing the base case 1175 ihp at 51 mph, with 332ºTs, the SSC was 15.3 lbs. At 350/183 it is 9% better, at 375/189, 14% better, and at 400/189 18% better, at which SSC is 12.6. Cut off is only slightly reduced in achieving these better SSCs, over this range, from 32% to 31%; the main reason for the reduced SSC is the greater energy in each pound of steam. This economy can also be seen in terms of coal fired. As the fire is made more intense by higher rates of firing, evaporation increases at about 0.85 times the rate of increase in the firing rate (ie double the firing rate, about 1.7 times the amount of steam will be evaporated) because smalls in the coal are carried away unburned by the increasing draft, but the heat in the steam above saturation temperature (ie the superheat) increases at about 1.15 times the rate of the firing, notwithstanding the coal carried away unburned (ie double the firing rate and the superheat increases 2.3 times). (These comparative rates, strictly elasticities, are from postwar BR tests). The superheat makes up for the loss in evaporation even before the economy in steam use. The improved efficiency would have reduced the requirement for evaporation, and the loss from carrying away fuel unburned. On the other hand, it reduces the need for an intense fire and the gains from the higher temperature resulting from that. The higher Ts therefore has to be obtained by a higher proportion of the FGA being devoted to superheating.

Higher pressure contributes very little to the output of the engine, and requires a stronger and therefore heavier boiler. Within given weight restrictions, a higher pressure boiler therefore has to be smaller, of smaller evaporative capacity. Any gains from higher pressure are also subject to the gain being obtained at only about 0.85 times the increase in firing rate given above.

Moving the full goods load at 10 mph, 540 ihp, required just under 60% cut off if the steam was saturated, ie immediately after start and used 14,100 lbs of steam per hour. When Ts reached 230ºC steam consumption was reduced to 13,000 lbs at about the same cut off, and when 313ºC, the maximum for this ihp, to 11,700 lbs at 57½% cut off. But the tractive effort, ihp and steam rate required to move the full goods load at up to 14 mph is well within those required at higher speeds on passenger trains, circa 765 ihp and 16,200 lbs. As suggested in the book, the 18¼s should have been able to move their full loads faster than most engines could move theirs.

A driver once advised that he took 170 to 180 tons passenger trains from Cambooya to Wyreema with these engines when running to time, with speed 25 mph at the top of the climb, so limited by 10 chs radius curves, by using about half a lever with half regulator. I calculated that such would have required about 690 ihp. Sure enough, in Perform, with 317ºC Ts and associated 170ºC Tc, if that regulator provided 105 lbs in the chest, 50% cut off would provide that power. Or, 122 lbs in the chest would do so with 40% cut-off. Steam consumption would have been lower with full regulator and a shorter cut off.

B18¼

These earlier Pacifics had two essential differences from the later - a shorter lap and lead, and a smaller blast nozzle. Originally they also had smaller diameter valves (8 ins) and 160 lbs pressure, but by the time I knew them, they all had 9½ ins and 170 lbs. I was always given to believe they suffered from high back pressure if full regulator and modest cut-offs were used, and that was demonstrated to me by vibration at 45 mph when so driven. There were nevertheless men who drove them with full regulator and whatever cut-off was needed to do the job, including one of the special mail train drivers working on the NCL between Bundaberg and Rockhampton.

One of the first production versions of these engines was indicated on the Toowoomba Range and its foothills in 1931, when the engines had 8 ins valves and 160 lbs pressure. The indicator diagrams and a chart comparing the runs, including steam temperatures, survive. No water consumption is given.

Chest pressure has to be inferred from the figures given for the regulator opening or measured from the diagrams. Perform reproduces the average ihp of the four test trips over the section where Ts had reached its maximum, assuming an average chest pressure of 134 lbs (see above).

No 843, one of the 1936 version built with 9½ ins valves and 170 lbs pressure, was initially fitted with an ACFI feedwater heater. This was tested by requiring the same task on the Sydney Mail between Brisbane and Toowoomba with the same coal and crew, and comparing coal and water consumption with the FWH feeding the boiler on one day and the live steam injector on the other. The FWH saved both coal and water, but less than expected. Using calculated ihp, the water consumption with the live steam injector water yields an SSC of 23.9 lbs from Helidon to Toowoomba (the basis of most of my reckoning in the book for steam consumption of QR superheated engines at slow speeds/longer cut-offs).

The load and timetable were the same as on the tests of No 16 in 1931, so the cut offs recorded for No 16 can be used for No. 843 with the live steam injector. This results in an average cut off of 45%. For the calculated ihp of 560, an average Ts of 314^0C, an average Tc of 168^0C, this required an average 119 lbs steam chest pressure, and a cylinder steam consumption of 10,000 lbs per hour. That compares with an actual rate of 13383 lbs per hour, 34% more. For both these figures to be correct, 25% of the water used went to operating the brakes and the water supply in the carriages, or was wasted at the safety valves or injector overflow, which seems to be an unrealistically high rate of loss. From Ipswich to Helidon, the rate of loss on the same basis was even greater.

The ihp, steam consumption and conclusions on moving the full goods load by the 1930s engines correspond to those of the BB18¼.

On one day, two different drivers showed me their ways of doing things on the same engine. One worked at full regulator with a short cut off, and produced 50 mph where I calculated about 690 ihp. With a Ts of 318ºC, Perform reproduces this well - 167 lbs in the chest, 23½% cut-off, 692 ihp, 2.7 lbs back pressure and a steam consumption of 10,600 lbs The reason for the low hp and modest speed is that so little steam was getting into the cylinders. With so little steam, the back pressure was very low. There was no vibration.

The other drove with part regulator and adjusted the cut-off to produce 60 mph, some 880 ihp. With 150 lbs in the chest, a cut off of 30% was required, steam consumption was 13,950 lbs, SSC 15.9, and the back pressure 4.9 lbs. At 140 lbs chest, these figures were 34%, 14,200, 16.1 and 5.1 lbs. There was no vibration; Perform shows that only at 130 lbs and below in the steam chest were compression loops present, which would have been felt as vibration on the engine, and avoided. The best way to have worked the engine would have been with full regulator, 167 lbs in the chest, 28.2% cut off, using 13,600 lbs of steam per hour (SSC of 15.6) giving 4.7 lbs back pressure, and with no compression loop, ie least steam is used, back pressure is minimised, and SSC is highest with full regulator.

With the same 28.2% cut off, increasing the nozzle diameter to the BB18¼ figure would save 3% of steam, increasing the lap and lead to BB dimensions as well another 1.5% more. At that short cut-off the increased length of the BB port made no further difference.

This comparison can go further. I note that I did not record outputs from the B18¼ as high as from the BB18¼. How can this be explained when back pressure is not the reason after all? Here are a series of comparisons based on Perform.

		BB18¼ (b)	B18¼ (as built)(a)	B18¼ 1930s (b)	B18¼ (1930s) ihp at BB18¼ cut-off (d)	B18¼ (1930s) ihp at BB18¼ steam rate (d)
1175 ihp 51 mph Ts 332ºC	Steam (lbs/hour) Cut-off (%) Back pressure (lb/sq in) SSC	18,100 32 6.5 15.3	21,600 47 13.7(c) 18.3	20,000 39 11.2 17.0	988 32 6.4 15.9	18,100 1100 ihp
1000 ihp 40 mph Ts 327ºC	Steam Cut-off Back pressure SSC	15,300 30 4.5 15.3	16,600 41 7.4 16.6	15,900 35 6.6 15.9	872 30 4.6 15.5	15,300 970 ihp
900 ihp 35 mph Ts 324ºC	Steam Cut-off Back pressure SSC	13,800 29.5 3.6 15.3	14,500 38.5 5.4 16.0	14,100 33 5.0 15.6	797 29.5 3.8 15.6	13,800 875 ihp
800 ihp 30 mph Ts 321ºC	Steam Cut-off Back pressure	12,600 29 3.0	13,400 37 4.3	12,800 32 4.1	729 29 3.1	12,600 795 ihp

	SSC	15.7	16.7	16.0	15.5	
700 ihp 25 mph Ts 318ºC	Steam Cut-off Back pressure SSC	11,200 29 2.4 16.0	11,400 36 3.3 16.3	11,200 31.5 3.1 16.0	652 29 2.6 16.0	11,200 700 ihp

All calculated at full regulator, steam chest pressure assumed to be 3 lbs below boiler pressure

(a) 160 lbs boiler pressure, 8 ins diameter valves

(b) 170 lbs boiler pressure, 9½ ins diameter valves

(c) a 9 lbs compression loop in exhaust line above inlet pressure; pressure relief valves will have been set off; a 4.7 ins diameter nozzle would have cured that.

(d) Ts adjusted as necessary for change in ihp

So a B18¼ in its final form could produce the best horsepower (1175) I knew from a BB18¼, but with a considerably longer cut-off and back pressure, therefore using about 10.5% more steam. No compression loop was present, but would have occurred at any higher output.

Perform does not detect compression loops at lower rates of output with the Ts used in the above exemplifications. They nevertheless occurred, leading to vibration. The vibration was a disincentive to drivers to work them at high rates. Such loops can be detected by Perform, however. They occurred at lower Ts, ie before full superheat was available, when the output concerned needed a greater quantity of steam. The loops therefore seem to have been mostly the result of using a large quantity of steam at high rpm with a relatively long cut off which reduced the proportion of the stroke over which compression occurred. All of this was mostly the opposite of what the drivers believed.

C19

The Ts for these engines has to be judged. The PFGA is the same as on the C17s, and the FGA a third larger. On the other hand, the grate is only a fifth bigger, and the nozzle is the same diameter, points which lead to an intense, high temperature fire, and hot gases passing the elements. The judged Ts is 340ºC at 500 ihp and 400 at 1000 ihp. These high temperatures result in good SSC figures.

From loads observed in photographs and set down for these engines, and speeds from timetables, they were expected to develop about 820 ihp on passenger trains at 24 mph, and that when they were fitted with 8 ins valves (finally 9½). This would have required 41.5% cut-off and conferred an SSC of about 14.1. At this speed, increasing the valve diameter to 9.5 inches allowed a cut off of 40% but the SSC was not affected.

Making various changes to the engines made little difference at this speed, but a larger blast nozzle would have reduced back pressure. If it had been intended that they haul heavy passenger trains at a reasonable speed, say to develop 950 ihp at 45 mph, then the effect of improvements is marked. With 8 ins valves, 38% cut off is needed, large compression loops form, and SSC is 14.20. Even with 9.5 ins valves, compression loops are almost as bad. These disappear with a 4.33 ins nozzle, which, with 32.5% cut off, reduces SSC to 13.16. Increasing pressure to 175 lbs and reducing cylinder diameter to 18 ins (as on the C18 as built), reduces cut off needed to 30% and SSC to 12.54. Lengthening the lap makes a modest further improvement, but increasing the port width makes no difference at these cut offs. As Ts is already high, a larger blast nozzle and 18 ins cylinders with 175 lbs pressure would have been very desirable improvements. Following the reasoning underlying the judged Ts of these engines, the larger nozzle must result in lower Ts.

With the final valves, moving the full goods load at 10 mph required some 620 ihp, and a cylinder tractive effort of 23,500 lbs. This turns out to be impossible. With full (75%) cut off and saturated steam, the best possible is just over 600 ihp. Drivers would not work at such a long cut off for a prolonged period. Even with various degrees of superheat, 67% cut off yields ihps in the high 500s and just over 600. A C19 with a full load could therefore have managed only about 9 mph. SSCs were in the 19 to 20 range, even at 9 mph.

DD17

Ts has been judged at 316ºC at 500 ihp, and 360 at 1000 ihp, when continuously developed.

One of these engines, worked beyond its normal suburban lines, developed some 900 calculated ihp on the Little Liverpool Range at 35 mph. With Ts considered to have been a modest 340ºC because the effort was short and sudden, Perform would have it that this required 34% cut-off, and that is about what was used. SSC by Perform was 15.2. Back pressure was about 7 lbs, a high figure resulting from the 4.125 ins nozzle, intended to make the smokebox self cleaning. Enlarging the nozzle to 4.4 ins would have improved SSC by about 3.5%. On that particular effort, increasing the lead, and enlarging the valve diameter have very small positive effects on the SSC.

Beyer Garratt

My calculated effort of 1360 ihp on the Blackall Range was obtained with 55% cut-off (one QR engine with an indicator of cut-off) at 31 mph but only part regulator. Working back through Perform using 336ºC Ts and 178ºC Tc, the inlet pressure needed was 127 lbs, with which SSC was about 18. Had full regulator and 30.5% cut off been used, the SSC would have been 15.26. Other calculated efforts would have given SSCs of 14 to 15 if full regulator was used.

Moving their full goods load at 10 mph required 900 ihp. This required 75% or full cut off, at which the SSC was 29 lbs while the steam was saturated, and 27 when the Ts reached 303ºC. It is unlikely that drivers would have worked them at full cut off for extended periods, even at 10 mph. Had 67% cut off been used, 10 mph could not have been held.

AC16

These American designed and built engines had short lap to increase the maximum cut-off and improve starting. They also had restricted 3¾ ins blast nozzles to ensure their smokeboxes were self cleaning. The valves were only 8 ins diameter.

My best calculated output of 886 ihp at 28 mph with 345ºC Ts and 181ºC Tc would have required 48% cut-off according to Perform, and given 15 lbs back pressure. Not surprisingly the SSC with that was high, 17.7.

Another calculated ihp of 823 at 47 mph with much the same temperatures would have required a 34.5% cut-off, and back pressure would have been 15.6 lbs.

The full goods load, the same as that of a C17, required 510 ihp at 10 mph (higher than that of the C17 because the engine was heavier). This would have required 70% cut-off, at which back pressure would have been 9.8 lbs and the SSC 24.5 lbs.

The suspicion of locomotive men that despite their excellent reputation for steaming, these engines were heavy on steam is borne out. Compared with the C17, an AC16 used 15% more steam to move the full goods load at 10 mph, and 6% more to produce 760 ihp at 23 mph. The constricted blast nozzle was a heavy price to pay for rendering the smokebox self cleaning. The usual QR arrangement of a steam powered blow out chute of the smokebox ash sump used a tiny amount of steam. That constricted nozzle was the reason for the sharp blast of these engines.

The high steam consumption would have been considerably reduced by larger nozzles, probably sufficient to provide draft on the fire, but not to keep the smokebox clear of ash. Taking the 886 ihp effort, a 4 inches nozzle would have saved 8% of steam, a 4¼ ins nozzle 11% and a 4½ ins nozzle (as on the B18¼, 13%. Increasing the lap to 1¼ ins would have improved the saving to 14%. With these improvements in place, increasing the diameter of the valves would have had no effect on steam consumption.

Snifting

It was not made clear in the book in what circumstances hot gases and char (HGC) could flow from the smokebox to the valves and cylinders. With the engine shut off, air or low pressure steam entering the cylinders during the inlet phase up to the point of cut-off was reduced in pressure as the piston advanced, so that it was less than atmospheric pressure up to release. During this phase, there is no connection between the smokebox and the partial vacuum in the cylinders, and HGC is not attracted to the cylinders. At release, the exhaust opens to the smokebox. There is then a large difference in pressure between the smokebox, at atmospheric pressure, and the cylinders at partial vacuum, barely affected by the action of the blower, and HGC are attracted by the partial vacuum, and flow through the exhaust passages and ports to the cylinder, until the pressure in the cylinder is brought to atmospheric in the exhaust phase by the reduction in volume as the piston returns. Using a long cut off increases the amount of the snifting medium admitted before the valve cuts off, and thereby minimises the vacuum formed, but also maximises the period the valve is open to exhaust. In turn the latter maximises the flow of HGC.

Any low pressure medium such as air results in a brake on the engine while coasting, the less so the longer the cut-off. The braking effect can be neutralised by using steam so that the cylinders produce zero power. This will not necessarily eliminate the partial vacuum at the opening of the valves at release. That can be eliminated only by admitting steam at an even higher suitable pressure and cut-off. The consumption of steam is minimised by operating at the shortest possible cut off (mid gear) with relatively high pressure, a large regulator opening, if not full, regulator.

Despite my views on the satisfactory behaviour of some snifting, Perform shows it was generally not good enough. On all engines with hydrostatic lubricators and air snifting, the air snifters came into operation as soon as steam was shut off, at any speed at all. On the DD17, and AC16 with mechanical lubricators which atomised the oil with steam, the same applied. On the BG I cannot say, as the arrangements changed over time.

In simulating air snifting, the Perform coefficients for expansion and compression of saturated steam were used, and 28^{0} C air and cylinder temperatures were assumed. Using steam, 175^{0} C steam temperature was used, and 90^{0} C cylinder, the latter justified by the low to modest quantity of steam involved.

For the PB15, C17 and BB18¼, the effect of each of following was tried:

- a plentiful supply of air; as the automatic steam snifter can have operated at only about the same pressure, this illustrates its working also;

- a low steam pressure to show the effect of the NSW practice of driving with a certain low pressure in the steam chest; this will also illustrate the effect of driving with the regulator slightly cracked as recommended on the QR at times;

- the pressure which gave zero ihp so that the engine was not braked by the suction in the cylinders;

- the pressure which resulted in a positive pressure in the cylinder between release and the end of the compression stroke, eliminating all vacuum.

All were tested at 30 and 40 mph, plus 50 for the BB18¼, and for three approaches - full gear cut-off, half full (45%) cut-off, and 5% cut-off. The results are tabulated in full for the PB15.

PB15

Steam chest pressure	Ihp #	Steam pressure lbs				Steam consump- tion lbs/hr
		At cut-off	At release	During compression stroke	Of compression loop	
full (76%) cut-off, 40 mph						
0 (air)	-46	-9	-13	2	6	nil
4	0	0	-3	0	8	3320
8*	30	3	1	1	5	4100
45% cut-off, 40 mph						
0 (air)	-119	-12	-20	1	22	nil
5	-76	-7	-11	1	18	880
20	0	6	-3	0	10	2140
35*	79	15	3	0	10	3525
5% cut-off, 40 mph						
5	-176	3	-1	0	4	
137*	0	76	2	0	12	815
76% cut-off, 30 mph						
0 (air)	-24	-7	-3	0	3	nil
3	0	0	-2	1	5	2600
8*	31	3	0	1	5	3360
45% cut-off, 30 mph						
0 (air)	-77	-11	-18	0	13	nil

5	-44	-2	-7	0	8	940
15	0	5	-4	0	8	1675
30*	62	15	2	0	10	2775
5% cut-off, 30 mph						
5	-129	-1	-9	2	15	
125*	0	80	3	0	14	650

a negative ihp indicates that the formation of the partial vacuum acts as a brake on the engine.

* at this chest pressure, the pressure in the cylinder between release and the end of the forward stroke and during the whole of the compression or return stroke is at all times above atmospheric, ie there is no partial vacuum to attract HGC from the smokebox..

The compression loops are in some cases considerable, but the pressure at the opening of the valves to the power stroke is still well below that encountered while powering.

It is easy to see why drivers did not use steam while coasting to eliminate the drag on the train or the prevent a vacuum forming in the cylinder at any stage - such action used lots of steam, especially at long cut offs, and powered the train in circumstances where that was possibly not desirable.

It is also clear that no single chest pressure is ideal for snifting at all speeds. Further, the chest pressure which results in the engine developing zero power does not remove vacuum. That which eliminates vacuum powers the train and uses a lot of steam. But coasting in mid gear with a large regulator opening is the most favourable method of coasting.

On a C17 coasting at 40 mph and full cut-off with air snifters, the pumping of the pistons absorbed about 53 ihp. This is rather less than the 142 ihp had cut-off been only 45%. At 30 mph, these figures are 30 and 92 ihp. To obtain about zero ihp when coasting, some positive steam pressure was needed in the steam chest. With full cut off this was 4.5 lbs at 40 mph and 3 lbs at 30 mph. With 45% cut off it was 19 lbs 40 mph and 15 lbs at 30 mph. To operate at 5% cut off, 172 lbs (full regulator) was required in the chest and at both 40 and 30 mph.

On a BB18¼ coasting with full cut-off with air snifters, the pumping of the pistons absorbed about 74 ihp at 50 mph, 48 at 40 mph and 27 at 30 mph. With 45% cut off, these figures are all higher, at 200, 144 and 94 ihp. To obtain about zero ihp when coasting, some positive pressure was needed in the steam chest, here assumed provided by saturated steam. With full cut off this was 4.5 lbs at 50 mph, 3.5 lbs at 40 and 3 lbs at 30 mph. With 45% cut off these pressures were 19 lbs at 50 mph, 16 lbs at 40 and 13 lbs at 30. To operate at 5% cut off, 167 lbs was required in the chest (full regulator) and at all of 50, 40 and 30 mph.

So far as the entry of HGC is concerned, the best measure is to shut off the exhaust while coasting, as was done on the rack engines when they were working in compressing brake mode. This might be made automatic - the nozzle closed whenever no steam was present in the main steam pipe. Air could then be used for snifting, so long as the braking effect on the train was suitable for the circumstances. The air used for snifting would then recirculate, the compression loop from one return stroke providing most of the air for the next stroke, without any bypass being needed. If the driver wished to steam to remove the braking effect on the train, then he should be encouraged to used the mid gear position with full regulator. In that case, the blast nozzle would not have to be covered.

The Drifting Gauge when fitted allowed the driver to crack the regulator open a little to avoid a vacuum in the branch steam pipes while the engine was coasting. As this vacuum was induced by the suction during the inlet phase until admission was cut off at full cut off, the steam admitted could not have been sufficient to prevent a partial vacuum after cut-off.

Where the steam provided lubricator atomisation, its adequacy for snifting would depend on its quantity and pressure, which are not known.

PB15 Trial superheating

It is stated in the book that in view of the work these engines were to do well into the 1960s, they should have been superheated, or at least the 1924 version built as such. The following exercise shows that at low degrees of superheat, the superheat contributes little to reducing SSC, and that until there is more than 60^0C superheat, longer cut offs are needed on the superheated engines to achieve the same ihp than with the engine saturated, on account of the lower density of the steam, hence mass of steam admitted at each stroke.

All these calculations reproduce 460 ihp. The cylinder temperature (Tc) is that used for all superheated engines. The port length given by the slide valves was retained in the exercise.

Steam temp ^{0}C	Cylinder temp ^{0}C	Cut-off %	Steam/hr	SSC

188 (saturated)	141	35½	11,700	25.4
228 (40^0C of s/h)	141	41½	11,300	24.5
248 (60^0C of s/h)	144	40½	11,000	23.8
268 (80^0C of s/h)	151	38½	9920	21.4
288 (100^0C of s/h)	158	37	8910	19.3
308 (120^0C of s/h)	166	35	7810	17.0
328 (140^0C of s/h)	174	33	6840	14.9
348 (160^0C of s/h)	182	33	6220	13.5
368 (180^0C of s/h)	185	33½	6020	13.1

The highest steam chest temperature observed in any QR tests in which it was measured was ca 340^0C on C17s in 1934 tests on the Toowoomba Range, so the upper end of the temperatures used above is outside QR experience.

Up to 268^0C steam temperature, a liquid film is present on the cylinder walls for the whole stroke; by 368^0C only a vestige of such a film is left at the beginning of the stroke. Condensation absorbs much of the pressure of the inlet steam at the lower temperatures, the chest pressure of 157 lbs falling to as low as 142 lbs at inlet at most temperatures even with considerable superheat, rising to 144 at 328^0C, 150 at 348 and 156 at 368^0C. At the last of these temperatures, 368^0C, the cut-off requires lengthening again, despite the boost to pressure at admission, because of the lower density of the steam, hence lower mass admitted per stroke.

This explains why the saturated C16 with its full load climbed the Toowoomba Range with shorter cut offs in critical places than did the superheated C17 with its full load (data in QR tests).

Real economy comes about with steam temperatures of over 300^0C. At such temperatures, piston valves would have been necessary, to avoid the cutting of valve faces which occurred when some of the slide valve C16 and B17 engines were superheated. In that case, a longer equivalent width of valve could have been employed in the exercise, probably with further slight improvements in economy.

Effect of Carbonising Ports

C17s and the D17 tank engines, which had the same cylinders and hydrostatic lubrication of the valves and cylinders, suffered most of all QR engines from carbonising of the ports. This led to them being "weak on banks" on occasions, while there were occasions when they were failed because they could not start their trains. At 10 mph with a full goods load, the carbon could be up to 0.15 inch on the inside edge of the port before there was any noticeable effect, according to Perform. At 0.2 inch there was some effect, but if 0.3 in thick, the opening was delayed until about 10% of the stroke had occurred, and tractive effort was 7½% down. More importantly, that delay to obtaining steam could mean inability to start.

At 19 mph and 700 ihp, there was a loss of tractive effort from 0.1 inch, and by 0.2 inch, the admission pressure did not reach even 90% of chest. If the carbon was 0.3 in thick, admission pressure was at best 50% of chest.

At 33 mph, even 0.1 inch build up reduced admission pressure to 90% of chest, 0.2 inch to 70%, and 0.3 in to 40%, and very little steam entered the cylinders.

Coasting

The Perform program throws light on the usefulness of various coasting practices. As explained in Section 5.06, and above under Snifting, when an engine is shut off and coasts, the cylinders act as vacuum pumps, the more so the shorter the cut off. Perform allows the resistance from this to be estimated, with the cylinders fed by air at atmospheric pressure. With full cut off used, the resistance is over 400 lbs for a C17, for example, increasing slightly with speed. It is relatively high on QR engines, on account of their relatively large cylinders, and small coupled wheels. This is solely the effect of the pumping; the friction effects on mechanical parts are in addition. On the other hand, the friction on those parts when steaming is saved.

On steep gradients with sharp curves, hence low speeds, coasting was the rule. On gradual down gradients with easy curves (eg Grandchester to Lanefield), light trains or light engines (ie without train attached) had to steam lightly - the pumping resistance and the head end atmospheric resistance of the train moving through the air overcame the effect of the down gradient. When steaming lightly, ie with the regulator just cracked, with low steam chest pressure and low superheat, the SSC is high.

If the train was heavier, the free coasting of the vehicles of the train overcame the pumping and head end atmospheric resistance, the more so the heavier the train. If the schedule was easy and the gradient steep enough, it was possible to let the train run. It might well accelerate, depending on the gradient and its weight. If the schedule was tighter, it was best to accelerate under power until the train reached the speed limit, and then to have let it coast.

It was also possible to steam lightly, in much the same manner as with a light engine. Which method used the least steam depends on the gradient and the time available, also how hard the steaming to accelerate the train could be without overly exciting the fire during a long shut off. But in the case of Grandchester to Lanefield and west of Gowrie, with 250 tons or so of passenger train, accelerating to the speed limit and then coasting used slightly the lesser quantity of steam

Compounding, Not an Error

The review of the book in the Australian Railway Historical Society Bulletin, February 2003 p 67, claims that I advocated compounding for the QR. I did not. In Sections 11.03 to 11.05, I reviewed various alternative forms of locomotive and possible general improvements to steam locomotives which might have been considered for the QR. Comparison with what was done elsewhere in the world to improve the steam locomotive is an essential part of such a review. After the review, only the tender booster was advocated - see p 287 LH para 3, also Part 12 p 289 LH, in which my ideas on worthwhile improvements to QR steam locomotion are summarised. Compounding is not mentioned.

I advocated superheating the saturated engines and persisting with the exhaust steam injector. Otherwise, I did not advocate seeking thermal efficiency improvements (see p 286 LH para 2). While experience with the ESI has indeed varied (see BIRC May 1930, April 1937 and January and April 1954), it was successfully used in New Zealand, and in the UK, where it remains in use. NZ locomotive men certainly considered it to be a success there. See also E J McClare, Steam Locomotives of New Zealand, Vol 3, p 171f. QR enginemen who used it found it advantageous where steaming was continuous, eg Chinchilla to Toowoomba. By reducing the fresh water injected into the boiler for a given task, the ESI also reduced boiler maintenance.

Driving Compound Engines

While I did not advocate compounding, I know that the remarks in the above review about the need for highly trained drivers to operate compound locomotives, on the basis of French practice, are wrong. Steam locomotive drivers in France were trained as a specific grade, rather than promoted from firemen. That applied to drivers of simple expansion steam locomotives as well as to those driving compounds. It is explained in the Journal of the Stephenson Locomotive Society 847, Sept/Oct 2007, p 220, that not every main line driver (mecru - mecanicien de route) was trained in the workshops, those that were had to do at least two years main line firing, that some who joined to be firemen chose later to become drivers, that all had to go through training courses and pass the exams, which included an exam for each class, compound or not. While such training was different from that in almost all other countries, it was similar to the way drivers are trained in the 2000s in Australia. There was nothing especially complicated about driving compound locomotives. The few extra things which had to be known in addition to those needed to drive simple locomotives were straightforward in comparison with the A6-ET Westinghouse brake equipment which QR drivers mastered.

The unsuperheated D50 class engines in NSW (p 133), also referred to in the review, did more than shunt - they hauled a lot of coal in the Newcastle area.

Albert Bridge

Attention was drawn in the above review to my statement that not much more could have been done to strengthen the Albert bridge over the Brisbane River at Indooroopilly under traffic (p 23, also caption to Fig 1.05 p 15, and p 11). Details of the site and 1895 bridge are given in MPICE 1898 p 288. It is of course useful to ask whether there was some way of strengthening this bridge with minimal interruption to traffic. The example was given of replacing the very high main span of the Trisanna viaduct in Austria in a matter of hours (this was even longer than each of the spans at Indooroopilly). In that case, the new span was built on girders erected above the valley floor to one side of the existing span. The existing span was slid out to similar girders on the other side in the same movement as the new span was slid in from the side on which it was built. The main span was replaced to avoid having to replace the considerable approach viaducts and reroute the line on a hillside.

The Trisanna case does not provide an example for what might have been done at Indooroopilly. The Brisbane River at Indooroopilly is deep and has a seven feet tidal range. Doing something similar to what was done at Trisanna would have required a central pier and abutments on each side of the existing bridge. That would have been double the number of each needed for a replacement or second bridge. Sinking the piers for both the 1895 and 1958 bridges was a considerable task.

Replacing the existing spans with even short (say weekend) interruptions to traffic would have been difficult. Floating out the old spans and floating in the new would have required specialised vessels, and a construction yard with a large crane somewhere upstream of Victoria (or later, Grey Street) bridges on account of height limitations, solely for the job. As the abutments and pier had not been designed for jacking the spans from the level of even the bottom of the abutments, which are high up the bank, it would have been necessary to carry the spans at about their level above water after erection.

The existing spans (300 tons each) could have been hauled in to the banks, the reverse of the way the northern spans of both the 1895 and 1958 bridges were placed. The masonry towers at the entrance to the bridge would have been demolished in advance. The existing spans, supported at the pier end by a vessel, would have been lifted at that end with the tide, and by jacks at the land end, and hauled in while the vessel supported the pier end until the span cantilevered from the shore side. On completion of the hauling to the shore, the existing spans would have been slid to one side and the new slid from the other side to the same place for hauling and floating out into place. This arrangement would have required four "pads" for the spans, three of which would have been considerable structures. The process would have taken several days for each span, largely on account of the need to build and remove the slipways, and on the southern side, to build a considerable supporting structure above some of the existing track, on account of the gradient falling from the bridge to the south and there being a curve there.

Building a new bridge around the old in situ (as done at Antigua on the NCL - see W J Doak, Strengthening Railway Bridges, 1918 paper reprinted in SE April 1974) was limited by the inability to support the new spans from beneath while under construction, the lack of sites for jacks to lift the new spans, and by the relative narrowness of the central pier.

The Albert bridge carried water mains. There was also a river-bed main to the Chelmer to Oxley peninsula from the pipeline from Mt Crosby. I do not know if it could have maintained supplies while the bridge was removed and replaced, even for a period of only a few days.

A new, stronger, bridge was probably the best solution even in the days of double track. Once quadruplication of the route became policy from the late 1940s, building a second stronger bridge became the best policy.

The reviewer said that various of my remarks will not have won me many friends in the QR of 2002. That is an odd statement from the editor of an historical journal. I was not writing about 2002 but about 1900 to 1969. It is the task of the historian to report the facts of the past and if possible reach judgements on them, whatever people in the organisation in later years might think. His complimentary comments on the QR of 2002 had no relevance to my book.

My grateful thanks to Messrs N Condon, M J Ghee, E D Hills, P G Kennedy and N G Linnehan for assistance with these corrections.

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amended 12 January 2007, 28 April 2008 and 10 October 2008

Further amendments can be found on the "John Knowles Railway History Web Page", at http://freespace.virgin.net/johnk.pb15

10 Oct 2008