AST Trust
Advancing Steam Traction

NEWSLETTER No. 29 FEB 2026



Jamie Keyte showing the NYMR's

Paul Middleton (aka Piglet) Revolution

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CHAIRMAN'S PIECE

John Hind

Back in 2019 Revolution was a dream, now in 2026, that dream is turning into reality and the results are starting to see the light day of day with exposure to the critical eyes of the enthusiast and the railway press.

The first public appearance was at the Greatest Gathering, with further exposure at our 2025 conference and now we have an article coming up in 'Trackside' Magazine and, once we can run it on air, the prospect of exposure on YouTube.

None of this would have not been possible without funding. It's all very well having an idea, but without money, hardware is not going to happen, so a BIG THANK YOU to anyone who has donated to Revolution, whether that is by a lump sum or a regular monthly donation.

Advance Notice Conference 2026

At the 2025 conference, Paul Middleton suggested using the NYMR's new 'Mutual Improvement Class' facility at Grosmont, for our 2026 conference.

Mike Stockbridge and Ian Cook (ASTT Member and Director of the NYMR) have taken that idea up are developing plans for accommodation, dinner venues and facilities based around Pickering and Grosmont. Further information will follow, by separate announcement.

Dates are being confirmed with the NYMR, but it is looking like the 1st week in October – Saturday 3rd and Sunday 4th of October.

We already have offers of papers from members and can expect a follow up talk by Paul Middleton on the S160.

If anyone else would like to present a paper, please contact me.

AGM

Our 2026 AGM will be taking place on a Saturday afternoon in March or April. The date is to be confirmed at our next committee meeting. Details will be sent out by a separate notice. As in previous years it will be a 'virtual meeting' on Zoom.

DARJEELING HIMALAYAN RAILWAY STEAM LOCO REVIVAL

John Hind



We have been approached by the Darjeeling Himalayan Railway about modernising one of their B Class Locomotives to solve several operational problems.

One of them was imported into the UK by Adrian Shooter and ran on his Beeches Light Railway for a few years. It is now under overhaul at Statfold Light Railway.

Before we start formal discussions or get involved further, Jamie Keyte is going to make arrangements with Statfold to have a look at the locomotive. Anyone who is interested in getting involved, please contact me, so that we can include you on the visit to Statfold. Although a small locomotive, studies like this can very 'resource' intensive and without people to support it, nothing will happen, so if you are able to help with technical, financial or administrative support, please let us know.

Photo By Chris Allen, CC BY-SA 2.0, https://commons.wikimedia.org/w/index.php?curid=47012801

MEMBERSHIP MATTERS
Chris Newman

Committee Members

ASTT's management committee :

John Hind	Chairman & Trustee	Jamie Keyte	Trustee
Hendrik Kaptein	Secretary & Trustee	Alex Powell	Trustee
Chris Newman	Treasurer & Trustee	Grant Soden	Ex-officio
Mike Stockbridge	Ex-officio		

New Members:

Sylvain Tasiaux from Roizy, France – mechanical engineer specialising in Fluid Mechanics. Has been an apprentice fireman working on ex-PLM Pacific 231.K.8 in main-line serice. Since 2016, he has volunteered as fireman and driver on the CFBS metre-gauge heritage railway in France. He has worked with Nigel Day on a project to fit a Lempor exhaust on one of the CFBS engine which has not yet been given a go-ahead.

Joseph Burgard from Minnesota, USA – a mechanical engineer with military experience now working in the automotive industry. A life-long steam enthusiast, he built first live-steamer at the age of 14. He has worked for the T1 Project, doing heat flow calculations and material/information gathering.

Membership Numbers

Two past members have withdrawn from membership for reasons of age and declining health. John Duncan who will be 87 his year has been actively involved with ASTT since its establishment and with the 5AT Project before that. It is proposed to recognise his contribution by proposing that he be awarded honorary membership at our 2026 AGM.

Our nominal membership now stands at 119 subscribing members plus 12 non-subscribing student members (total 131). Of the former, 85 (71%) have paid their 2026 membership fees. We urge the remaining 29% to pay their fees which fell due on 1st January, and remind them that our subscription fees increased by 25% this year. For information about renewing your subscription, please visit https://advanced-steam.org/membership-renewal/.
The current breakdown of our nominal 131 members is as follows (in the hope that all resubscribe!):

Full Members:	36	UK Members:	80	Age Ranges	
Associate Members:	83	EU:	20	Over 60	Approx. 41%
Student Members:	12	North America:	15	30 to 60	Approx. 44%
		South America:	1	Under 30	Approx. 15%
		Australasia:	13	Av. age	Approx. 55
Total Membership:	131	Asia & Africa:	2		
Overseas members	**51 = 39%**				

PUBLICATIONS PAGE
Chris Newman

Book Sales

As expected, book sales have fallen since September with only 24 books sold. Sales are likely to remain low without new titles to place on offer so we urge members to consider putting forward new titles that we can publish and post on our website.

Books sales since September 2025 have been as follows:

Publisher	Author	Title	Sales since N/L 28	Total Sales
ASTT	L.D. Porta	Porta's Papers Vol 1	1	162
	L.D. Porta	Porta's Papers Vol 2	1	153
	L.D. Porta	Porta's Papers Vol 3	1	120
	Martyn Bane	Porta's Papers Vol 4	3	95
	L.D. Porta	Porta's Papers Vol 5	5	66
	C. Newman (Editor)	Porta's Centenary Compendium Vol 1	2	92
	C. Newman (Editor)	Porta's Centenary Compendium Vol 2	3	58
	Ian Gaylor	Lyn Design Calculations	1	114
	David Wardale	5AT FDCs	1	226
	David Wardale	Reminiscences of a Trainspotter	1	45
	Alan Fozard	5AT Feasibility Study	0	47
	Chris Newman	ASTT Prospectus[¶]	3	91
Camden*	*David Wardale*	*The Red Devil and Other Tales from the Age of Steam*	*0*	*260*
	Phil Girdlestone	*Here be Dragons*	*0*	*33*
	Jos Koopmans	*The Fire Burns Better …*	*0*	*11*
	L.D. Porta	*Advanced Steam Design*	*0*	*5*
Crimson Lake	Adrian Tester	Physiology of the Loco Boiler – Part 1	2	41
	Adrian Tester	A Defence of the MR/LMS 4F 0-6-0	0	44
	Adrian Tester	Introduction to Large Lap Valves	0	24
		Total sales	24	1690

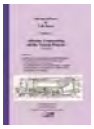
* Camden is now selling these titles in digital format and are no longer sold by ASTT.

[¶] ASTT's "Prospectus" was produced for distribution at the Greatest Gathering event in Derby (August 2025). It is however available for sale "at cost" for £2 per copy.

Book sales provide a valuable source of income to ASTT which helps to fund activities, not least our *Revolution* project. **Members are offered 20% discount on our in-house titles and are urged to support our endeavours by purchasing books from us.**

Note: We'd like to expand our book selection, so we welcome any suggestions for additional titles that we might be able to sell, or new texts that we might publish under ASTT's logo.

Titles published by the Advanced Steam Traction Trust

	Year	Author	Title	Pages	RRP*
	2015	David Wardale	The 5AT Fundamental Design Calculations	556 (b&w)	£49.00
	2016	Alan Fozard and others	The 5AT Feasibility Study edited	230 (colour)	£35.00
	2018	Ian Gaylor	Steam Locomotive Design Specifications and Calculations for New Build Baldwin 2-4-2T 'LYN'	604 (colour)	£52.00
	2018	L.D. Porta transcribed and edited by Chris Newman	Selection of Papers by L.D. Porta – Vol 1 - Tribology and Lubrication	250 (b&w)	£25.00
	2019	L.D. Porta transcribed and edited by Chris Newman	Selection of Papers by L.D. Porta – Vol 2 - Adhesion, Compounding and the Tornado Proposal	256 (b&w)	£25.00
	2021	L.D. Porta transcribed and edited by Chris Newman	Selection of Papers by L.D. Porta – Vol 3 - - Steam Locomotive Boilers, Fireboxes and Combustion	290 (b&w)	£25.00
	2024	L.D. Porta and others Compiled by Martyn Bane	Selection of Papers by L.D. Porta - Vol 4 - Locomotive Boiler Water – Treatment,	513 (colour)	£40.00 (£48 hard cover)
	2025	L.D. Porta transcribed and edited by Chris Newman	Selection of Papers by L.D. Porta – Vol 5 - - Exhaust Systems and Other Aspects of Loco Design, Performance and Operation	340 (some colour)	£27.00
	2022	Compiled and edited by Chris Newman	A Compendium of Articles and Papers to celebrate the Centenary of the birth of Livio Dante Porta - Volume 1	226 (colour)	£25.00
	2023	Compiled and edited by Chris Newman	A Compendium of Articles and Papers to celebrate the Centenary of the birth of Livio Dante Porta – Volume 2	331 (colour)	£27.00
	2023	David Wardale	Reminiscences of a Trainspotter	67 (b&w)	£16.00
	2025	Chris Newman	ASTT 2025 Prospectus	22 (colour)	£2.00 (at cost)

* The RRPs (recommended retail prices) shown do not include postage. Paid-up members of ASTT are entitled to a 20% discount on these prices.

These books can be purchased through ASTT's website at http://advanced-steam.org/books-forsale/.

REVOLUTION UPDATE
Jamie Keyte

Work is progressing towards a submission for approval of the boiler design. This is an important first step and is in the capable hands of Richard Coleby. There are detail drawings to be completed but the actual design is substantially complete in CAD form.

Meanwhile manufacture has progressed steadily on various bits and pieces. These are the time consuming little items which somtimes get lost within the bulk of the locomotive – hence they are not always in mind when thinking of jobs to do! They include:



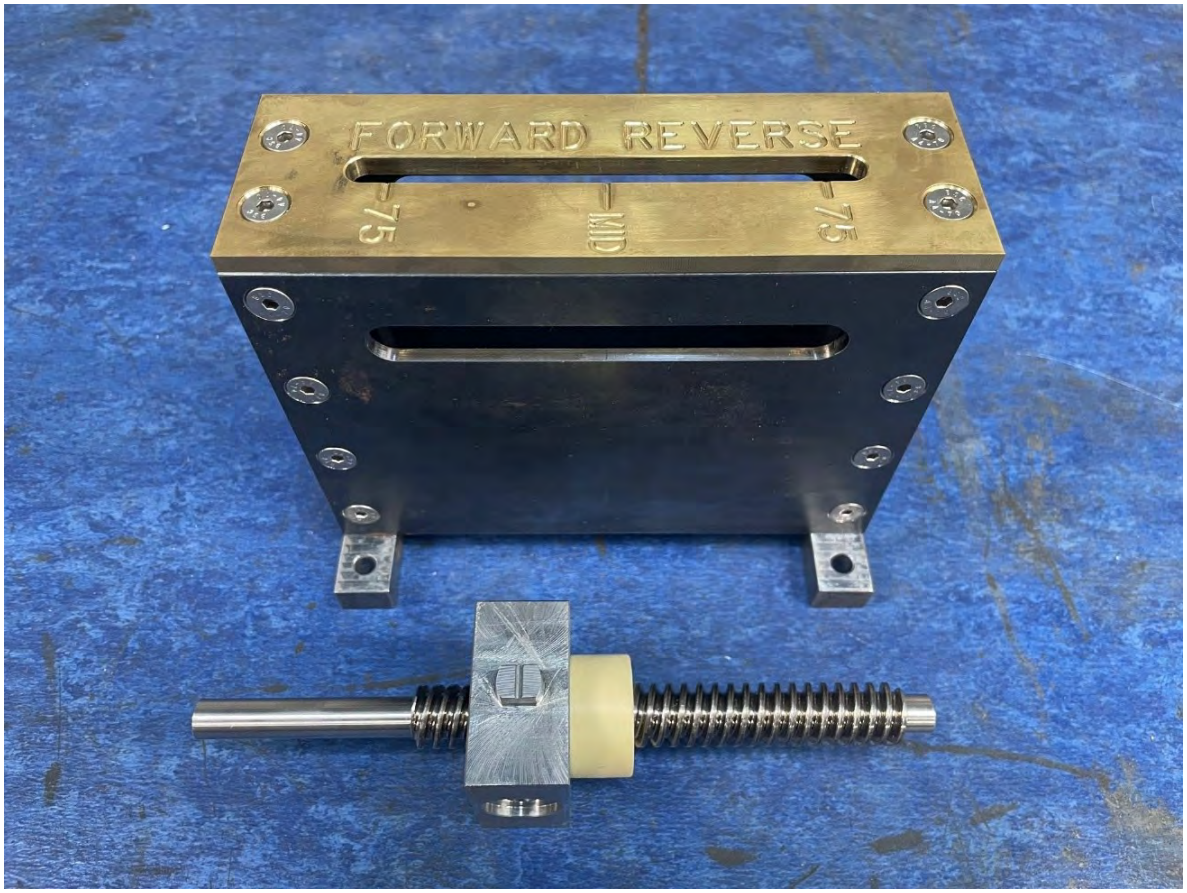
The mechanical lubricator casing, lid and valve blocks. This has evolved to include a dedicated drain port and additional ports to permit a small steam heat coil for warming the oil. The casing was machined from a billet of aluminium and will be hard anodised. Of course, no opportunity was missed to advertise the AST! Richard has largely schemed out most of the pipe runs on the locomotive.



The cylinder drain cock bodies from bronze. These are based on a design successfully used at Stapleford.



The smokebox door and hinges. This was machined from a solid disc of steel - you can get away with this in 10.25” gauge – and the extra weight is very useful! Richard designed an innovative method of producing the hinge straps which look the part but are far more robust. Just the dart and handle required to complete the “face” of Revolution.



The reverser pedestal and weigh shaft have been produced. There are still some turned components required to complete the assembly, along with the reach rod.



Funding

At the end of last year, we did one of our cost to complete reviews. By then we had spent and committed £47642. We have £19022 left to spend of the £66664 that has been raised to date, from donations, ASTT funds and Gift Aid. We estimate that we need another £22214 to finish Revolution, excluding any contingency for any remaining unknowns.

Without contingency the estimated total cost of Revolution is £88878.

Pleasingly, we are within the original budget, we set out in 2020 of £99300, which included a 10% contingency. In getting here, there have swings and roundabouts within the budget, with some items costing more, some less and one for extra CAD software not needed.

Revolution still needs funding to complete it so donations are still needed.

We will be applying to the Charities Commission for recognition as a charity and this releases more Gift Aid, which we reclaim from HMRC, reducing but not dropping the need for more donations.

Publicity

We received a visit on 27/01/26 from Nick Brodrick of Trackside magazine. Nick is planning to write an article for a future issue of the magazine which will hopefully create some useful publicity.

On the subject of publicity, the LNWR George the Fifth Steam Locomotive Trust is planning to have an Open Day for its members at Keyte Smith Ltd's workshop in Kirkby in Ashfield, Notts, NG17 7LA. This is provisionally set for Saturday 16th May between 10am and 3pm. On display will be the (12":1ft) main frame assembly of the new build No. 2013 "Prince George" 4-4-0. "Revolution" will also be on display and AST members are welcome to come and look at both.

Jamie will be bringing Revolution to the Hengist Locomotive Works Open Day on Saturday 11th April 2026 at Diamond Group (formerly CTL Seal), Butterthwaite Lane, Ecclesfield, Sheffield S35 9WA. Standard Gauge new builds BR Standard Class 6 'Hengist' 72010 and LNER B17 ' Spirit of Sandringham' 61673 will be on display as well as narrow gauge new 'Gowrie'.

In addition, there are live steam models and trade displays. There is only one Open Day a year and the event attracts between 500 and 600 people but never seems crowded. Entry is just £3.00. If anyone would like to man the stand with Jamie, then entrance will be free!



A GREAT FIRE EXTINGUISHED: IN MEMORIAM JOS KOOPMANS

On 22 December 2025, Jos Koopmans passed away at the age of 85, after a short illness. Jos was born in 1940, in Amsterdam, and studied Naval Engineering at Delft University. Still most of his professional career was in IT and in logistics.

Although a member of the Dutch Rail- and Tramway Enthusiasts' Society (NVBS) since 1969, it was only in the second half of the 1980ties that Jos became really involved in both steam and tramway documentation and preservation. By that time his four children – one daughter and three sons – had grown up, so his family life became a bit less demanding. In more recent years Jos became a happy grandfather of nine grandchildren.

Jos' first steam related publication dates from 1988. This is an extensive review of a book on the Netherlands Railways class 6300 4-8-4 locomotives, built by Henschel in 1930 and by then the biggest tank engines in Europe. The book states quite a number of technical errors and questionable assumptions, which Jos, in the style that later became his trademark, incisively pointed out and corrected. The review also contained a compact and elegant treatise on the basic thermodynamics and functioning of steam locomotives in general, just to further enlighten the interested public. Many more most interesting publications followed suit.

Earlier on Jos also started a 4-6-0 steam loco replica project, based on the last and largest express steam locomotive in The Netherlands (built in the 1930ties) but then the enterprise, like plans for a smaller 4-4-4 tank engine and however highly developed, came to an end as higher authorities duly blocked service by any such "useless engines".

Jos was most active in nationwide tramway restoration and preservation as well. Thus in 1992 he was one of the recreators of the Arnhem electric bogie tramcar nr. 76. Through his perseverance the Open Air Museum in Arnhem got an electric tramway. By now Nr. 76's most successful museum service life easily outdoes the original's 15 year lifespan (prematurely ended by the Second World War).

His last contribution to tramway preservation was the restoration of motor car 1 of the Twentsche Electrische Tram. He started this in a small workshop in Overloon. He was just able to witness the completion of this project.

Outside The Netherlands, Jos was mostly known by his work on steam locomotive front-end design. In 2005 Jos successfully defended his Ph.D. thesis on exhaust theory and optimisation, published as The Fire Burns Much Better. In this thesis Jos explained why great steam locomotive engineers like Kylälä, Chapelon and Porta were on the right track on exhaust performance – think here of Kylchap and Kylpor style exhausts, great improvements upon classic exhaust systems – but that they did not yet fully express the theory behind it. Jos did so after all. Remarkably, his thesis was reprinted a couple of times: a rare but then in this case most deserved distinction.

Next and on the basis of his scientific findings Jos actively pursued front end optimization for preserved steam locomotives. Heritage steam conservatism notwithstanding Jos greatly improved steam locomotives in The Netherlands, in Canada and in the United Kingdom. Outstanding here is his new front end for King Edward II (a former GWR King class 4-6-0 locomotive): steaming was much improved, within an even smaller than original loading gauge.

Jos may not always have been a model of polite subtlety in debate. But then this most positively contributed to clear communication of his most original and critically constructive views. Thus his great and sometimes fiery polemic with David Wardale will be remembered by many in awe. Strongly opposing views notwithstanding these two steam wizards still went for the very same cause.

His achievements did not go unnoticed: in 2024 Jos was appointed Member of the Royal Order of Orange Nassau.

In his last months, Jos was still actively searching DB archives in Koblenz, Germany, hoping to find unpublished Test Bulletins on the Western Germany Railways class 23 locomotives.

We are most grateful for his many years of hard work and great efforts, and for his most instructive, entertaining and jovial company. A great fire has been extinguished. But Jos will not be forgotten at all. Hereby go all our good wishes for his beloved wife Petra, for his children and grandchildren and for all those close to him.

Hendrik Kaptein, with many thanks to Wim Beukenkamp & Eelco de Bode.

REVOLUTION
A LITTLE LESS REVOLUTIONARY THAN WE'D THOUGHT!

Chris Newman

When putting together the conceptual design for Revolution, we all thought that Jamie's and Richard's idea of fitting a removable chimney mounted on a plate forming the top of the smokebox was both clever and novel. Yet it seems that BR had the idea first having (partially) put into practice 75 years ago when they planned the testing of B1 No. 61353 on the Rugby test plant.

The Gresley Observer (journal of the Gresley Society) No 197 of Autumn 2025 carries an article titled *A Mystery Solved" which focussed on the appearance of this same Thompson B1 which had been photographed by Eric Trecy in 1951 when double-heading a Coronation Pacific leaving Tebay at the start of the climb to Shap. With some clever research, the author of the article (Chris Nettleton) was able to deduce that the B1 was heading north to undertake running trials after being tested on the Rugby test plant.

The second page of the article carried an equally interesting anecdote as follows:

"So why had 61353 been sent to Rugby for testing? Referring to the relevant RCTS Green Book (Part 2B page 141), here is the answer:

> When No. 61353 was built in September 1949 it was specially adapted for testing a variety of blast arrangements at the new Rugby Testing Station. To allow for the tests, the chimney and anti-vacuum valve were mounted on detachable plates on the smoke-box (fig. 227). Four blast arrangements had earlier been proposed for trial, as follows:-
>
> Double chimney with a diverging blast-pipe branching into two nozzles, each inclined from the vertical at an angle of 1 in 8. (This experiment was later deleted from the schedules).
>
> L.M.S.-type double blastpipe and chim-ney. Provision for the fitting of 2in., 4in. or 6in. high blastpipe tops. The nozzles to be 3½ in. diameter and capable of boring out later to 4in. if required.
>
> A4-class type double blastpipe and chimney with Kylchap cowls.
>
> S.R. type single chimney with five-jet Lemaitre multiple blastpipe, each jet inclined from the vertical at an angle of 1 in 12. Provision for the fitting of 2½ in., 2¾ in. or 3in. diameter blastpipe tops.

No. 61353 had to wait until October 1950 before it could be accepted for testing at Rugby. In addition to the stationary tests, road tests were conducted between Carlisle and Skipton in April and August 1951. Altogether the tests occupied a year. Upon its arrival at Rugby, preliminary experiments were carried out with dimensional variations in the blastpipe and chimney, but these produced no improvement. The remainder of the tests were therefore conducted using the standard B1 class smokebox arrangement. No evidence has been found that any of the above special blastpipe and chimney arrangements were actually tried out on this engine. On the road tests, the loads varied between 116 and 436 tons according to circumstances, and normal express passenger timings were gener-ally adhered to. It was found that provided the fire was kept relatively thin and of an even thickness, the boiler was very free-steaming though rather sensitive to any departure from the correct method and rate of firing, and slow to recover from the effects of such a departure. The engine rode well on the line and on the test plant too. For the purpose of the tests however, No. 61353 had been speci-ally rebalanced with 70 per cent of the reciprocating masses, primarily to prevent excessive forces on the drawbar of the station-ary test plant.

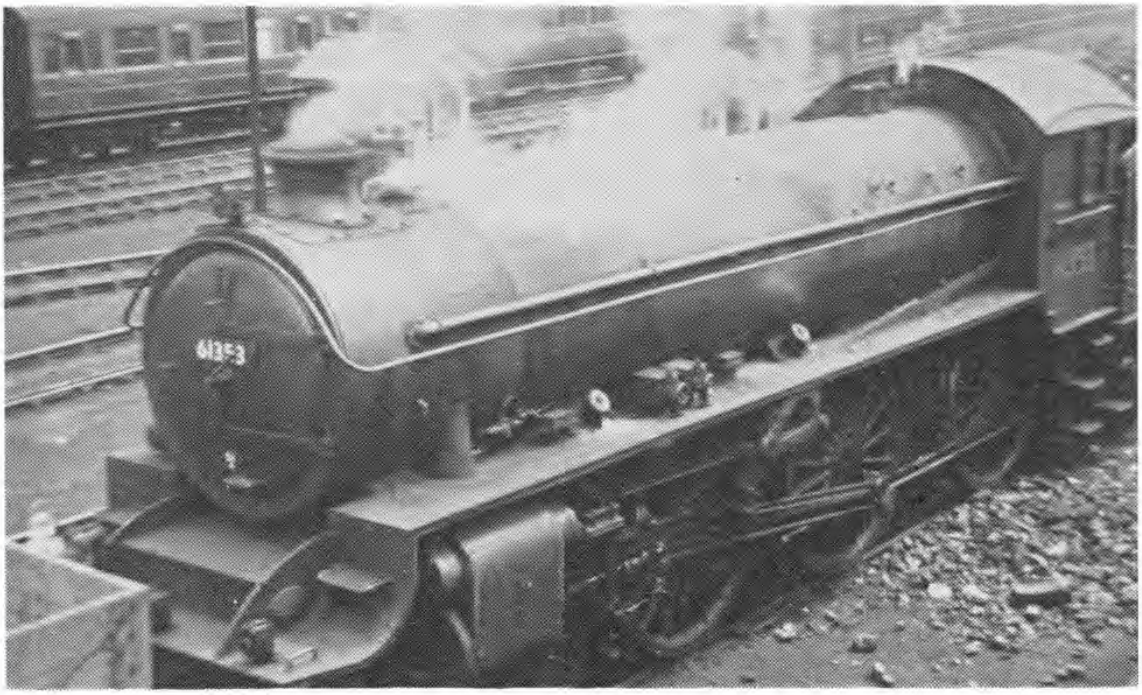


Fig. 227 Class B1 No. 61353 at Scarborough, July 1952.

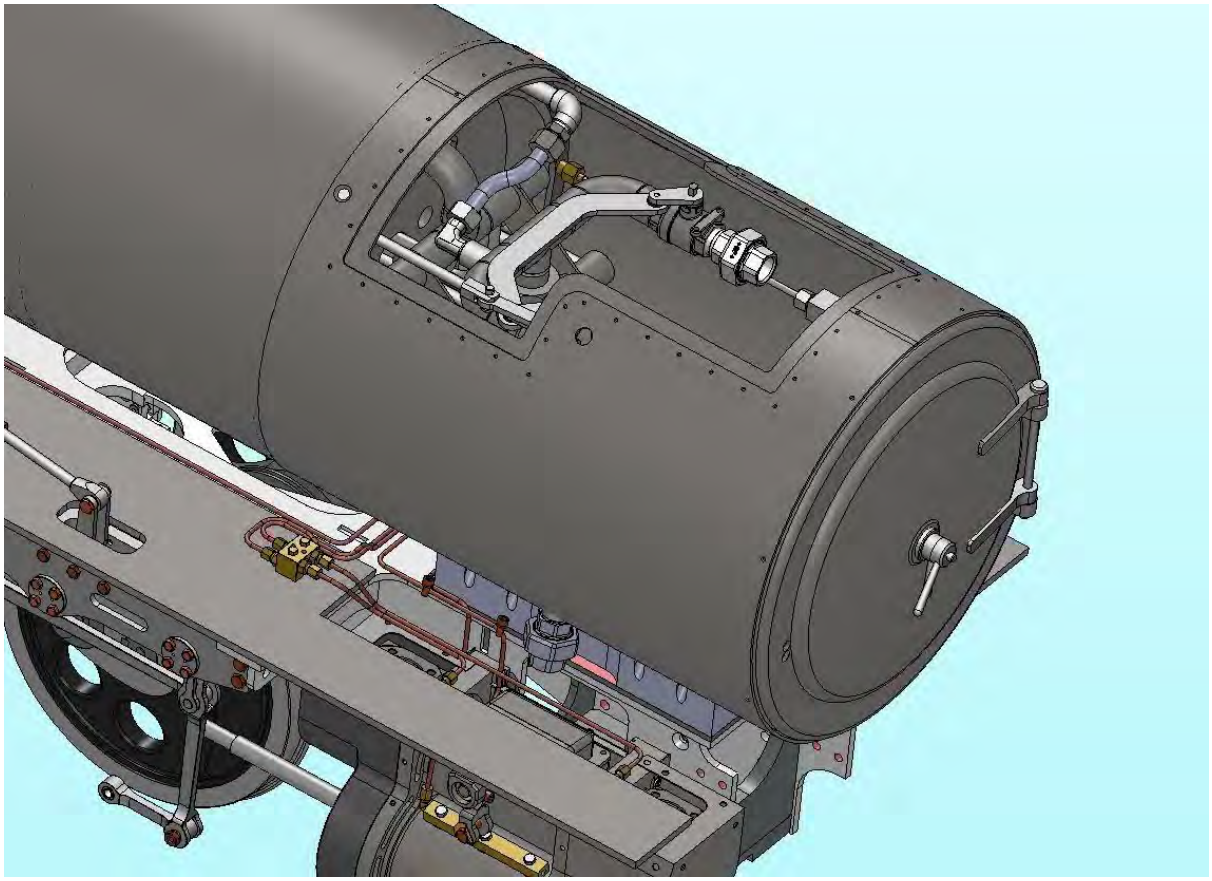
Chimney mounted on removable panel to allow experiments to be made.

It's a shame that BR chose to abandon the comparative tests and to test only the standard B1 smokebox arrangement. But at least it leaves ASTT with the opportunity to break new ground by actually conducting comparative testing of exhaust systems!

The gap in the top-front of *Revolution*'s smokebox where the replaceable chimney will be fitted can be seen in the photo and drawing below. The gap is widened towards the rear of the smokebox to provide access to the superheater header and regulator mechanism.



Revolution partially assembled in Jamie Keyte's workshop. Jamie can be seen behind.



3D drawing showing the gap at the top of the smokebox to facilitate changes of chimney and access to the superheater header and regulator mechanism © Richard Coleby

THE WORK OF D.A. COSTA ON LOCOMOTIVE POPPET VALVE GEAR AND INDICATOR DIAGRAMS

Graham Shirley, CEng, FIMechE



Figure 1 Italian Caprotti Valve gear

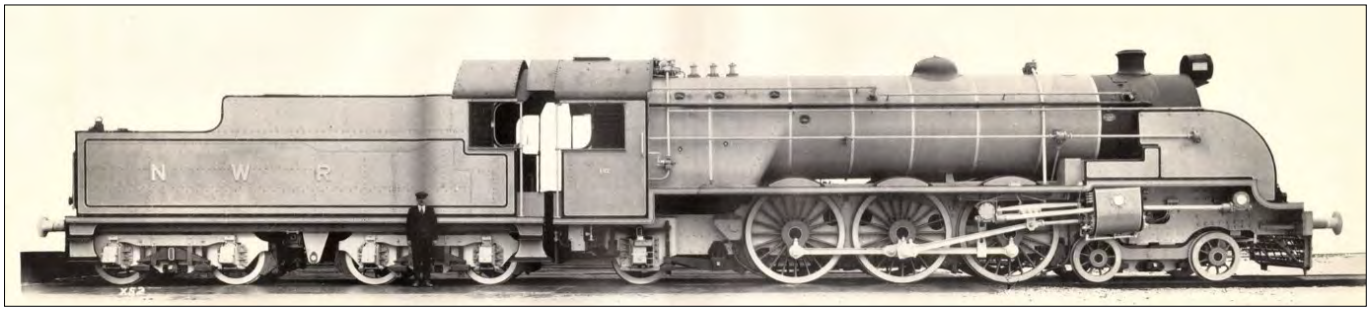


Figure 2 London based ALE RC poppet Valve gear

The poppet valve gear manufacturers in the UK mainly depended on locomotive sales to the British Empire through companies like William Beardmore, Vulcan, North British, and Beyer Peacock, as few UK companies equipped their engines with poppet valve gear. As a result, much of the development work occurred overseas, with India playing a significant role. Vulcan supplied 4-6-2 XS class engines to the North Western State Railway (ISR). Engines 760 and 761 were fitted with Caprotti valve gear from Italy, while Engines 780 and 781 were fitted with Rotary Cam (RC) valve gear from Associated Locomotive Equipment (ALE). In 1932, these engines showed improved operational availability compared to those fitted with piston valve gear, though no fuel economy savings were reported.

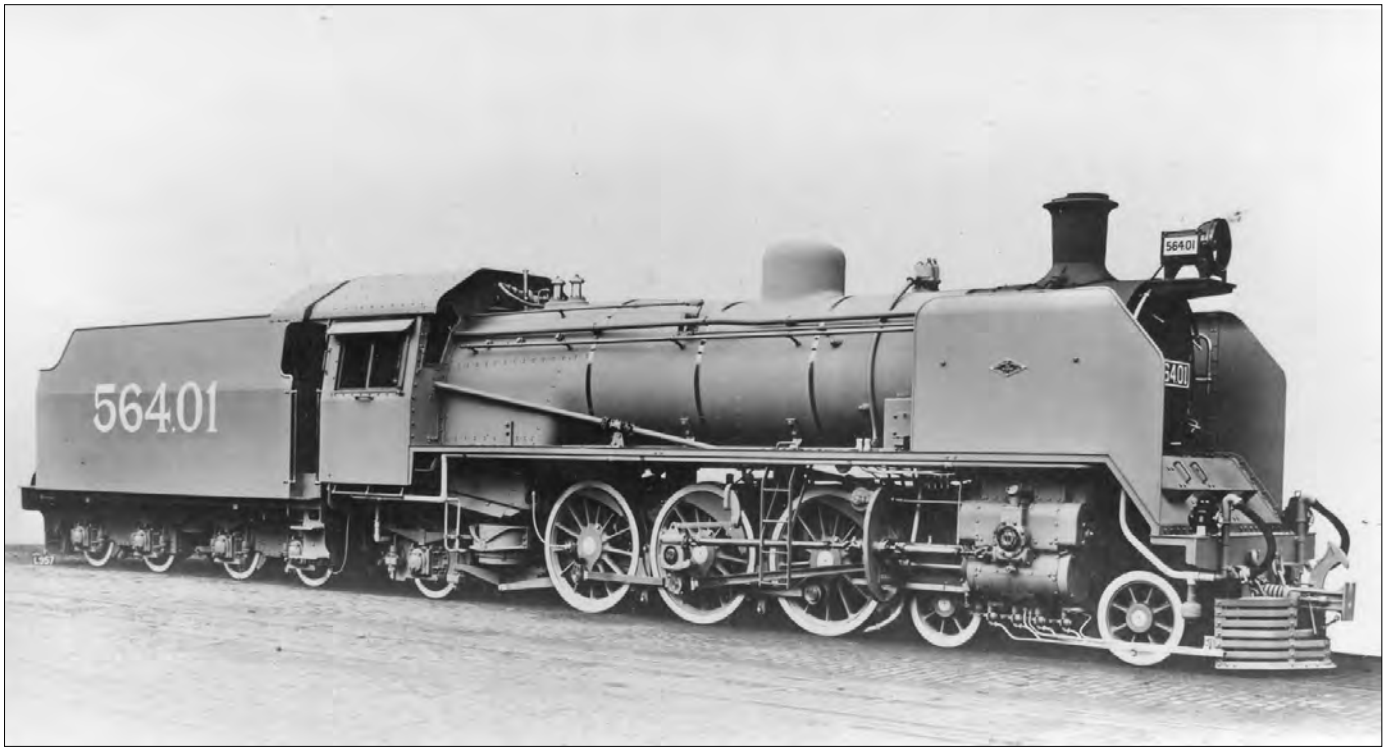


Figure 3 Malay Railway 3 cylinder ALE RC poppet valve gear.



Figure 4 The D class on GIPR were part of the BESA (British Engineering Standards Association) standard designs introduced around 1905–1928 to unify locomotive specifications across Indian railways

The Great Indian Peninsula Railway (GIPR) was one of India's earliest railways, operating broad-gauge lines from Bombay into the Deccan plateau and beyond. It played a major role in connecting Bombay to central and northern India. In 1907, Class HP

(Heavy Passenger) broad gauge 4-6-0 locomotives were introduced according to British Engineering Standards Association (BESA) design, some of which included poppet valve gear. D. A. Costa is known for his pioneering work in steam locomotive engineering, especially in locomotive poppet valve gear and indicator diagrams, detailed in his 1938 paper to the Institution of Locomotive Engineers #149. The indicator diagram demonstrated the efficiency of non-condensing simple expansion steam locomotives. His contributions significantly influenced the efficiency and functionality of British Caprotti valve gear for steam locomotives, which played a central role in the 1950s redesign of Italian Caprotti valve gear. Costa's pre-war predictions were confirmed in the test results published in the British Transport Commission Bulletin 15, detailing the performance and efficiency tests of the new British Railways Class 8 locomotive 71000 Duke of Gloucester by Sam Ell at Swindon.

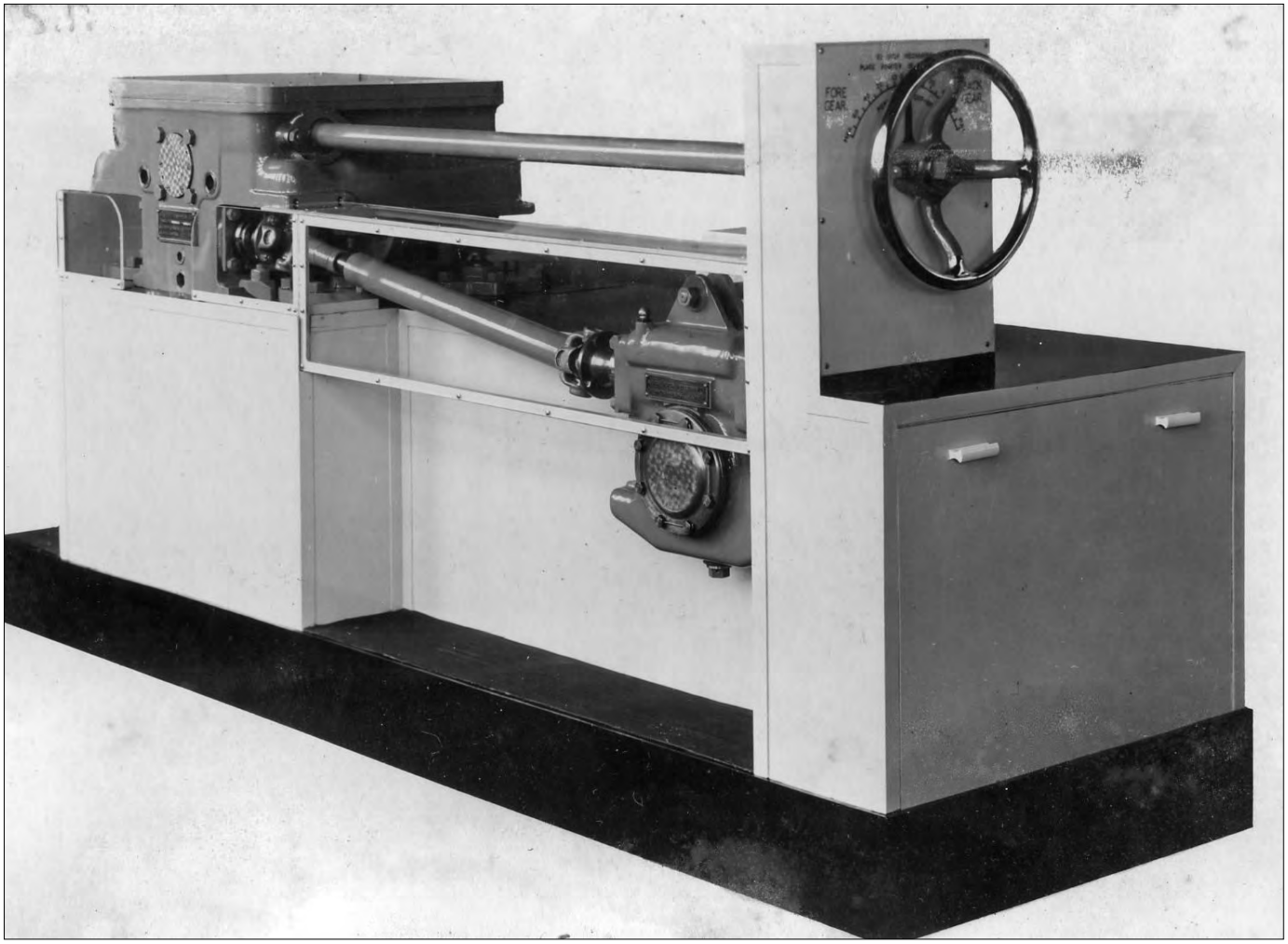


Figure 5 British Caprotti valve gear 1951 festival of Britain display

Locomotive Poppet Valve Gear

Traditional slide and piston valves in steam engines had inefficiencies due to their geometric constraints, leading to poor steam expansion and wear from friction especially with highly superheated steam which carburised the lubricating oil. Poppet valve gear, using cam-operated poppet valves, offers a more efficient solution by allowing separate control of steam admission and exhaust timing. Traditional valves only control admission cut-off, while poppet valves optimize the exhaust start point. D. A. Costa addressed this in his paper.

Overview

D. A. Costa's paper analyses the efficiency of non-condensing simple expansion steam locomotives, focusing on performance and operational improvements.

Introduction

- The paper, presented by G. Da Costa in 1938, discusses inefficiencies in non-condensing simple expansion steam locomotives.
- It emphasizes the limited scope for improving modern locomotive boiler efficiency.
  The analysis aims to enhance locomotive efficiency through a detailed examination of the motive power circuit.

Methodology

- Da Costa employs an actual theoretical cycle to derive ideal indicator diagrams and steam rate curves.
- The study uses various tables and figures to present calculations related to steam consumption and work done. Key parameters include clearance volume, steam chest pressure, and exhaust valve closure percentages.

Key Calculations following

- **Clearance Volume**: 11% for the 71000 Duke locomotive.
- **Steam Chest Pressure**: 240 psi, with a final pressure of 0 psi.
- **Steam Temperature**: 750 °F, with exhaust valve closure at 50%.
- **Cut Off**: Selected at 1%, with three cylinders and a piston stroke of 28 inches.
- **Driving Wheel Diameter**: 74 inches, with a piston diameter of 18 inches.

Performance Analysis

- The paper includes multiple tables detailing the weight of steam at various cut-off percentages and exhaust valve closures.
- **Weight of Cushion Steam**: Calculated based on temperature assumptions and steam pressure.
- **Weight of Cylinder Steam Feed**: Derived from the cut-off and exhaust valve closure data.
- **Final Expansion and Compression Pressure**: Analysed to determine work done per pound in a complete cycle.

Efficiency Metrics

- The document presents a series of tables (II to IX) summarizing the weight of steam and work done per cycle.
- **Work Done**: Calculated for various cut-off and exhaust valve closure scenarios, with specific values for each configuration.
- The maximum work done per lb of steam is highlighted, indicating optimal operational parameters.

D A Costa's Calculations

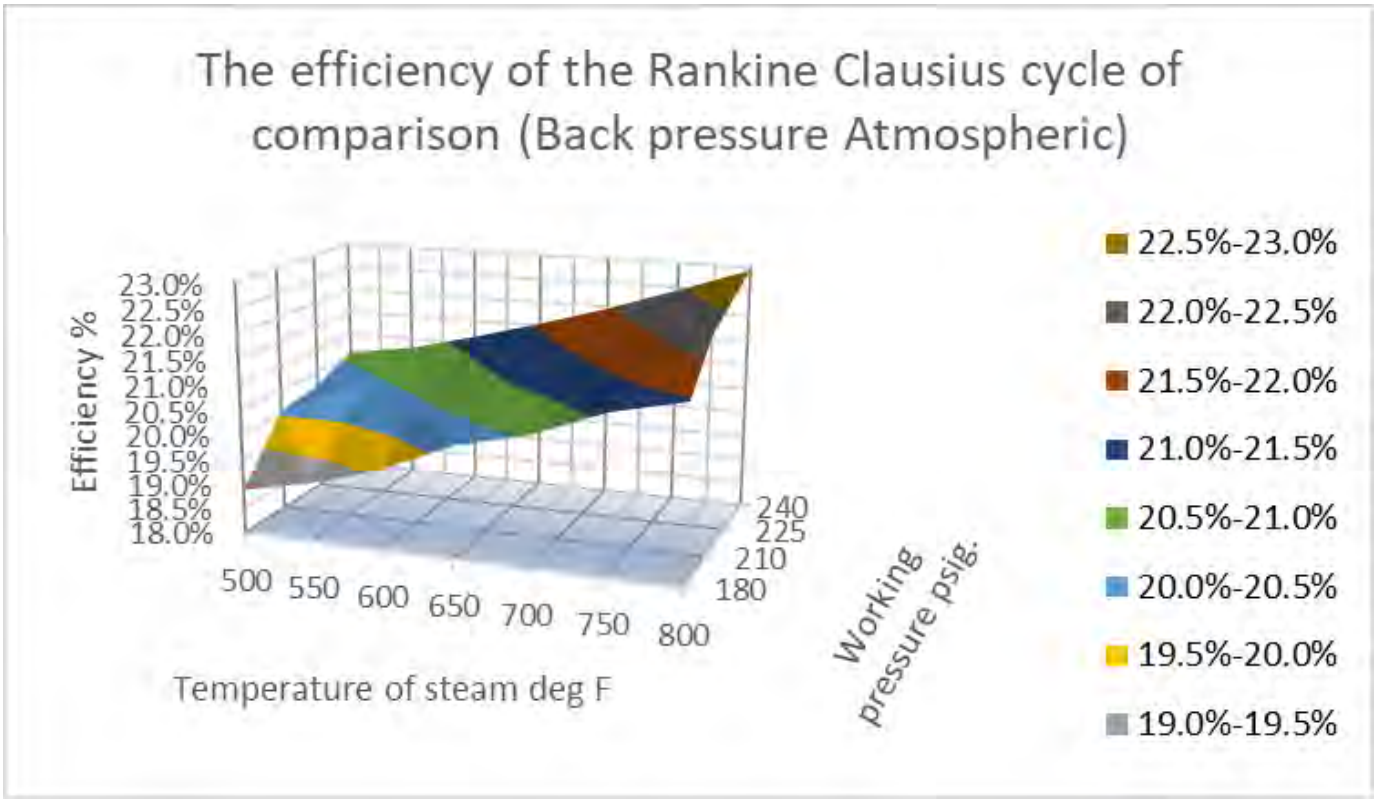


Figure 6

D A Costa's paper begins by exploring the efficiency of steam locomotives, focusing on the Rankine-Clausius cycle from the 1930s. Efficiency increases with higher boiler pressure and increasing steam superheating. However, this cycle is not ideal for studying locomotive performance since it requires complete expansion, which isn't achievable in all cut-offs with fixed dimensions. The cycle does not accurately reflect the actual steam cycles, especially at early cut-offs where Caprotti valve gear ex-

cels, making direct comparisons unhelpful.

A new “ideal” standard of comparison is proposed by D A Costa, a standard which represents the “ideal” performance throughout the working range of cut-off. The theoretical indicator diagram of the actual cycle of operations in the locomotive cylinder is represented by “abcdefa” in Figs. 3a and 3b.

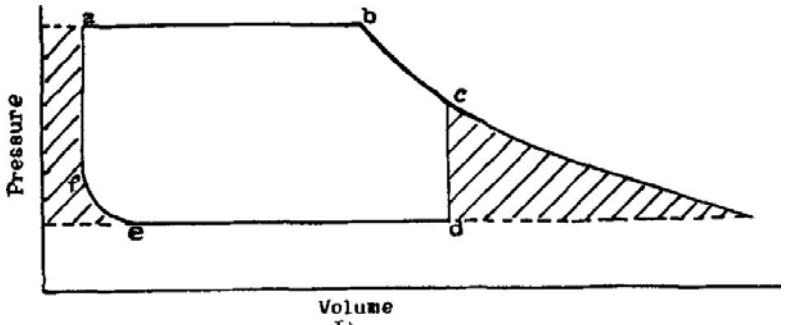


FIG. 3a.

LATE CUT-OFF DIAGRAM OF THEORETICAL STEAM CYCLE.

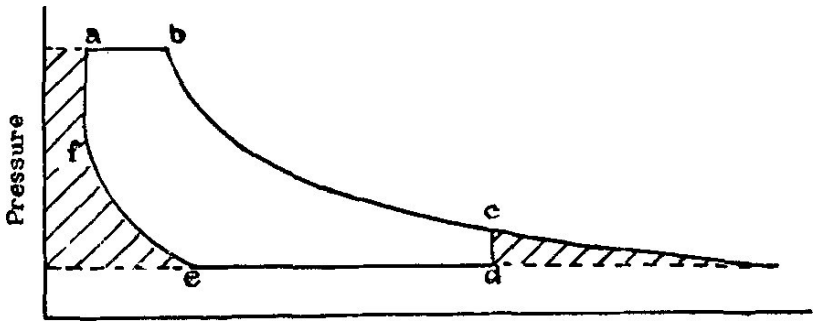


FIG. 3b.

EARLY CUT-OFF DIAGRAM OF THEORETICAL STEAM CYCLE.

In this theoretical cycle, steam is admitted to the cylinder at a constant maximum temperature and pressure along “ab”. At point “b”, cut-off occurs, and the steam expands along segment “bc”.

In 1924, Arthuro Caprotti published a paper titled "A New Locomotive Distributing Gear Using Poppet Valves," wherein he applied polytropic indices for the expansion and compression segments. During the return stroke of the piston, steam is exhausted at a constant minimum temperature and pressure until reaching point “e”. At this juncture, the exhaust valve closes, leading to compression along segment “ef”. At point “f,” pressure is restored along segment “fa” to the initial maximum admission pressure. Testing on Indian locomotives demonstrated that these indices varied with cut-off and speed. The expansion index ranged from 1.13 to 1.25, and the compression index varied from 1.09 to 1.4. The values utilized by Caprotti for the expansion curve segment “bc” $PV^{1.2} = Constant_1$ and the compression segment “ef” $PV^{1.3} = Constant_2$ were closely approximated. In the ideal case of a non-condensing engine, “cd” must lie along the atmospheric pressure line, thereby defining point “d.” Therefore, for a given working pressure and cut-off, there exists only one variable: the compression line “ef”. Determining the optimal point at which the exhaust valve should close, or where “e” should be located to ensure the most efficient use of steam at a specific horsepower, will resolve this issue. Consequently, the ideal indicator card for any given boiler pressure, cut-off, and cylinder clearance volume will have been established.

Cushion steam weight is crucial for determining the optimal cycle based on cylinder temperature. Accurate measurements identify the most economical compression point. In the 1930s, D A Costa estimated temperatures from 250°F at 10% cut-off to 350°F at 80% cut-off, with early errors of ±20°F and later errors up to ±100°F. These impacted steam consumption calculations by ±1% and exhaust valve closure by ±2% he claimed. The science of locomotive understanding progressed, and Sam Ell's 1950s tests provided chart 23 (Ref:BR Test Bulletin 15) showing exhaust steam temperature closely related to steam flow rate, and chart 11 offered cut-off data indicating a boiler steam flow. The inference of the two charts is shown below and maybe different to D A Costa assumption. In the meantime, we proceed with D A Costa’s assumption to replicate the figures in his paper.

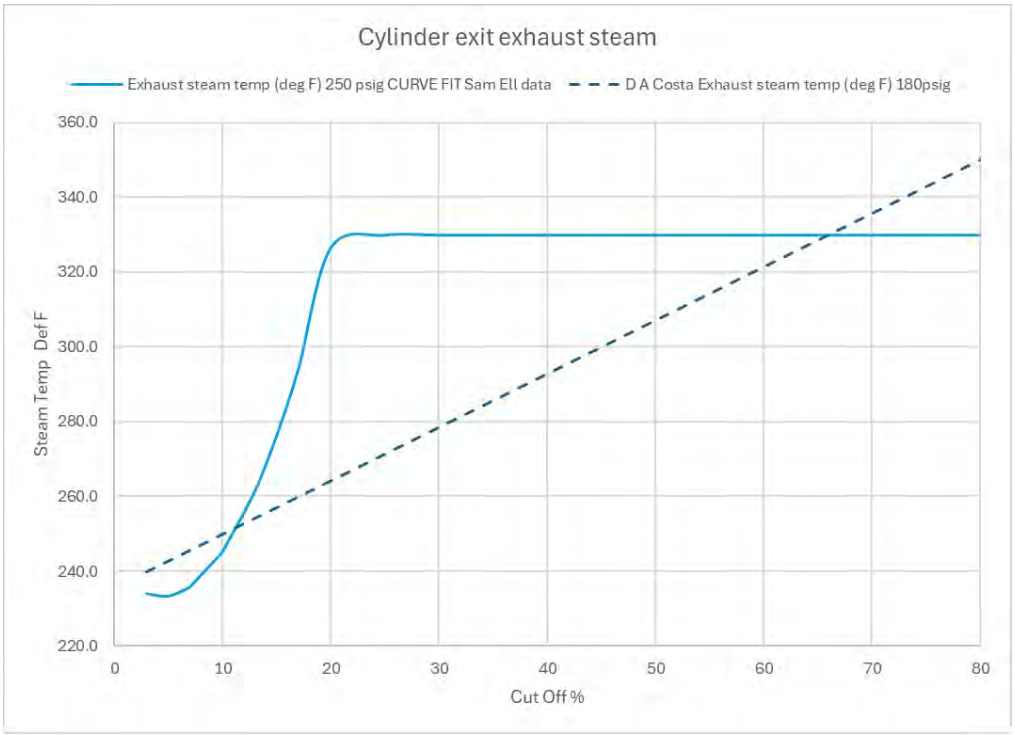


Figure 7

Key principles crucial to the steam locomotive's economy are:

Early cut-off is essential for steam efficiency.
Higher boiler pressures improve cycle efficiency.
Smaller clearance volume enhances economy.
The ideal compression point depends on cut-off and clearance volume, but not significantly on boiler pressure. Minimal compression is needed at late cut-offs, increasing as the reverse lever is notched up.

Three corollaries can be deduced from the above principles:

Any pressure drops between the boiler and cylinder, including part regulator working, is inefficient.
Valve gear should allow for an "infinitely variable" cut-off to use the earliest cut-off for power development.
Figure 8 shows higher speeds enable earlier cut-offs, resulting in greater efficiency at given horsepower compared to lower speeds.

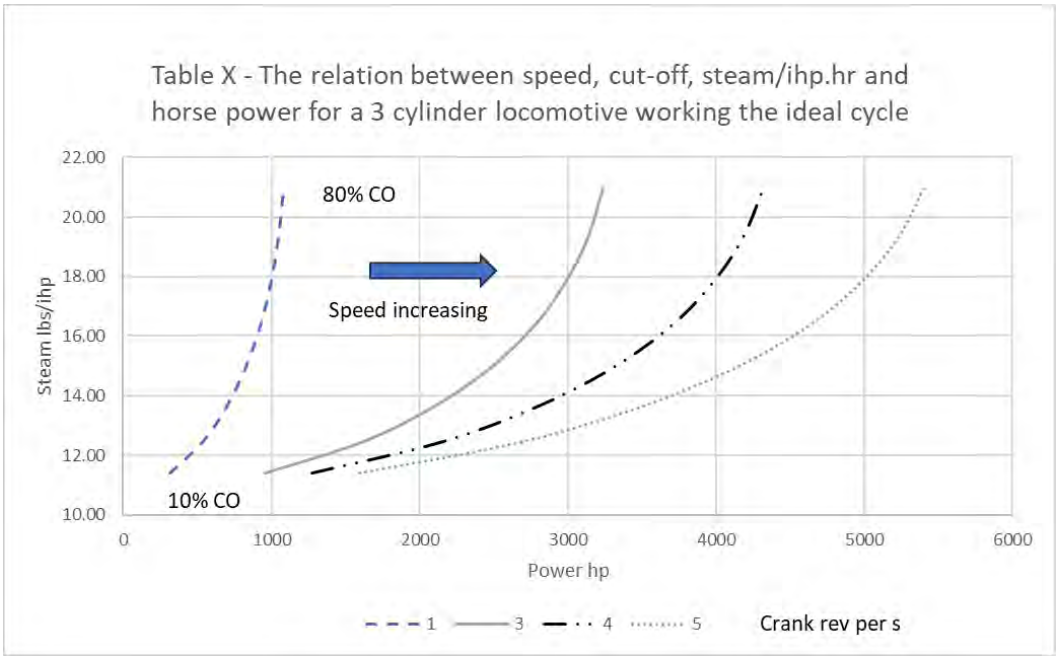


Figure 8

The calculation proceeds through the following stages:

Determining the weight of the cushion steam (Table II).
Calculating the weight of steam at cut-off (Tables III and IIIa).
Measuring the weight of the cylinder feed (Table IV).
Assessing final expansion and compression pressures (Tables V and VI).
Computing the work done in one complete cycle (Table VII).
Evaluating the work done per pound of cylinder feed (Table VIII).
Plotting curves that relate steam consumption and work performed for various degrees of compression and identifying the maximum values (D A Costa's Figure 4 now Figure 25 below).
Introducing a correction to account for differences in horsepower and locating the point of closure of the exhaust valve to achieve optimal efficiency (D A Costa's Figure 4 and Table IX).

The steps above were originally completed with laborious work with 6 figure log tables in the 1930's. Today they can be quickly completed with an Smath studio work sheet. (https://smath.com/en-us/view/smathstudio/summary) and exported the data to Excel for some graphics. Initially we will work with the parameters discussed in D A Costa's paper to verify the numbers.

CALCULATION DATA

$ClearanceVolume\% := 10$

$Steam_chest_Pressure := 180\ psi$ Initial Presure (Boiler Pressure gauge)

$Final_Pressure := 0\ psi$ Final Presure (Atmosheric Pressure guage)

$Steam_Temp := 650\ °F$ Temperature of steam after super heater

Figure 9 Smath worksheet

With identification to the point in the indicator diagram in figure 3 above.

DA Costa calculations are based on a cylinder with a swept volume of 100 cu ft.

$P_a := Steam_chest_Pressure$ See diagram above for identification

$P_b := Steam_chest_Pressure$

$P_d := Final_Pressure$

$P_e := Final_Pressure$

$Piston_Area_100 := 1\ ft^2$ Calculation for 100 cu ft.

$Cylinder_Length_100 := 100\ ft$

$Vol_dispacement_100 := Cylinder_Length_100 \cdot Piston_Area_100 = 100\ ft^3$

$V_a := \frac{ClearanceVolume\%}{100} \cdot Vol_dispacement_100 = 10\ ft^3$

$V_b(CutOff\%) := \frac{(ClearanceVolume\% + CutOff\%)}{100} \cdot Vol_dispacement_100$ Initial Volume in cylinder for a cylinder of 100 cu ft

$V_c := Vol_dispacement_100 \cdot \left(\frac{(100 + ClearanceVolume\%)}{100}\right) = 110\ ft^3$ Final Volume in cylinder

$ne := 1.2$ ne = 1.2 for expandsion

$nc := 1.3$ nc = 1.3 for compression

$V_d := V_c$ End of stroke.

$V_e(Exh_Valve_Close\%) := \frac{(Exh_Valve_Close\% + ClearanceVolume\%)}{100} \cdot Vol_dispacement_100$

$Exh_Valve_Close\% = 40$ $V_e(Exh_Valve_Close\%) = 50\ ft^3$ Volume of steam at atmospheric pressure

$V_f := V_a$

Figure 10 Smath worksheet

Table II Weight of cushion steam

The final expansion temperature of the steam is assumed to be 250 degrees F at 10% cut off and increase in a straight line to 350 degrees F at 80% cut off for a boiler pressure of 180 psi.

$$Temp_of_steam := (1.429 \cdot CutOff\% + 235.7) \cdot 1\ °F$$

$$Specific_Volume_at_compression := 29.65\ \frac{ft^3}{lb}$$

Figure 11 Smath worksheet

Superheated steam specific volume assuming the steam chest is at boiler pressure gauge and 650 degrees F of super heat. Steam table can be found online at https://www.spiraxsarco.com/resources-and-design-tools

For our requirements the table looks like this:

Pressure	0	psi gauge
Superheat Temperature	279	°F
	Calculate / Reset / Print	
Saturation Temperature	99.9743	°C
Degrees Superheat	37.2479	°C
Specific Enthalpy of Water (h_f)	418546	J/kg
Specific Enthalpy of Evaporation (h_{fg})	2.25647E06	J/kg
Specific Enthalpy of Superheated Steam (h)	2.75056E06	J/kg
Density of Steam	0.540240	kg/m³
Specific Volume of Steam (v)	29.6506	ft³/lb
Specific Entropy of Water (s_f)	1307.05	J/kg K
Specific Entropy of Evaporation (s_{fg})	6047.51	J/kg K
Specific Entropy of Superheated Steam (s)	7547.59	J/kg K
Specific Heat of Steam (c_v)	1501.76	J/kg K
Specific Heat of Steam (c_p)	1996.29	J/kg K
Speed of sound	496.826	m/s
Dynamic Viscosity of Steam	1.36841E-05	Pa s
Isentropic Coefficient (k)	1.31598	
Compressibility Factor of Steam	0.990291	

Figure 12 Steam Tables online

So, the values in D A Costa's paper can be checked for conformity with Exhaust Valve closure 40% before dead centre.

$$Wt_of_cushion_steam(Exh_Valve_Close\%) := \frac{V_e(Exh_Valve_Close\%)}{Specific_Volume_at_compression}$$

$$Exh_Valve_Close\% = 40 \qquad Wt_of_cushion_steam(Exh_Valve_Close\%) = 1.69\ lb$$

Figure 13 Smath worksheet

TABLE III The weight of steam at cut off

Inputs	Pressure and Superheat Temperature	
Output	Single Value / Table	
Pressure	180	psi gauge
Superheat Temperature	650	°F
	Calculate / Reset / Print	
Saturation Temperature	379.553	°F
Degrees Superheat	270.447	°F
Specific Enthalpy of Water (h_f)	352.838	Btu/lb
Specific Enthalpy of Evaporation (h_{fg})	845.354	Btu/lb
Specific Enthalpy of Superheated Steam (h)	1348.39	Btu/lb
Density of Steam	0.302086	lb/ft³
Specific Volume of Steam (v)	3.31032	ft³/lb
Specific Entropy of Water (s_f)	0.540999	Btu/lb °F
Specific Entropy of Evaporation (s_{fg})	1.00731	Btu/lb °F
Specific Entropy of Superheated Steam (s)	1.70430	Btu/lb °F
Specific Heat of Steam (c_v)	0.389392	Btu/lb °F
Specific Heat of Steam (c_p)	0.516576	Btu/lb °F
Speed of sound	598.805	m/s
Dynamic Viscosity of Steam	2.20065E-05	Pa s
Isentropic Coefficient (k)	1.29239	
Compressibility Factor of Steam	0.975023	

Figure 14 Steam Tables online

$$Specific_Volume_at_cut_off := 3.36\ \frac{ft^3}{lb}$$

$$Wt_of_steam_at_cut_off(CutOff\%) := \frac{V_b(CutOff\%)}{Specific_Volume_at_cut_off}$$

Mass of steam at cut off.

Figure 15 Smath worksheet

TABLE IV Weight of steam feed to cylinder

$$Wt_of_cylinder_feed_steam(CutOff\%, Exh_Valve_Close\%) := Wt_of_steam_at_cut_off(CutOff\%) - _Wt_of_cushion_steam(Exh_Valve_Close\%)$$

Figure 16 Smath worksheet

TABLEIV Weight (lb) of cylinder steam feed in lb per cycle for a cylinder of 100 ft^3.

	3	5	7	10	15	20	30	40	50	60	70	80	85
5	3.36	3.96	4.55	5.45	6.93	8.42	11.4	14.37	17.35	20.33	23.3	26.28	27.77
10	3.19	3.79	4.38	5.28	6.77	8.25	11.23	14.21	17.18	20.16	23.13	26.11	27.6
15	3.03	3.62	4.22	5.11	6.6	8.08	11.06	14.04	17.01	19.99	22.97	25.94	27.43
20	2.86	3.45	4.05	4.94	6.43	7.92	10.89	13.87	16.84	19.82	22.8	25.77	27.26
25	2.69	3.28	3.88	4.77	6.26	7.75	10.72	13.7	16.68	19.65	22.63	25.6	27.09
30	2.52	3.11	3.71	4.6	6.09	7.58	10.55	13.53	16.51	19.48	22.46	25.44	26.92
35	2.35	2.95	3.54	4.43	5.92	7.41	10.39	13.36	16.34	19.31	22.29	25.27	26.76
40	2.18	2.78	3.37	4.26	5.75	7.24	10.22	13.19	16.17	19.15	22.12	25.1	26.59
45	2.01	2.61	3.2	4.1	5.58	7.07	10.05	13.02	16	18.98	21.95	24.93	26.42
50	1.84	2.44	3.03	3.93	5.42	6.9	9.88	12.86	15.83	18.81	21.78	24.76	26.25
55	1.68	2.27	2.87	3.76	5.25	6.73	9.71	12.69	15.66	18.64	21.62	24.59	26.08
60	1.51	2.1	2.7	3.59	5.08	6.57	9.54	12.52	15.49	18.47	21.45	24.42	25.91

$$\frac{TABLEIV}{lb}$$

TABLE V Final Expansion

$$P_c(CutOff\%) := \frac{\left(V_b(CutOff\%)\right)^{ne}}{\left(V_c\right)^{ne}} \cdot \left(P_b + 14.7\ psi\right) - 14.7\ psi$$

Figure 17 Smath worksheet

TABLE VI Compression pressure

$$V_f := \frac{ClearanceVolume\%}{100} \cdot Vol_dispacement_100 = 10\ ft^3$$

$$P_f(Exh_Valve_Close\%) := \frac{\left(V_e(Exh_Valve_Close\%)\right)^{nc}}{\left(V_f\right)^{nc}} \cdot \left(P_e + 1\ atm\right) - 1\ atm$$

$$P_f(Exh_Valve_Close\%) = 104.4\ psi$$

Figure 18 Smath worksheet

TABLE VII Work done per lb of steam in one stroke

$$Work_admission(CutOff\%) := \left(P_a + 1\ atm\right) \cdot Cylinder_Length_100 \cdot Piston_Area_100 \cdot \frac{CutOff\%}{100}$$

Figure 19 Smath worksheet

Work done during expansion between points b and c on the indicator diagram.

$$Work_Expandsion(CutOff\%) := \frac{(P_b + atm)\cdot V_b(CutOff\%) - (P_c(CutOff\%) + 1\ atm)\cdot V_c}{(ne - 1)}$$

Figure 20 Smath worksheet

Work done during exhaust between points d and e on the indicator diagram.

$$Work_exhaust(CutOff\%) := (P_d + 1\ atm)\cdot Vol_dispacement_100\cdot\left(-\frac{(100 - Exh_Valve_Close\%)}{100}\right)$$

$$Work_exhaust(CutOff\%) = -1.2697\cdot 10^5\ ft\ lbf$$

Figure 21 Smath worksheet

Work done during expansion between points e and f on the indicator diagram

$$Work_Comp(Exh_Valve_Close\%) := \frac{(P_e + 1\ atm)\cdot V_e(Exh_Valve_Close\%) - (P_f(Exh_Valve_Close\%) + 1\ atm)\cdot V_f}{(nc - 1)}$$

$$Work_Comp(Exh_Valve_Close\%) = -2.1891\cdot 10^5\ ft\ lbf$$

Figure 22 Smath worksheet

The total work done is the summation

$$cycle_work_done_seg(CutOff\%, Exh_Valve_Close\%) := \begin{bmatrix} Work_admission(CutOff\%) \\ Work_Expandsion(CutOff\%) \\ Work_exhaust(CutOff\%) \\ Work_Comp(Exh_Valve_Close\%) \end{bmatrix}$$

$$cycle_work_done(CutOff\%, Exh_Valve_Close\%) := \sum cycle_work_done_seg(CutOff\%, Exh_Valve_Close\%)$$

For the example in the paper

$CutOff\% := 30 \qquad Exh_Valve_Close\% = 40$

$$\frac{cycle_work_done(CutOff\%, Exh_Valve_Close\%)}{1000} = 1522\ ft\ lbf$$

Sum up all segments of the indicator diagram.

Figure 23 Smath worksheet

TABLE VII. Work done x1000 for 1 revolution of a cylinder of 100 ft^3. Across the columns is Cut off % and down the rows Exhaust valve closure before Dead centre.

	3	5	7	10	15	20	30	40	50	60	70	80	85
5	577.2	690.9	798.6	950	1178.9	1382.7	1727.7	2003.5	2221.6	2389.8	2514	2598.6	2627.3
10	558.3	672	779.6	931	1160	1363.8	1708.8	1984.6	2202.6	2370.8	2495	2579.7	2608.3
15	535.1	648.8	756.5	907.9	1136.8	1340.6	1685.6	1961.4	2179.5	2347.7	2471.9	2556.5	2585.2
20	508.3	622	729.6	881	1110	1313.8	1658.8	1934.6	2152.6	2320.8	2445	2529.7	2558.3
25	478.3	592	699.6	851	1079.9	1283.7	1628.8	1904.6	2122.6	2290.8	2415	2499.7	2528.3
30	445.4	559.1	666.7	818.1	1047.1	1250.9	1595.9	1871.7	2089.7	2257.9	2382.1	2466.8	2495.4
35	409.9	523.6	631.2	782.6	1011.6	1215.4	1560.4	1836.2	2054.2	2222.4	2346.6	2431.3	2459.9
40	372	485.7	593.3	744.7	973.7	1177.5	1522.5	1798.3	2016.3	2184.6	2308.7	2393.4	2422.1
45	331.9	445.6	553.2	704.6	933.5	1137.3	1482.4	1758.2	1976.2	2144.4	2268.6	2353.3	2381.9
50	289.6	403.3	511	662.4	891.3	1095.1	1440.1	1715.9	1934	2102.2	2226.4	2311.1	2339.7
55	245.5	359.2	466.8	618.2	847.1	1050.9	1396	1671.8	1889.8	2058	2182.2	2266.9	2295.5
60	199.4	313.1	420.8	572.2	801.1	1004.9	1349.9	1625.7	1843.8	2012	2136.2	2220.8	2249.5

$\frac{Cycle\ WD}{1000\ ft\ lbf}$

$$Specific_WD(CutOff\%, Exh_Valve_Close\%) := \frac{cycle_work_done(CutOff\%, Exh_Valve_Close\%)}{Wt_of_cylinder_feed_steam(CutOff\%, Exh_Valve_Close\%)}$$

TABLE VIII. Work done x1000 ft.lb per lb of steam for 1 revolution of a cylinder of 100 ft^3. Across the columns is Cut off % and down the rows Exhaust valve closure before Dead centre.

	3	5	7	10	15	20	30	40	50	60	70	80	85
5	171.6	174.6	175.4	174.4	170	164.2	151.6	139.4	128	117.6	107.9	98.9	94.6
10	174.8	177.3	177.8	176.4	171.5	165.2	152.2	139.7	128.2	117.6	107.8	98.8	94.5
15	176.9	179.2	179.4	177.7	172.3	165.8	152.4	139.7	128.1	117.4	107.6	98.5	94.2
20	177.9	180.2	180.3	178.4	172.7	166	152.3	139.5	127.8	117.1	107.3	98.2	93.8
25	177.9	180.3	180.4	178.4	172.5	165.7	151.9	139	127.3	116.6	106.7	97.6	93.3
30	176.8	179.5	179.7	177.8	171.9	165.1	151.2	138.3	126.6	115.9	106.1	97	92.7
35	174.4	177.8	178.3	176.5	170.8	164	150.2	137.4	125.7	115.1	105.3	96.2	91.9
40	170.5	174.9	176	174.6	169.2	162.6	149	136.3	124.7	114.1	104.4	95.4	91.1
45	164.9	170.8	172.7	172	167.2	160.8	147.5	135	123.5	113	103.3	94.4	90.2
50	157.1	165.3	168.4	168.7	164.6	158.6	145.8	133.5	122.2	111.8	102.2	93.3	89.1
55	146.5	158.2	162.9	164.5	161.5	156	143.8	131.8	120.7	110.4	101	92.2	88
60	132.4	149	156	159.4	157.8	153	141.5	129.9	119	108.9	99.6	90.9	86.8

$$\frac{Specific_Cycle_WD}{1000\ \frac{ft\ lbf}{lb}}$$

Figure 24 Smath worksheet

In D A Costa paper figure 4 shows the maxima on the curve plot versus the work done for various Exhaust valve closure points. This is repeated here as Figure 25 below.

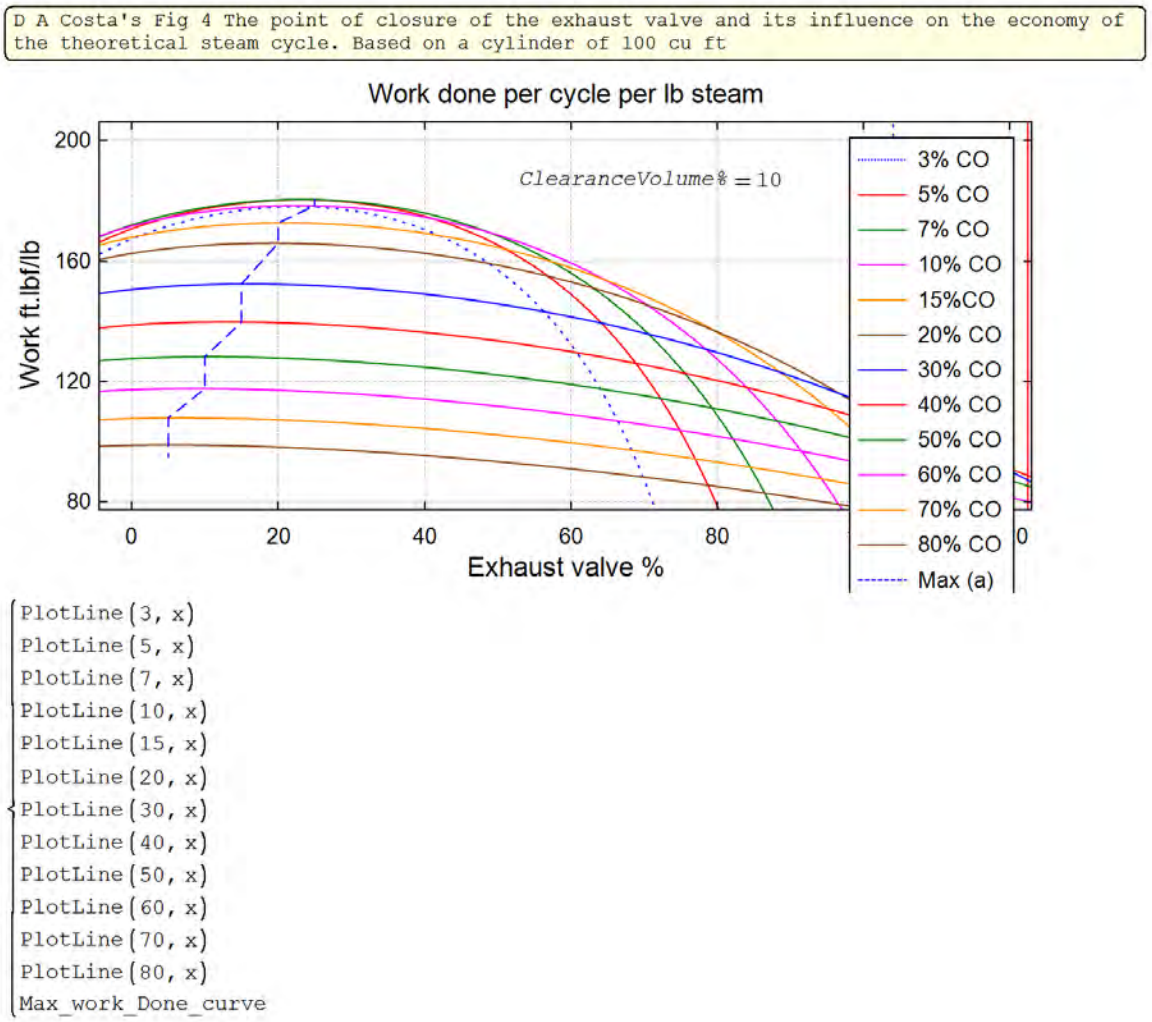


Figure 25 Chart of maxima power 180 psi, 650 deg F

Extracting the locus of maxima for Cut off and Exhaust valve closure points.

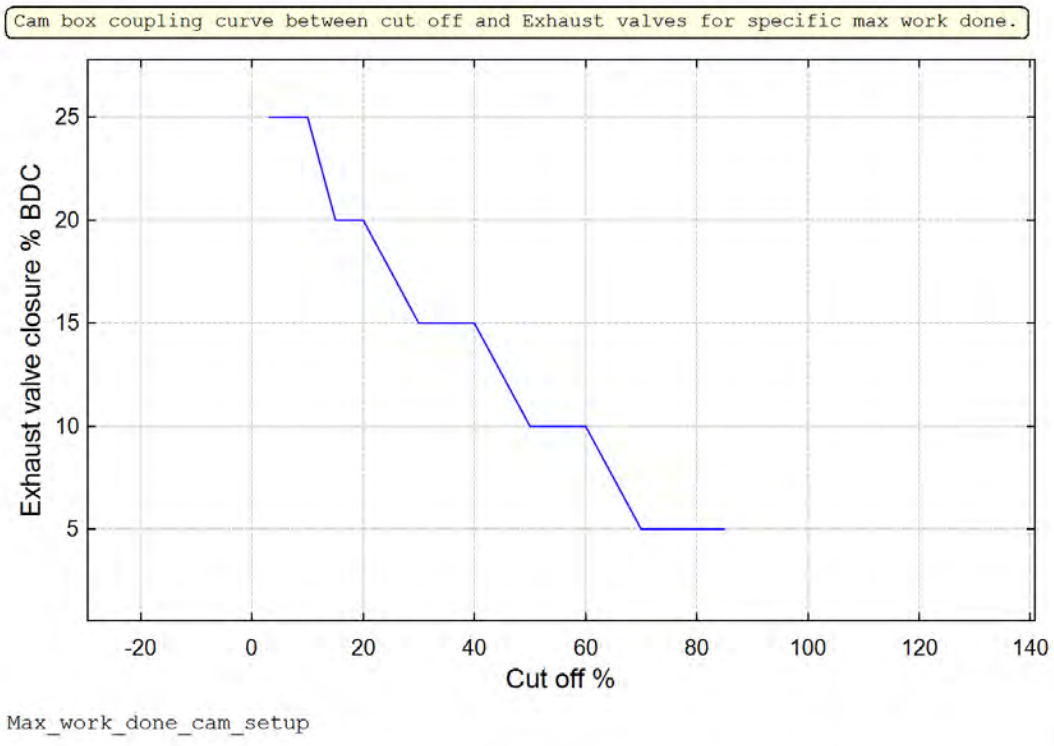


Figure 26 Valve sequence programming for 180 psi, 650 deg F

The operating envelope for the poppet valve gears is implied by D A Costa's calculations. He did not publish this chart, possibly due to the scale of computational effort required. Today a modern computer makes easy work of a task that may have been impossible with log tables and hand calculations.

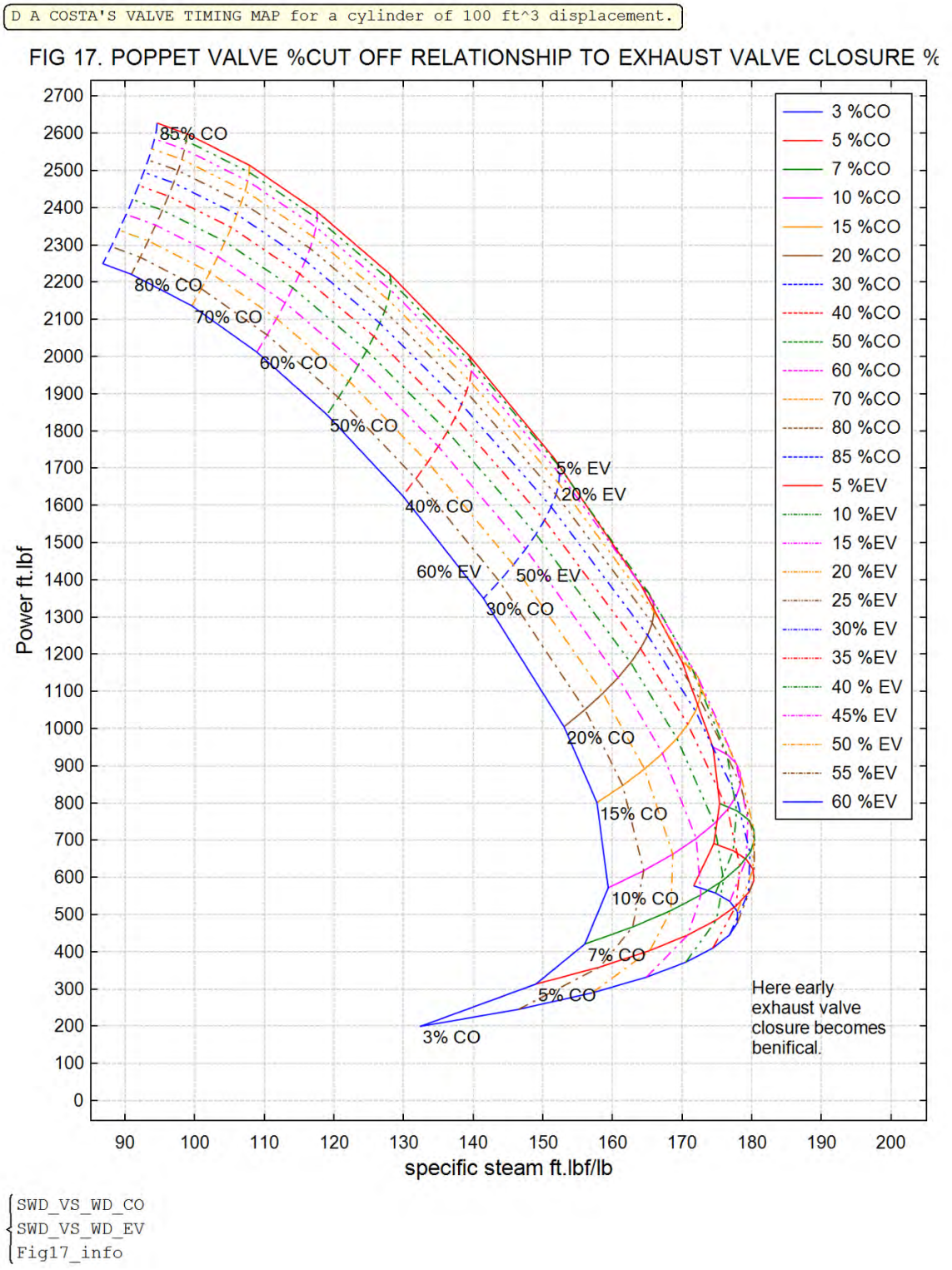
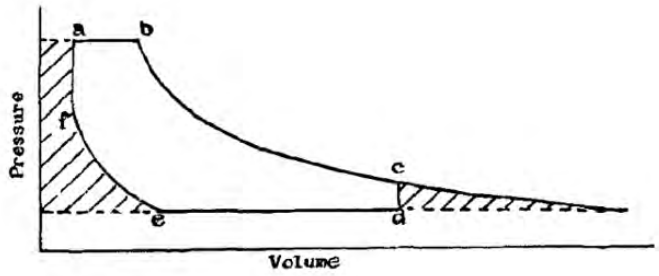


Figure 27 Valve gear operating space for 180 psi, 650 deg f

The curve of constant 85% cut off increases power as the Exhaust valve is closed later in the cycle, whereas the curve of constant 3% cut off increases specific steam, or efficiency significantly as the exhaust valve closes earlier in the cycle. For the best performance the valve gear programming should follow the cut off and exhaust valve locus on the far-right hand edge of the operating envelope. For most of the time the exhaust valve closure is 5% BTDC, but at 20% Cut off the exhaust valve should begin closure much earlier. For the earliest condition up to 30 % BTDC. The blue dashed vertical curve meets the Blue solid 3% cut off locus at specific steam flow 178 ft lbf/lb, and power 500 ft.lbf.

D A Costa's IDEAL indicator diagram.



Define functions to plot the polytropic curve lines segments of the indicator diagram.

$$Expand(x) := \text{if}\left[x \geq \frac{V_b(CutOff\%)}{ft^3}\right] \wedge \left[x < \frac{V_c}{ft^3}\right] \quad \frac{\left(V_b(CutOff\%)\right)^{ne}}{\left(x\ ft^3\right)^{ne}} \cdot \left(P_b + 1\ atm\right) - 1\ atm \over psi$$

else
break

$$Compress(x) := \text{if}\left[x \geq \frac{V_f}{ft^3}\right] \wedge \left[x < \frac{V_e(Exh_Valve_Close\%)}{ft^3}\right] \quad \frac{\left(V_e(Exh_Valve_Close\%)\right)^{nc}}{\left(x\ ft^3\right)^{nc}} \cdot \left(P_e + 1\ atm\right) - 1\ atm \over psi$$

Indicator fixed points.

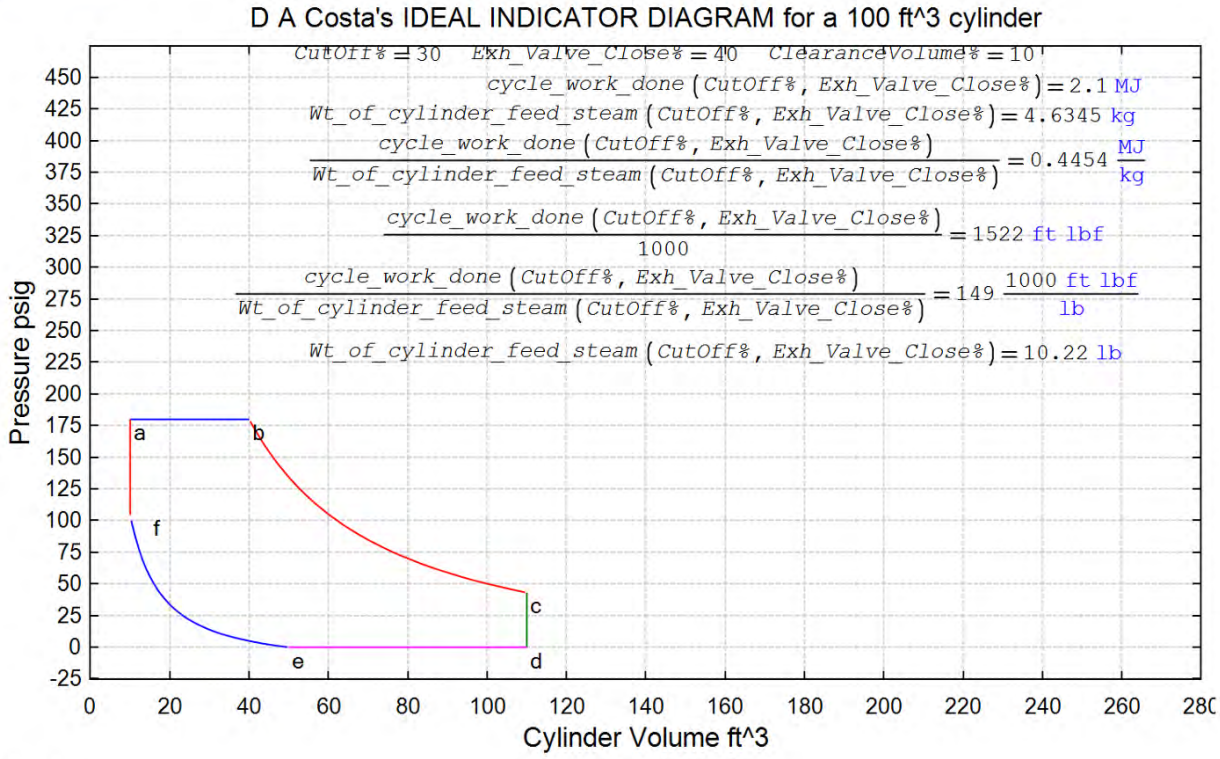
$$Fill := \begin{bmatrix} \frac{V_a}{ft^3} & \frac{P_a}{psi} \\ \frac{V_b(CutOff\%)}{ft^3} & \frac{P_b}{psi} \end{bmatrix} \qquad Exhaust := \begin{bmatrix} \frac{V_d}{ft^3} & \frac{P_d}{psi} \\ \frac{V_e(Exh_Valve_Close\%)}{ft^3} & \frac{P_e}{psi} \end{bmatrix}$$

$$Release := \begin{bmatrix} \frac{V_c}{ft^3} & \frac{P_c(CutOff\%)}{psi} \\ \frac{V_d}{ft^3} & \frac{P_d}{psi} \end{bmatrix}$$

$$Refill := \begin{bmatrix} \frac{V_f}{ft^3} & \frac{P_f(Exh_Valve_Close\%)}{psi} \\ \frac{V_a}{ft^3} & \frac{P_a}{psi} \end{bmatrix}$$

Figure 28 Smath Ideal cycle calculation

Figure 5




```
Fill
Expand(x)
Release
Exhaust
if x = V_c
  P_d
if (x ≥ V_e(Exh_Valve_Close%)) ∧ (x < V_c)
  P_e
Compress(x)
Refill
Indicator_info
```

Figure 29 Ideal Cycle 180 psi

The data in D A Costa's paper relates to an engine with a boiler pressure of 180 psi, by the time the Duke of Gloucester was designed in the 1950's pressures had converged on 250 psi. Repeating the calculations with more modern data, and a 3-cylinder engine configuration.



Figure 30 71000 Duke of Gloucester 4-6-2 3 cyl with British Caprotti valve gear

CALCULATION DATA

$ClearanceVolume\% := 11$	Clearance volume. 71000 Duke has 11%
$Steam_chest_Pressure := 240\ psi$	Initial Pressure (Boiler Pressure gauge)
$Final_Pressure := 0\ psi$	Final Pressure (Atmospheric Pressure gauge)
$Steam_Temp := 750\ °F$	Temperature of steam after super heater
$Exh_Valve_Close\% := 50$	Closure of the exhaust valve % of stroke from dead centre.
$CutOff\% := 7$	Cut off selected
$No_cylinders := 3$	
$Piston_stroke := 28\ in$	Data for 71000 Duke of Gloucester. BR Technical bulletin No15 BR Class 8 Performance and efficiency.
$Piston_Dia := 18\ in$	
$Piston_rod_Dia := 4\ in$	
$Driving_wheel_dia := 74\ in$	

Figure 31 Smath worksheet

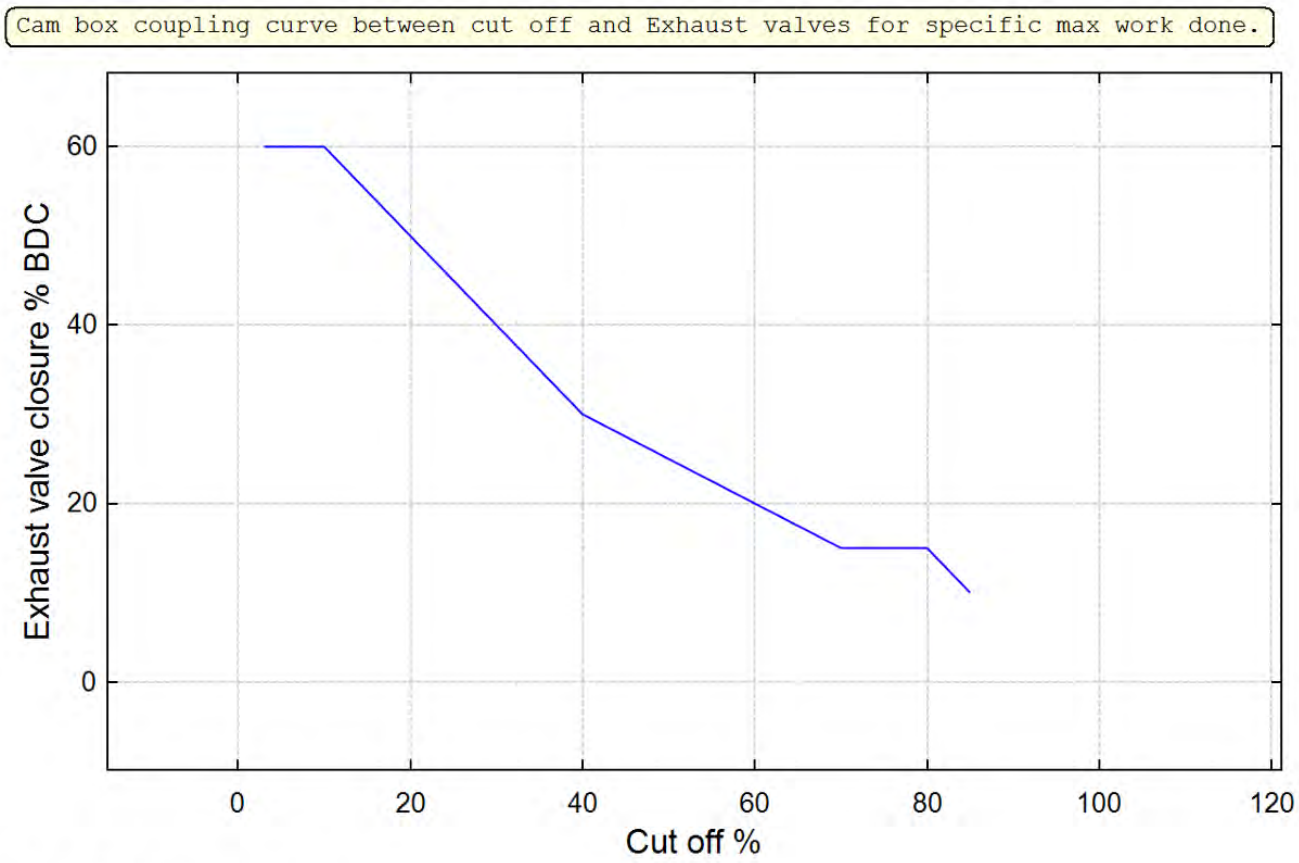


Figure 32 Valve sequence programming for 240 psi, 750 deg F

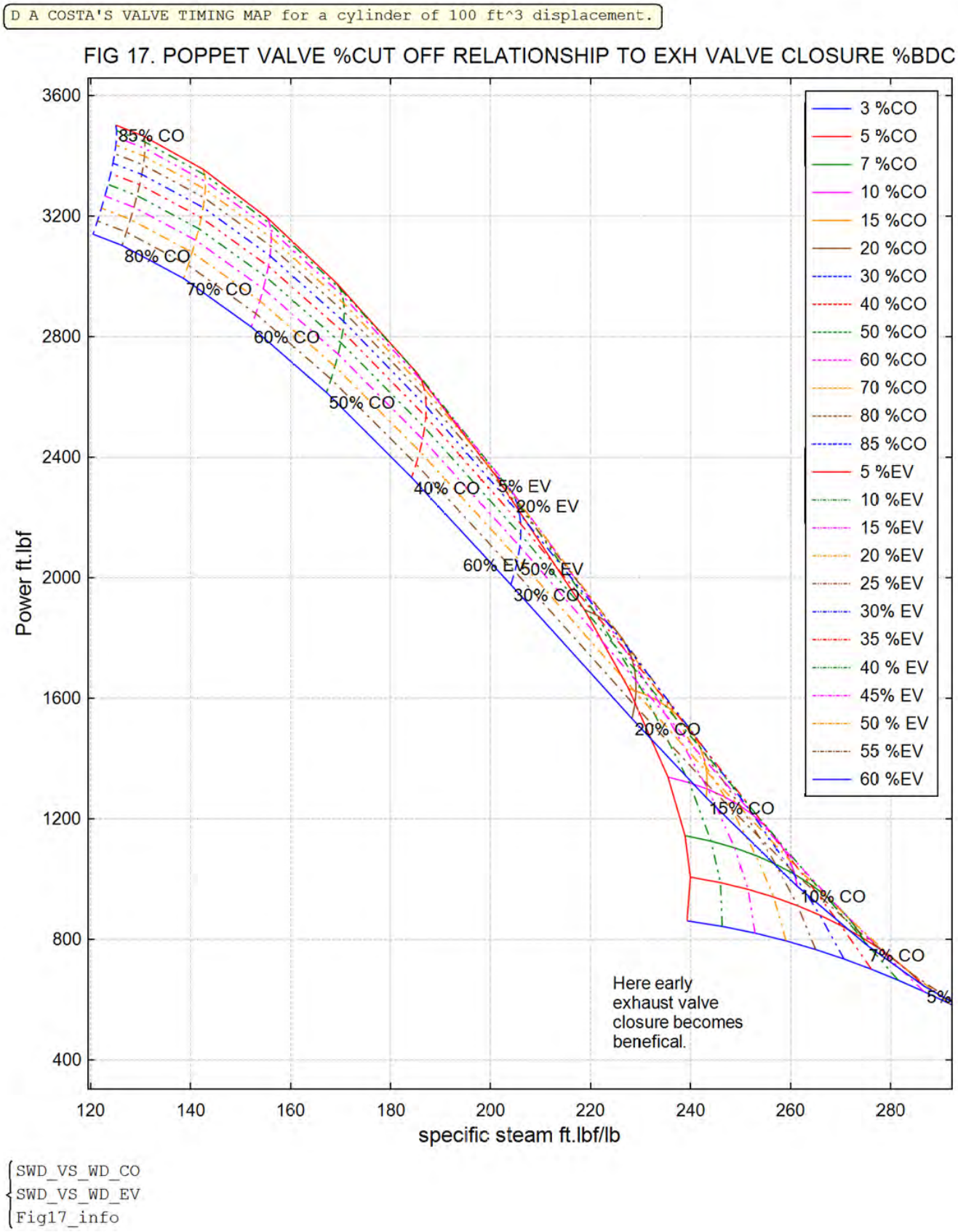


Figure 33 Valve gear operating space for 240 psi, 750 deg F

The increase in boiler pressure from 180 to 240 psi and increase in super heat has extended the operating envelope on the specific steam axis. The curve of constant 5% cut off now effects a reduction in power but significant saving in specific steam or operational efficiency as the exhaust valve is closed earlier in the cycle. This is the zone where express locomotives work; a link gear is unable to operate at 5% cut off due to piston valve geometry. For a freight locomotive operating at low speed, and higher haul effort, using cut off greater than 25% there is no benefit to be gained from the application with poppet valve gear in terms of steam economy, but there may be operational shed service time savings.

$ClearanceVolume\% = 11$

$No_cylinders = 3$

$Piston_stroke = 28\ in$

$Piston_Dia = 18\ in$

$Piston_rod_Dia = 4\ in$

$Driving_wheel_dia = 74\ in$

Data from the first page repeated here.

$Steam_rate := 21740\ \frac{lb}{hr}$

Rate of boiler steam production selected to match 1957 test results

$RS := 75\ mph$

Inpect point on Rail speed for graphic. Fig 10. matches 1957test data

$$Total_piston_displacemet := \left(\frac{Piston_Dia}{2}\right)^2 \cdot \pi \cdot Piston_stroke = 4.1\ ft^3$$

$$Piston_rod_displacement := \left(\frac{Piston_rod_Dia}{2}\right)^2 \cdot \pi \cdot Piston_stroke = 0.2\ ft^3$$

$$Net_displacement := No_cylinders \cdot \left(\left(2 \cdot Total_piston_displacemet - Piston_rod_displacement\right)\right)$$

$$Net_displacement = 24.1\ ft^3$$

$$Volume_Cylinder_entry := Specific_Volume_at_cut_off \cdot Steam_rate = 20.2907\ \frac{ft^3}{s}$$

$$CUT_OFF_RAIL\left(RS\right) := \frac{Volume_Cylinder_entry}{Net_displacement \cdot \frac{1}{Driving_wheel_dia \cdot \pi} \cdot RS}$$

$$Exhaust_valve_rail\left(RS\right) := \frac{cinterp\left(col\left(TABLEIX, 1\right), col\left(TABLEIX, 2\right), CUT_OFF_RAIL\left(RS\right) \cdot 100\right)}{100}$$

$$Exhaust_valve_rail\left(RS\right) = 55.2\ \%$$

$$Pwr_Table_VII := Cycle_WD \cdot \frac{Net_displacement}{100\ ft^3}$$

2D Interpolation of data in Power_Table_VII using cut off and exhaust valve close point to find work done per rev.

$$Power_Rail_1957 := InterpBilinear\left(\frac{CO}{100}, \frac{EV}{100}, Pwr_Table_VII, 0.07, 0.50\right) \cdot \frac{RS}{\pi \cdot Driving_wheel_dia}$$

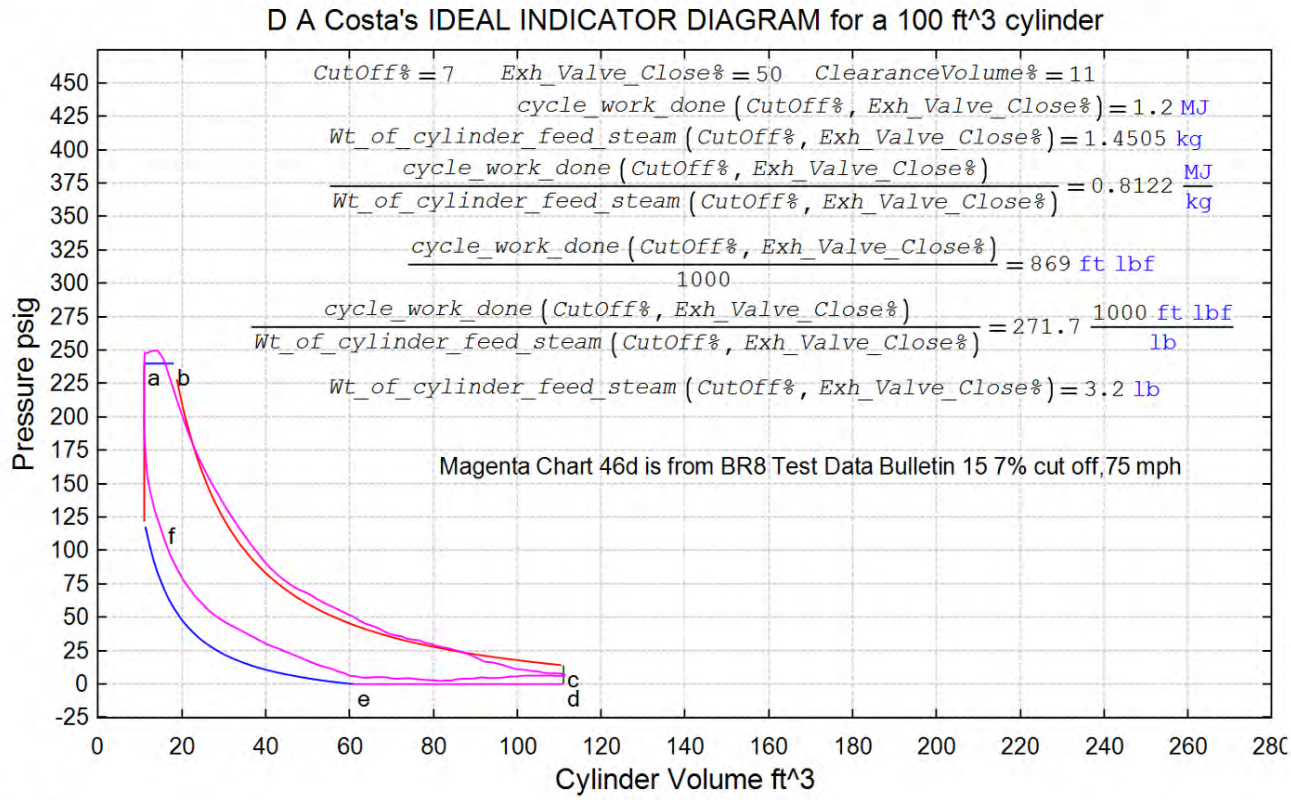
$Power_Rail_1957 = 2164\ hp$

$$Actual := \frac{1712\ hp}{Power_Rail_1957} = 79\ \%$$

In 1957 the British Railway Commission published Bulletin 15. Performance and efficiency tests of 3 cylinder 4-6-2 express passenger locomotive No 71000. At 250 psi and a steaming rate of 21740 lb/hr, 7% cut off, the Swindon test plant measured an indicated horse power of 1712. This is the magenta curve inside the calculated diagram.

Figure 34 Smath worksheet with data for Chart 46d

Figure 5




```
Fill
Expand(x)
Release
Exhaust
if x = V_c
  P_d
if (x ≥ V_e(Exh_Valve_Close%)) ∧ (x < V_c)
  P_e
Compress(x)
Refill
Indicator_info
Chart_46D_7PC_FWD
```

Figure 35 Ideal cycle vs measured data Chart 46d

In 1957 the British Railway Commission published Bulletin 15. Performance and efficiency tests of 3-cylinder 4-6-2 express passenger locomotive No 71000. At 250 psi and a steaming rate of 21740 lb/hr, 7% cut off, the Swindon test plant measured an indicated horsepower of 1712. This is the magenta curve inside the calculated diagram. The power level measured on the Swindon test plant was 79% of the figure given by D A Costa's Ideal cycle.

Now changing the steaming rate to 30000 lb/hr to generate the graphs on traction data in D A Costa's report.

$Steam_rate := 30000 \frac{lb}{hr}$

$Volume_Cylinder_entry := Specific_Volume_at_cut_off \cdot Steam_rate = 28 \frac{ft^3}{s}$

$Speed_revs_road := \frac{Volume_Cylinder_entry}{Net_displacement} = 1.1604\ Hz$

Full cut off
$CO_{13} = 85$

$Full_cut_off_Speed := \frac{Speed_revs_road \cdot \pi \cdot Driving_wheel_dia}{\frac{CO_{13}}{100}} = 8.0614 \frac{m}{s}$

$Full_cut_off_Speed = 18.03\ mph$

Need an array of speeds from this to 90 mph

$RS := 75\ mph$

Inpect point on Rail speed for graphic. Fig 10

$CUT_OFF_RAIL(RS) := \frac{Volume_Cylinder_entry}{Net_displacement \cdot \frac{1}{Driving_wheel_dia \cdot \pi} \cdot RS}$

$Exhaust_valve_rail(RS) := \frac{cinterp(col(TABLEIX, 1), col(TABLEIX, 2), CUT_OFF_RAIL(RS) \cdot 100)}{100}$

$Exhaust_valve_rail(RS) = 49.6\ \%$

Determine Exhaust valve for rail speed.

$Pwr_Table_VII := Cycle_WD \cdot \frac{Net_displacement}{100\ ft^3}$

2D Interpolation of data in Power_Table_VII using cut off and exhaust valve close point to find work done per rev.

$Work_done_per_rev(RS) := InterpBilinear\left(\frac{CO}{100}, \frac{EV}{100}, Pwr_Table_VII, CUT_OFF_RAIL(RS), Exhaust_valve_rail(RS)\right)$

$Work_done_per_rev(RS) = 3.9612 \cdot 10^5\ ft\ lbf$

$Work_done_per_rev(RS) = 5.3707 \cdot 10^5\ J$

$Power_rail(RS) := Work_done_per_rev(RS) \cdot \frac{RS}{\pi \cdot Driving_wheel_dia}$

$Force(RS) := \frac{Power_rail(RS)}{RS}$

$Power_rail(RS) = 4089\ hp$

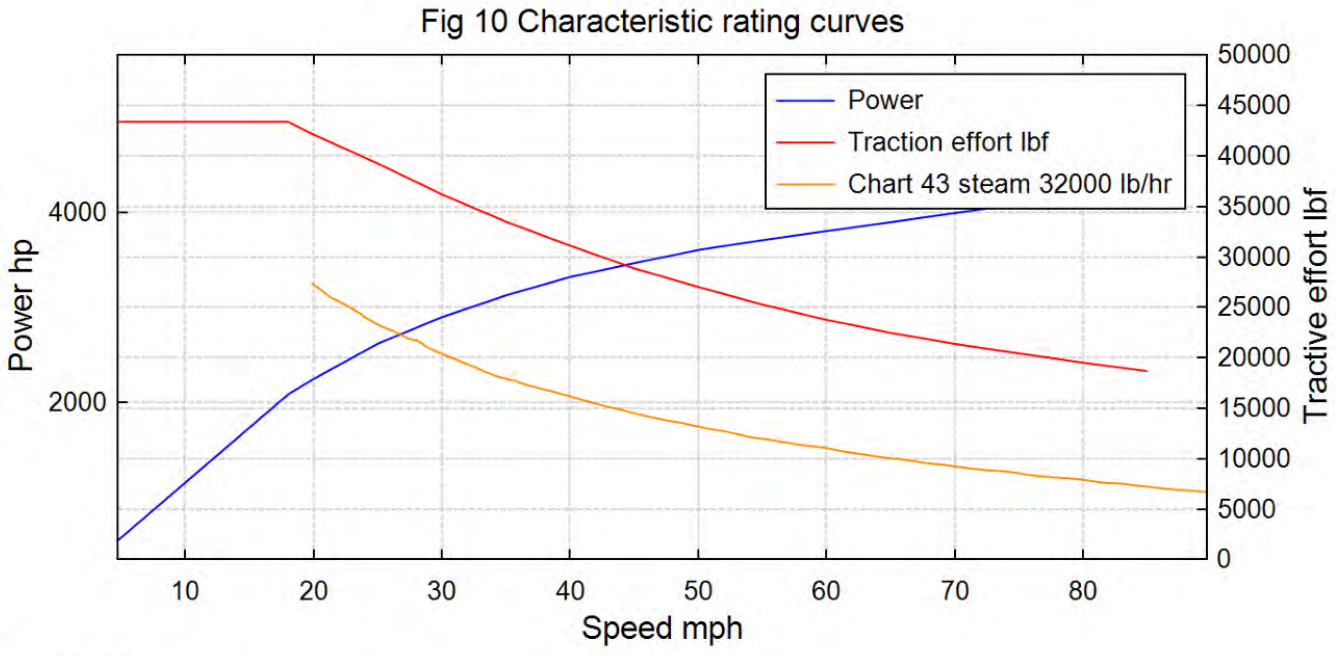
$Force(RS) = 20447\ lbf$

Figure 36 Smath worksheet for Traction curves

```
"Fig 10 Characteristic rating curves for steam  loco"
"Steam flow limited straight line segment"
Fig10_Line_Segment_power := [ 0 mph   0 hp ; Full_cut_off_Speed mph   Power_rail(Full_cut_off_Speed) hp ]
Fig10_Line_Segment_traction := [ 0 mph   Force(Full_cut_off_Speed) lbf ; Full_cut_off_Speed mph   Force(Full_cut_off_Speed) lbf ]
fig10_CO 1 := CO length(CO) %
fig10_EV 1 := EV 1 %
"Constant steam flow curve segment"
"Generate a vector for x axis speed"
Speed_mph := stack( Full_cut_off_Speed mph , [ Full_cut_off_Speed mph |5 , Full_cut_off_Speed mph |5 + 5..85 ] )
"Generate data for figure 10"
for cnt ∈ [1..length(Speed_mph)]
    Fig10_Power cnt := Power_rail(Speed_mph cnt · 1 mph)
    fig10_TE cnt := Force(Speed_mph cnt · 1 mph)
    fig10_CO cnt+1 := CUT_OFF_RAIL(Speed_mph cnt · 1 mph)
    fig10_EV cnt+1 := Exhaust_valve_rail(Speed_mph cnt · 1 mph)
```

Figure 37 Smath worksheet for generating Figure 10.

The 1957 data published as Chart 43 gives the actual measurements for a steaming rate of 32000 lb/hr. These are about 50% of the values given by D A Costa's ideal cycle based on the power given by indicated horsepower, using the optimum maximum valve timing settings.




```
{ Fig10_Line_Segment_power
  Fig10_Line_Segment_traction
  augment(Speed_mph, Fig10_Power hp)
  augment(Speed_mph, fig10_TE lbf)
  Chart_43_32K_Traction
```

Figure 38 D A Costa's Traction curves

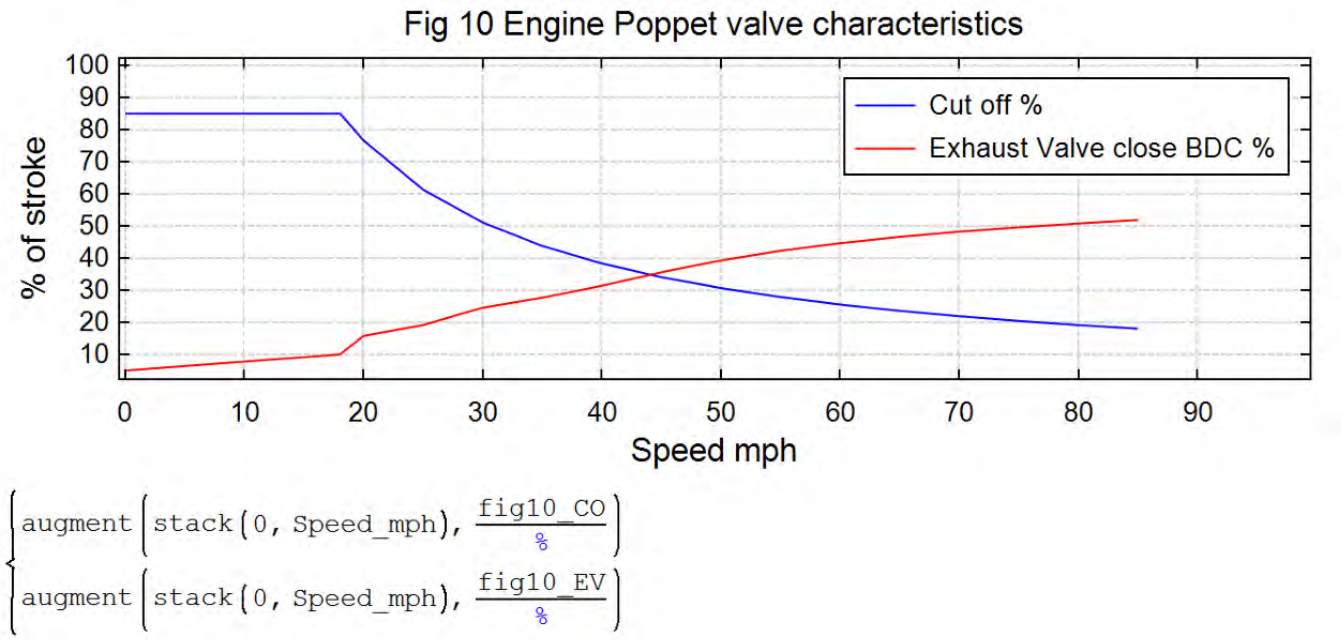


Figure 39 Optimised speed versus valve timing

Design and Functionality

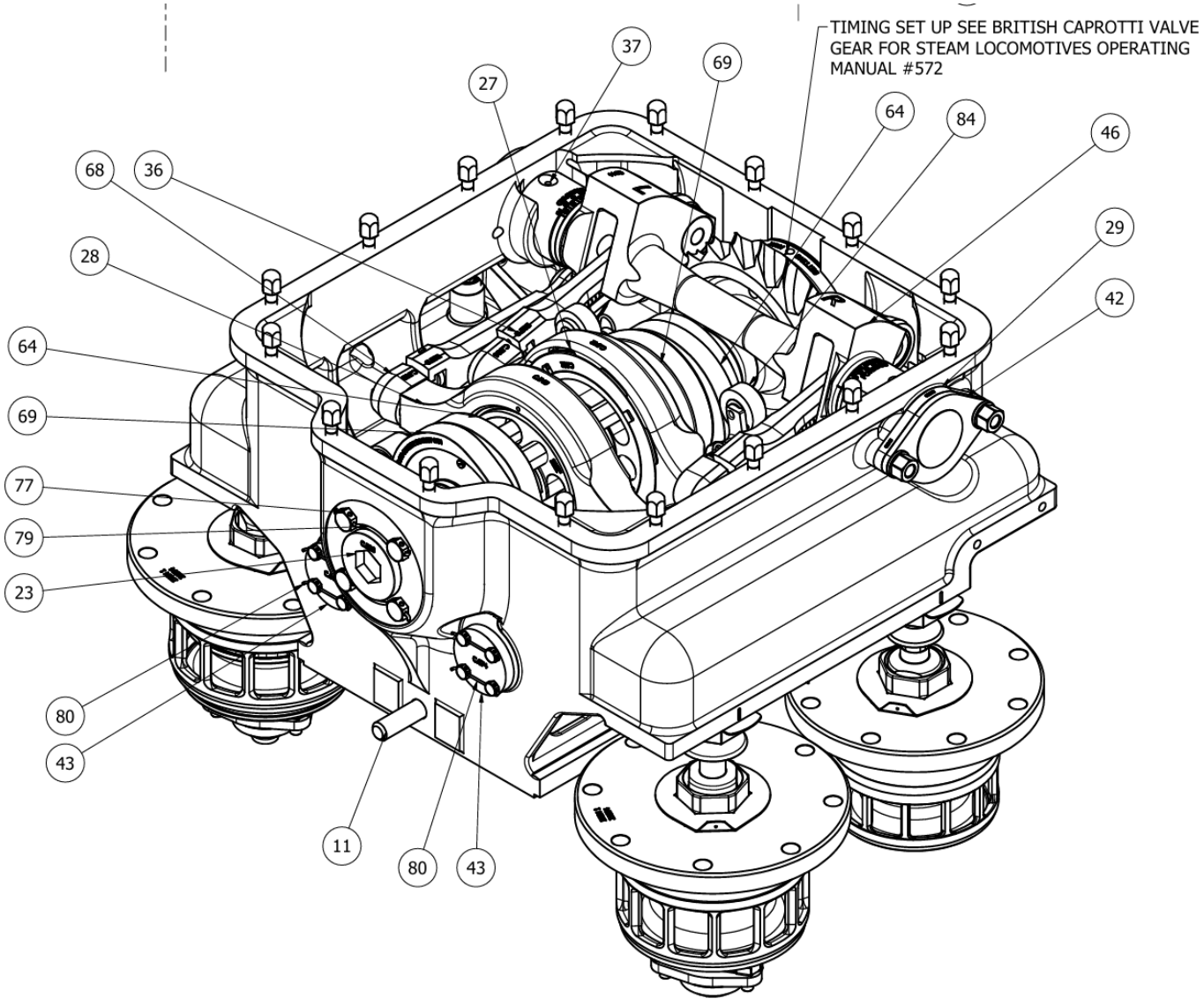


Figure 40 Caprotti Valve gear Cam box

Poppet valves, which are commonly found in internal combustion engines, control the flow of gases by opening and closing at precise intervals. In Caprotti's innovative design, these valves were adapted for use in steam locomotives. By employing camshafts to regulate valve movement, Caprotti's system enabled more accurate and efficient steam distribution within the cylinders. This improvement reduced energy losses and enhanced overall engine performance. The cambox; a housing for the camshaft and mechanism, was mounted on top of the locomotive's cylinder casting, with the valves positioned in passages directly below the mounting surface. The cambox contained two cams which were moved relative to each other by the driver cut off controls. The two cam profiles were summed together to produce a new variable profile to actuate the valves.

An alternative to traditional valve gear is the Rotary Cam (RC) poppet valve gear developed by ALE. Initially, Caprotti Valve Gear Ltd and ALE were competitors. However, in 1936, Caprotti acquired ALE and adopted the ALE name for future business activities. From that point onward, Caprotti equipment was marketed in preference to the RC valve gear.

A unique challenge arose when the Federated Malay States Railway required valve gear for its metre-gauge, three-cylinder 4-6-2 locomotives. The standard Caprotti gear could not be fitted, prompting a redesign of the RC cam valve system. This redesign incorporated improvements such as updated modern cam profiles, smoother valve dynamics, and revised timing events. The final batch of engines built by North British, equipped with this updated system, proved successful.

The ALE RC gear features a series of cams, each providing a different cut-off, stacked side by side on the camshaft. The desired cut-off cam is selected by moving the camshaft assembly along its axis of rotation using a vertically rising stirrup lever. While the worm drive gear remains fixed, it transmits motion to the camshaft via internal splines.

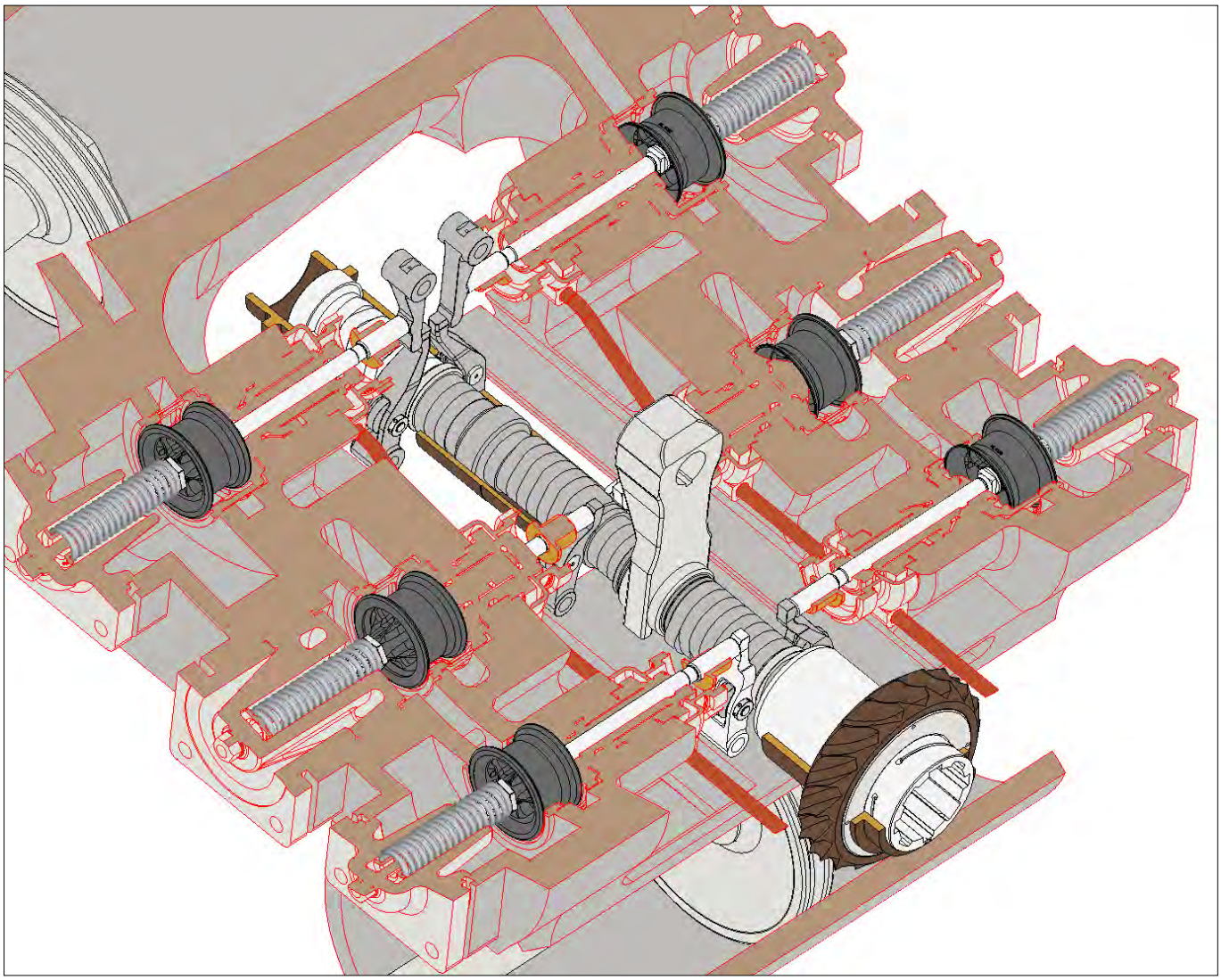


Figure 41 FMSR Metre gauge 4-6-2 564 Class 3 Cyl RC poppet valve implementation

The Federated Malay States Railways (FMSR) Class O1 4-6-2 locomotives were part of the famous Pacific type engines introduced for express passenger services in British Malaya. These locomotives hauled prestigious trains like the Penang Mail and were considered the top-link express engines of their time.

Several members of the class survive in preservation, but the most notable preserved example from the original O1 series is:

562.04 “Dungun” – Currently under restoration and displayed at Bukit Mertajam Railway Station

Other preserved locomotives from later batches include:

564.21 “Selama” – Port Dickson Army Museum.
564.34 “Pekan” – Royal Pahang Polo Club.
564.36 “Temerloh” – Old Johor Bahru Railway Station

Advantages

The poppet valve gear provided several advantages over traditional slide valves:

Reduced Friction: The cam-operated mechanism minimized direct contact between moving parts, significantly reducing friction and wear being enclosed in an oil bath. The power required to drive the valve gear is 2~3 HP, significantly less than link gear.

Improved Steam Flow: The design facilitated a smoother and more direct steam flow, enhancing engine efficiency.

Enhanced Durability: With reduced wear and tear, the poppet valve gear increased the longevity of locomotive engines between valve inspection.

Better Control: The precise camshaft control allowed for more accurate valve timing, optimizing engine performance under various operating conditions.

Standardisation allowed common valve gear parts different engines.

Indicator Diagrams

The contribution of D. A. Costa paper to steam locomotive engineering was his methods of design analysis and utilization of indicator diagrams in locomotive design stage. These diagrams are graphical representations of the pressure inside a steam engine's cylinder over the course of its cycle, providing valuable insights into engine performance and efficiency at the drawing board.

Importance of Indicator Diagrams

Indicator diagrams serve several critical functions in steam engine analysis:

Performance Assessment: By analysing the pressure variations within the cylinder, engineers can assess the efficiency of steam usage and identify potential areas for improvement.

Fault Diagnosis: Deviations from expected pressure patterns can indicate mechanical issues or inefficiencies, allowing for timely maintenance and repairs.

Optimization: These diagrams help engineers fine-tune engine parameters to achieve optimal performance, balancing power output and fuel consumption.

Legacy and Influence

The contributions of D. A. Costa to locomotive poppet valve gear and indicator diagrams have left a legacy in the field of steam engineering. His innovative designs and analytical methods paved the way for more efficient and reliable steam locomotives, which played a crucial role in the industrialization and economic development of the 19th and early 20th centuries.

Costa's work continues to be studied and referenced by engineers and historians, highlighting the enduring impact of his contributions. His advancements in valve gear technology and performance analysis remain relevant in the context of modern engineering, demonstrating the timeless value of his innovations.

In conclusion, His pioneering efforts in design and analysis have not only enhanced the efficiency and functionality of steam engines but also set the stage for future developments in the field. Costa's legacy serves as a testament to the importance of innovation and precision in engineering, inspiring future generations to continue exploring and advancing the boundaries of technology.

- The findings suggest that early exhaust valve closure can enhance efficiency, particularly at lower cut-off percentages.
- The analysis provides a framework for improving future steam locomotive performance through careful adjustment of operational parameters.
- The results are compared with historical data from the 1957 British Railways tests, validating the calculations and conclusions drawn in the paper.

A DEBATE ABOUT THE MERITS OF SUPERHEATING - PART 2
compiled by Chris Newman

As expected, the debate that was started in Part 1 of this article (published in Newsletter No 28), has spawned two responses so far: one from Terry McMenamin and the other a 24-page dissertation from Dr David Pawson. I thank them both for taking the time and trouble in formulating their responses.

Far be it for me – a humble civil engineer – to make judgments as to whose opinions are right and whose are wrong, but both Terry and David come down heavily in support of Jamie Keyte's view that cylinder wall heating is no substitute for superheating. At any rate, both articles will repay the effort of reading them – David's especially as he has gone to great trouble to back up his arguments with data from the Altoona test plant (USA) in the 1920s, and with results of simulations run on Bill Hall's "Perform" software which David has, over many years, verified as consistent with the data recorded on the Rugby test plant in the 1950s and 1960s.

It's a really worthwhile read!

MERITS OF SUPERHEATING

Terry McMenamin

The discussion on the merits and demerits of superheating triggered by Joseph Cliffe is certainly interesting. Chapelon's claims suggest that the losses due to condensation in non-jacketed cylinders must considerably exceed the additional energy added to steam by superheating. A first calculation suggests this is unlikely: at 250psi gauge and 350°C superheating has added 1.234MJ/kg of energy. At 10psi gauge the enthalpy change from saturated steam to water is 1.732MJ/kg. Expanding from 250psi 350°C to saturated steam at 10psi requires an expansion ration of 6.76, or 15% cut-off. Heat lost to condensation at that pressure equal to that input by superheat would seem to imply over 70% of steam condensing. Obviously that did not happen. My figures are from Spirax online steam calculator, which is easy to use, but it is possible that there is a mistake somewhere, if so please correct me. It is also true that conditions in the 160A1 were very different, but the point is that the initial steam condition was not unusual in locomotive practice.

In practical experience, the application of superheating increased power and reduced specific fuel consumption. Early adopters encountered lubrication problems, but bought into the benefits. Churchward famously limited the degree of superheat on the GWR, a practice Stanier took to the LMS. Hundreds of his locomotives had to be modified to higher superheat, an exception being the Duchess which was designed by Coleman while Stanier was in India.

Reciprocating marine engines of similar power to locomotives almost all operated at lower steam pressures and obtained double the efficiency. Power stations such as Ratcliffe, visible from my back bedroom window, more than doubled the efficiency again. Both obtained their results by courtesy of the sea or rivers, a virtually infinite heat sink enabling power generation at low temperature and pressure due to condensing operation, the latter combining it with very high steam pressure and temperature.

Jamie Keyte nails it: ¾ of the input energy of a steam locomotive is blown straight up the chimney in the exhaust steam.

However the business of hauling trains needs more than ideal thermal efficiency, it has to be as economical as possible and reliable. Joseph says water is cheap, and that is true of tap water. However, the LMS in one of its films 1930s stated that even then it was spending £3.5million a year on water. Not only that, but enormous tanks and water cranes at every significant station, water troughs (steam heated in some places) tender tanks of ever-increasing size, all cost money and the tender capacity limited range, so that frequent water stops caused delay. Having got this water to the engine its sediment collected in the boiler, even with the best water treatment, so less water use would translate into longer washout intervals. It appears to me that Maunsell's lackadaisical approach to the trial of vapour recompression showed that he failed to appreciate the benefits from what we would consider a relatively simple system enabling vastly reduced water consumption with increased efficiency.

Summary

- There is no reason to disbelieve any of the findings relating to the historic benefits of superheating. Chapelon's reported results are puzzling.
- Modelling of initial steam conditions, expansion ratios, and heat loss to cylinder walls may help reconcile the apparent contradictions.
- The unavoidable problem of the Stephenson locomotive is the loss of energy caused by disposal of exhaust steam to atmosphere. This loss dwarfs any other losses and limits the scope of improvement
- Water as delivered to the boiler requires expensive infrastructure and limits locomotive operating range.
- High rates of water consumption caused unwanted contaminants to build up in the delivery system including tanks, and precipitating in the boiler. A closed water cycle would ease washout requirements.
- Vapour recompression should be worth trying again, with more determination than Maunsell showed.

Provocative thought: should a high-speed geared engine with vapour recompression be tried on Revolution? It would reduce the time for steam to leak during expansion, and allow smaller, lighter wheels.

THE MERIT OF SUPERHEATING

David Pawson

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Introduction

In the last Newsletter (no 28 of October 25), Chris Newman compiled a paper 'A Debate about the merits of Superheating', based on correspondence with Joe Cliffe, Hendrik Kaptein, Jamie Keyte and Richard Coleby. In this the merits of superheating were challenged, and an alternative of using steam jackets to heat cylinders was proposed.

Among the verbatim comments quoted casting doubt on superheating were:

A. Chapelon in his later writings found superheating to be unnecessary with steam jackets.
B. The original purpose of superheat was to prevent condensation inside cylinders, known as the "missing quantity"
C. Superheating steam is a very inefficient way to heat cylinder walls due to its poor heat transfer coefficient and the lack of turbulence in a large cylinder.
D. The (originally expected) efficiency benefit of superheating is curtailed by the rather limited amount of steam available for heating up cylinder walls.
E. Superheating, though undoubtedly effective, falls short of original theoretical expectations
F. Superheat undoubtedly reduces water consumption. Once condensation is eliminated the thermal gains are very limited, but water is cheaper than the coal needed to superheat.
G. The base case also rests on an optimum all-saturated boiler drafted against a less efficient superheated version, which was never done because the false belief in superheating was so strongly held.

There is to my mind no debate at all about the merits of superheating. In simple engines, it is the single largest improvement to the efficiency of the steam engine ever achieved. I find it unfortunate that so late in the day this topic is on the table, but nevertheless feel it needs to be shown that masses of test data, supported by the best quantitative theory of engine efficiency together confirm this. It's an open and shut case. If you already firmly hold this to be true, you may want to back out now. On the other hand, you may want to be delighted by an interesting tale of how old data and modern theory perfectly complement each other to demonstrate the benefits. If so, read on.

I will then show that PRR test plant data from Altoona show cylinder wall temperatures get hotter at higher superheat- so there's no problem with heat transfer with superheated steam. The idea that superheated steam can't heat cylinder walls is nonsense. However, when cylinder wall temperatures are below a certain level, steam condensation occurs during the cylinder cycle, reducing efficiency. Condensation is a contributor to the 'missing quantity'. Steam jackets can prevent this.

I will then show a combination of data and theory says the efficiency loss in a full scale simple expansion engine due to condensation is, even in the worst case scenario of a saturated engine, rather small. The majority of the benefit of superheating is because hotter steam expands more efficiently. Steam jacketing will not benefit even moderately superheated simples because walls are hot enough to prevent condensation. The benefit of jacketing in smaller scale saturated engines is however likely much larger.

Chapelon was interested in steam jackets because when superheating was applied to Compounds it did not deliver as expected, because cool low pressure (LP) cylinder inlet steam gave cool cylinders hence efficiency loss through condensation, sufficient to negate the Compound benefit. I think this is where the notion that superheat is ineffective comes from. It took Chapelon's best efforts in the 1920s to sort this out principally via very high superheat. LP steam jackets are another way to deal with this problem.
As for the claim that in Chapelon's Compound 2-12-0, superheating could be replaced by steam jackets, this one is quite easy. He did not say that. He said something different, which is actually quite surprising. To work out what was going on, I will have to take you on a typical nightmare journey through inadequate French data, and misleading claims. I conclude there is possibly a rational explanation of his finding, little to do with steam jacketing.

This 'debate' is to my mind is about the merits of the widely adopted British/American simple expansion approach, versus those who think Chapelon had some transformational wisdom and technologies in his Compounds that could be applied to simples. He did not. However, an optimised superheated Compound, bugs sorted, ought to be

more efficient than a simple. For reasons unclear however, even the best French Compounds are barely more efficient at the drawbar than a simple.

As to whether Porta was correct to advocate higher levels of superheat than those latterly achieved- one of counter protagonists Joe Cliffe's starting points- that is a more open question as I will show.

This is all a rather tall order, so don't read on if you are feeling drowsy.

'Hot steam' Schmidt (1858-1924)

The German engineer Wilhelm Schmidt is widely credited as the father of modern superheaters. Others had attempted to reach about 250^{o} C, but Schmidt aimed for 350. Joe argues that the aim of superheating was to deal with 'the missing quantity'- the disparity between the steam entering the cylinder and the pressure revealed in the indicator diagram. The missing quantity was large in 1900.

At the time, it was recognised that one explanation was that steam could condense onto cold cylinder surfaces, and re-evaporate at the end of the cycle having done no useful work. Perry outlined the basic theory in the early 20th century, causing some controversy. Joe's suggestion that one driver for superheating was to deal with this aspect of the missing quantity may have some truth, but Schmidt studied under Professor Gustav Zeuner, founder of the Dresden thermodynamics School, and thermodynamics was clear by the 1880s that the hotter the working fluid in a heat engine, the higher in principle its efficiency. In Zeuner's care, 'Hot Steam' Schmidt must have seen this as a greater driver for his endeavours. Schmidt produced his first superheater in ca. 1890.

One of the problems with employing highly superheated steam was that this made lubrication of the slide valves in use at the end of the 19th Century difficult. Schmidt therefore also made advances to produce today's commonly used piston valve, where this problem is largely overcome.

By 1908 Schmidt had formed a company to sell his superheaters in Britain and in 1910, the progress in Europe resulted in the formation of The Locomotive Superheater Company in the USA.

The Superheater was thus pioneered in Europe, being first applied in this country in 1906 on Saint 4-6-0 2901, and its use quickly spread. The USA was somewhat behind, but nonetheless it is to the United States that we must look for a massive programme of testing which unequivocally quantified the benefit of superheating accurately over a wide range of conditions.

Astounding benefits of superheating at the Pennsylvania Railroad (PRR) test plant at Altoona

The Altoona test plant

By 1882 the PRR was the largest railway in the world by both traffic and revenue, and by the 20th Century second only to the US federal government in turnover in that country. It was brash, confident, and only the best would do. They therefore set out to build a locomotive testing station to fully characterise a steam locomotive's performance, and find out what improvements could be made. The Advisory Committee for the project included many of the worthies of the day such as Professor Goss. Their work up to 1924 can be found online in 32 Bulletins, every single data point recorded, all assessed clearly and dispassionately and in meticulous detail, to the point of tedium. All facets of the engine and boiler performance on a test were recorded, not just ones of special interest. They were not trying to confirm established theories- there were no quantitative ones- so all that mattered was the data- what happened?

Their interests were shaped by the concepts of the day of course, some of which were misguided- for example the obsession with heating surface and evaporative capacity- but these did not dictate their conclusions. These 32 Bulletins cover all major aspects of locomotive performance improvements, and I would go so far as to say we have not really learned anything new since, though a combination of incremental improvements did lead to significant later advances beyond the efficiency and power levels they established by 1914.

The only major error in their results was power at the wheelrims, which means their assessment of the power needed to drive the machinery between the pistons and wheelrims which they called 'machine friction' are all high, and often irrational. They later admitted this in Bulletin 32 (p70).
#43 - Locomotive testing plant at Altoona, Penna. : bulletins ... v.30-32. - Full View | HathiTrust Digital Library
The bulletins are all essential reading for steam locomotive connoisseurs. Forgot some of the dated concepts therein, just focus on the treasure trove of carefully documented and reported data. There is no better place to learn about superheating.

Altoona plant tests- Initial testing
The PRR test plant was first erected at the 1904 St Louis exhibition, a celebration of the Louisiana purchase, and America's growing triumphs in arts, engineering and science.
At the time, the competing traction types were principally 2 cylinder saturated simples and 4 cylinder saturated compounds. The first Bulletin reports tests in St Louis on a variety of saturated simples, and Compounds from different railways-Vauclain, Cole (New York Central) and French De Glehn types. They also tested a superheated Prussian S8 Hannover Compound. The results showed that the superheated Hannover Compound, albeit the puniest design, rather odd to American eyes, was head and shoulders above the rest in terms of engine efficiency. More generally, the saturated Compounds were consistently more efficient than the saturated simples. Compounding worked.
#12 - The Pennsylvania railroad system at the Louisiana purchase ... - Full View | HathiTrust Digital Library
With the plant moved back to Altoona, the PRR therefore tested their best passenger design, a simple saturated Atlantic, of similar dimensions and weight to the NYC Compound tested earlier. They soberly concluded the arch rival Compound had a more efficient engine at all powers, and that its maximum horsepower was 1600 compared to the simple's 1300. Oh no! 5-0 to the competition.
#56 - Pennsylvania Railroad Company test department; locomotive ... - Full View | HathiTrust Digital Library

Defining tests on superheating
After footling around with duff ideas from suppliers and shed foremen in Bulletins 6-9, came the big breakthrough in **Bulletin 10.** By 1912, superheaters were already being fitted on the PRR based on their reputation, and in that year, a class H8 Consolidation freight type with suitably modified cylinders was fitted with a Schmidt superheater, and tested on the plant. Steam temperatures of 320^{o} C were reached and the report concluded, rather blandly 'the application of a superheater….. has materially increased the economy in coal….at 500ihp the saving is approximately 24%, while at 1500IHp the saving is 45% . Bingo! (The report is much more balanced than that but that was what they felt). The maximum saving in water was 35%. A corresponding increase in power was obtained, but for freight locos, maximum tonnage rating – the big money saver- is generally about the TE required to get up the ruling grade at low speed, and this didn't change much.
#9 - Locomotive testing plant at Altoona, Penna. : bulletins ... v.5-10. - Full View | HathiTrust Digital Library

There followed a series tests of Schmidt superheated vs saturated versions of various express types.

Bulletin 11 showed that a superheated E3 Atlantic had a 38% increase in power at speed, saving up to 48% coal compared to its non-superheated counterpart. Inlet steam temperatures up to 330^{o} C were again achieved The higher power meant that tonnage rating at passenger speeds increased by similar amounts, and the question now became whether there was sufficient TE to start such trains in all circumstances.
#9 - Locomotive testing plant at Altoona, Penna. : bulletins ... v.11-16. - Full View | HathiTrust Digital Library
Bulletin 18- tested a superheated K2 Pacific, and large benefits in power and efficiency, were again shown over the saturated version though the exact benefit was difficult to state because of paucity of saturated data.
#11 - Locomotive testing plant at Altoona, Penna. : bulletins ... v.18-21. - Full View | HathiTrust Digital Library
Bulletin 21 describes tests on an E6 Schmidt Atlantic. A 32% benefit in coal economy in the normal working range was attained over the saturated version. If pushed to abnormal high outputs benefits were even higher. A similar increase in maximum power was attained. This Bulletin was presented at the 1913 Convention of the Master Mechanics Association, and I feel sure that Schmidt's superheater company was in clover from that point on. The report like others focuses on all aspects of design, and they are particularly concerned with good draught and low exhaust pressure. They understood the importance of chimney exhaust velocity profile on draught and so measured this.
Bulletin 19 describes tests on a K29 Pacific, an Alco built forerunner of the K4 Pacific which became the PRR standard. This showed the efficiency of the other superheated types, and its larger grate and 6 coupled axles meant it had the TE to start heavy trains and power to run them at speed. A milestone level of performance had been reached.

Of course there had been concerns that the loss of evaporative heating surface when fitting a superheater might be disadvantageous; or that the additional heat needed to superheat the steam would increase coal consumption. But it was abundantly clear that these were not substantive. The additional enthalpy needed to superheat steam to a high value of 370^{o}C is no more than 15% if heat transfer efficiency is unchanged .The benefit in engine efficiency gained is far greater than this.

The end of Compounding in the United States?

Bulletin 19 then goes on to compare the efficiency of superheated simples with the superheated Hannover Compound which had triumphed in earlier efficiency tests. They showed that at speeds above 30mph, the superheated simple was superior to the Compound, whilst below that speed the Compound was superior.

I think the reason is that, particularly at low speeds, if you try to steam to the boiler limit, with typical simple designs you need to work in excessively high, inefficient cut offs. You need high HP and LP cylinder cut offs in a compound as well, but with double expansion these do not lead to such a steep loss of efficiency, hence the low speed crossover the PRR observed.

This lack of efficiency loss at long cut offs is why French Compounds were able to deliver such eye catching power outputs when pressed to the boiler limit at express speeds. To my mind these power claims were of little practical relevance, since working at the boiler limit was extraordinarily wasteful in coal economy, and probably bad for boiler maintenance too. If you want to produce these powers you can, but no serious railway operator would ever plan to do so. Pushing locomotives beyond their economic limit to see what they could do was also in vogue in this country at the time- see the LMS performance tests to establish the limits of the Jubilee, Duchess, and original low Superheat Black 5. The LNER went for very high speeds.

Bulletin 19 then goes on to make the throwaway statement 'as the normal speeds of our passenger trains are above the (efficiency crossover) speed, it would seem from the test of this locomotive that the compounding of a passenger locomotive is not justified'. Given the slender nature of the data on the superheated compound, this is an appropriately judicious statement.

But make no mistake, the door on Compounding had been firmly shut in the USA except for some specific tasks; the attractions of simple to build and maintain two cylinder designs were too great.

The Definitive tests on superheating and cylinder condensation.

We then come to **Bulletin 24**, a must read for anyone interested in superheat. Having established a practicable design philosophy based on high superheat, this series of tests on a K2 Pacific explored the topic of Superheat more broadly.
#9 - Locomotive testing plant at Altoona, Penna. : bulletins ... v.22-25. - Full View | HathiTrust Digital Library

They tested a whole range of superheater designs of various lengths and passes, including one with the header only in use, delivering saturated steam. They tested over a range of operating conditions. – up to ten tests on each of a dozen superheater configurations. Speed was 56mph. In each test, steam temperature rose quickly to a steady value. However, it then drifted up slightly towards the end of the 90 minute test so true equilibrium takes a long while to establish.

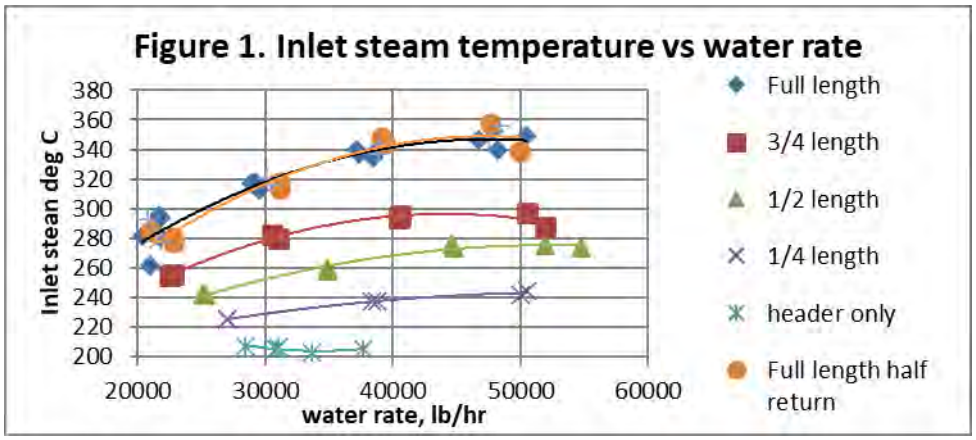
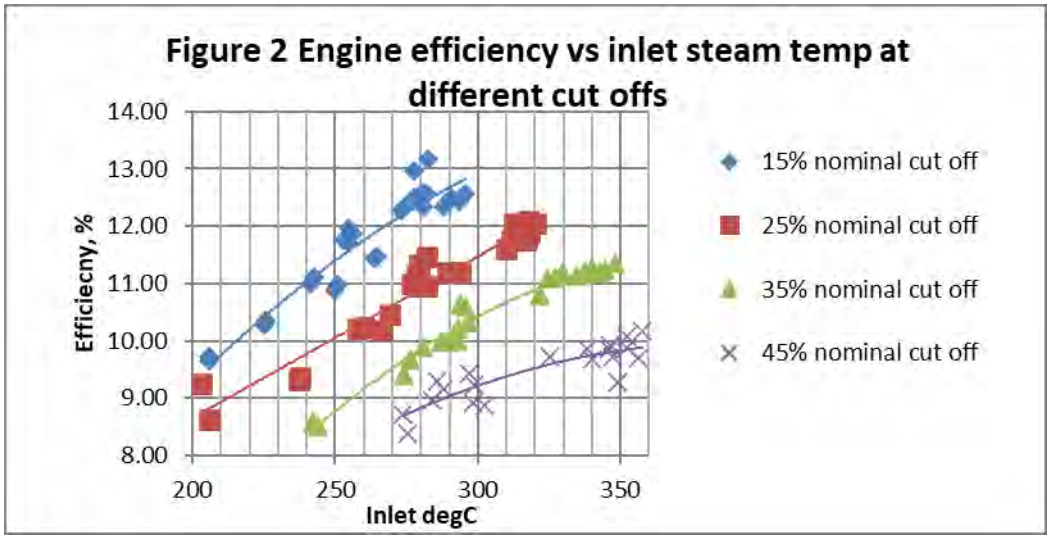
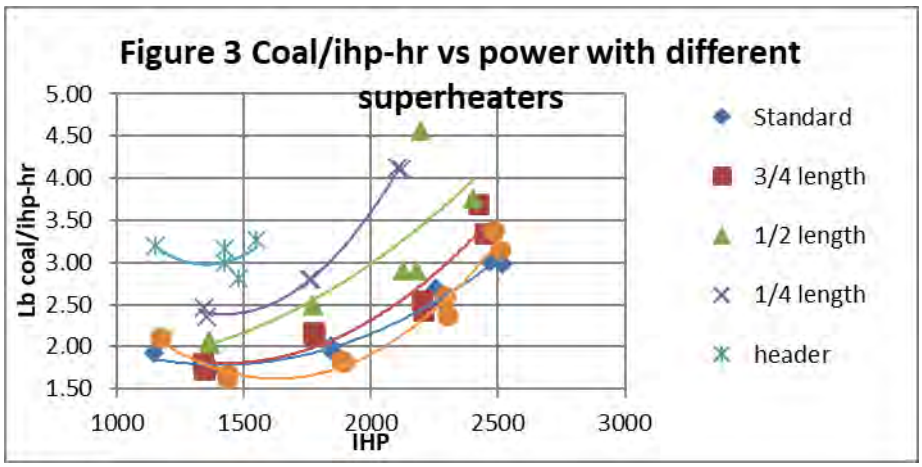


Figure 1 shows some typical results. Inlet steam temperatures ranged from 205° C (close to saturated steam) up to 350°C according to the superheater used. The results for the different superheaters are much as you would expect I think, with superheat increasing smoothly, albeit at different rates, with increasing steam rate. The most interesting finding was that a half-length return loop gave as good superheat as a full return loop, a finding only the LNER took up over here.

Altoona showed that over the range of temperatures studied, the best engine efficiency was always obtained at the highest superheat, and all efficiencies were higher than with a saturated four cylinder Compound. In 25% cut off, the improvement in efficiency going from a near saturated engine to the standard superheater was an astounding 40%. (Figure2).



What matters of course is the effect on coal consumption. Figure 3 shows how this varies at different superheat levels. At a typical 2000IHP for the K2, the coal saving between the standard and ¼ length superheater was 38%. (The header only consumption results are not reliable because the lack of elements killed the boiler efficiency). They laughed all the way to the bank.

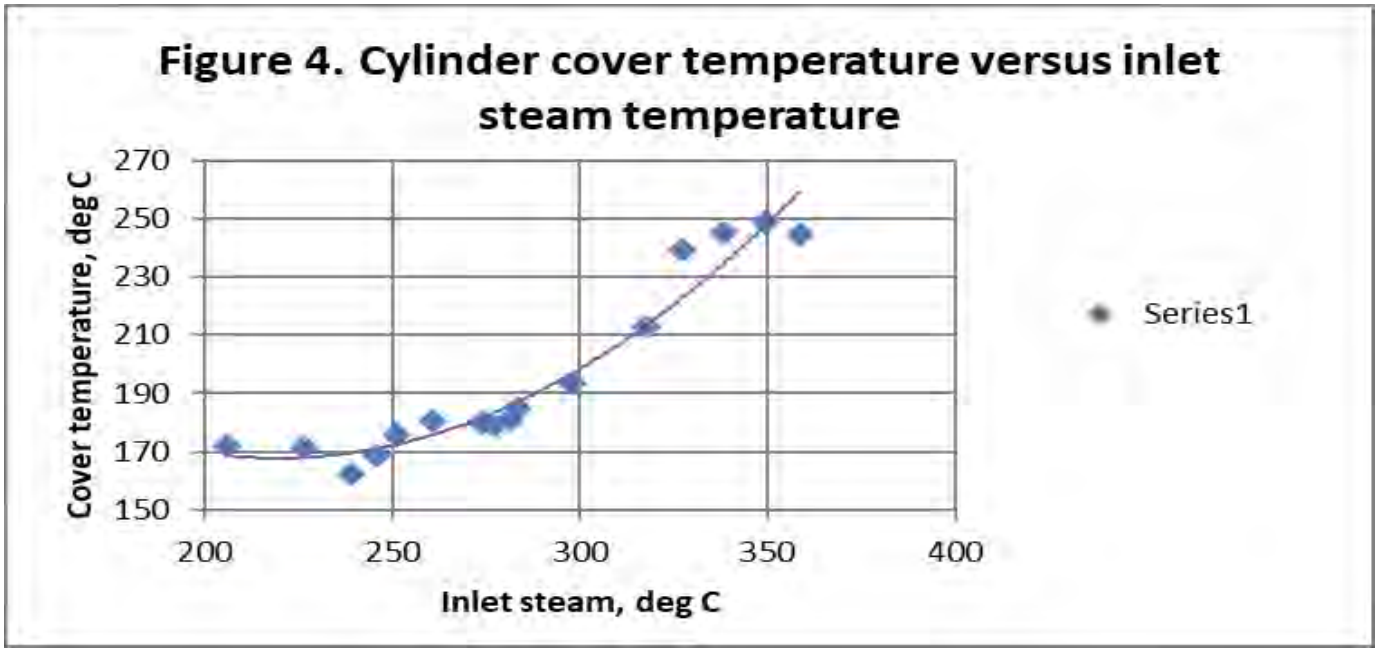


All PRR Mainline locomotives were quickly fitted with the largest practicable superheater, and with the odd exception, Compounds disappeared from US mainlines, Superheating was the single greatest improvement in efficiency in the history of the reciprocating engine. PRR test bulletin 24, is perhaps a defining moment in attempts to improve steam locomotive efficiency.

As a rather sad footnote to this wonderful series of papers, PRR practice then stagnated at K4 level for nearly 20 years. Their rivals moved on with more powerful and efficient designs through a series of incremental improvements- boiler pressure, superheat, back pressure, feed heaters etc. In the 1930s, the arch competitor NYC's J3 Hudson was more efficient and in a different tractive league to the PRR K4. When the PRR tried to recapture the high ground in the late 1930s, they misguidedly put their faith in Franklin poppet valves. Their reports moved from the confident 'let's find out what's going on' mode to the arrogant 'let's prove we're right and the best' mode. This blinkered approach ignored the fact that contemporary poppet valves of leaked like sieves and led them to misinterpret their Altoona tests on a Franklin valve K4, to prove it was a saviour. Further, at Baldwin's RP Johnson's prompting, they also adopted Duplex drives. Together, these ideas led to the catastrophic failures of their final steam designs, the T1 4-4-4-4 and Q2 4-4-6-4. Both perished quickly.

The measurement of cylinder wall temperatures

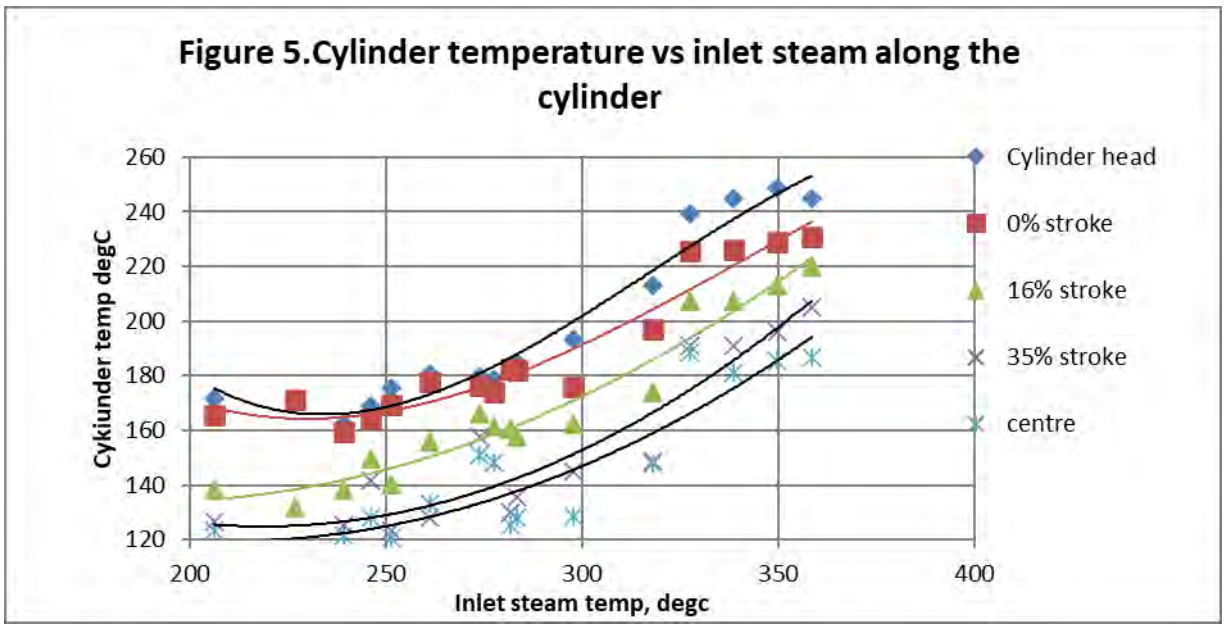
Given the theories on condensation around at the time, the PRR were naturally also interested in the effect of superheat on cylinder wall temperatures. Perry's theory said that condensation would occur if cylinder walls were below the saturation temperature of the steam. They therefore measured cylinder wall temperatures on about a dozen of the above tests, covering the range of superheats tested. Temperatures were measured on the cylinder cover, and on the wall at 0, 16, 33 and 50% of stroke. There was a clear increase in surface temperatures as superheat increased at all points. Figure 4 shows the temperatures of the cover.



Thus on the cylinder cover, and likely the piston face, even with saturated steam the lowest temperature observed was 170°C. Once inlet steam reached 300°C, the cover temperature reached the inlet saturation temperature of around 200°C, so no condensation would occur on the cover above this inlet temperature. On the cover, the temperature does not depend on the cut off used.

This is consistent with Swindon tests on their first Hall, which found no evidence for condensation on the cylinder cover when working in 40% cut off.

Along the length of the cylinder there was a continuous fall in surface temperature to as low as 120°C (Figure 5)., There is quite a bit of scatter in the 35% cut off and centre of cylinder lines in Figure 5, unlike the smooth line in Figure 4. Here, temperatures at a given inlet temperature get higher at longer cut offs, which will tend to lessen the condensation at higher cut offs.



The pressure at points along the stroke was measured from the indicator diagram. The steam saturation temperature at each point can then be estimated from this pressure, and can be compared to the cylinder temperature at that point.

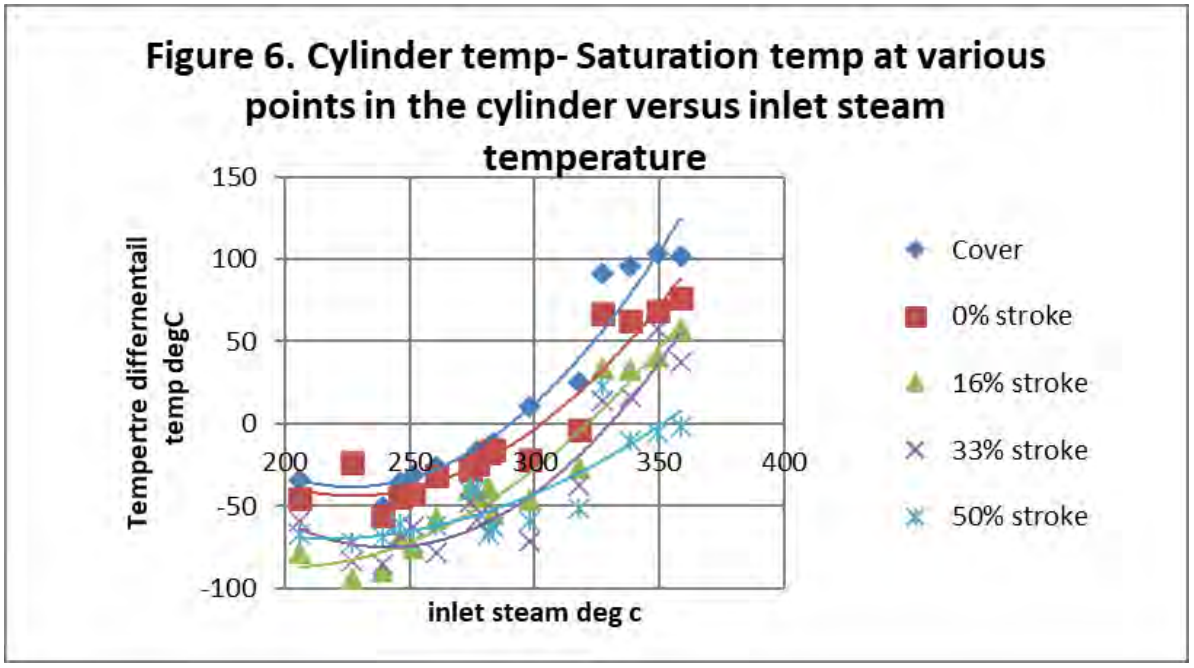


Figure 6 plots the difference between cylinder temperature and saturation temperature at different points along the stroke. Where the difference between cylinder temperature and saturation temperature is negative, condensation is to be expected. It can be seen that, as noted above, once inlet steam reaches about 300° C, condensation on the cylinder cover and at 0% stroke will cease. However, only at 350°C will condensation cease entirely at the centre of the cylinder.

These were ground-breaking results, but they had no way to calculate how much condensation would occur at a particular cylinder temperature in the condensing region. However, given that the superheaters of the day were already delivering sufficiently well to eliminate condensation in most circumstances, this was rather academic. The benefits of increased superheating had been demonstrated, so the results were not of much practical value, and interest in the topic waned.

What is clear from all this is that the assertions made at the beginning that ' *Superheating steam is a very inefficient way to heat cylinder walls due to its poor heat transfer coefficient and the lack of turbulence in a large cylinder'* .and *'The (originally expected) efficiency of superheating is curtailed by the rather limited amount of steam available for heating up cylinder walls'* are a demonstrably false theory of Chapelon's. Like most of us exploring complex phenomena with no clear theoretical framework, Chapelon was full of theories that he thought explained what was going on. They have traction in the 21st century only because they are repeated ad nauseam by his supporters.

I see no reason to doubt that the basic picture shown in Figure 5 applies to the cylinders of simple expansion types of similar dimensions, such as latter day European types. There will be some differences due to somewhat higher inlet pressures than the 205 psi used here, and perhaps the effectiveness of cylinder lagging, but nothing fundamental.

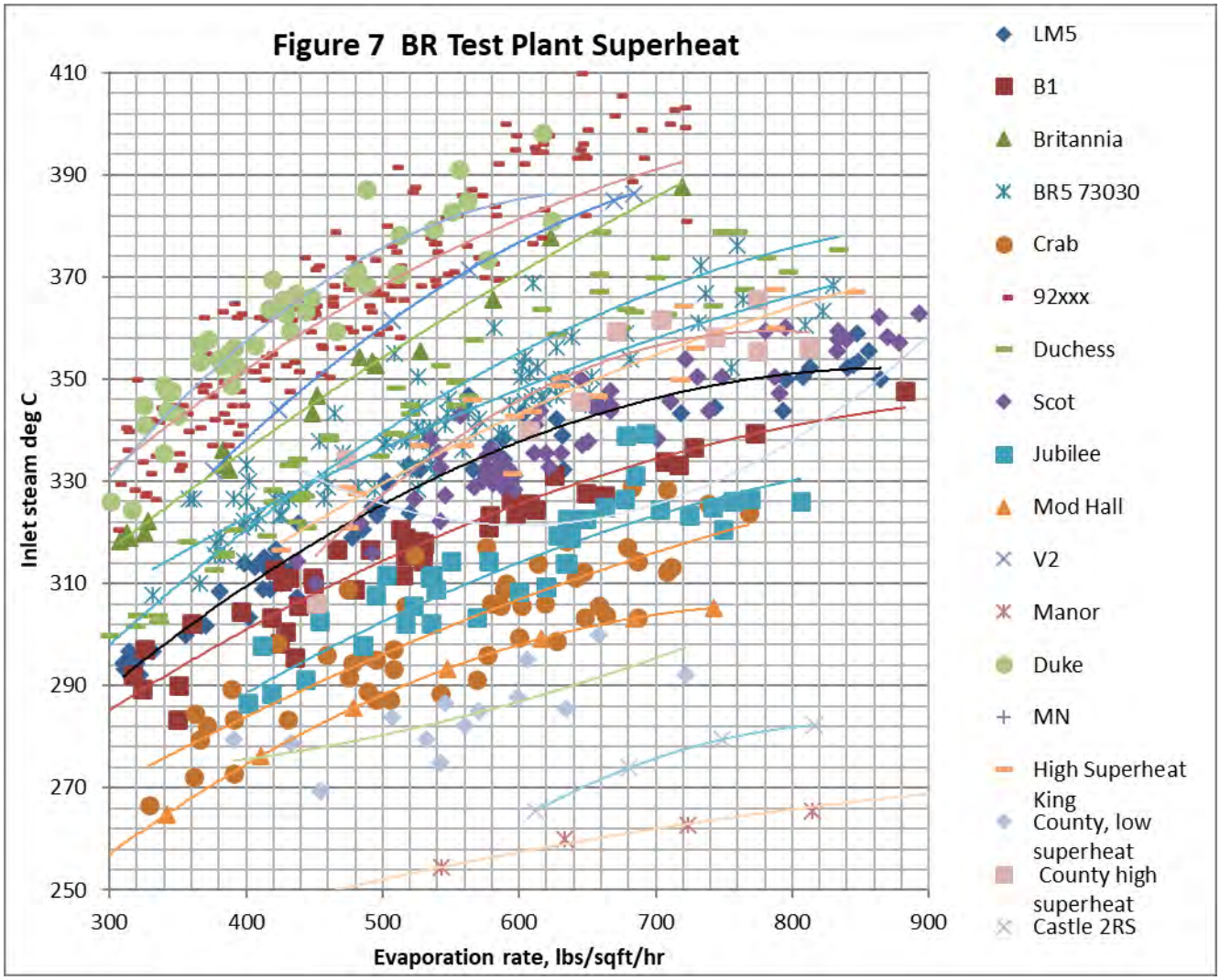
UK Superheating practice and tests

In this country high superheat became the norm, except on the GWR, which like the PRR became fossilised in its Edwardian past at Churchward superheat levels. I've seen it said that Churchward's reason for not to go higher was because a feature of GW engines was their low clearance volumes, which could give problems with over compression with the higher exhaust pressures from hotter exhaust steam. Had a King been given high superheat in the jumper cap era, they might have been ok but with the narrow plain blastpipe fitted in 1952, the low clearance did I think cause problems, even after some relief was provided with double chimneys. To my mind the Kings have never achieved their full potential as a consequence. Compression is not so pronounced on the Castles.

There was also a cherished belief at Swindon, evidenced in some of the early BR Test Bulletins, that if the exhaust became superheated, you were wasting energy. The Mod Hall level of superheating is about as high as you can go without running serious risk of this. This is in flat contradiction to basic thermodynamics. The higher the superheat, the greater the proportion of the inlet steam's energy you can extract. High exhaust temperatures are a largely irrelevant consequence.

There was constant friction between Swindon and the BR hierarchy in the 1950s, low GW superheat being a particularly sore point. Figure 7 shows the big picture on BR superheating. What you should be able to see is that if you plot against evaporation/sqft (a surrogate for combustion rate/sqft, hence gas temperature), you get a series of nested curves showing increased superheat with evaporation rate. Looking at the detail, the principal design variable is the number of elements- the higher the number, the higher the superheat. There is a secondary effect of boiler length, shorter being better. If you go longer

than needed, as on the Duchess, you underperform. At the top of the class are the larger BR types and LNER V2/Pacific boiler. The bad boys at the bottom are the Manor, and 2 row Castle, typical of other GW 2 row types.



BR therefore decided in 1956 (knowingly or not) to repeat the Altoona tests on the K2 using a BR5[1]. The old BR and Swindon steam titans were still duking it out, despite the Modernisation Plan making it all irrelevant. They tested inlet steam temperatures from 230°C, (30 deg above saturated) to 400°C. Figure 8 shows that coal consumption at 800IHP was 31% higher at 230 degrees (similar to the ¼ length superheater in Figure 1), compared to the standard boiler, which delivered about 340°C. These coal/ihp-hr figures follow a very similar pattern to Figure 3. Benefits would have been greater at higher power. The 280°C line in Figure 8 would be typical of GW practice.

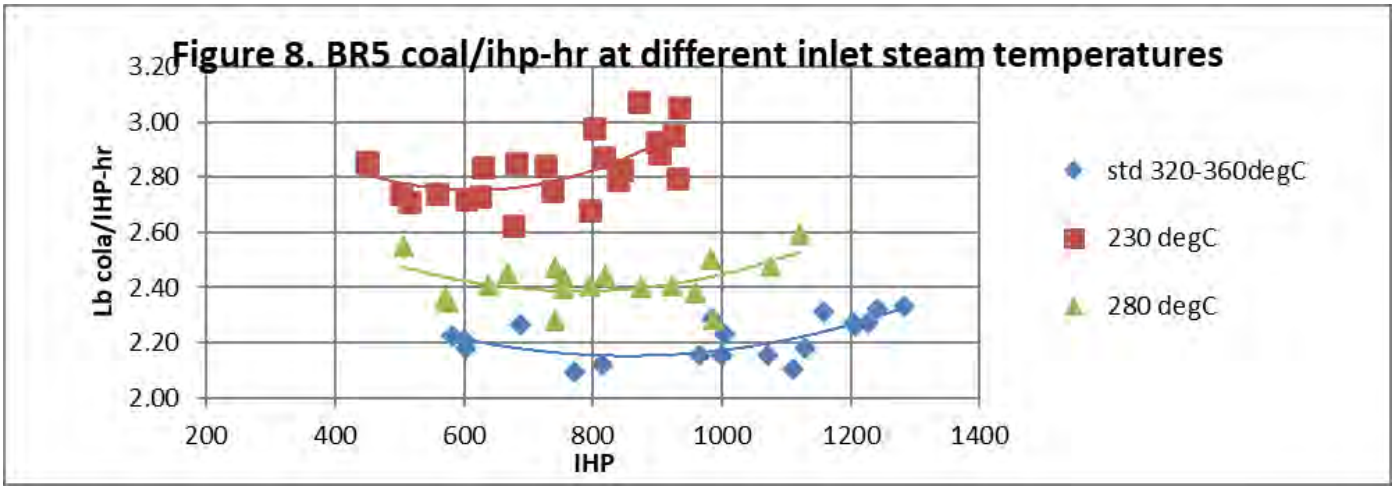
There never had been any real doubt about the benefit of superheating, but entrenched views were difficult to overcome.

A quantitative theory of the benefit of superheating

The steam engine may have spawned the topic of thermodynamics, whose three laws we now know determine reality as we know it, but thermodynamics was unable to repay the compliment by defining how efficient an engine would be. This is because classical thermodynamics deals with reversible processes, and unfortunately in real steam engines there are irreversible processes when the steam enters the cylinders, and at release, which make direct calculation impossible.

Enter another hero in this story, Professor Bill Hall. He was a distinguished academic, having been the Dean of Science at Manchester, and amongst other things chairman of the British Heat Transfer Society. He was also an outstanding practical engineer, having made vital contributions to the development of the British Nuclear power Industry. He was also a supporter of the Ravenglass and Eskdale railway, and was intrigued why their locomotives were so inefficient.

He understood full well that engineers tend to distrust first principles academic theories. In steam days this didn't matter, because those theories were too complex to solve analytically. Hence the steam locomotive developed,



largely successfully, through native wit and trial and error. Empirical theories abounded, not always based on sound science. Many of them seemed to work.

The Perform programme for calculating engine power and efficiency

Nonetheless, Bill set out to calculate engine efficiency. The goal was quantitative, testable first principles predictions -an end to steam age empiricism and guesswork. The power of modern computing enabled him to solve the governing equations. This was a several year project, and he made a remarkable contribution to our understanding of the reciprocating steam engine. His computer programme, Perform[2], calculates the power and efficiency of a locomotive engine at different cut offs and speeds, from its dimensions and the inlet steam temperature and pressure. The initial programme took no account of cylinder heat transfer effects-condensation.

He got round the problem of irreversibility by calculating the flow of steam into the cylinders using compressible flow equations, plus equations that accurately calculate port opening at various cut offs and speeds, and finally the backpressure created in the blastpipe. He then used steam tables to calculate what the indicator diagram would look like at this flow, so flow and power in the calculation were directly linked. The programme needs knowledge of lap, lead, exhaust clearance, clearance volume, valve diameter, piston dimensions and steam chest pressure and temperature. Given these, you can calculate IHP and steam flow at any cut off and speed combination.

Like any programme, it's garbage in, garbage out. There are two principal uncertainties, steam chest pressure and the discharge coefficient through the ports. This latter is a genuine unknown, and may vary with the size of the port opening. All you can do is pick a value and see from the data what works. It turns out the result is not that sensitive to this factor, so it's easy to pick a sensible value.

The problem with steam chest pressure is that you may know its mean value from test plant data, but what you want to know is how it falls during admission- again not known with any certainty. All you can do is work with measured steam chest pressure and see what happens. This is not too much of a problem in this country, where the fall is small, but in the USA, where steam flow/cylinder was up to 3 times higher, pressure drops during admission are much higher, so greater uncertainty is present. If I had started with US data, I might not have been so bullish about Perform.

The programme also assumes that the valve gear delivers exactly what the drawing office says it should. Calibration of the valve gear on two different BR5s at Rugby showed differences of up to 2% in actual cut off, so there could be as much as a 7% difference in power at a given nominal setting. So absolutely precise agreement between the programme and the test data is not to be expected.

Bill compared its predictions with the BR test results for the Britannia. He found that he could reproduce the indicator diagram well. I have the raw IHP data for over 500 tests at the Rugby test plant, on a wide range of designs, and went on to show, except for BR5 73031 and Scot 46165, where the cut off scale does not appear to have accurately reflected what was going on in the valves at low values, there is amazing agreement between Perform's power predictions and test results.
So, a major breakthrough from Perform.

The missing quantity rears its ugly head

However, whilst Perform predicted IHPs well, I found that on the same 500 tests, it consistently underestimated the amount of steam consumed- the missing quantity- just as Bill had done[2]. What you see for each type is a more or less constant discrepancy across all engine conditions between predicted and reported cylinder steam. The exact value varied between types, and on the most tested type, the BR9, it varied between class members, so the missing quantity is a slippery beast.

The question then was whether this discrepancy was due to condensation, or leakage or some combination. The 'discrepancy' aka the missing quantity will feature large in what follows.

Cylinder condensation principles and the Bill Hall model

Bill therefore set about trying to quantify the process of condensation, an almost obsessive retirement project. His background in the science of heat transfer made him the right man for the job. The principal tenets of the theory of cylinder condensation first propounded by John Perry[3] were:

- Steam will condense on cylinder surfaces of the cylinder (cover and walls) when their temperatures are below the saturation temperature of the steam at any given point.
- Conversely, when cylinder steam pressure falls to a point when its saturation temperature is below that of the cylinder, the condensed layer will re-evaporate. This normally occurs after the exhaust ports are open, so the steam which has condensed does no useful work.

The formation a condensed layer leads to a reduction in the pressure in the cylinder which means steam flow increases to compensate. The indicator diagram changes little, so the power remains similar, so the net is more steam at the same power, hence a reduction in efficiency. Given the higher steam flow, back pressure will go up slightly, which reduces efficiency a bit more.

Bill first built a small scale steam engine and instrumented it so he could study cylinder temperatures, and he measured the small fluctuations in temperature at the cylinder surface as the steam condensed and evaporated during the cylinder cycle, thus verifying Perry's theory[4].

Bill then developed a modification of the Perform programme[5], a first attempt to estimate how much condensation would occur in different scenarios, based on first principles heat transfer theory. The programme can be found on the ASTT website (Members library section). It calculates where in the piston cycle the condensed film forms, how thick it becomes, and when it evaporates. From this the resultant increase in steam flow is calculated, plus changes in power and back pressure.

All these things can be done across a range of cut offs and speeds, and the dependency of condensation on superheat can be established.

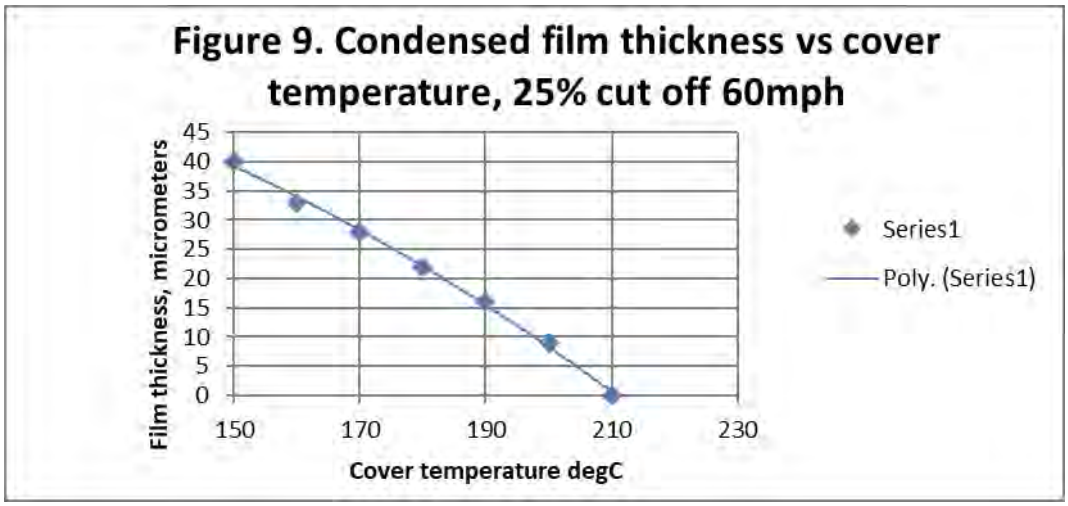


Figure 9 shows the estimated film thickness for a Britannia working in 25% cut off at 60 mph and inlet steam of 350°C. The film thickness increases as cover temperature falls. The thicker the film, the greater the condensation loss when it re-evaporates. At about 145°C, there is a discontinuity. The model predicts that the film will no longer evaporate, so a stable insulating film forms on the surface of the cylinder, meaning that condensation will in effect stop. This is what the heat transfer maths says, but Bill doubted it would ever occur in practice. That this is indeed the case is confirmed by the Altoona data. Measured cylinder temperatures (figure 5) closely approach but are always higher than the temperature at which this phenomenon would occur. I find it quite remarkable validation of the model that temperatures at which this would occur are never reached in practice and are predicted accurately.

Somewhat surprisingly, the model also shows that the thickness of the film does not change directly with the temperature of

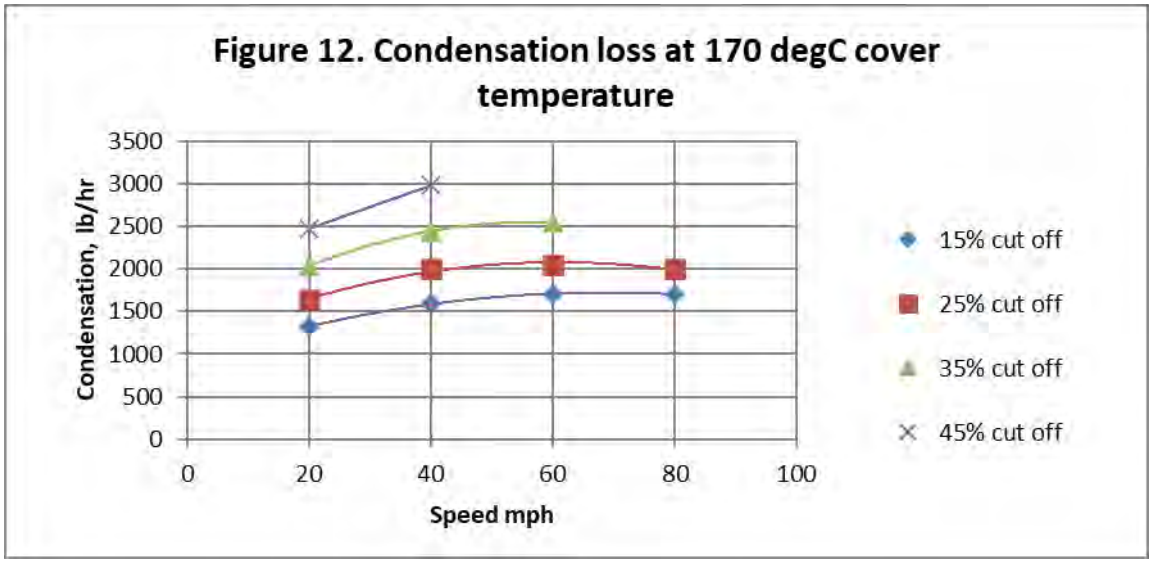
the inlet steam, holding cover temperature constant (graph not shown), only indirectly through the effect of the inlet steam temperature on the cover temperature.

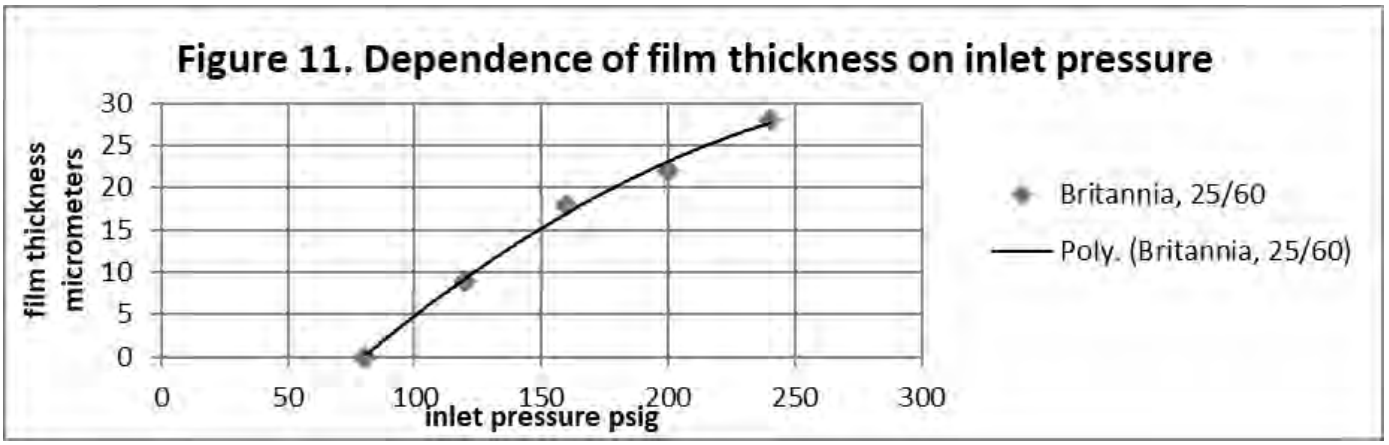
The film thickness does however change with engine speed. Figure 10 shows estimated film thickness on a Britannia at a cover temperature of 170 deg C. The film becomes thinner as speed increases at a given cut off. This is likely a kinetic effect. The rate of condensation is determined by the kinetic theory of gases, and with less time at a given cut off as speed rises, condensation in less. This is maybe as well, because otherwise with more strokes/second the rate of condensation would increase rapidly with speed(!) The film is also slightly thicker as cut off increases, again likely due to the longer time available for the film to form before cut off.

A final consideration is the effect of inlet pressure on film thickness. Figure 11 shows that the thickness of the film falls with falling inlet pressure-the thickness depends on $p^{1/2}$, so the impact of condensation on efficiency is reduced. This is not to advocate the use of lower pressure- the overall efficiency at 160 psi working at the much longer cut off needed to achieve the same power as at 240 psi would be far, far worse.

The important fundamental point emerging is that the maximum thickness over the film is limited by the poor heat transfer characteristics of the film itself. In the worst case cut off, speed and inlet steam scenario it is never more than 100µm when re-evaporation is occurring. The maximum amount of condensation that can occur is thus severely limited.

Initial estimates of the likely maximum level of condensation losses using the Hall model

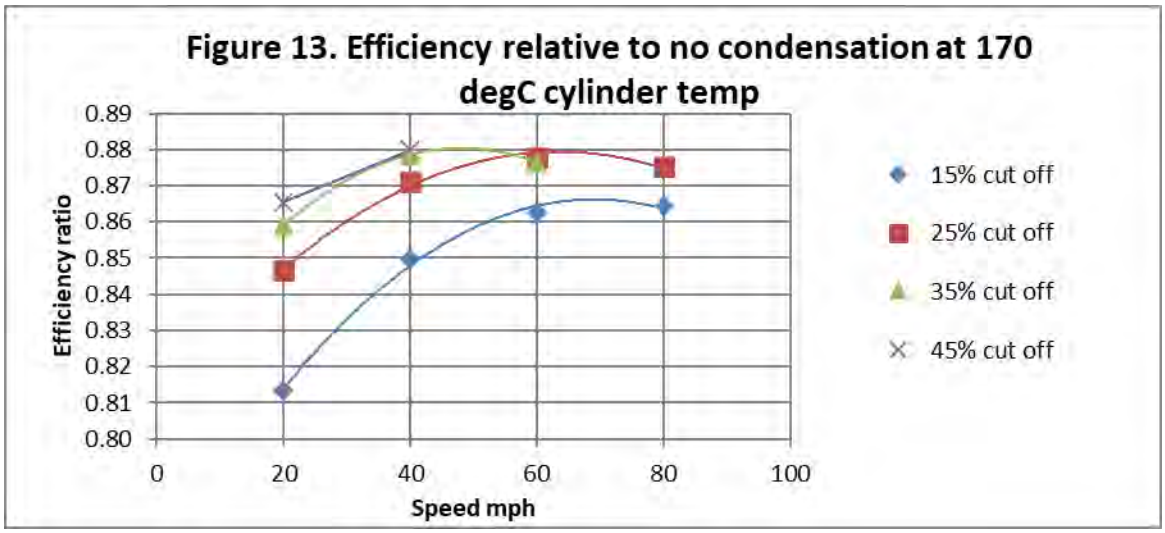




The Hall model can then be used to estimate the worst case scenario losses that can be expected from condensation. This is done by calculating the loss at the minimum cover temperature of 170°C identified at Altoona, given by saturated steam, on our trusty Britannia (Figure 12). Losses increase with cut off at a given speed. At typical cut off and speed combinations, the model says worst case scenario condensation loss is likely little more than 2250lbs/hr in these cylinders..

On this same basis Figure 13 shows what the net effect on engine efficiency would be at this same worst case scenario cover temperature. Expressed in % of tender feed, there would be a loss of 12-15% efficiency in the normal operating range due to condensation at the very low superheats- near saturated- at which this cover temperature would occur. % Losses are predicted

to be higher in short cut off working, because total steam flow is lower in shorter cut offs.



To be clear, the numbers in Figures 12 and 13 are maximum worst case scenario losses. Thus, whilst some of the improvement in efficiency with increasing superheat is due to a reduction in condensation, the majority of the benefit is down to simple thermodynamics- a hotter working fluid gives better efficiency.

These estimates are for a two cylinder design. Losses will be greater with 3 and 4 cylinder designs, though not proportionately, because the surface area of a single cylinder in a 3 or 4 cylinder design will usually be less than in a 2 cylinder design of similar tractive effort.

How reliable is the Bill Hall condensation quantification in Figures 12-13?

There is no reason to doubt the principles illustrated in Figures 9, 10 and 11, and the finding that film thickness does not depend directly on inlet steam temperature are correct. However, Figures 12 and 13, which attempt to define the worst case scenario level of condensation are much more questionable- here Bill's model is just a first pass at quantifying a very complex question.

Bill did not have to hand the Altoona data on how actual cylinder temperatures along the length vary, and believed that any attempt to calculate them from first principles would be fruitless. What his programme therefore does is calculate how much condensation would occur on the cylinder cover at a given temperature, and then doubles this to account for the piston face. He then assumes that a film of the same thickness will form on the cylinder walls up to the point of cut off, and on the clearance spaces. The increase in condensation due to the cylinder walls over and above the cover and piston total is thus estimated to be about 30% at 16% cut off, 60% at 33% cut off and 90% at 50% cut off. The exact figures will depend on the size of the cylinders and the operating pressure. From this, it can be seen that, the predicted increases in condensation as cut off increases (as shown in Figure 12) are not surprising.

This is however a gross simplification. The Altoona data show clearly that the cylinder wall temperature falls along the length of the stroke, which would increase condensation (see figure 5),. On the other hand the pressure falls, meaning condensation is less favoured,(figure 11) and Figure 5 shows that cylinders tend to be hotter towards the centre in longer cut offs, reducing condensation. So the question is how these factors balance out-an impossible job from a theoretical perspective.

So, the above analyses do not give a precise estimate of condensation, though they do put a clear magnitude on its probable size, given to the limiting thickness of the condensed film. Anyone wanting to argue that condensation losses are higher than this, or that steam jacketing is going to give large benefits will need to change the heat conductivity of the condensed water layer or increase the rate of condensation. Good luck with that.

An experimental determination of the amount of condensation and its contribution to the missing quantity

So, Bill's calculations of the amount of condensation are not precise. However, direct confirmation of its value can be gleaned from the aforementioned tests on BR5 73031 with differing levels of superheat[1]. Five levels were tested ca 380, 360, 310-340 (standard), 280 and 230°C. The two higher levels were obtained by electrical heating so the benefit in coal consumption is not known. From these tests, the relative contributions of leakage and condensation can then be estimated.

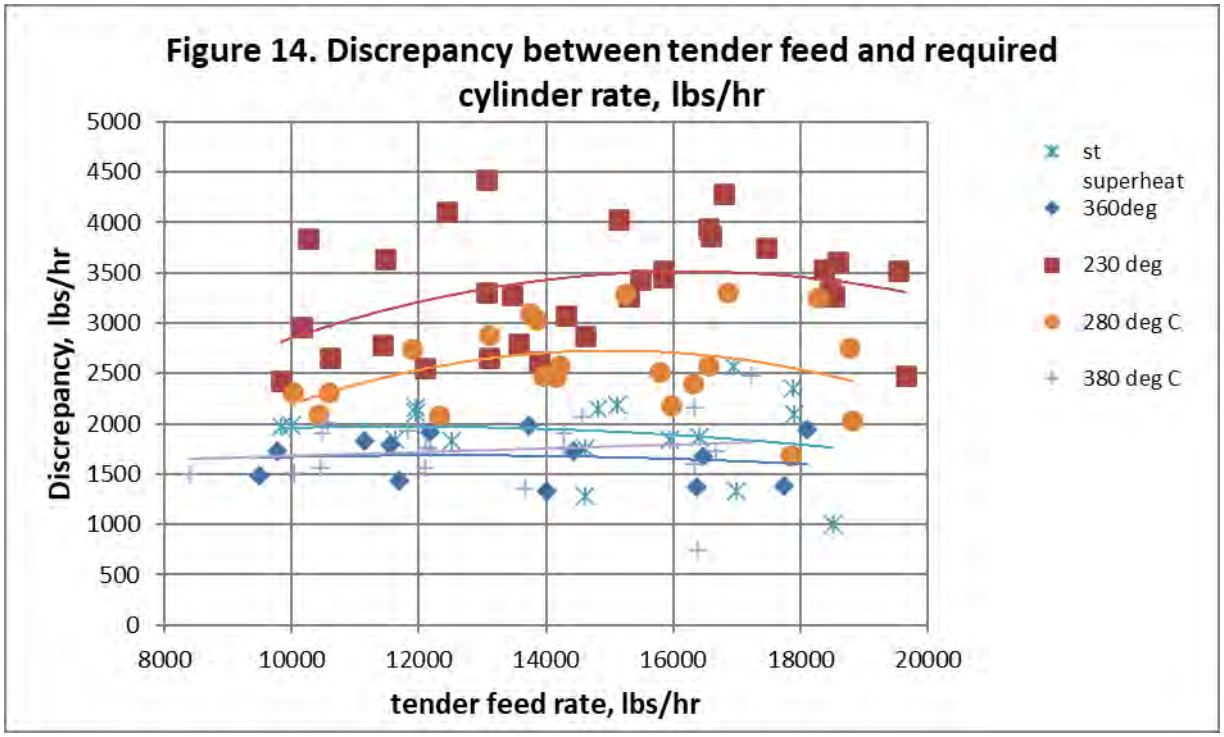
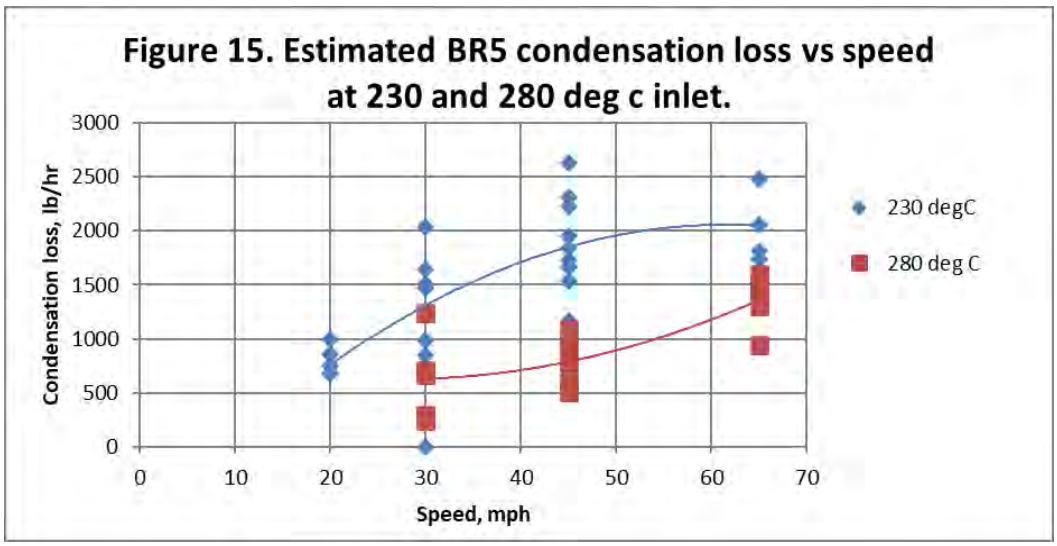


Figure 14 plots the discrepancy or missing quantity between the steam flow estimated by Perform to produce the measured power, and the tender water feed, at different superheat levels.

At the three highest superheat levels- standard, 360 and 380°C, the discrepancy is almost constant across the range of feed rates at about 1800lb/hr. The foregoing analysis says that at these inlet temperatures we can be sure no condensation is occurring, and so these must be leakage losses. However, at 280°C, the discrepancy has risen to about 2600lbs/hr, and at 230° C to about 3400lb/hr.

We can thus conclude that the level of condensation is about 800lb/hr at 280°C, and 1600lb/hr at 230°C. Given that this steam was somewhat above saturated, this is entirely in line with the estimate of about 2250lb/hr worst case scenario condensation shown in Figure 12. Thus, Bill's predictions on the magnitude of the condensation losses have been directly validated by the data to hand.



In Figure 14, there is considerable scatter in individual points at low superheat, due to the variety of operating conditions tested. Figure 15 replots this data vs speed, and shows that with 230°C steam the condensation loss increases to ca. 2000lb/hr. With GW 2row level steam the loss reaches about 1500lb/hr. In agreement with figure 12, the losses are lower at low speeds, and splitting the points in Figure 15 by cut off shows that losses are higher at higher cut off, also as shown in Figure 12.

So, whilst the Bill Hall model does not predict the condensation losses with great accuracy, it does provide a clear understanding both of the magnitude of losses and how cut off and speed influence them. The maximum measured loss is entirely consistent with Bill's estimates of film thickness. . We can therefore be sure that we now have a good handle on when and where condensation will occur, and the kind of impact on efficiency it can have.

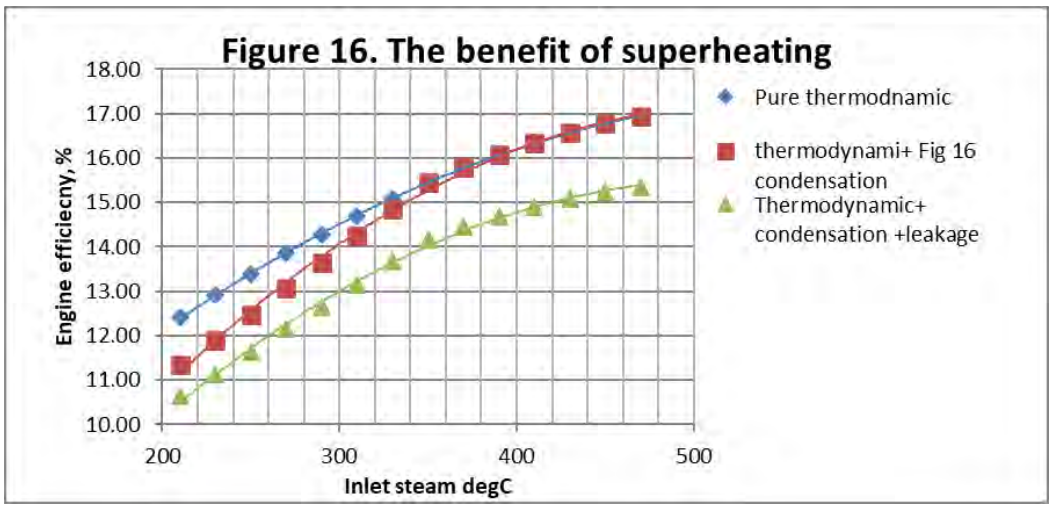
Leakage losses and the missing quantity

Looking at the other Rugby test results, mostly at inlet temperatures where little or no condensation would be expected, says that leakage of the order of 600-900lb/hr per cylinder and valve pairing is about the norm for these test plant condition locomotives. The 'missing quantity is a result of the incorrect assumption that tender feed= cylinder steam rate. Don't use more cylinders than you need is one conclusion. The design of the BR standards mean they are subject to high piston ring wear however, and on some BR9 tests this figure rose to 1500lbs/hr per cylinder and valve pair. David Wardale opined that steam leakage 'was by no means negligible, yet hardly ever quantified throughout steam locomotive[6]. The above results give an indication of its magnitude in ex works condition BR locomotives.

We may conclude that the majority of the missing quantity in types tested by BR is due to leakage. So the missing quantity still sits there grinning at us, impossible to quantify from first principles.

In earlier times, as Joe points out, piston ring design likely led to much worse losses. I have analysed test runs with Caledonian and L&Y 4-6-0s. The economy is horrendous, surely caused by poor piston and valve ring design. No wonder the missing quantity was a concern. Analysis of pre1930s UK test data is fraught with difficulty because of this.

The benefit of superheating



Given the foregoing, Figure 16 is a schematic of how superheat influences efficiency in Britannia cylinders operating at 21% cut off, 70mph- their efficiency sweet spot, as calculated by Perform.

The Blue line shows the pure thermodynamic benefit for hotter steam in leakage and condensation free cylinders- values of efficiency approaching 17% can be achieved, about 88.5% Isentropic Efficiency (IE). (100% IE is the maximum that thermodynamics allows under for given inlet and exhaust conditions). At the pressure and temperature limits of a firetube boiler, and low backpressure, the maximum efficiency possible at 100% IE is about 20%. Even then, about 80% of the heat input is going straight up the chimney. Expanding steam down to atmospheric pressure is a poor idea. Making it 10% better does not change the game, if there are alternative tractions in town.

The brown line shows the negative of impact of condensation as given by Figure 15 on efficiency. There is a significant loss of efficiency at low inlet temperatures, which gradually disappears by about 340°C, pretty negligible above 320°C.Below this temperature, steam jacketing would eliminate condensation, and lead to efficiency going back to the blue line, thus eliminating the negative.

If these cylinders operating with saturated steam were steam jacketed, the efficiency would go up from 11.3 to 12.4%. Alternatively, you could get this same benefit from non-jacketed cylinders going to an inlet steam of but 250°C. Any higher superheat and you win all the way over the jacketed ones. **Steam jackets are no substitute for superheating in simples.**

The green line shows the harsh reality of what happens when typical levels of leakage hit. At maximum inlet temperatures of 380°C typical in this country, you get an efficiency of about 14.5%, just over 80% IE. The benefit achieved at this point over a saturated engine comes out at about 40%- entirely consistent with the test plant data presented earlier. If you try to push this further to, say 430°C, the benefit is only a 4% improvement at most to 15.1%. In this sense, I think Joe was right to question Porta's wisdom in arguing for even higher superheat- diminishing returns have set in.

Further, raising superheat brings complications, particularly on the lubrication front. A salutary tale is the testing of the PRR T1 at Altoona. They threw the book at this design, and with a 69 element A type superheater, got inlet steam temperatures of 425°C in their striving to reach 6500IHP. They quickly realised this was more trouble than it was worth, and so blanked off some of the elements! These extreme high power outputs would never be achieved or required in practice, and the 69

element superheater likely gave acceptable lower levels at the lower outputs of their daily work.
I'm not arguing that these problems with very high superheat are insuperable, I just believe, as with most attempts to improve efficiency beyond latter day steam standards one needs to ask whether the trouble is worth it for the small benefit achieved. A simple to maintain loco, with fewest bells and whistles is a good loco in my view.

The relevance of test plant data and computer models to in service working

Given the clear cut evidence of the large benefits of superheating from test plants, fully supported by the best quantitative theory we have, the question might be asked as to how relevant these conclusions are to actual running. BR road test running confirmed the test plant efficiency findings on superheat- for example the BR4 outperformed the lower superheat Mod Hall at all steam rates on drawbar efficiency, but these tests were also under possibly unrealistic steady state conditions[7,8].
In actual service running two things are clear; firstly on many lines, steaming rate is usually not constant for periods of more than 10 or 15 minutes at best. This can lead to time lags between actual and expected superheat. For example, the highest superheat on a service test run of double chimney Castle 7018 was attained working easily down from Whiteball summit after a vigorous climb up the other side- presumably because the fire was still very hot[9].
Secondly, it takes some time for the fire to heat up and produce the flue gas temperatures which produce the superheat shown Figure 7. Wardale showed that it took over 10 minutes for the Red Devil to get up to a high level of superheat starting a heavy train cold from Pretoria[10]. The results on cylinder temperature shown in Figures 4-6 were obtained after a warming up period of 15 minutes, so we can be confident cylinders will be mostly warmed through in that period. The tests then ran for more than an hour with no further significant changes, albeit with a slight uptick at the end.
During warming up, steam will be being used to warm the engine through, and additional coal will need to be burned. Some estimates from a Mod Hall run said that it used about 2500lb steam to heat things through before Maidenhead, during which time superheat would surely be less than the expected value[10]. On service runs, steam might also be used for train heating, and there will be other extraneous losses, particularly when drifting. Both Wardale and BR showed that as time progressed after overhaul, engine efficiency decreased due to increased leakage[6,11]. All these additional losses will tend to blunt the benefit of superheat, proportionately greater as superheat increases
But in BR's exchange trials in 1948 it was clear that in the passenger tests, the high superheat A4 outperformed the other types with the low superheat King being worst, and in the mixed traffic types the higher superheat LM5 and B1 were superior to the GW Mod Hall[12].
On the 1948 freight engine tests, things were less clear cut, for these slow speed coal drags were running on low power or drifting a lot of the time, when the superheater would not kick in. It took 7 hours for an 880 ton coal load to be worked over the 116 miles up to Acton from Severn Tunnel Junction, more or less all downhill for the last 95miles. Further, all trains were badly checked, and as a consequence standby coal losses will have further eaten into superheat's advantage. The skill of the fireman in dealing with very uneven power requirements will also have been in play.
Overall, we might expect in service benefits of superheating to be less pronounced than on the test plant due to the exigencies of operating conditions, but nonetheless there can be little doubt that from other tests that whatever the variations from test plant ideals, superheat provided large benefits in daily express and fast freight running.

Do you need a superheater in all circumstances?

There are some types where it is not worth fitting a superheater, e.g. shunters, and small freight locos used for trip working, shuffling s from one signal check to the next. These often do not sustain high steam rates long enough for the fire to burn hot enough to produce the hot gases needed to allow the superheater to work. They probably waste most of their coal in standby losses in any case.

The effects of scale

The foregoing all relates to mainline locos. An important subsidiary question is how these conclusions are affected by scale. As noted above one of the drivers for Bill Hall's work on condensation was the poor efficiency of Ravenglass and Eskdale Locos. He firmly believed that these would suffer far more condensation than main line locos because of the small size of the cylinders.
There is no data I know of measuring the cylinder wall temperatures of such types. All that one can do is estimate how much condensation would occur at various cover temperatures.
Modelling a saturated version of RHDR Pacific Northern Chief, working with saturated steam of 125 psig with a saturation temperature of 180^{o}C, says that at a cover cylinder temperature of 170^{o}C efficiency would fall by 12% due to condensation, but at 150^{o} C efficiency would be 46% lower.

So Bill was likely right about the pernicious effects of condensation in small scale types. No wonder Northern Chief had a superheater added in 1952! This loco would steam hard enough and long enough for the superheater to be effective. It follows that small scale locos are not good models for large ones when it comes to condensation.

Summary to this point

- The benefits of increasing superheat in simple engines are clear from both BR and Altoona data. It is the largest single benefit to engine efficiency ever achieved.
- The benefits relative to saturated steam are due to both the thermodynamic effects, and reduced condensation, and engine efficiency values of up to 50% higher can be achieved.
- Condensation can occur if inlet steam is below about 320^{o}C.
- Bill Hall's theory plus Altoona data say the properties of the condensed film put a clear maximum on the efficiency loss which it can cause. In the worst case scenario of a saturated engine the loss is less than 15% for full scale two cylinder types. The 73031 data confirm this. So the benefit of superheating is a largely due to hotter steam expanding more efficiently.
- A 15% efficiency benefit could be delivered by jacketing the cylinders of main line saturated engines, lesser amounts until inlet steam rises to ca. 320^{o}C.
- The benefit of jacketing a small scale saturated engine is likely much higher.
- Steam jacketing removes the negative of condensation. It cannot provide a positive benefit to efficiency of itself. This is important when discussing the 160A1 results below
- Bill's perform model is the best theoretical guide to power and thermal efficiency, even though it cannot estimate the amount of leakage present in any given loco..
- In real operation the benefit of superheating on coal consumption may not be as high as test plant and theory suggest, because of operational factors that blunt the benefit.
- The idea that superheated steam is unable to heat cylinder walls- a justification for the results of 160A1 discussed below is flatly contradicted by the Altoona data which shows that cylinder wall temperatures increase at all points as steam temperature rises above saturated after no more than 15 minutes warm up, perhaps less. The idea is totally out of step with the fundamentals of heat transfer theory, which says that if the cylinder walls were below inlet steam saturation temperature, the first thing that would happen is that a condensed layer would form and start heating the cylinders.

The Chapelon tests on steam jacketing on 160A1

So it seems we have been led down a rabbit hole, by the claim that steam jacketing is a viable alternative to superheating by Chapelon's work on 2-12-0 160A1. We therefore need to examine the data to see what might have been going on.

The context - the condensation problem with Compounds

In contrast to British and American practice, some French Railways, notably the Nord and PLM decided to stick with Compounding after the introduction of superheat. The majority of French locos were simples, but the Compound vs simple debate was never fully resolved there, in part because the simples tested did not to follow best practices from elsewhere[16]. Chapelon was a PLM man originally, so a Compounder through and through. He estimated its efficiency benefit was in principle about 9%- well worth having[13]. But in the 1930s he was struggling to make some ancient P-O 4 cylinder Compounds- 2 HP feeding 2 LP - efficient[14]. (The P-O did not build compounds after WW1). He succeeded in raising their engine efficiency in normal operating to almost match UK simples[15]- the originals must have been truly awful. Superheating had not delivered in this narrow context. even though with saturated locos, Compounds were more efficient than simples.

Chapelon's improvement work followed the precepts that Woodard, Head of Lima Locomotive works in the USA, used for his ground breaking 'Superpower' designs of the 1920s, precepts derived from the Altoona tests described earlier. Thus he went to higher steam temperatures, striving for values in excess of 400^{o}C, higher pressures, feed heaters and improved steam passages. These were not Chapelon's own ideas as often stated. He did create an excellent exhaust design, the Kylchap.

The problem he was dealing with was that in the superheated Compound set up, the exhaust steam temperature from the HP cylinders (i.e. inlet temperature to the LP cylinders) can be quite low, and so condensation on the large surfaces of the LP cylinders is then a real possibility[15]. So, superheat was not delivering its potential, and the cause was 'wall effects', better put, condensation.

From the data it looks like that this problem can be eliminated with inlet steam temperatures in excess of 400^{o}C as on the famed 4-8-0, or by going to the IHP+2LP set up of 242A1, which tends to give higher LP inlet temperatures. These temperatures

also depend on the ratio of HP:LP cut offs used, so LP inlet temperature is a complex subject. His 4-8-0, 141P and likely 242A1 seem to have solved the problem, and this gave them engine efficiencies superior to best UK simples. Some spectacular high power running was without doubt achieved with these types. This allowed Chapelon to position his Compound designs as a world beaters[17], but just as with the PRR claims of 6500IHP for a T1, these were obtained at firing rates that were totally uneconomic and out of the question as a practical proposition. For publicity purposes only.

Thus, Chapelon's experience was that superheating had not delivered its potential In Compounds, and that 'wall effects' were the problem. This left him acutely interested in a subject that had become largely irrelevant elsewhere. Problems avoided if they had focussed on optimised simples!

Analysing French data

I spent five frustrating tears a decade ago trying to get to grips with French Compound data, believing at first that the anomalies therein were due to the language barrier. However, it gradually became clear that, unlike the early Altoona test described above, the purpose of French reporting was usually to show how clever they had been at making design improvements. Only the data supporting these claims were reported, which makes it difficult if not impossible to figure out what was actually going on. This is annoying since some of the data look odd[15].

For example, the reported locomotive resistances (LR) of these Compounds are so high – about 1200HP at 75 mph compared to 650 for a similar sized Duchess, that even the best Compounds are not convincingly superior to simples in drawbar efficiency. I have seen no convincing explanation of why this should be. It could be they are erroneous values, a result of inadequate road test protocols being used, or adjustment of data to fit cherished views about what LR was- this happened in this country. I tried in vain to make these cases stick, so had to accept high Compound LR as the reason for their sub-par drawbar efficiency[15].

Further there were a number of instances where, within the same report, contradictory versions of the same data points were reported. This sloppy reporting does not matter if you can reasonably claim to have made improvements-and in most cases they had- everyone likes feel good stories

For many steam fans, Chapelon's highly enthusiastic feel good stories about Compounds have made him a saint. Over the years he has had many hagiographers re-trumpeting his triumphs. In his final writings his enthusiasm led him off into fantasy land with claims he could design an 8000IHP 4-8-4 and a 15000IHP 4-8-8-4, based on his interpretation of the 160A1 results. All good copy for enthusiasts. Personally, I am no fan of claims which cannot be validated from published data.

It is with these perspectives that I will look at his claims for his steam jackets on 160A1.

The Chapelon 2-12-0 design, 160A1

I have seven sources; Locomotive a Vapeur (LAV)[17], Stora's website[18], 'La Centenaire de la Locomotive Crampton' (p251-254)[19], Steam Locomotive.com and two extracts sent to me by Joe, which I can only label 'p24'[20] and 'p254-256'[21]. In addition there are the Chapelon letters published recently in the ASTT newsletter[22].

160A1 was a six axle Compound freight locomotive, designed to haul heavy loads up steep gradients. It was a 1940 rebuild of an older 2-10-0 P-O freight locomotive. Adding a sixth coupled axle was key to this goal, allowing higher tractive effort at an acceptable adhesion ratio.

The 160A1 design used the 1HP feeding 2LP principle, six cylinders in total. It was believed that wall effects' would be particularly pronounced at low speeds**,** (I am not sure why), which made it important to eliminate wall effects once and for all. Chapelon therefore decided to address the problem by re-superheating the steam between the HP and LP cylinders. An excellent idea.

He also steam jacketed both HP and LP cylinders. With hindsight, this is a bit belt and braces. The saturation temperature of 270 psi HP inlet steam is about 212^{o}C. At the superheat achieved, the exhaust temperature from the HP cylinders likely exceeds 212^{o}C, so the wall temperatures would surely exceed this temperature, so no condensation. Similarly, with re-superheat, at inlet temperatures of 310+^{o}C to the LP cylinders, no condensation would occur there either. There would however be some benefit if saturated steam were used.

The design comprised

- Two inside H.P cylinders:, 20.47'' bore, 21.25'' stroke, driving the fourth pair of wheels;
- Two inside L.P cylinders of 20.47'') bore and 21.25'' stroke, driving the second axle
- Two outside LP cylinders, 25.1'' bore, 25.59' stroke driving the third pair of wheels.

This was a great lumbering beast weighing over 220 tons, and with 55" drivers and a 100 + ton adhesion weight, the resistance

of machinery alone between the pistons and wheelrims alone would likely be of the order of 200HP even at 40mph. It really was a low speed design. And four inside cylinders to look after!

The boiler was divided into two sections, with the front two meters housing the pre-heater. A 39 element Houlet superheater with 72 sq m of surface served the HP cylinders while a 68 element modified Schmidt Type A (110.46 sq m) supported the LP cylinders. However, the presence of two superheaters was a pain and there was every reason to try to eliminate one. On the 1HP+2LP set up, the HP cylinder produces a relatively small power output, so this was the one to check out.

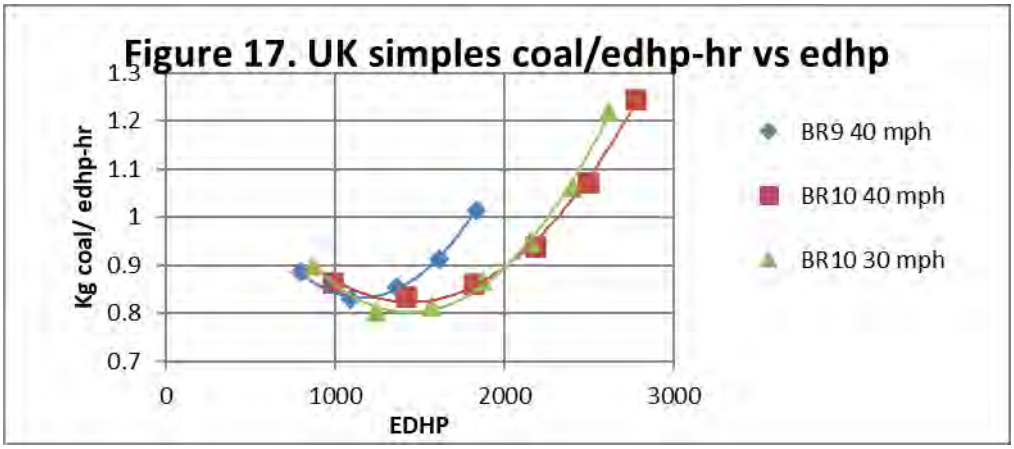
The steam temperatures delivered to the HP cylinders were said to be 290-340^{o} C, to the LP 270-380^{o}C. Chapelon said that a virtue of this design was that it did not need 'exaggerated' HP inlet temperatures, as found on e.g. 242A1, required to eliminate LP condensation These had caused carbonisation problems -getting an efficient superheated Compound is hard!. So, the lower HP superheat on this loco was a good thing.

Reported test results on 160A1

Meaningful data is scanty.

Road test efficiency and power

'Crampton[19]' illustrates drawbar data on a number of 1948 test runs, most notably southbound on the PLM main from Laroche to Dijon, where it hauled 1600tons in 2 ¾ hours over the summit at Blais y Bas, at an average DHP of 1700 and speed of 35mph. The graphic shows nearly 2000DHP being sustained at ca 37 mph on the easier grades from Flogny to Les Laumes, with a couple of short 2500DHP peaks, presumably for show. 160A1 did what it was designed to do. The reported TE of 36 tons on the return was only a momentary spike after a slip, a typical over optimistic Chapelon claim.



The striking thing about this run is the assertion that the coal drawbar economy was 1.23kg/ dhp-hr. Chapelon thought this figure 'very low', but it is awful. Figure 17 shows actual data for a BR9[23], adjusted to coal of the Bthu/lb normally used in French tests, and a hypothetical BR10 2-12-0 simple with suitably increased TE and grate to allow it to compete more fairly with 160A1. At 1700 EDHP, the BR9 was near its limit, and gave 0.95kg/dhp-hr, but the BR10 would consume little more than 0.8 Kg/edhp-hr. I am confident that the relatively minor extrapolation from the BR9 to BR10 gives a reasonable estimate of economy. The BR9 figures are based on live steam injector only, whereas 160A1 benefitted from a feed heater, which should have provided ca. 10% benefit.

That same year there was repeated running on the mountainous routes from Lyon to Roanne and St Etienne. 160A1 developed enough power to haul the maximum allowed tonnages on four different routes without any helper engines, but here the drawbar economy had worsened to 1.51kg/dhp-hr. Even so, the elimination of helper engines would have led to a net improvement in coal used.

An efficiency of 0.85 kg/ dhp-hr is quoted in the letter of 20/2/51., also by Stora[18], and mentioned in LAV as being at 20km/h[17]. Drawbar efficiency can be best at low speeds, especially in Compounds, so the contrast with the 1.3-1.5kg/dhp-hr quoted earlier is not too surprising.

The water consumption southbound was 8.54kg (18.8 lb)/dhp-hr. Even allowing a very high 300HP for locomotive resistance at the average speed at 35mph, the cylinder efficiency would be about 16lbs water/ihp-hr, again totally less than stellar. I am confident that a 6 axle BR10 working at 275psi would have done at least 10% better. Nonetheless, in his letters[22], the claim is made that 160A1 'may well be' 30-40% more efficient than ordinary locomotives- which ones not stated.

One possible reason for the poor efficiency was that 160A1 was fitted with Walschaerts driven Lentz oscillating cam poppet valve gear, a technology already rejected in this country by Gresley 10 years previously. Oscillating cam Poppet valves of that

era tended to leak like sieves, so their use could well explain the low efficiencies measured on the road tests.
So, we have a heavy complicated monster meeting the haulage brief, but with poor engine efficiency
Boiler efficiency is said to be 'remarkably' better than Pacifics (20th June 1950)[22]. This is hardly credible-how could the heat transfer efficiency have improved 'remarkably'? Heat transfer efficiency is already around 80% in well-designed boilers. There's not a lot of scope for getting better.

Reduced superheat tests

From the Chapelon letters[22] we can infer that the tests with reduced superheating took place in late 1950. The testing began on constant speed road tests[17]. Table 1 shows the only known data points on IHP and efficiency with de-superheating, all at constant speed and cut off[21]. IHPs are quoted which says they may well be measures done later at Vitry.

Table 1. 160A1 power and efficiency with different superheats.

The first claim made is that if the HP cylinders are fed with saturated steam and LP still re- superheated, power is little affected, and there is only a slight increase in calories/ihp-hr (a sensible efficiency metric for this set up) of 1.5% at 60km/h and 3% at 20 km/h. It is said the LP superheat increases somewhat in this scenario[21].It is then said that the increase in consumption with the LP also de-superheated is 6% at 60km/h and an 13% at 20km/h (final column)
Given this, Chapelon's conclusion is clear:
… In conclusion, the tests on 160A1 have shown that the ideal machine would be a compound admitting saturated steam to jacketed HP cylinders, and superheated steam to the LP cylinders'.[21]
The assertion above is repeated in his letters (10th May 1955). So Chapelon's own words and data say the claim made above that steam jacketing allows complete de-superheating is false.

Modelling this system with Perform

I find the report that eliminating HP superheat does not reduce efficiency surprising, and worthy of investigation. I therefore attempted to model this with Perform to see what was going on. There is unfortunately no reporting of important data inputs- inlet steam pressures and temperatures in either HP or LP cylinders, or coal and water consumptions.
Because of this, you have to make too many guesses to draw precise conclusions. However, you can get a sense of what was going on, using some arbitrarily chosen inlet temperatures, and assuming that the HP exit pressure= LP inlet pressure, as Chapelon claims[17].
These calculations say that:

- The first oddity is the choice of 48HP+32LP cut offs, which contrasts with the equal cut offs advocated by Chapelon himself[14], and the PLM view that the LP should be fixed at a high value of 62.5%. Perform says that the steam flow at 60km/h at 45HP/45LP would be similar to 48/32 but the power 10% higher. Getting top efficiency from Compounds is complicated!
- The use of 48HP/32LP leads to very high receiver pressures- in excess of 120 psi- with the likelihood of large HP compression loops not present in the 45HP/45LP configuration. This could lead to relief valve losses further killing efficiency. High receiver pressures had been a target in the past to raise receiver steam temperatures. This could explain why 48HP/32LP was chosen, but with re-superheat this driver would no longer be relevant.
- Even at 48HP/32LP, the efficiency of the fully superheated engine should have been far better than the best simples at around 12lbs steam/ihp-hr, not worse as the road tests seem to show. This supports the leaky poppet valve idea.
- The consequence of eliminating HP superheat is that at a given cut off combination, as in Table 1, steam flow goes up, because saturated steam is denser- nearly 40%. Consequently, the steam flow at equal cut off with the LP still superheated increases by ca.10%.

Speed	Cut off %		IHP with or without HP superheat	IHP all Saturated	saturated: super-heated power	Deterioration in economy with no superheat
Km/h	HP	LP				
60	48	32	2600	2375	0.91	13%
40	48	32	2120	1920	0.91	
20	48	32	1260	1185	0.94	6%
40	62.5	40	2590	2405	0.93	

- The HP power is much diminished with the loss of superheat, but with 10% higher steam flow to the LP cylinders there is a large gain in power. The net is that there is little difference in power, just as Chapelon claims.
- However the higher steam flow means that at equal power, efficiency should go down, contrary to what Chapelon claims.
- There is a similar picture when the LP is de-superheated- steam flow is now nearly 20% higher, and the power falls by about 8%, as Chapelon admits.

There is little doubt that if you lower superheat, steam flow will go up at equal cut off settings in normal circumstances, so if IHP remains similar, the claim of little efficiency loss remains puzzling.

The explanation given in the introductory assertions, which imply that wall effects somehow depress the efficiency of the HP cylinder when using superheated steam is surely a non starter. It is in the first instance quite simply untrue that cylinder wall temperatures are lower when superheated steam is used as shown at Altoona. And, in any case, in this locomotive the HP cylinders were steam jacketed so would certainly be hot. And even if the cylinder walls were cold, how would this affect the power produced? The steam still would expand according to the laws of thermodynamics. I think we can rule out the idea that raising cylinder wall temperature by ca 40^{o}C through steam jacketing can directly improve efficiency- how could it? As Mr Spock would have said, 'It's all quite illogical, Captain'.

There are two possible explanations, I think. Firstly, the use of the calories in steam/ihp-hr metric for efficiency involves some tricky measurements. You need to know the steam flow through the HP cylinders and HP inlet temperatures and pressures accurately. You also need to know the HP outlet temperature and pressure, so you can calculate the enthalpy added by the re-superheater. Chapelon does provide data of this nature in his tome on 4-8-0[24], but it is neither self-consistent nor compatible with other measurements. Further Perform says that the HP exhaust will be wet, with an enthalpy below dry steam at the measured temperature. I can see no reliable way of estimating the exhaust enthalpy in this circumstance, and if this is in error, the efficiency estimate will likely be too high. Coal consumption would be easier to measure, and more definitive, but figures are not quoted.

A second possibility starts from Chapelon's statement that when the HP is de-superheated, the LP superheat goes up. There's no data to go on, but Perform says this would help bring efficiency back up, and with the steam jacketing, the HP cylinder efficiency would not be as bad as it might otherwise be when using saturated steam. A perfectly reasonable explanation of the finding.

If I might add a final wild, unsupported speculation- is it possible that if the Poppet valves do leak, leakage is less when superheat (and possibly speed) is low?- less thermal stress on the valve seats?

So an increase in LP superheat when HP superheat is reduced provides a possible rational explanation as to why there is no significant loss of efficiency in this very specific circumstance-nothing to do with wall effects. Extrapolating this finding to make the general assertion that steam jacketing can replace superheat is however ludicrous.

The above commentary is a perfect example of the frustration of trying to analyse French data- lots of claims, often unlikely or confused, with no hard data to back them. Such claims are not safe. What you are reading is a sales brochure for the ideas of an ardent advocate of steam, not a trustworthy engineering report. . The reports are only good for making ardent Chapelonistas happy.

In conclusion I would also say that my view of the definition of a good design is one that can be driven by idiots with little maintenance. 160A1 is the very antithesis of this. It has complexities on its complexities. Keep it simple, stupid. 160A1 ran 25000 miles in its 14 year life before being scrapped in 1954. The SNCF, at least, was not stupid. Why are we still talking about this inefficient monster?

Where might steam jacketing help?

So, whilst steam jacketing can never be a substitute for superheating in simples, if fitted to the LP cylinders of a conventional Compound, it ought to solve the problem of LP condensation that had bugged the designs for decades. Work with high but not excessive HP inlet steam temperatures, and the full benefit of Compounding would be delivered. If only Chapelon had tested jackets on them! Re-superheating would add a bit more benefit, but add significant complexity. The 160A1 set up was an over-elaborate response to a relatively simple problem.

Steam jacketing should also help on non-superheated types, especially smaller scale ones, and those whose work does not demand the high steam rates that are required for superheat to kick in- for example trundling coal loads up to London at low speed. Jackets had been all the rage in saturated steam times[3].

Conclusions

Among the points made casting doubt on superheating were:

A. *Chapelon in his later writings found superheating to be unnecessary with steam jackets.* This is not what Chapelon said. What the 160A1 data do point to is a somewhat surprising maintenance of efficiency when superheating was removed from the HP cylinder, quite specific to this oddball design.

B. *The original purpose of superheat was to prevent condensation inside cylinders, known in as the "missing quantity"* In part, maybe, but it was also clear by the 1880s that increasing the temperature of the working fluid in a heat engine would give significantly improve efficiency.

C. *Superheating steam is a very inefficient way to heat cylinder walls due to its poor heat transfer coefficient and the lack of turbulence in a large cylinder.* This concept is completely refuted by the Altoona cylinder temperature data. The theoretical premise on which this idea is based is not sound.

D. *The (originally expected) efficiency of superheating is curtailed by the rather limited amount of steam available for heating up cylinder walls.* The evidence that superheating provides a large benefit is overwhelming. It is likely true that in early superheated Compounds that superheating was insufficient to heat the LP cylinder walls enough to eliminate condensation

E. *Superheating, though undoubtedly effective, falls short of original theoretical expectations.* Patently incorrect for simples. The benefits attained are high and exactly in line with theory. However, in the specific case of a superheated Compound, condensation in the LP cylinders may lead to lower efficiency than hoped for.

F. *Superheat undoubtedly reduces water consumption. Once condensation is eliminated the thermal gains are very limited, but water is cheaper than the coal needed to superheat.* Incorrect. The gains from reduced condensation are relatively small and the benefit of superheat continues to increase way beyond the point at which condensation ceases.

G. *The base case also rests on an optimum all-saturated boiler drafted against a less efficient superheated version, which was never done because the false belief in superheating was so strongly held.* The evaporation achieved by a saturated boiler is indeed likely to be higher than a superheated one at a given combustion rate. However the % difference in enthalpy between superheated and saturated steam is small, so the loss will be small. The % improvement in engine efficiency from superheating is much larger than this enthalpy differential, hence major improvements in coal consumption from superheating.

Postlude

I hope it is clear from the foregoing that my only interest is to analyse reported data, summarise them, and cross check them against the laws of physics as we know them.

The question for me is why Chapelon was so excited by the 160A1 tests. Here's my take.

Chapelon was a passionate steam fan, fighting in Quixotic fashion against electrification in the 1930s. For him, to beat electrics, more power was the real driver, with more efficiency a means to this end. His background with the PLM made him a diehard Compound man. Compounding for him was the way to go. Read your LAV. He was able to show some quite remarkable power outputs from Compound designs, albeit under hopelessly uneconomic conditions.

He had however been struggling for a decade or more in the 1920s to overcome the Achilles heel of superheated Compounds, LP cylinder condensation- wall effects. Superheating had improved Compound efficiency, but not delivered the hoped for benefit over superheated simples. The first solution he came up with was very high HP superheat, later going to the 1HP+2LP set up. High HP superheat led to carbonisation problems however.

One logical way to tackle LP condensation was through re-superheat, with cylinder jackets for full security. Hence 160A1. Having two superheaters was a pain however, but he was delighted to discover that there was no power loss when the HP superheat was removed. Nor, apparently was there any loss of efficiency. Crucially, the power delivered met expectations. So there you have it. Chapelon concluded that 'ineffective' superheat could be overcome with steam jackets, giving a template for future Compound designs- magical 4-8-4s and 4-8-8-4s. He claimed incorrectly the efficiency of 160A1 was great, but it was poor, possibly a consequence of poppet valve use.

Chapelon had no quantitative understanding of the heat transfer process – this was not possible in his day. Modern theory, backed by data says his claims are suspect and in any case his solution to the condensation problem was way more complex than needed. All that was needed for high efficiency was LP steam jackets, high but not excessively high HP superheat, and the

hoped for 9% Compound efficiency boost would be gained, assuming Locomotive Resistance was at UK levels.

Would Compound designs be thus perfected? Could be debated endlessly. For my money, I would simply try a 280 psi fire tube A1 boiler on a Gresley Hush hush with LP steam jackets to find out. Possibly 14% more efficient than Tornado -that would be the one to beat.

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STEAM TRACTION - THE CASE FOR STATUTORY PROTECTION

Owen Jordan

In the threequarters of a century since the start of the railway preservation movement, what has now become branded as the 'Heritage Rail' industry has never operated within a framework modelled upon and intended to support and nourish its activities. More to the point, Steam Traction, a key feature of the business model of heritage rail for over half a century, has experienced mixed fortunes in its battle to survive, on the national network, never having had the automatic right to access. Is it time this was changed?

In the Beginning…

The preservation of the steam locomotive started in modest circumstances; a handful of historic items set aside by the commercial railway companies as being worthy of a place in a museum. And for the most part, placed in one. But even at the beginning there was always a 'let's get that old engine out and run it' tendency of loco owners. After a century of steam traction, the skills and infrastructure of the steam railway – within which the locomotive developed, lived and breathed – remained a core part of society, supplying the essentials – transport of goods and people – that were needed. This infrastructure was not to have its existence questioned, let alone threatened, at this stage, indeed successive generations of steam traction would continue to use – and develop – this essential infrastructure. There was not however to be a second century of steam traction in line with the first.

Preservation of the steam railway – as an integrated infrastructure including everything the locomotive needed as a matter of course – got off to a bright, if modest, start following the creation of a nationally-owned network in the lead-in to the second half of the twentieth century. At that point however the locomotive, whilst far from being the critical transport traction it was half a century earlier, was still secure on its home ground. A ground built around the needs of the steam locomotive. The preservation movement, as a consequence of having that infrastructure and the locomotives to operate upon and within it, gave no thought to a future that was going to be anything other than more of the same. Closing railways piecemeal was not, in itself, a threat to the steam locomotive except in the total numbers and distribution over the country.

Fast forward then two decades, and reality was biting everyone with an interest in the steam locomotive. Hard. And then some. The national rail network was home to just four operational steam locomotives, and the one with an all-line ticket could rely upon a servicing infrastructure that equated to absolutely nothing; no water points, no coaling plant, no turntables. In spite of continued enthusiasm, there was no welcome from British Railways. A whirlwind of change, equating the steam locomotive with everything that was wrong with the railway at that point – in ignorance of a correct diagnosis – had swept the steam locomotive and its infrastructure into the dustbin of history. It was no surprise that the unlucky locomotive decided to emigrate…

That said, despite the open hostility of the national rail operator, and the scorched earth approach to steam traction facilities, the UK rail system in 1970 was still very much a place where the inbuilt requirements for operation of the steam locomotive were very much in evidence. Despite the war on steam locomotive infrastructure, there was an excess of facilities that allowed - had it been allowed – the steam locomotive to run and without too much by way of the hindrances created by the deprivation of fuel, water and servicing. Even the loco works were still full of steam age machines busily being found wanting for repair of diesels… The continued presence of suitable facilities was made patently clear in short order as the steam ban was reversed, and while turning, watering and coaling locomotives was a bit of a challenge in some places, the heritage main line steam traction business on the national network was in business. After a fashion at any rate.

In parallel with this, the preservation railways, well provided with expertise in addition to the enthusiasm that was critical – if generally a bit long in the tooth – set up shop and some thirty full sized train sets, complete with numbers of steam locomotives, were soon in business, too. Starting literally from scratch, the heritage rail industry hitched itself up the learning curve with remarkable alacrity. A number managed to salvage various service items for steam traction on their lines, but no-one managed to get a working turntable. The serious engineering kit of the pre-electronic age - lathes, presses etc – needed to keep the locos running was fortuitously long lived and multi-functional, and not all of it went for scrap. Enough landed in places where it was made available for the steam locomotive to survive, if on rather short commons. Steam locomotive engineering on a commercial basis was also up and running outside the nationalised railway workshops.

The Third Century

Fast forward then, half a century or so after the Revocation of the Edict of Nonsense. All of the players who took action to prevent the steam locomotive becoming static history are now well past pensionable age. There has, overall, been a successful handing on of the duties and responsibilities from one generation to the next, without which the operational steam locomotive would indeed have become history. But.

Times have changed. The era of not-very-accurate heavy engineering, epitomised by the steam locomotive, has been totally superceded by the rapidly shrinking size – and cost – of things made in the era of tolerance defined manufacture. It is not permissible to 'think sloppy' in terms of how a steam locomotive rides as a vehicle; modelling of how a vehicle rides, to take just one example, is now essential.

The national rail network has been ruthlessly pruned of most of its non-essential bits. And some essential bits as well… Welded stiff section rails, granite ballast, concrete ties are now the hard road norm virtually everywhere; not what once was... Some termini don't have a run-round loop, signals have appeared in places where you can't stop a steam locomotive. And wouldn't want to re-start one, either; slipping – part of the challenge of the steam locomotive at low speed – can lead to all sorts of frowned-upon things. Many lines have traffic at speeds the steam locomotive cannot even meet halfway. Catenary and power supply are in the way of chimneys, coal bunkers, injector overflows. Throw a bit of clinker – or perish the thought, fire – out at your peril, but the day to day experience that would - and did – allow crews to chance it and arrive at a water point with no water in the tender and the gauge glasses empty, is long gone. As, of course, are the water points… Dry weather leads to bans, due to non-maintenance of the lineside that amounts generally to total neglect.

The heritage lines started off with locomotives they could afford, but very quickly discovered they couldn't haul enough people to make ends meet. What were once branch lines, used to seeing two, three or four coach trains and a small – usually tank – engine, now hosted standard sets of eight coaches plus a main line mixed traffic loco – or larger - as standard. After running through the options, traction in the third decade of the twenty first century is now mostly what was 'modern' in 1955. Very little heritage traction is under pension age, and most of it is doing a job it was not designed to do; low speed short runs, frequent stops. Hired-in express locos run half the time tender first, since the turntable problem has taken over half a century to solve.

In 1975 there were several hundred ex-BR steam locomotives under thirty years old, and pre-nationalisation stock only marginally older; even a decade and more in the scrapyards and awaiting restoration on heritage sites had not eliminated the inherent repairability of the steam locomotive. Today there is just one loco less than three decades old on the standard gauge. If all current plans proceed as planned – which they will not – there might be ten new build full size steam locomotives by 2030, of which two might be suitable for the heritage railway's operational parameters. The costs of building new to standards that will be acceptable on the national rail network are very high indeed. Rebuilding a twenty year old ex-Barry wreck in 1975 – which taxed the preservation movement sorely at the time (half took more than a decade to complete, some several decades, some are still twinkles in an ageing eye or three..) – cost small change compared to today's new builds.

The rail industry in the 1950's was sore taxed with dealing with the locomotive coals supplied by British mines; it was a factor in the decision to end steam haulage. Today we have no UK steam coal sources. Environmental considerations are threatening to ban the use of coal altogether. Water fit for locomotive use is not widely available; all the lineside treatment plants that supplied loco water - are history, and most river extraction potable water in lowland England is chemically treated to make it fit for domestic use. Good clean untreated water suitable for locomotives to drink neat comes out of my taps in West Wales, but I am somewhat of an exception.

The Case for Protection

It is very easy to miss the point. The steam locomotive was a - the - pinnacle technology of the second phase of the Industrial Revolution 1820-1890. The technology of steam power made the transition from stationary power by machines made of wood and stone with a few metal bits, to an all-metal mobile self-contained power unit. The objective of the designers and builders was to make – as best they could at the time - a machine without limit of life; relatively high levels of wear but repairable at reasonable cost. Because of its high use profile – visible to all, not just those who could afford a ticket to ride – it was designed as an aesthetic composition, especially so in Britain. It was also subject to a maintenance regime for many decades that equated to much the same treatment as a thoroughbred horse would expect; until the motor age the two mobile power units were valued for appearance, not just work output. As such, the Steam Lcomotive became the model for all similar machines built subsequently, however much you or I might decry today's products; from cars to trucks to airliners, if it moves in the public domain, it has style. Or ought to…

The steam locomotive was made by machines, yes, but machines controlled by hand. The steam locomotive was assembled by hand, individually, not on an automated production line. The steam locomotive was designed to be repaired at several levels, including the hand operated hammer and chisel. Overhaul was a machine assisted manual process, and remains so.

The steam locomotive is controlled by hand. All the controls were originally designed to be hand operated. Firing was – is – a manual skill. Mucking out a locomotive is not a great deal different from mucking out a horse – all hand labour, some of it very hard and dirty indeed. The 'profession' of 'bar boy' is – I think – extinct; though I have been in a loco firebox, it was not to clean it out, hot… Coaling and watering were originally hand operated tasks, and remain predominately that today.

The vast majority of the products made today are untouched by the hand of man, designed to be superceded in a relatively short span of years (or weeks), have varying degrees of built-in obsolescence, are normally either very difficult or impossible to repair by hand, are functionally built of components whose size is such that many parts are invisible to the human eye. We have moved out of the age of the man-made machine, into the age of the machine made micro-machine.

The steam locomotive is under threat simply because it is Not Of This Age. We could lose it – the sight and sounds of it operating as it was designed to do – down to a whole host of - hopefully - well-intentioned elves. The case therefore for the Listing of the Operation of the Steam Locomotive as a Cultural Icon, within the framework of the systems that sustain it and that should also be scheduled, across the full range of its capabilities – in Britain – is on the basis that it needs statutory protection from the age it now finds itself in.

What Would Statutory Protection Achieve?

Protection would not in itself apply to the steam locomotive per se; how could it?, since it is quite a challenge today to identify a single original part on many heritage locomotives; they are the spirit of the original, not the original... The only way for a locomotive to survive is for people to want to own and run it; protection's purpose is to guarantee spaces for the locomotive to run as originally designed. Without that - the will to get up steam - there is no future beyond the museum. What protection would do – should do – is provide for say, a minimum of 30% of the national rail network, including access to all the major towns and cities, to be kept free of obstructions to the running of steam locomotives, and the dedication of areas for lineside viewing, something that has been lost as rail has sought to hide from society. The presumption should be that the steam locomotive can run over that line, not that it cannot. That implies a whole raft of enabling measures:-

Maintenance of the kinematic envelope to allow steam to run without fouling the structure gauge

Minimum specification for OHLE to keep the contact wire a certain minimum distance above the chimney top/ or offset at low structures. (contact wires are not in a constant relationship with the track, but vary in height and position to prevent grooving of the pickup, etc, so can be offset for some distance to compensate for a low structure)

Siting of 3rd rail contacts out of risk areas – injector overflows etc – where locomotives could stand for some time

Ditto for signalling fixed installations, cables etc.

Provision of run-round and turntables/wyes at suitable locations

Provision of adequate numbers of watering/fuelling points as part of the fixed assets of the national rail network

Procedures for preventing loss of lay-by, loops etc, or their provision, if line speeds exceed more than 10% of the steam locomotive's capability – say 80mph + 10% (steam typically will lose time if checked to low speed, as power/weight factor and adhesion are much lower than modern traction, and timetable paths thus need to be more generous, even when top speed is not an issue)

Provision by the national network, of equipment to enable the steam locomotive to interface with all current and future signalling, without the cost falling on the locomotive owner. Ideally as a portable 'stand alone' piece of kit that can be used as needed. It should be borne in mind that this is not heritage kit, will obviously change over time, and the cost will never – ever – go down…

Maintenance of the wheel/rail interface at the optimum for steam traction. (UK gauge is slightly different today, and stiffer track means less load transfer onto driving wheels)

Legislate on such items as insurance, to prevent the steam locomotive being excluded by changes in the way risk is managed in society.

Provide a 'cover-all' clause that requires all present and future legislation to not place restrictions, by accident or design, upon the steam locomotive or its rights of way.

Legislative provision of a heritage coal industry to preserve the skills of mining we are losing at supersonic speed, and provide the fuel the heritage industry needs. (this can, and should, be carbon negative from the off…)

Legislation on a specific set of parameters for short line heritage rail to enable this sector to move eventually to steam locomotives better suited for the task. Current light rail provision is not exactly light in its operational parameters.

Identification and scheduling of the equipment needed to carry out major overhaul/ rebuilding/ new manufacture as heritage icons in themselves, so that they are not lost by carelessness.

Recognition that the working steam locomotive needs to be seen, and provision made for that without the sort of fencing in that resulted in football getting a bad name in the 1970's.

What we will lose if we don't…

We have already lost a lot; while much of the later steam locomotive infrastructure was essentially brutalism at best – don't even think about some of the worst - much of the kit was, like the locos themselves, an aesthetic composition. Form and function went hand in hand until quite late in the steam locomotive's day, and form was not wanting in proportions and grace

of design. These losses – especially of the older, well-designed kit - were avoidable; any future losses are equally avoidable, provided we keep our eyes open and wits handy. Statutory protection would materially shift the balance of power from the forces who would eliminate steam traction by the casual application of indifference. In places there is already something worse than that; the lineside fencing in open country on the WCML is a case in point.

We don't have a complete train set that reflects the development of the steam railway and its traction, despite the efforts of many, it is something that remains out of reach. But Britain started the whole thing, after all, and Britain has made a much better fist of hanging on to the heritage in many aspects than the rest of the world. But that is no reason for either complacency or smugness; the national rail network has things that the steam locomotive needs, and the alternative to national access to the sights, sounds and total experience of that traction form, is the allocation of a single long line – say the S&C from Skipton to Carlisle – and the gradual elimination of the steam locomotive from everywhere else, barring only the short haul heritage sector. Perhaps only two of those lines are up to main line scale of operation, and it is doubtful if their current arrangements could sustain that kind of operation long term; their business plan is primarily the one that allows survival. Just.

Following such a path would be tantamount to putting all of one's eggs in a basket, then balancing it on the edge of a precipice.

Some sectors today are not obviously in any serious financial difficulties; miniature and narrow gauge steam are largely insulated from the heavy costs of heavy rail, and speeds and risks are relatively low. It is notable to point out, though, that where the narrow gauge decided to use what were essentially heavy haul locos – the ex-SAR NGG16's – the plan didn't exactly work out, and the railway as built, or rebuilt, isn't doing what the planners optimistically thought it would do when they backed it. But the standard gauge branch line based heritage rail sector, with almost total reliance upon steam traction not optimised for the task, is quite clearly the most fragile egg in the steam locomotive basket; some lines even look as if they are basket cases…

It has to be remembered also, that while full size steam enthusiasts can and do travel to the ends of the earth, almost, for their fix, this addiction invariably starts young and small, and that means local to where they live. Without that local access, the new generations of enthusiasts – of all levels of enthusiasm – will neither arrive nor mature.

A Future Imperfect

There are many people, the author included, who are very wary indeed of both the legislatures and executive sections of government, not least because diktats from those authorities wrought the havoc – or at least contributed significantly to that havoc – with the UK's railway infrastructure at the end of steam traction. That they did so in almost complete ignorance of the underlying economics – not wanting to look for the reasons for decline but selecting the convenient – and wrong - bogyman in steam traction – is unlikely to make this author an enthusiast for statutory protection without some pretty powerful motivation. Not least because these parts of governance – UK-style - have a long track record of not knowing about that which they are responsible for.

However, the actual wider infrastructure the steam locomotive needs to operate in its full glory, is for the most part outside the zone of influence – and pocket - of the locomotive owner/operator. That is the nature of a national transport network, even with an influential operators association to put their issues forward. Placing the operators of that network under a statutory obligation to provide and maintain that network in a condition fit to operate steam traction sounds onerous. But typically, as history has shown, it's simply a matter of not doing – usually smashing – historic items that are only in the way marginally, where a little tlc would probably be cheaper anyway. It is a state of mind that needs to be cultivated; heritage is not usually an expensive drag on business; quite the reverse normally.

Statutory protection – making people think before they do something that might ultimately prevent steam traction – is thus a tool for the short, medium and long term. Many enthusiasts know their home patch of railway well, and are well able to engage in consultations if key infrastructure items come under review. Authorities, such as the national network operator, when not subject to any sort of formal consultation about change, have a tendency to the 'big brother knows best and is always right' view and discount inputs from other knowledgeable sources. The author was able to stop the singling of the Swansea District Line, for example, simply by pointing out that a major structure – the Tawe River Bridge – was an integrated double track Warren Truss, and the proposed singling saved not a penny of the needed repairs to the truss bottom chords that had rusted through. Subsequent experience suggests it was not a very welcome intervention… This process of engagement happens today on the canal network, where after decades of removing a single structure and putting out of use - or potential use – many miles of route, everyone with an interest is consulted. Why not the Steam Locomotive Railway? And why not a statutory duty?

The canal, of course, is a technically simple construct. There are only a few sets of signals on the whole UK network. The kinematic envelope is easily defined; if the boat gets stuck, it's the user's fault and no-one else is going to be inconvenienced or even injured in all likelihood. Railways are a different order of technology and risk, one that changes over time, and it is that technology change – outside the parameters of steam technology – that presents major issues, of both suitability and cost, and as a consequence, access. There is not a clean air system – to keep the signal interface electronics free of dust and muck – on Tornado, for example. Should the loco have one? Or should the loco be given a cab bracket to mount a fully portable device

– not a plug-in - handed to the driver on its way onto the signalled network?

It is decisions like that – which currently are being presented as fait accompli to the loco operator – that statutory protection should – at the very least – get an appropriate level of consultation, a level playing field between the parties even, before the imposition of a new technology. The rail industry spends £billions on signalling, track and power supplies and it is extremely doubtful if a single shekel of that regular and essential expenditure considers the steam locomotive and its operation. Long term that attitude, across the whole gamut of rail technologies, is almost certainly going to put the main line steam locomotive out of business. Statutory Protection might not be a cure-all, but it does offer a potential way forward to long term steam locomotive operation in the manner of the times that the locomotive was designed in.

There appears to the author to be a sound case for the Statutory Protection of the Steam Locomotive, its rights of way and supporting infrastructure, extending to and inclusive of the engineering that builds and maintains it. That Statutory Protection is on the grounds that the steam locomotive is a key technology in the history of mankind, and Britain as originator and holder of a major portion of the surviving material has a humanity-wide duty to ensure its survival and operation. That operation should be as a 19th century technology, not as an adaptation to suit any present or future age.