AST Trust

Advancing Steam Traction

NEWSLETTER No. 21 - MARCH 2023

Lancashire & Yorkshire Railway 0-8-0 No. 1435, with corrugated firebox and large dome.

CONTENTS

AGM 2023

As in previous years, this was an online event, followed by an update on Revolution by Jamie Keyte. We had 19 attendees including one who attended intermittently from his phone at the 'Four Castles Event' at Didcot!
Alex Powell was elected to be a director and trustee in place of Mike Horne, who stood down. Gwion Clark has volunteered to help Chris with some of the many tasks that Chris does to support us.
Finances remain robust and we have received our first tranche of Gift Aid from HMRC, but it is clear that we need to more donations to finish Revolution.

COMMITTEE MEMBERS:
Following the AGM, the following members now sit on ASTT's management committee:

John Hind	Chairman & Trustee	Jamie Keyte	Trustee
Cedric Lodge	Secretary & Trustee	David Nicholson	Trustee
Chris Newman	Treasurer & Trustee	Alex Powell	Trustee
Richard Coleby	Trustee	Grant Soden	Ex-Officio

CHAIRMAN'S PIECE
John Hind

Change in life is inevitable. One of these is a change in membership of the committee. Mike Horne has been a member of the ASTT committee since our foundation in 2014, first as Secretary and then as a committee member. Mike is stepping down from the committee, but remaining a member and our instrumentation guru.

To be a sustainable group, we need to ensure that others join us on the committee. Fortunately, we do not have many complex matters to manage and do not need to meet often – currently just twice a year, by video, though with flurries of e-mails from time to time. We have had a policy of encouraging our younger members to take part in running of the group as ex-officio members of the committee and then when an opportunity arises as full members of the committee. Mike stepping down means that there is a vacancy and Alex Powell volunteered to take his place and was elected at the AGM – brief report follows later. If anyone else would like to be an ex-officio member as step to being a committee member, please let me know.

We are just past our annual renewals window. If you have not paid our membership secretary Chris Newman will have chased you for up annual subscriptions. The simplest way to pay is by a direct debit, then it's never forgotten. Our policy is to send out one reminder and one newsletter once a subscription has lapsed. If you have not renewed, this will be your last newsletter!!!!

Just prior to the AGM we sent out an appeal for funding for Revolution and that has resulted in an offer of generous lump sum donation plus a regular monthly contribution. We still need more donations to complete, so please support this interesting project.

MIKE HORNE - GRATEFUL THANKS

John Hind

I can remember first meeting Mike, but not the year, though probably in the late 2000s! It was in Alan Fozard's Garden one sunny day and we sat chatting in the sunshine not realising what it would lead to.

Alan had gathered all who were interested in supporting the development of the 5AT. When we pulled stumps on the 5AT project in 2012, several of us wanted to stay together as a group and out of that the Advanced Steam Traction Trust grew, first as an unstructured 'Action Group' without a constitution, but later with the more formal structure of the Trust, that we have today.

Mike was our first Secretary and heavily involved with Paul Hibberd in developing the constitution, which to someone with Mike's action orientated nature was a slow and long process!

Away from the 'back office' activities, Mike was involved in a number of our initiatives; presentations on the 5AT, survey of Heritage Railway Association members to see how many were interested in new build steam - the result was zero!!, Mexican Tourist railway with various meetings in Stoke-on-Trent and the City of London, project managing the manufacture of the Lempor exhaust for the KWVR's S160, developing our test instrumentation and using it on the Kirklees Light Railway 15" Gauge Badger and then the complete contrast in using on the S160s at the KWVR and the Churnet Valley Railway and helping organise our conferences in Bury in 2017 and 2018.

For the last couple of years Mike has been involved with the Avro Heritage Museum https://www.avroheritagemuseum.co.uk/ at Woodford near Manchester. Mike's background in the RAF as an instrument fitter continues to be invaluable to them as it has been to us in developing our instrumentation.

Thank you Mike!!!!

Outside the Houses of Parliament prior to meeting the HRA

Checking steam pipe temperatures on the KLR's Badger

A quick fix on Badger and by way of a contrast in size on the KWVR's S160

MEMBERSHIP MATTERS

Chris Newman

We welcome three new members who have joined since our No 20 Newsletter was circulated last November:

Howard Bolton from Derby, UK. A retired professional engineer (BSC C.Eng. M.I.Mech.E., AMICME), he was a trainee student apprenticeship with British Rail Engineer Ltd at Crewe Works, held managerial roles at Horwich, Wolverton, Derby Loco and BREL HQ, ultimately rising to Group Manufacturing Systems Manager for New Build. He then became a management consultant (mostly to the manufacturing sector) working in 27 countries on four continents. He owns, maintains and operates a Fowler ploughing engine.

Wayne Layton from NSW, Australia. Wayne trained as a fitter and turner and has been a member of a steam society in Victoria, Australia for 30 years.

Shay Sullivan from Arkansas, USA. Shay is studying Mechanical Engineering Student at the Matsu Community College. He expects to graduate in 2027. He volunteers with 557 Restoration Company, Alaska Railroad No 557 being an ex-US army class S160 2-8-0.

Full Membership

The following Associate Members have accepted an invitation to become Full Members of ASTT:

Pep Alemany	Andrew Barnes	Nigel Barnes	Peter Best
Wolf Fengler	Andrew Hartland	Xavier Jiménez	Iain McCall
Terry McMenamin	Steve Rapley	John Scott	Mani Venkataramanan

Six others were invited of whom two declined and three have not yet responded.

Student Membership

Student numbers have declined from 11 to 8 because two have completed their studies, and one has temporarily dropped out of university. We are thankful that all have opted to retain their association with ASTT by becoming Associate Members. One new student member has joined us (as noted above).

Non Renewals

Five members have not yet renewed their membership. A further reminder will be sent out to them.

Membership Numbers

Membership numbers are now:

Full Members:	37	UK members:	75
Associate Members:	62	EU:	14
Student Members:	8	USA	7
		Australasia:	9
Total Membership:	107	Asia:	2

PUBLICATIONS PAGE

Chris Newman

Book Sales

28 books have been sold in the three months since November when Newsletter No 20 was circulated. The sales numbers are listed as follows:

Publisher	Author	Title	Sales since N/ L 19	Total Sales
ASTT	L.D. Porta	Porta's Papers Vol 1	5	133
	L.D. Porta	Porta's Papers Vol 2	5	126
	L.D. Porta	Porta's Papers Vol 3	6	85
	C. Newman (Editor)	Porta's Centenary Compendium Vol 1	5	62
	Ian Gaylor	Lyn Design Calculations	2	104
	David Wardale	5AT FDCs	1	209
	Alan Fozard	5AT Feasibility Study	0	40
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	7
	L.D. Porta	Advanced Steam Design	0	5
Crimson Lake	Adrian Tester	A Defence of the MR/LMS 4F 0-6-0	2	33
	Adrian Tester	Introduction to Large Lap Valves	2	19

New Books

Two new books have appeared on the horizon:

Reminiscences of a Trainspotter by David Wardale: David has been reticent about having this book published because of concerns that the printed quality of his photographs would not meet his demanding standards. Now content with the quality of a test print that he has had made in Inverness, he's planning to send me a finished draft for a trial printing by Lightning Source.

It's a very different book to what those that we have come to associate with him, being only 68-odd pages long (many filled with photos) and focussing entirely on his teenage years when he graduated from "trainspotter" to "gricer" and finally to "enthusiast" during the last years of steam in the UK. In other words, it is a non-technical book.

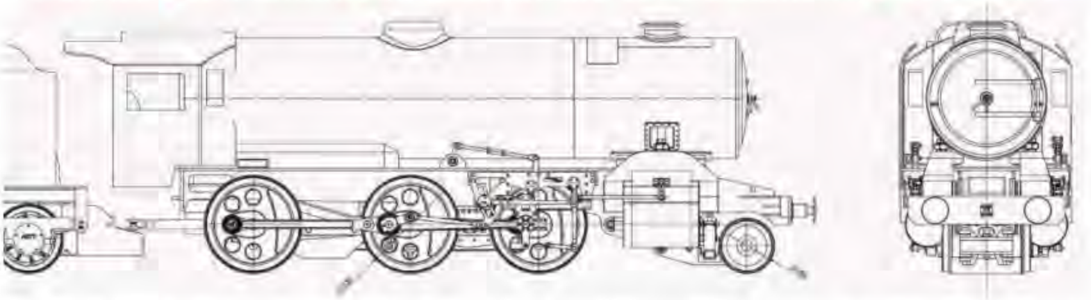
At David's request, the book will be published without his name appearing on it, and (sadly) without any teenage photos of himself. Notwithstanding I hope that we can look forward to lots of purchase orders from ASTT's membership.

Porta's Centennial Compendium Volume 2 (compiled by myself): having just today received from Martyn Bane his summary of Porta's work on his guinea-pig locomotive FCGB class C16 4-8-2 No 1802, I will now be able to begin assembling the chapters of this book in preparation for publication. Martyn has indicated that he needs to do further work on his draft text, so it may yet be some time before the book is published. But this is a big step forward, so sincere thanks to Martyn.

New books needed! Member (and non-members) are encouraged to get in touch if they would like to consider allowing us to publish a book for them. Book publishing provides ASTT with a useful source of income, and helps us to disseminate information about steam traction.

REVOLUTION APPEAL
Chris Newman

Can we get it running in time for the 2025 bicentenary of steam?



Those of you who attended the Darlington conference will have seen the progress that Jamie Keyte has been making on the *Revolution* project. Jamie brought with him the latest parts that he has manufactured, including:

- The assembled frame,
- Suspension components including axle bearings,
- Coupling and connecting rods (manufactured in aluminium),
- Pony truck including axle and wheels.

Since then, he has assembled the three driving wheelsets including axles and tyres.

The idea behind *Revolution* is to build a 10¼" gauge (fifth-scale) demonstrator to test out new "revolutionary" ideas that might be developed for use in full-sized locomotives with the aim of reducing their operating and/or maintenance costs. Ultimately, the concept could be expanded into an all-purpose low-cost low-maintenance locomotive to operate on heritage railways.

Whilst the prototype takes the form of a 2-6-0 Mogul, the design can be easily adapted to take on a variety of wheel arrangements and externally guises – for instance a 2-6-2T or a 2-8-0, styled in a way to replicate any of Big Four railways or BR Standards - or anything else.

Its revolutionary features include a simple unstressed frame structure mounted on spring beams which house the axles, bearings and wheelsets all at fixed distances from one another, and which transmits the traction forces directly from cylinder block to drawgear, rather than through the frame. Its cylinders will be fitted with steam jackets, while its valves and valve gear will be a novel combination of Walschaerts and Bulleid concepts. Its suspension system will be almost entirely formed from elastomers developing ideas first tried out in the early days of the railways and since used on modern traction.

We have a small team of members working on the design of the locomotive including:

- Jamie Keyte who has designed (and built) the frames, wheelsets, rods, suspension system and pony truck;
- Richard Coleby who has designed the cylinder block and valve gear;
- Grant Soden who is working on the design of the tender;
- Alex Powell who has undertaken preliminary design work on the boiler, and

- John Hind who is overseeing planning, coordination and budgeting.

So far, ASTT has accumulated donations amounting to £11,500 from which £9,700 has been spent. A further £2500 (for the wheels) has been committed which will be drawn from ASTT's general funds. It is estimated that a further £2000 is needed to complete the rolling chassis which will represent the completion of the first stage of the project and allow the novel suspension system to be tested on Stapleford's tracks.

We've estimated a "worst case" baseline cost of £99,300 to complete the locomotive (including a 10% contingency), of which some £45,000 will need to be raised through additional donations, the balance being covered through gift-aid and VAT rebates.

One of our members has generously committed to donating £75 per month (£900 per year) towards the project through a standing order. If each of our other members could donate £10 per month (or if half our members could donate £20 per month), the locomotive could be completed within four years - and even quicker with more generous donations.

This is a challenging, exciting and eminently achievable project that will focus attention on ASTT and its aims. It will also provide an opportunity for members to participate in taking steam traction forward beyond where its development ended. Equally importantly, it will provide us with a testbed of our own that will allow us to try out new ideas that might allow further advances to be developed. To that end, it is being designed with built-in facilities for instrumentation and data capture.

We appeal to members to support this endeavour. If you all our members could donate £10 per month for the next four years, this should suffice to bring *Revolution* into steam by the end of 2026. Better still, £20 per month could see it running by the end of 2024 – in time for it to be displayed at the celebrations for the 200th anniversary of the birth of railways that are planned to be held in Darlington in 2025.

Please help by sending a donation or by setting up a standing order with your bank to send a monthly contribution to the Advanced Steam Traction Trust's HSBC bank account: Sort Code 40-28-14, Account No 4176 0947.

If you are a UK tax payer and have not yet signed a Gift Aid declaration, we would be grateful if you could complete and sign the form below, and either email a copy to info@advanced-steam.org, or post it to Chris Newman, Flat 4, 2 Kimmerghame View, Edinburgh EH4 2GP. By so doing, we should be able to claim a rebate of 25 pence for every pound that you donate.

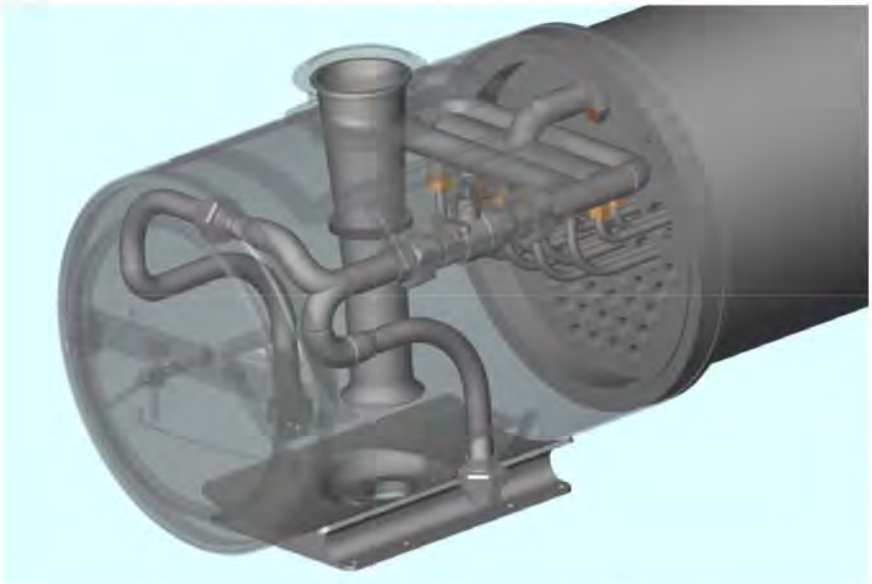
GIFT AID DECLARATION

Please treat all gifts of money that I make today and in the future as Gift Aid donations. I understand that I must pay an amount of Income Tax and/or Capital Gains Tax to the UK government each tax year that is at least equal to the amount of tax that the **Advanced Steam Traction Trust** will reclaim on my gifts for that tax year. I confirm that I have paid or will pay an amount of Income Tax and/or Capital Gains Tax for each tax year (6th April - 5th April) that is at least equal to the amount of tax to **all** the charities or Community Amateur Sports Clubs that I donate to will reclaim on my gifts for that tax year. I understand that VAT and Council Tax do not qualify.

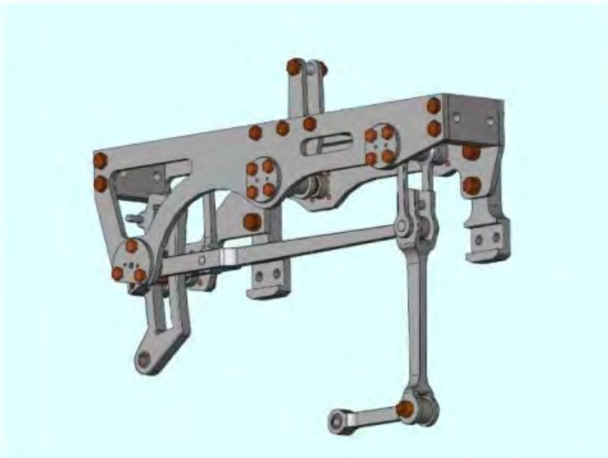
I confirm that I would like all my donations to the Advanced Steam Traction Trust to be treated as Gift Aid:

Signed: ______________________________ **Date:** ________

CAD Drawings of Revolution Components by Richard Coleby

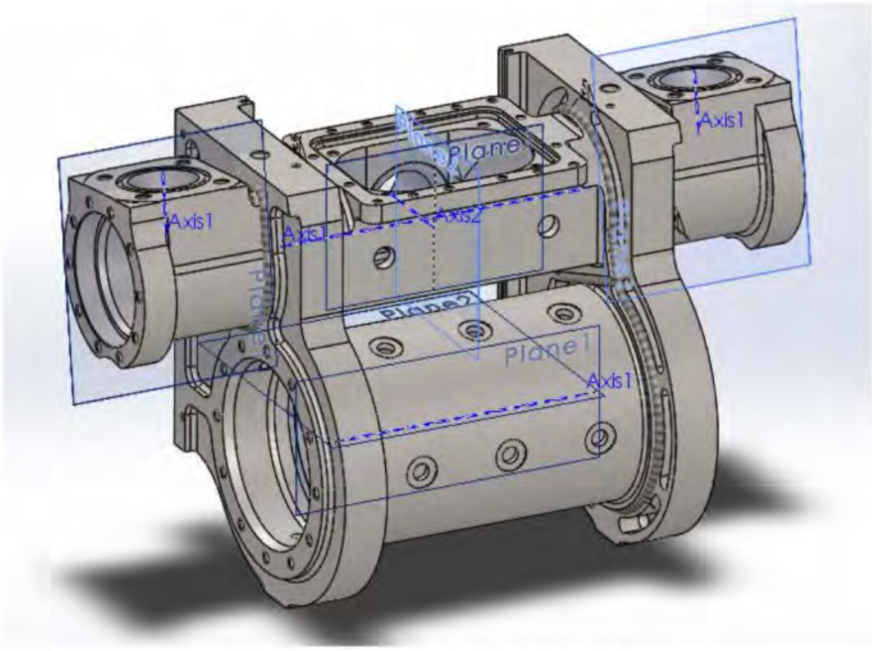


Smokebox, Superheater and Exhaust System

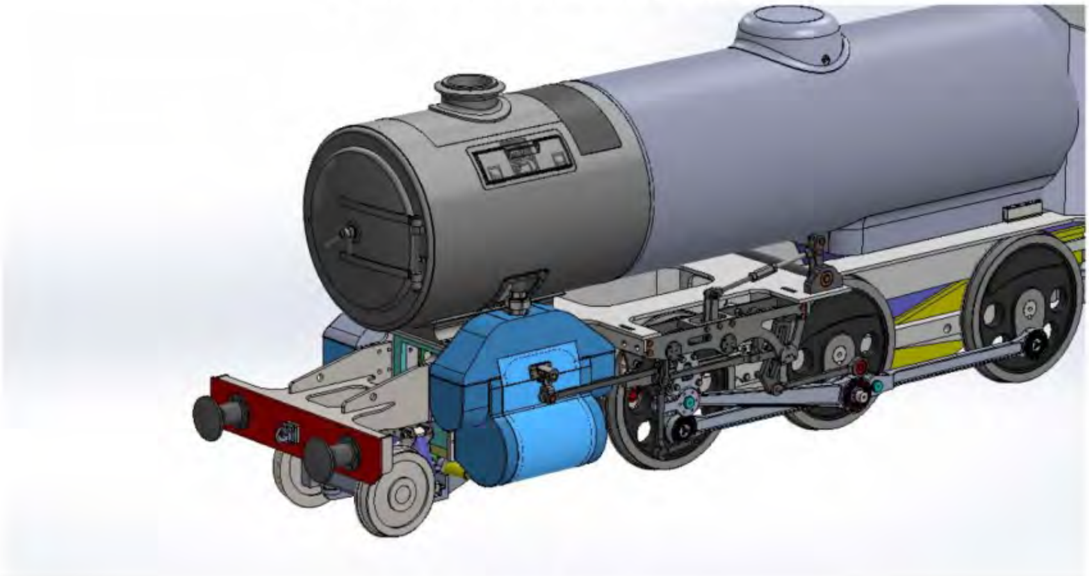


2,174.42Walschaerts valve gear assemblage

Slide bar and roller-supported crosshead



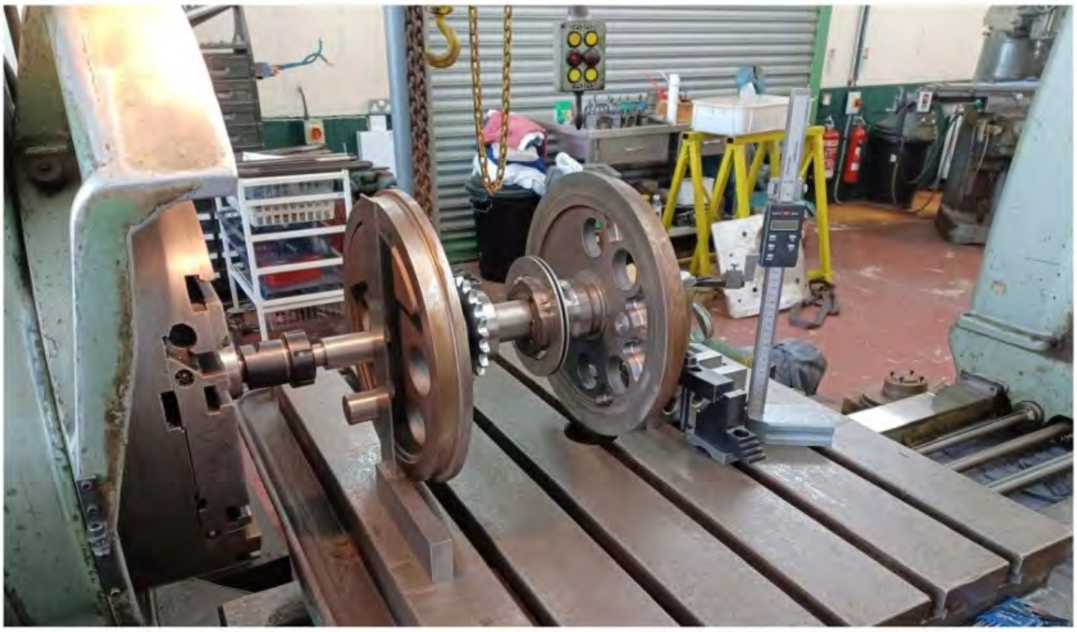
Cylinder and valve chest drawing



Front-end layout

Photos of Manufactured Components

Main frame assembly (upside down)



Assembled wheelset with axle and crank pins, and axle-mounted brake disk.

“Arcor-coated” suspension components.

Pony truck assemby.

ALTERNATIVE FUELS
John Hind

Next Bure Valley Trials

Thanks to the generosity of Andrew Barnes and his team at the Bure Valley Railway, the next set of trials of alternative fuels are being planned for the latter half of April. Because of locomotive availability, the trials will be taking place during the BVR's running season and therefore will start after the last train has run, so some late nights.

We are going to trial; Coal Products latest fuel, which compared with last year's products has 30% biomass in its composition, Arigna Fuels 100% renewable biomass-based coal substitute 'Harvest Flame' made via the torrefaction process from biomass – in this case olive stones and from Phoenix Oils rapeseed briquettes made from the waste from producing rapeseed oil.

The Arigna Fuels and the Phoenix Oils have been trialled in steam locomotives on lines in the UK and Ireland, but not under the demanding work and firing rates that are needed at the Bure Valley, where we use the same locomotive, same load, same test methodology and same driver and get to as near as practically possible to consistent trial conditions.

Our test method compares the fuels on a quantitative and qualitative basis, so meaningful comparisons can be made that are useful to both developers and users. We are now at a position where one standard gauge railway will not try a fuel, till it has been tested at the Bure Valley.

Visit to Coal Products plant at Immingham

John Scott and I had an invite from the Heritage Railway Association to be part of a delegation from the HRA on a visit to Coal Products at their Immingham plant to see how they make, test and develop their products. John was not available on the day and I could not tempt other members of our Alternative Fuels group out on a day trip to Immingham! Immingham was built by the Great Central Railway for the export of coal and now imports coal. Also on the visit was Andrew Barnes who is MD of the Bure Valley Railway and is one of our members and a great supporter of the search for an alternative to coal.

Headed by Steve Oates the HRA's CEO, delegates came from the Bure Valley, Isle of Wight, Keighley and Worth Valley, North Yorkshire Moors, Middleton, North Norfolk and Ffestiniog and Welsh Highland Railways.

We were given an overview of CPL's view of the global coal supply situation both now and in the long term. While coal is still coming into the country from Colombia and now Indonesia, it is not always in lump coal sizes. Ffos-y-fran is still available, but from stocks that are on the ground. Steelworks are now trying e-coke and the long-term future for coal remains bleak.

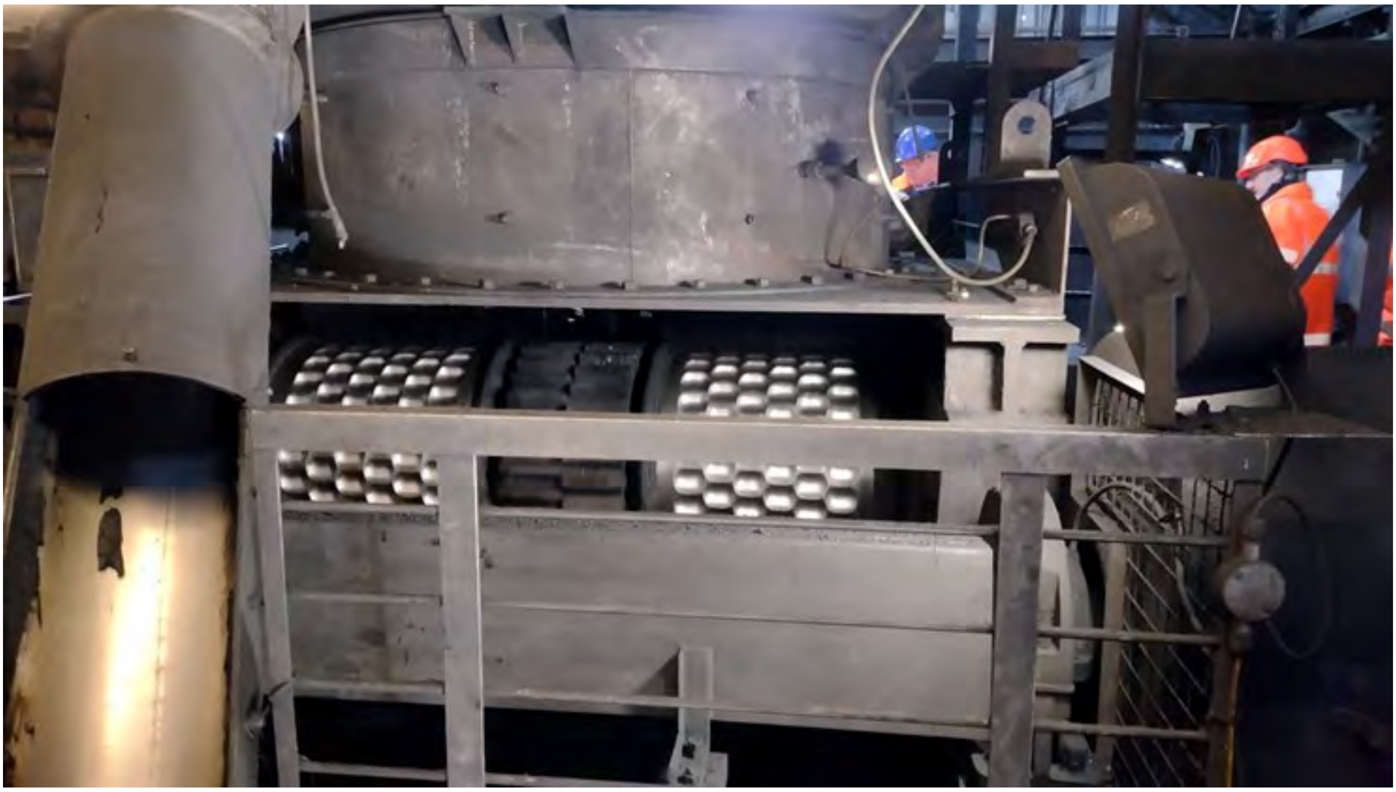
CPL previewed their latest fuel with 30% biomass (which we will be trialling in April) and showed us round the plant. Although not a massive plant, it does show the level of capital investment needed to produce a manufactured fuel. The plant has a capacity of 500,000 tons per year, but because of downtime to change over products the realistic capacity is 300,000 tons per year. The plant has been busy this year as to avoid high gas and electricity costs, there has been a switch to burning Manufactured Solid Fuels (MSF). Fifteen products are produced with different characteristics. Their raw materials include anthracite and molasses and prices are driven by worldwide commodity prices. Anthracite is sourced worldwide and the metallurgical coal from the planned coal mine in West Cumbria is too hard and abrasive to go through the plant – particularly the steel tyres for forming the ovoids. The tyres wear and need replacing on a regular basis.

Also on site is their research laboratory, with 3 staff devoted to research into briquette products and a pilot plant capable of producing 1 ton/day. A lot of work goes into perfecting the fuels to develop; calorific value, crush strength, structure in storage and structure in combustion. Some products for the domestic market need a small quantity of bituminous fuel to give a flame and a small amount of pet coke for reactivity while still maintaining sulphur emissions below 2%. Their additive to keep structure in storage needed over 100 trials to find and is now regarded as Intellectual Property.

One of CPL's two plants at Immingham

Quality Control Lab

Quality of the product is tested on a domestic fire with natural draught and an array of thermocouples is used to measure radiant heat. Calorific value, ash content and moisture are also measured.



‘Tyres’ for forming the Ovoids

The tyres are steel and limit the size of ovoid that can be produced. They are in opposed pairs and rotate in synchronisation and raw material is fed in at one end and the product is formed in the pockets and ejected onto a conveyor. Too big a pocket and the ovoids do not release from the tyre.

Screening Plant to sort coal into different sizes

DERIVATION OF THE LOCOMOTIVE ENERGY BALANCE

By Martin Johnson

1. Introduction

The definitive work on conducting locomotive energy balances was published by Lawford Fry (Ref. 9.1) and has been extensively used in analysing tests at Rugby and Altoona among other locations. The method is a relatively simple way of drawing up the complete energy balance including a derivation of gas flows and fuel lost without burning.

The Fry method assumes that the fuel lost is identical in composition to the fuel fired. However, in my experience what comes out of a smokebox is not the same as what one shovels in the firebox - the smokebox residue is a much lighter material more akin to coke. This paper will introduce some more scientific ways of confirming that common observation.

If you accept the Fry analysis and then back calculate a flue gas analysis from the derived air flow and fuel composition, you will not get the experimental results you started with. I could not solve this conundrum due to difficulty in assimilating flue gas analyses, which typically comprise 4 volumetric readings of CO_2, CO, O_2 & N_2. So there the matter rested until I became involved in helping Chris Newman edit Volume 3 of L.D. Porta's papers. It seems that Porta is also of the view that fuel lost is not the same composition as fuel fired, and fortuitously Porta provided a brief sketch that provided a lightbulb moment in terms of understanding flue gas analyses. This will be explained briefly here, but for the full explanation refer to Porta's paper along with commentary by C. Newman, John Boutwood, Prof. Wibberley and myself can be found at Ref. 9.2 .

This paper presents results of my subsequent work, on improving on Fry's method of energy balance. I have not included full step by step calculations in this paper as it would be too unwieldy, but I am happy to make spreadsheets and notes available to interested parties.

A second article will show how improved methodology in calculating the energy balance changes our view of combustion and fuel loss mechanisms within a boiler.

2. Composition of Fuels and Their Combustion

Coal, and most other commercial hydrocarbon fuels, are a mixture of multiple hydrocarbons. In the case of coal, in addition to "fixed" carbon there will be a range of alkenes, phenols and many other compounds.

As an example the alkanes (which used to be known as paraffins) show the complexity of the problem. The alkenes have a chain of carbon atoms with each carbon atom having two hydrogen atoms attached to the "sides" and another two hydrogen atoms at each end of the chain as shown in Figure 1, showing the structure of the first 4 alkanes, to give the general idea.

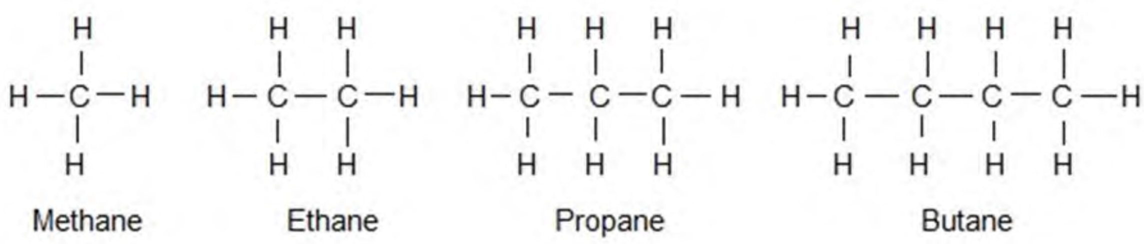


Figure 1: The first four alkanes in the sequence.

Well known alkanes with more carbon atoms include octane (C_8H_{18}), paraffins including jet fuel and diesel oil (mixtures between 6 and 20 carbon atoms), going up to bitumen which includes the longest alkane with 54 carbon atoms. The important point for combustion is that as the number of carbon atoms increases, the boiling point increases making the material less volatile. **So all those compounds will ignite and burn at different rates, the gases almost explosively, bitumen quite slowly.** In addition, the longer molecules may well break down into shorter molecular chains in the heat of a firebox.

The net result is that although we might know the average analysis of the fuel as fired, we do not know the analysis of the fuel burning at any given instant. For example, after a firing round the lighter (hydrogen rich) compounds burn as they are quickly released leaving a carbon rich bed of semi coke to burn more slowly. However, the semi coke fire bed gets the incoming air (and hence oxygen) first, potentially leaving the lighter compounds oxygen starved. Which is the scientific basis for "little and often" firing to give the lighter compounds a chance to get at the oxygen, as does allowing a bit of top air after a firing round.

3. The Combustion Triangle

Combustion gas analysis gives us a picture of what is left after combustion at any given time, but understanding quite what is going on from CO_2, CO, O_2 and N_2 figures is another matter! The combustion triangle gives a method of assessing combustion test results and was found as a small sketch in Porta's papers. In effect it is simply a graph of CO_2 against O_2 each flue gas analysis presenting as a point on that graph. Figure 2 shows a sample combustion triangle for four hydrocarbon fuels comprising 100, 90, 80 & 70 % by weight carbon, the balance being hydrogen. For each fuel a range of air ratios (actual / stoichiometric) have been assumed, from stoichiometric combustion (the points on the Y axis), with air increasing in 5% increments so 1.0, 1.05, 1.1 etc.

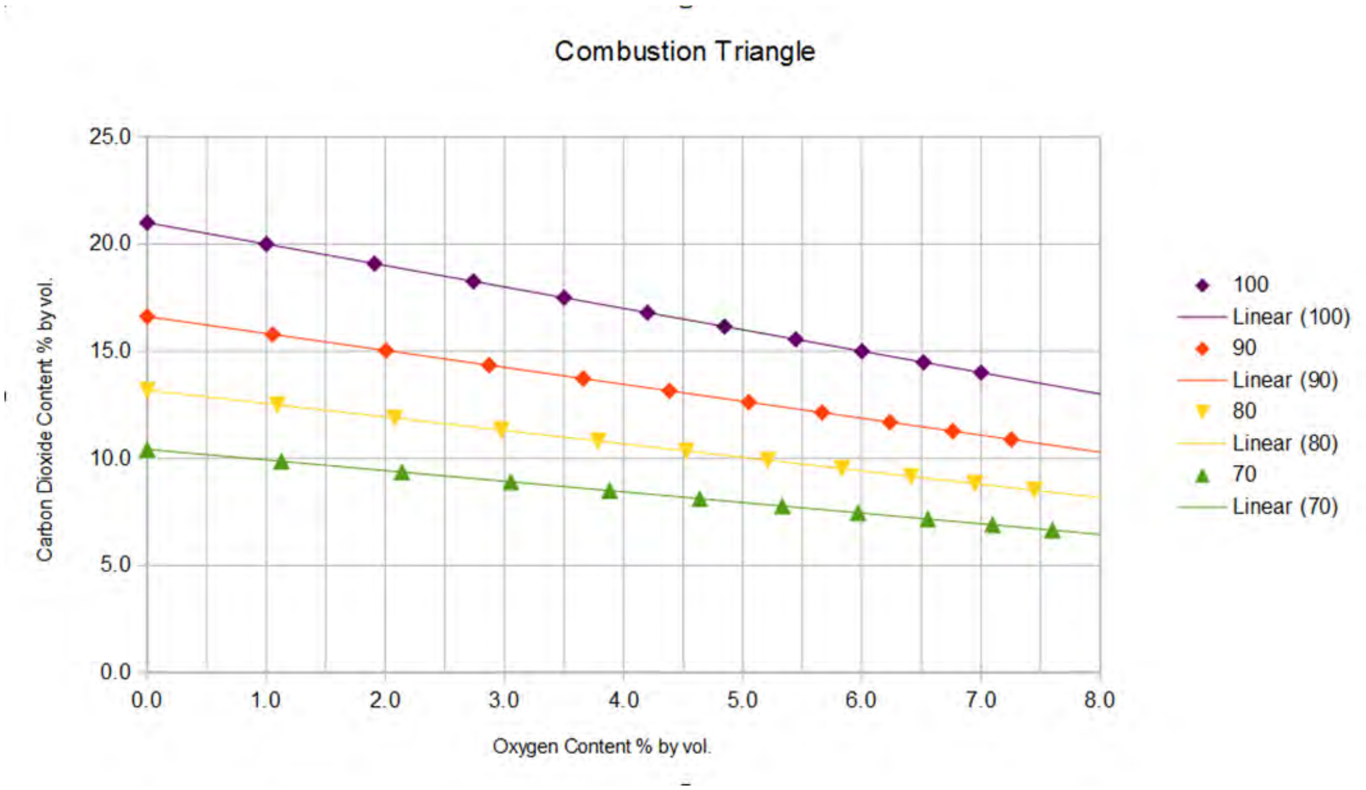


Figure 2: Combustion triangle drawn for four hydrocarbon fuels.

There are several things to note about the presentation:

1. Each fuel has a distinctive line. The more hydrogen in the fuel, the lower the line on the page.

2. The oxygen content of the combustion gas is strongly related to air ratio, and only marginally related to fuel composition, as witnessed by the near vertical families of points at constant air ratio. Stoichiometric combustion will always appear on the Y axis - because by definition there is zero surplus oxygen in the combustion products.

3. All lines radiate from a point at zero carbon dioxide and 21% oxygen, representing our atmosphere.

4. Figure 2 is derived from simple molar chemistry, the assumed fuel composition and an assumed air ratio.

3.1 What about Carbon Monoxide, Nitrogen, and Moisture?

In short, they don't matter:

- If CO is present in the flue gas, we correct the CO_2 reading using molar chemistry to give an equivalent CO_2 assuming the CO had all been burnt.
- Nitrogen is simply the balance of what is not oxygen locked up in CO_2 or O_2 (ignoring tiny amounts of atmospheric CO_2 or Argon).
- Moisture was not measured by the Orsat process, and as far as I know is not measured by modern gas analysers either. So stoichiometric combustion of pure hydrogen for example (giving only water vapour) would appear on Figure 2 as a point at the origin - no CO_2 (obviously) and no surplus O_2.

Ref. 9.2 gives the full derivation and method of calculation of the combustion triangle including allowance for oxygen sequestered in the fuel, so for brevity I shall not repeat it all here.

I have also extended the method since contributing to Ref. 9.2 to allow for sulphur in the fuel and the consequent effect on oxygen demand and the fact that sulphur dioxide registers as carbon dioxide in the Orsat apparatus. However, due to the relatively low sulphur content in the fuels used in the Rugby tests, the net effect is insignificant.

4. Experimental Results

I plotted the Rugby test data, courtesy of Dr. Pawson's transcription, on a combustion triangle and the results are shown in Figure 3. The experimental figures for CO_2 and O_2 have been corrected on the assumption that the CO would have burnt to CO_2 according to molar principles. In calculating the fuel line I have assumed a fuel which is simply the numerical average of all the fuel analyses recorded in Dr. Pawson's transcription; this closely corresponds to Blidworth coal as that was the majority fuel used in the tests. There are some departures from Fry's method to improve any corrections for heat in the ash and coal. Fry also uses a simple approximation to the total mass of gas per kg of fuel; I have used a longer but more precise approach. These changes are discussed in more detail in later sections.

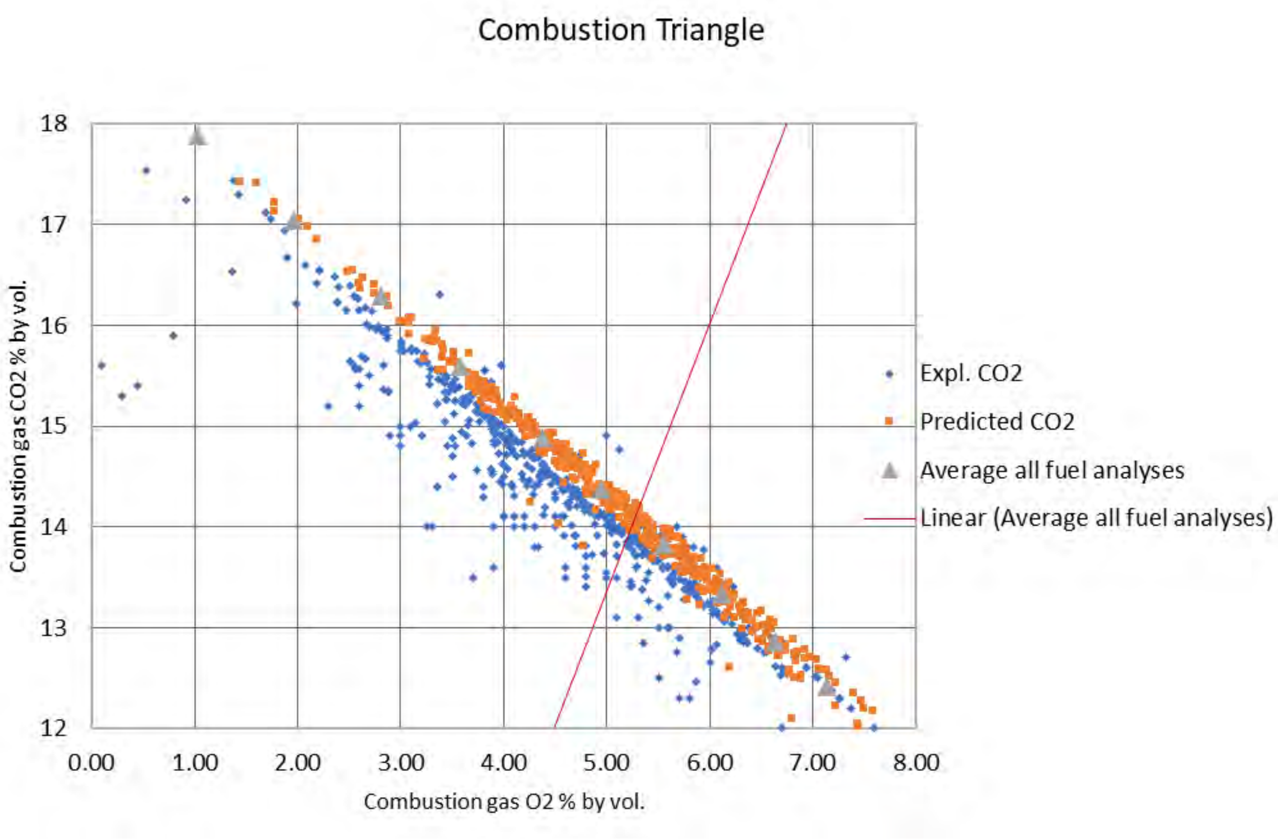


Figure 3: Rugby test results and combustion triangle for average coal.

The experimental data (orange points) in the above figure shows a fair scatter of points below a linear group of data following the gradient of the combustion line for average fuel. The lower scattered points will correspond to periods when hydrogen rich fuel was being burnt, such as after a firing round as discussed in Section 2. The linear grouping of points is likely to be periods when the fuel approximated the mean analysis and upper scattered points represent a carbon rich fuel.

Fry's method assumes that the fuel lost is identical in composition to the fuel fired. However, using that assumption and back calculating the CO_2 and O_2 values, gives the blue points which cluster around the average fuel line (because that is the assumed fuel they are calculated from), but have a noticeable shift from the original data. Figure 3 demonstrates that on average the fuel burnt is less carbon rich than the fuel fired, so Fry's assumption that fuel lost is of the same composition as the fuel fired is not correct.

Another interesting conclusion from the combustion triangle presentation of Figure 3 is that an air ratio (actual air / stoichiometric required air) of 1.2 equates to an O_2 value around 3.5 to 4% by volume and there are a significant number of points falling well below that ratio. Complete combustion is usually considered to require an air ratio of around 1.2.

5. Improving on the Fry Method

The Fry method is based on an energy balance across the whole boiler as summarised in Table 1. The balance was conducted considering one unit mass of fuel. Fry made great claims for the simplicity of his method, but in obtaining simplicity, a number of approximations were made which I will deal with as we look in detail at the energy terms in Table 1. To be fair to Fry, the full evaluation of test results would have been a considerable task using a slide rule but with today's computing power we can do better. Fry's detailed method can be found at Ref. 9.1 .

ENERGY IN	ENERGY OUT
Coal calorific value	
Air heat (temperature x specific heat)	
Water heat (temperature x specific heat)	
Coal heat (temperature x specific heat)	
	Steam produced (quantity x enthalpy)
	Combustion gas (temperature x specific heat)
	Energy lost due to partial combustion of fuel to CO
	Energy lost as water vapour in the exhaust
	Energy lost by radiation and convection to surroundings, plus steam leakage.
	Ash (temperature x specific heat)
	Lost fuel (temperature x specific heat)
Fuel loss is derived from the total of (ENERGY IN - ENERGY OUT)/CALORIFIC VALUE	

Table 1: The locomotive energy balance. Terms shown in red were not included by Fry.

Considering the above table in more detail:

- The flow of fuel in can be directly measured by weighing. The fuel chemical energy or calorific value can be measured directly in a bomb calorimeter, for example.
- The air into the ashpan has a heat value, being the product of temperature and specific heat but the air flow is not directly measured, being inferred from flue gas analysis and fuel flow.
- Water in can be readily measured by weighing or volumetric means, and has an enthalpy due to its temperature and specific heat.
- The fuel has a heat value due to its temperature and specific heat value; this is a tiny correction which Fry did

not include but I have made it just to be sure.

- Steam flow out is inferred from the water taken in. Steam enthalpy is determined from pressure and temperature readings.
- Combustion gas out has heat value due to it's temperature (measured) and specific heat. However, the flow is not measured but inferred from flue gas analysis.
- Not all the carbon is converted to CO_2, some being emitted as CO. The CO content of the exhaust gas is measured and the appropriate heat loss calculated from the calorific value of conversion of CO to CO_2
- Ashes out. The hot ashes have specific heat and hence a heat value although some of this will be transferred to the incoming air in the ashpan. I have assumed the ashes are ejected at the same temperature as smokebox gases. This is a tiny correction and was not done by Fry.
- Water vapour out is the sum of moisture in fuel and air, plus the product of hydrogen combustion. Once again, the flow is not directly measured but inferred from fuel hydrogen content and air flow. The Rugby tests did not include reliable measurements of atmospheric moisture, so that is not accounted for.
- Radiation, convection and leakage loss (sometimes known as standing loss) could be measured by tracking the loss in pressure of an engine with no fire but this is not usually done. Fry assumed this loss to be equal to 5% of the evaporative heat.
- Lost fuel heat is the loss due to the unburnt fuel temperature and its specific heat. Fry did not account for this term. The lost fuel is assumed to be ejected at flue gas temperature (measured). The quantity is inferred from all the other elements in the energy balance.

Finally, the lost fuel quantity is derived from the difference between the total energy in less the energy out of the system.

The following sections discuss shortcomings with Fry's method and how I have tried to address them.

5.1. Determination of Gas Flows

Air flow can be inferred from flue gas analysis results of % by volume of carbon dioxide, carbon monoxide, oxygen and nitrogen. The usual method is to base the calculation on a carbon balance - that is accounting for all carbon atoms entering the system as fuel and leaving the system as gas. Ref. 9.3 gives a good description of the method. The carbon balance method is used because carbon is the dominant reaction in the combustion, but Figure 3 demonstrates that carbon is being lost from the system in unknown quantities.

In addition, Fry's equations introduce an empirical factor, simplifying the dry gas flow to:

$C' \times 2.52/(CO + CO_2)$

Justifying this assumption by "*In all of the tests the value of* $1.33CO_2+1/3\cdot O_2+223$ *is always very nearly 252.*" (Chapter III, p.17). However, there are effectively two assumption in this equation: an **air ratio of 1.25**, and an elemental oxygen content (O) of about 3.8%.

I have used formulae as given in Ref. 9.3 which make no such arbitrary assumptions. They are:

$$\frac{Mass of Air}{Mass of Fuel} = 3.036 \times C' \times (\frac{N}{CO_2 + CO})$$

Similarly, the dry gas flow can be found from:

$$Mass of Dry Gas = C' \times [\frac{11 \times CO_2 + 8 \times O + 7(CO + N)}{3(CO_2 + CO)}]$$

Where:

C' = Percentage by weight of carbon in fuel

CO_2, CO, O & N = Percentages by volume of gas in combustion gas

In my revised method, I assume that all fuel loss is carbon. The above equations can then be used by adjusting C' to be net of carbon loss.

5.2. Determination of Standing Loss

Class	Grate loading lbs/sqft/hr	Heat loss kw	Average heat loss kw
BR5 73031	51.6	113.3	
	49.5	187.1	137.1
	49.5	111.0	
MN 35022 As Built	32.2	65.9	
	38.0	-525.5	-152.7
	43.6	1.386	
BR9 92013	38.1	399.8	
	39.6	494.1	298.4
	42.3	1.408	
BR 9 92050	39.6	383.8	
	40.9	518.4	301.2
	45.3	1.356	
Crosti BR9 92023	36.7	195.4	
	38.2	161.3	119.4
	38.6	1.324	
Mech stoked BR9 92166	46.9	909.4	
	60.7	1583.8	831.6
	56.3	1.490	
DC (A) and Giesl Fitted BR	42.2	473.4	
	42.9	542.5	339.1
	43.0	1.495	
Duchess (SK)	36.5	82.4	
South Kirby coal	38.3	511.4	198.4
	39.5	1.273	
Duchess (BL)	38.5	382.9	
Blidworth coal	38.5	382.9	255.7
	43.4	1.340	
Jubilee 45722	43.1	291.8	
	50.3	331.7	208.3
	53.9	1.436	
Scot 46165	60.9	463.7	
	61.3	557.9	341.0
	67.8	1.286	
King 6001	51.5	409.0	
	48.6	170.5	
	51.3	649.4	409.6
Duke 71000	23.9	166.4	
	25.9	-198.4	-10.2
	26.1	1.331	
Duke 71000	34.0	-17.7	
Road tests	37.9	137.3	40.3
	45.1	1.426	
Grand average	kW		237

Table 2: Summary of Standing Loss Determination.

Fry assumed this loss to be 5% of the heat of evaporation, however that would imply that such losses vary with the amount of water evaporated. In fact, the radiation and convection losses depend only on the boiler temperature (near constant at working pressures) and leakage (near constant at working steam pressure) This means the importance of this term decreases as steaming rate increases. It is not feasible to calculate the standing heat loss from a boiler, as the shape and surface area including all pipes, valves etc. is difficult to determine, as is the degree of conduction to other parts of the engine such as smokebox and frames. Similarly, substantial assumptions would be needed to calculate a better estimate of leakage loss.

I analysed the three lowest firing rate tests for each locomotive class and assumed zero carbon loss to estimate the standing loss. These covered a range of firing rates between 24 to 68 lbs/sqft/hr with an average of 44 lbs/sqft/hr; probably rather high to assume zero carbon loss, but I did anyway and derived Table 2 showing standing losses.

There is a wide scatter of results with an average of around 235 kW, but that probably includes some carbon loss at the higher firing rates.

No patterns are evident with regard to engine size or coal type and only a very weak association with firing rate. Losses for the mechanically stoked Class 9 are remarkably high, but widely scattered.

Table 1 suggests that an engine in the British classification 5 to 9 loses approximately 200 kW to heat loss and steam leakage. However, when this figure was tried in an energy balance for all the Rugby tests, an unacceptably high proportion of tests were indicating negative fuel loss. This is probably because the firing rate used for deriving the loss are certainly not zero ranging up to 70 lb/sqft/hour. Accordingly, I have settled on a loss figure of 100 kW as best representing losses in Class 5 to 9 locomotives. This is a very approximate estimate but is the best that I can derive from the available data.

5.3. Determination of Sensible Heat for Coal, Lost Carbon and Ash

I have allowed for the heat value due to the temperature of coal, lost carbon and ash which Fry does not. They are very small quantities (approximately 0.05% of the coal calorific value) and could be ignored, but I wanted to establish that they were small and having done so, I have left them in the calculation.

5.4. Determination of Fuel Loss

The fuel loss is defined here as a carbon loss (evident as visible smoke, soot, smokebox char and carbon sequestered in ash), being the ratio of lost carbon over total carbon fed to the system. Due to the much higher reactivity of hydrogen to oxygen, I have assumed that only unburned carbon is lost from a locomotive.

This loss should not be confused with the partial combustion of carbon to form carbon monoxide which gives a loss of heat release which is treated separately in the analysis.

Deriving carbon (or fuel) loss from an energy balance is subject to error as the loss is a relatively small quantity deduced from the difference of two large quantities (Energy in - Energy out). There is a further problem, in that the heat lost in flue gas requires a knowledge of the quantity of flue gas which, as Section 5.1 shows, depends on the carbon content of the fuel which is affected by the carbon loss.

The problem can be solved by an iterative calculation refining an assumed value for carbon loss. I have used an initial estimate of fuel loss as zero, which then renders the first iteration similar to the Fry calculation. Subsequent iterations use carbon masses net of unburnt carbon (as calculated in the prior iteration) for calculating gas flows. The calorific value of the fuel burned is assumed to be the measured value less the calorific value of the lost carbon.

After four iterations, the residual errors are well below the experimental scatter in the data, although there is no reason that more iterations could not be used if desired.

The results of this modified method are presented in combustion triangle form in Figure 4.

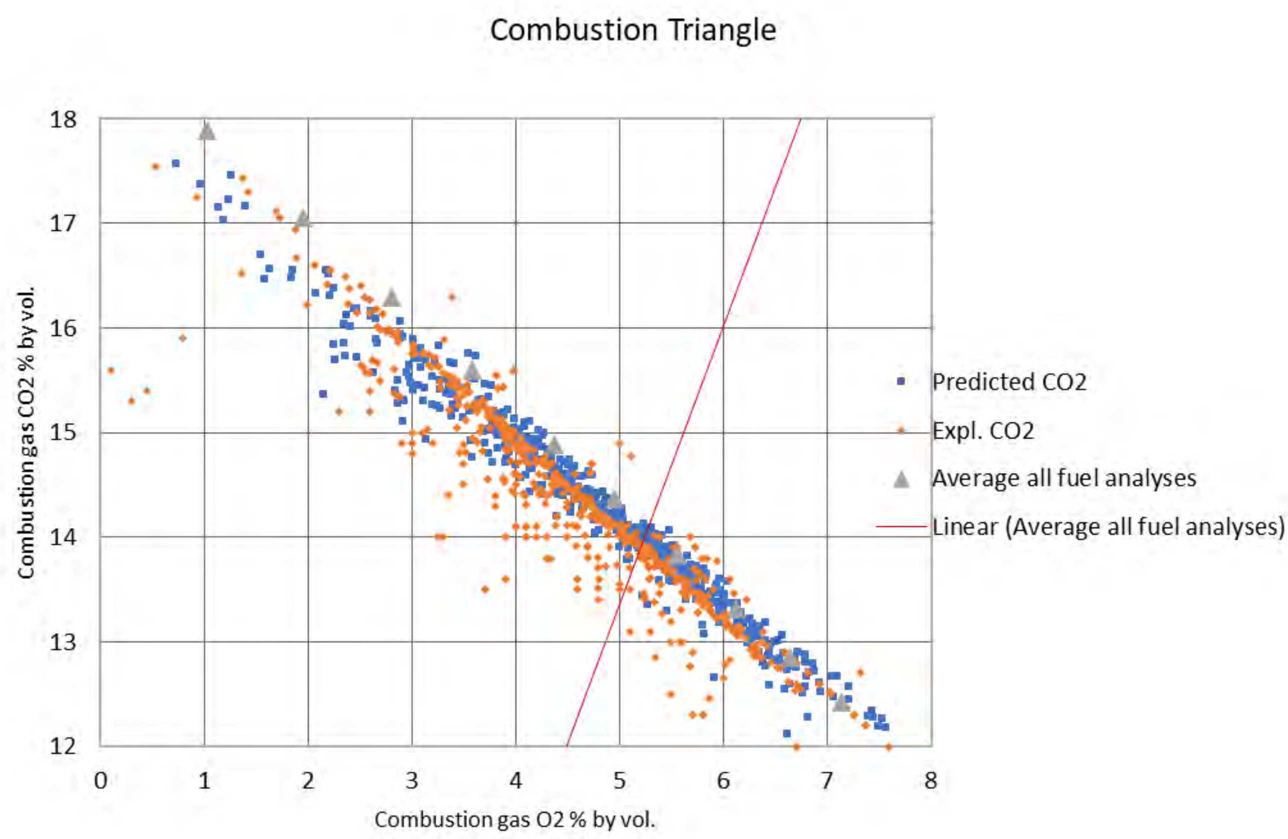


Figure 4: Results shown using the author's improved energy balance method.

Figure 4 shows that a much better match between experimental and back calculated points has been obtained, suggesting that the assumption of fuel loss being pure carbon is a reasonably accurate one. Compared to Figure 3 the separation of predicted and experimental readings is much reduced.

Figure 4 also shows that fuel burnt at high combustion gas oxygen content (associated with low grate loading) corresponds quite closely to the combustion line predicted from the fuel analysis. Conversely, at low combustion gas oxygen contents (higher grate loadings) the experimental and predicted points suggest the fuel being burnt is lower in carbon due to the carbon loss.

Figure 4 demonstrates that even though the fuel composition in the instant may not be known, it is possible to see changes to average fuel burnt as carbon is lost at low O_2 readings. However, that requires multiple readings of combustion gas composition to be sure that "average" fuel is being fired. That process took days to weeks at Rugby as each Orsat reading set takes about 5 minutes for a skilled operator. However, modern gas analyser instruments can produce multiple gas analyses each minute, hence it may be possible to develop better methods of assessing modern substitutes for coal using a combustion gas meter, data acquisition and a combustion triangle analysis. This would give a more reliable view of fuel lost which appears to be a particular problem with substitute fuels.

6. Can Gas and Air Flows be derived without relying on Carbon readings?

Gas flow can be derived from an oxygen balance (See Ref. 9.4 , for example). This would avoid the problem that air and gas flows are calculated from a knowledge of the quantity of carbon burnt, which is not actually known. The method is as follows:

$$Oxygen\ burnt\ with\ carbon = 32 \times (CO_2 + O_2 + CO/2)kg$$

$$Total\ oxygen\ burnt = N_2 \times 28 \times (\frac{23.3}{76.7})kg$$

$$Oxygen\ burnt\ with\ hydrogen = Total\ Oxygen\ burnt - Oxygen\ burnt\ with\ carbon\ kg$$

$$Fuel\ burnt = \frac{(Total\ oxygen\ burnt - oxygen\ burnt\ with\ carbon)}{(8 \times H_{fuel})}kg$$

$$Total\ air\ burnt = N_2 \times 28 \times (100/76.7)$$

From which the air to fuel ratio is known.

Where:

N_2, CO_2, O_2, CO	=	Percent by volume of gases in combustion gas
H_{fuel}	=	Percent by weight of hydrogen in fuel

Another method is to use the O_2 reading in the flue gas analysis which gives a good indicator of air ratio, as discussed in Section 3 and Figure 2. By plotting the O_2 content against air ratio calculated for the average fuel, it is then possible to derive the air content from a measured O_2 value. This would assume that the air ratio lines in Figure 3 are vertical but they are not absolutely vertical and slide to the right for fuels with decreasing carbon content. However, I have tried it as an approximate method.

The results of the above two calculations have been compared with the air / fuel ratio derived from the iterative carbon loss model of fuel loss and are shown in Figure 5. The abscissa represents the air / fuel ratio derived from the carbon balance method the subject of this article.

The blue data points represent the oxygen balance method and show a lot of scatter. The reasons for the amount of scatter are not completely clear but third equation would be very sensitive to errors in CO_2, O_2, CO readings, which in turn has a significant effect on the calculation of fuel burnt. The above equations also do not allow for nitrogen content in the fuel, which might appear as gaseous nitrogen in the exhaust or solid nitrous compounds in the ash; so there would be an indeterminate effect.

The orange points showing the approximate method relying on the combustion triangle show remarkably good agreement.

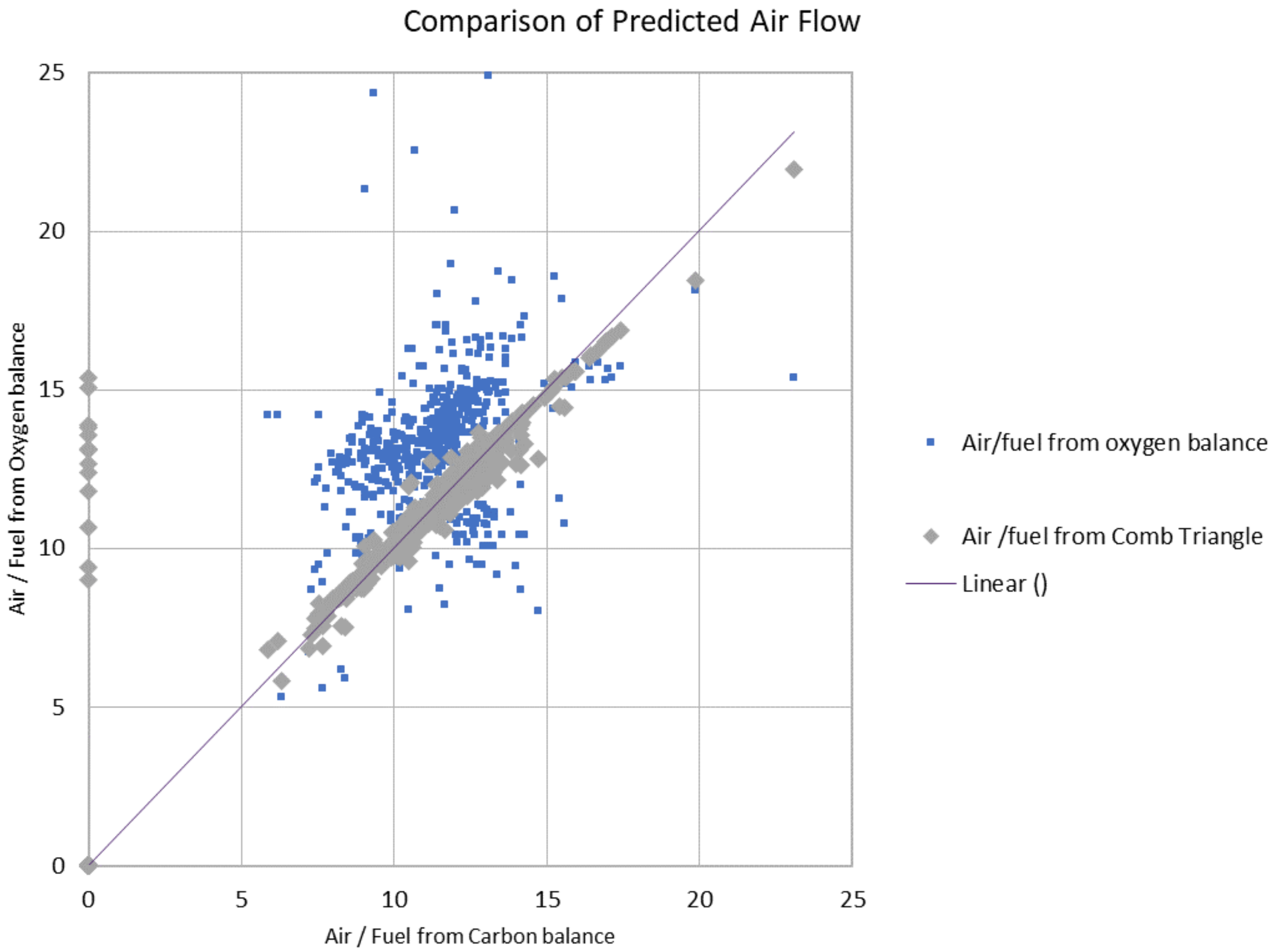


Figure 5: Comparison of air to fuel ratio derived from carbon balance, oxygen balance and combustion triangle method.

7. Is there other evidence to suggest the new method is correct?

It is possible to look at a complete mass balance for a combustion process. For this exercise I used a modification of the method set out in Example 15.5 of Ref. 9.4 . The method is very briefly summarised below based on unit mass of fuel. Chemical symbols relate to either the fuel analysis by mass or dry gas analysis by volume:

1	The mols of carbon burnt per kg of fuel is equal to the mass of carbon in the coal less that not burnt. $(C - C_{lost})/12$ [Mols] = A The mols of carbon per mol of dry gas is: $C_{Mols\ dry\ gas} = (CO_2 + CO)/100$ [Mols] = B Mols of dry gas per Kg of fuel = A / B
2	Hence, the mass of each gas component can be derived from the Orsat readings using the mols of dry gas per kg of fuel multiplied by the dry gas exhaust analysis.
3	Mass of dry air is found from a Nitrogen balance. Nitrogen in the fuel is N_2 / 28 mols (usually very small) Nitrogen in air is mols of N_2 derived from Step 2. Mols of air is (sum of Nitrogen in fuel + combustion gas)/0.79 Mass of air is mols of air x 29
4	Water vapour in the exhaust gas is: Moisture in fuel from fuel analysis + H_2O from combustion of H_2 = H_2 content of fuel x 9 Moisture entrained from the air is ignored as the test results did not contain suitable wet and try bulb thermometer readings.
5	The reactants are calculated as: 1kg of fuel (assumed a priori) Dry air from Step 3 Moisture from the air (ignored)
6	The mass of products is calculated as: Dry gases from Step 1 Moisture from Step 4 Unburnt carbon derived from energy balance calculation
7	The masses derived in steps 5 & 6 should balance. Any error is an indication of the accuracy of the analysis of the fuel burnt and the analysis of products of combustion.

The results of this calculation are summarised in the following table:

Iteration	1 "Fry method"	2	3	4 "Proposed method"
Max. imbalance kg/kg fuel	0.64	0.5	0.51	0.51
Min. imbalance kg/kg fuel	-0.11	-0.15	-0.22	-0.25
Average imbalance kg/kg fuel	0.15	0.09	0.08	0.07

The range between best and worst imbalance for a given iteration remains roughly constant between 0.65 to 0.76, probably reflecting the scatter in the original experimental readings. However, use of the iterative calculation method reduces the average mass imbalance from about 15% to 7%, suggesting the systematic errors have been halved by using the carbon loss assumption for locomotive fuel loss.

I cannot presently offer an explanation as to why the calculated fuel and air consistently exceed the calculated products of combustion. Further iterations might help, but are unlikely to reduce the average error to zero.

I have tried other more complex analyses of the residual mass imbalance, but the above is sufficient to show the improvement possible.

A development that has produced interesting results checks for test points that fall outside the median error +/- 2 standard deviations. That is excluding the 5% of tests showing the worst mass imbalance. This shows that mass imbalance on 14 of the 40 tests on Duke 71000 and 5 out of 50 tests on King 6001 exceeds 2 standard deviations. Other test series show one or two tests exceeding the 2 Std. deviation limits, which might reasonably be due experimental scatter. So, this suggests that the tests on Duke 71000 and King 6001 should be treated with caution. The data shown in Figure 4 excludes points that fail outside +/- 2 standard deviations.

8. Interim Conclusions

The Fry method of determining the energy balance for a locomotive is fundamentally flawed due to the assumption that fuel lost is the same composition as the fuel fired.

An improved method of energy balance determination has been developed which assumes pure carbon is the lost fuel. The method is still an energy balance, but requires an iterative solution.

The improved method has been applied to the Rugby test data set as digitised by D. Pawson.

The improved method has been cross checked by reference to a presentation of combustion gas analysis results used by L.D. Porta and re-invented by the author. The method, known as the combustion triangle, relies on plotting corrected CO_2 readings against O_2 readings.

Fuel loss is readily identified using the combustion triangle which suggests it may be possible to develop new techniques of analysis by using modern combustion gas instruments in conjunction with the combustion triangle analysis method. This may have application in assessing new fuels for replacing coal.

The improved method shows better correlation between experimental and back calculated flue gas composition than the Fry method. The improved method also reduces the mean error in the mass balance across the combustion process by 50% compared to the Fry method, confirming that the carbon loss model is a better one.

A method of using the residual mass imbalance as a means of checking the quality of individual test points for locomotive performance has been developed.

The development of an improved analysis has implications for our understanding of air / fuel ratio and combustion gas quantity, which may impact our understanding of combustion in a locomotive and the performance of the front end. This will be explored in detail in part 2 of this article.

9. References

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9.2. Porta, L.D. "Locomotive Boiler Combustion Calculations - The Heat Balance, A criticism of the Lawford Fry Method" in Selection of Papers by L.D. Porta, Vol. 3 Ed. by C. Newman, Publ. ASTT 2021

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CYLINDRICAL FIREBOXES – SORTING OUT THE CIRCULATION PROBLEM

Robin Pennie

Before anyone jumps on me to say that cylindrical fireboxes have been tried on several occasions and have been shown to be no good, I would like to say that I am well aware of the fact they have problems, in particular the time taken to raise steam. Those used on the L&YR could take 36 hours to do so, and that may well be a fact that should be in the Guinness Book of Records!

The Wikipedia entry for 'Launch-type boiler' gives a useful overview of boilers with cylindrical fireboxes, and their problems, but does not include any information to show if, or how, those problems have been dealt with, or at least ameliorated, in the past. Launch-type boiler - Wikipedia

OK, SO WHY THE INTEREST IN CYLINDRICAL FIREBOXES NOW?

There are a few reasons for this.

1) At the moment we have no idea how long steam locomotives will be allowed to work, but assuming that this is for another 40 to 50 years (with suitable fuel), then at least a few new locomotives will be required to maintain services, as many are already getting on for 100 years old, and any design with a cylindrical firebox is going to be a lot cheaper to build than one with a stayed, copper, firebox, and possibly cheaper than one of water-tube fireboxes promoted by Alan Haigh in his two books on locomotive boilers, besides being an integral part of a boiler with cylindrical firebox and stayed combustion that he also advocates.

2) I realise that an operating problem with cylindrical fireboxes is their very limited ashpan capacity. However, few of the heritage railways in this country are more than ten miles long, and they are all subject to a speed limit of 25mph, so the power output required for services on such lines is bound to be less than that needed for main line running. As a result, with less fuel being burned per mile because of low speeds, and on journeys of less than twenty miles, the limited ashpan capacity is not likely to be much of a problem.

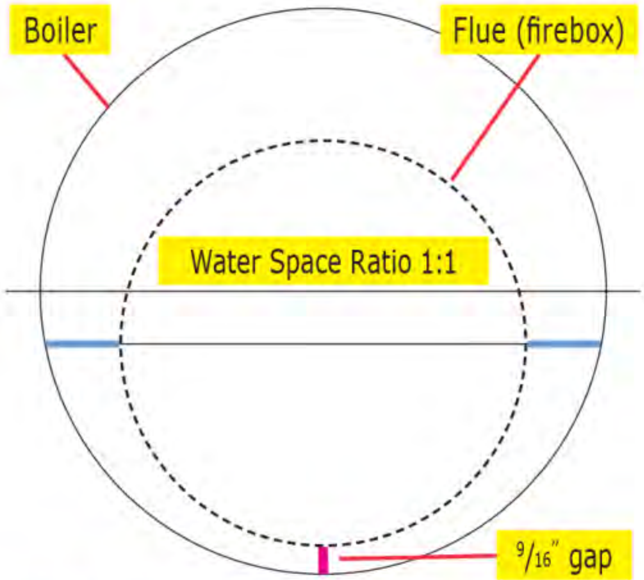
3) Many ways of improving the circulation of boilers with cylindrical fireboxes have been proposed over the years, two of which could be very useful, and have not to my knowledge been used on any locomotive design, even though they were used on stationary and marine boilers. I am not claiming that a boiler with a cylindrical firebox could be made to steam as well as a one with a conventional firebox, but I consider the deficiencies of the cylindrical one could be reduced enough for it to be worth considering in a limited number of applications (as indicated in point 2), but probably not enough for main line running.

Poor Circulation

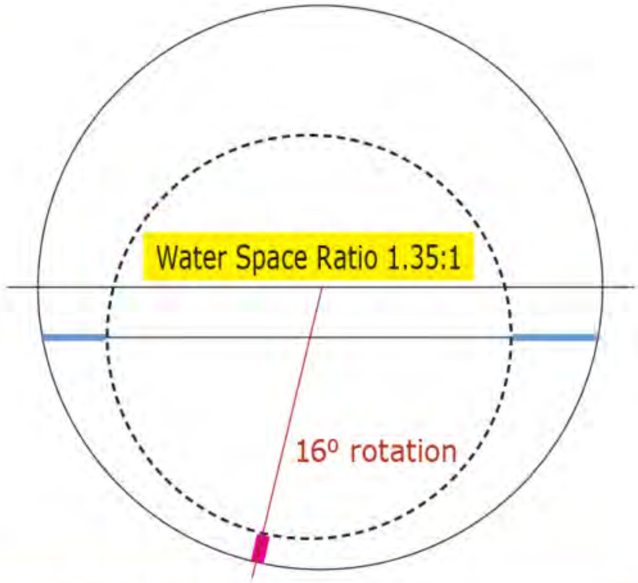
The problem of poor circulation with cylindrical fireboxes may have been mentioned in other papers to professional bodies but was certainly mentioned at some length by George Hughes in his 1909 paper to the I Mech E on "Locomotives Designed and Built at Horwich" (link to free download given later). Following comments from contributors about circulation in boilers, he said that, in his view, the problem was that "it was due to an equal body of water on each side of the flue". Some years prior to presenting the paper he had constructed a small boiler, 12in diameter and with a furnace of 8 5/8in diameter, which was a scale model of that used on the 0-8-0s involved, and carried out several tests. (See pp.640/641screens 80/81)

Tests on model boiler

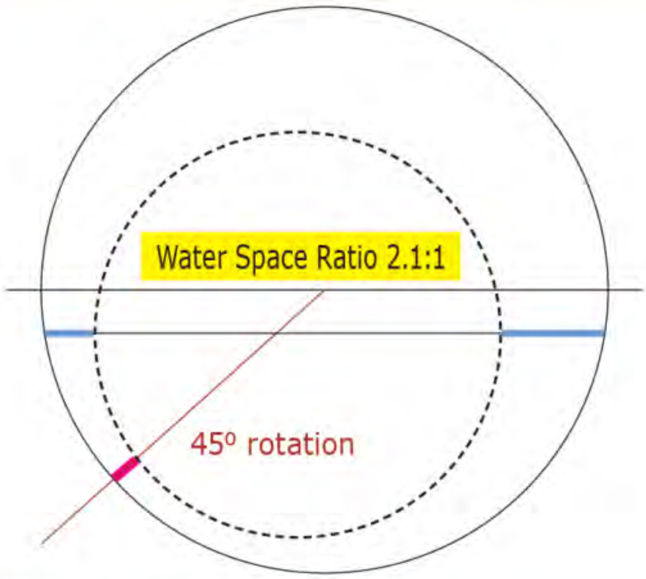
In the first test, with the model boiler as described, he was unable to create any circulation. In the second test, after rotating the boiler so that the flue was ½in off centre, circulation started when the top thermometer was reading 212 deg F and the bottom one 204 deg F. In the third test, after rotating the boiler 45 degrees, circulation started with the top thermometer reading 212 deg F and the bottom one 194 deg F. For the fourth test he inserted a diaphragm plate down the side of the boiler, which I have to assume gave the same ratio of channel widths as the third test, and circulation started at the same temperatures.



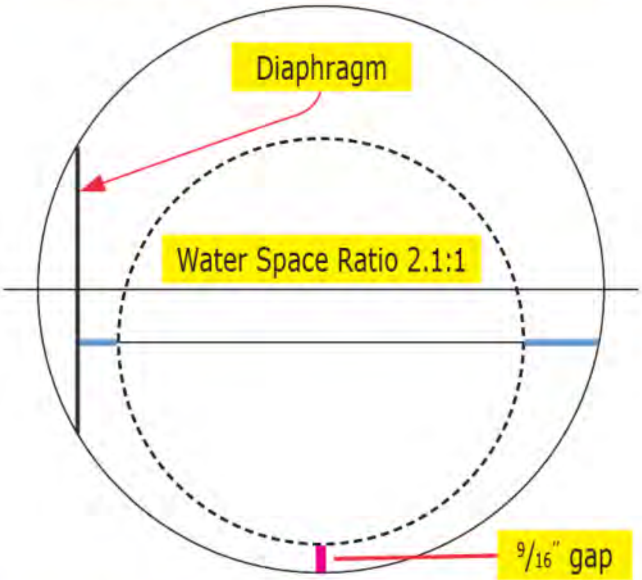
Hughes Experiment 1
Flue on vertical centre line and 9/16 inches above inner surface of boiler. No circulation was recorded.



Hughes Experiment 2
Unit rotated till flue centre line 1/2 inch to left of boiler centre line. Circulation started at 212^{0}F (top) and 204^{0}F (bottom).



Hughes Experiment 3
Unit rotated to an angle of 45^0 maintaining the 9/16 inches clearance. The ratio between the water space on the left and right of the flue was used to determine the position of the diaphragm. Circulation started when top thermometer recorded 212^{0}F and bottom thermometer recorded 194^{0}F.



Hughes Diaphragm Experiment
The flue was on the vertical centre line and 9/16 inches above the inner surface of the boiler. A plate was added along the length of the boiler (bracketed off the boiler shell) with the same ratio between the firebox and the diaphragm plate as in Experiment 3. Circulation started at 212^{0}F (top) and 194^{0}F (bottom).

Note: This is a historical reconstruction based on thin evidence. It is the simplest way of illustrating the tests that Hughes carried out, but there has to be an element of conjecture about it as he did not give full information.

While the results of the tests listed above contain a certain amount of information, they do not contain as much as they could have done had George Hughes been even slightly more methodical. A few examples:-

1) It is not possible to compare the effects of the rotation of the test boiler to different positions when one measurement is given as lateral displacement while the other is degrees of rotation. However, by rotating the boiler in steps of 5 degrees between 0 and 60 degrees, and measuring the temperature at which circulation started, it would have been possible to establish if there was an optimum angle was, and also the lateral displacement.

2) At no point did Hughes give any indication of how long it took for circulation to start for any position of the boiler.

3) No note appears to have been made of the additional time required for steam to be produced, not even at atmospheric pressure.

Unfortunately, the tests on the model boiler did not lead anywhere, as no attempt was made to test a diaphragm in a full size boiler to see what effect it had on circulation, let alone install diaphragms in the twenty-one locos fitted with cylindrical fireboxes. If even the first of these had happened, I would have expected some comment to that effect from Eric Mason, L&YR shedmaster, member of the I Loco E, and author of two books on L&YR matters.

(Free download: Paper - Hughes 1909 - I.Mech.E - Locomotives Designed and Built at Horwich.pdf (lyrs.org.uk))

Questions on Design

In the books available to me there is no information about the design of boilers with the simple layout of cylindrical fireboxes, only their various sizes and some constructional information. It seems likely that staff at various firms must have made design studies and used test models to work out the best relationship between the size of the firebox and the outer wrapper, and any impact this had on circulation, but I am not aware of any surviving evidence for this.

In order to get some understanding of how Cornish boilers, and other boilers with a single flue or firebox, were designed, I have gone through several books I have to see what I could find, and the following books have proved useful.

In chronological order, these are D.K. Clark, “The Steam Engine” for historical background; Ernest Pull, “Modern Steam Boilers” and A. Regnauld, “Modern Power Engineering” for what was current practice and design information; R. A. Whitehead, “Garrets of Leiston”; Alan J. Haigh, “The Design Construction and Working of Locomotive Boilers” and “Locomotive Boilers for the Twenty-first Century”. The other trawl has been through patents held on file.

Stationary Boilers

In the case of the Cornish boiler, Pull gives a table of typical dimensions from an unspecified manufacturer (or manufacturers) with the firebox (called a ‘flue’ in this case) being between 53.8% and 56.3% of the diameter of the boiler, for boilers ranging from 4ft diameter by 12ft long to 6ft 6in diameter by 24ft long, and with an average value of 54.9%. (As far as I can make out, the figures used are for the outside diameter of both the firebox and the boiler.)

MAKERS NOT KNOWN

Cornish Boiler

Typical Shell and Flue Ratios

Length	Shell (in.)	Flue (in.)	Ratio
12' 0"	48	27	0.563
12' 0"	54	30	0.556
14' 0"	54	30	0.556
16' 0"	60	33	0.550
18' 0"	60	33	0.550
20' 0"	60	33	0.550
16' 0"	66	36	0.545
18' 0"	66	36	0.545
20' 0"	66	36	0.545
18' 0"	72	39	0.542
20' 0"	72	39	0.542
20' 0"	78	42	0.538
24' 0"	78	42	0.538
Average Ratio			0.549

Average is of 6 firebox to shell ratios

Garretts of Leiston

Starting in 1909 Garretts introduced a range of superheated overtype, semi-portable units under the designation CCS (Compound Condensing Superheated) (*pictured below*). After some years this range was improved, becoming the MC range, for which relevant figures are available in Whitehead's book. Both types used a boiler described as "Cornish multi-tubular", because they had both a normal Cornish boiler firebox and a section of firetube boiler, thus avoiding the need for any brickwork setting to achieve good boiler efficiency.

In this case, the firebox to shell ratio is slightly higher, and falls between 56.1% and 62.5%, with an average value of 58.7%.

GARRETT MC RANGE

Shell and Flue Ratios

Length	Shell (in.)	Flue (in.)	Ratio
12' 6"	51	30	0.588
13' 0"	57	32	0.561
14' 6"	60	34	0.567
14' 6"	63	37	0.587
15' 6"	66	39	0.591
16' 6"	72	45	0.625
Average Ratio			0.587

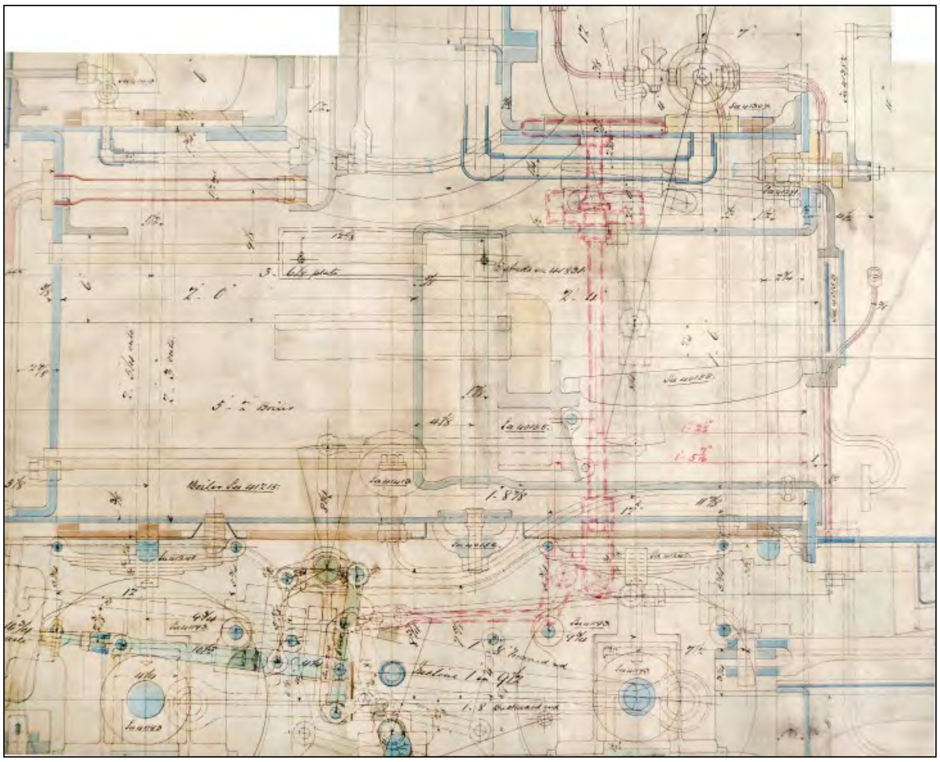
Average is of 6 firebox to shell ratios

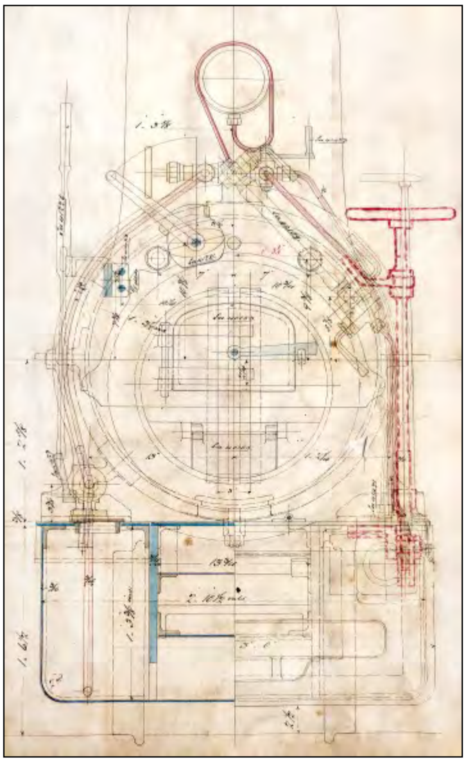
Locomotive Boilers

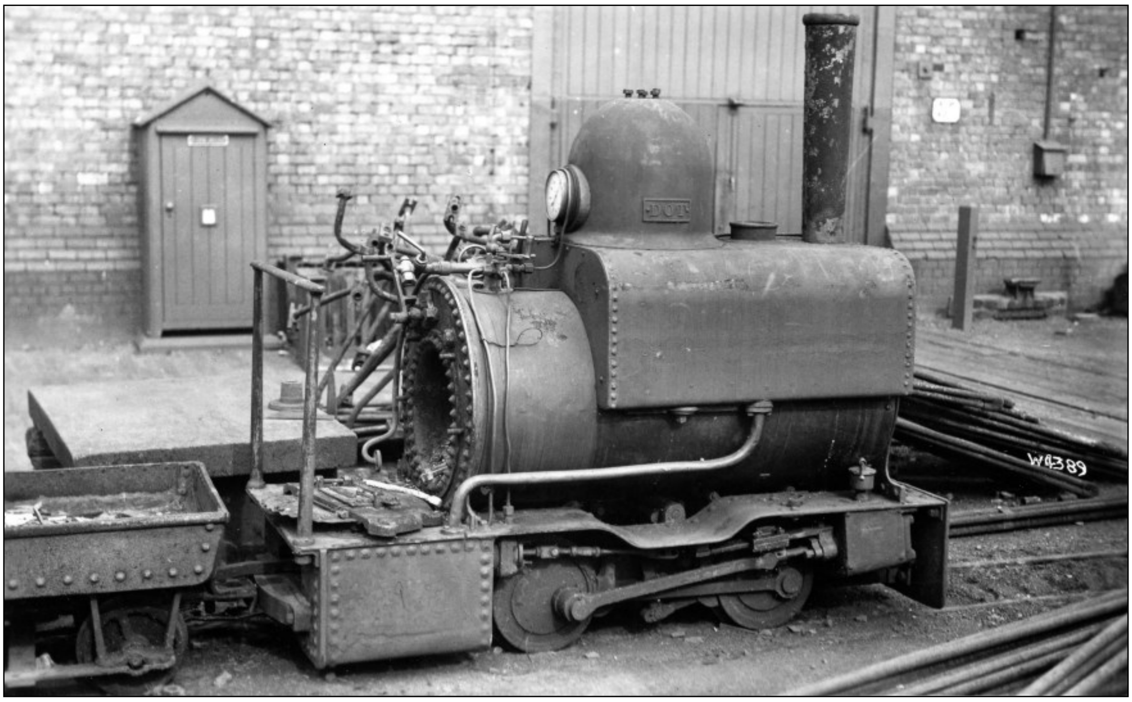
Regarding locomotive boilers, the only two designs for which I have photographs of drawings are at the opposite ends of the spectrum as far as size is concerned. The first is for a small 0-4-0 built by Beyer Peacock for use on their own 18in gauge railway, while the second is the L&YR 0-8-0 referred to above. One thing that is noticeable is that in both cases the cylindrical firebox is a larger proportion of the diameter of the firebox outer wrapper than is the case for stationary boilers, and this is true particularly of the L&YR 0-8-0.

The Beyer Peacock boiler had a firebox of ½in plate with an outside diameter of 1ft 6in, and a boiler of 3/8 in plate with an internal diameter of 2ft 2¼in, meaning that the firebox was 68.6% of the internal diameter, or 66.7% of the outside diameter.

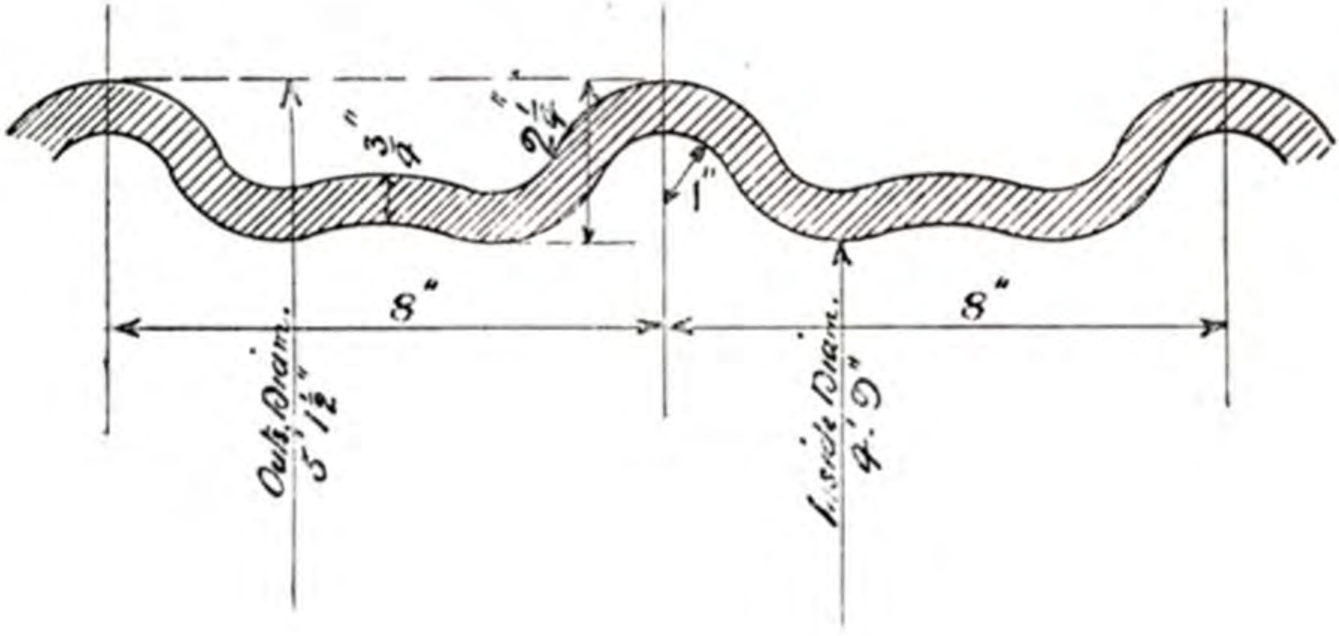
The L&YR 0-8-0 was altogether larger, with 13/16in plate for the outer wrapper, giving an outside diameter of 6ft 9¾in., and an internal diameter of 6ft 8 1/8in. The corrugated firebox had an internal diameter of 4ft 9in, and an outside diameter of 5ft 1½in. Depending on whether you measure the inside or outside diameter of the corrugated firebox against the inside diameter of the firebox wrapper, this gives a figure of either 69.7% or 75.2%, while the 8 5/8in. and 12in. diameters of Hughes's model boiler give a figure of 71.9%, which is closer to that of the average diameter of the corrugated firebox.

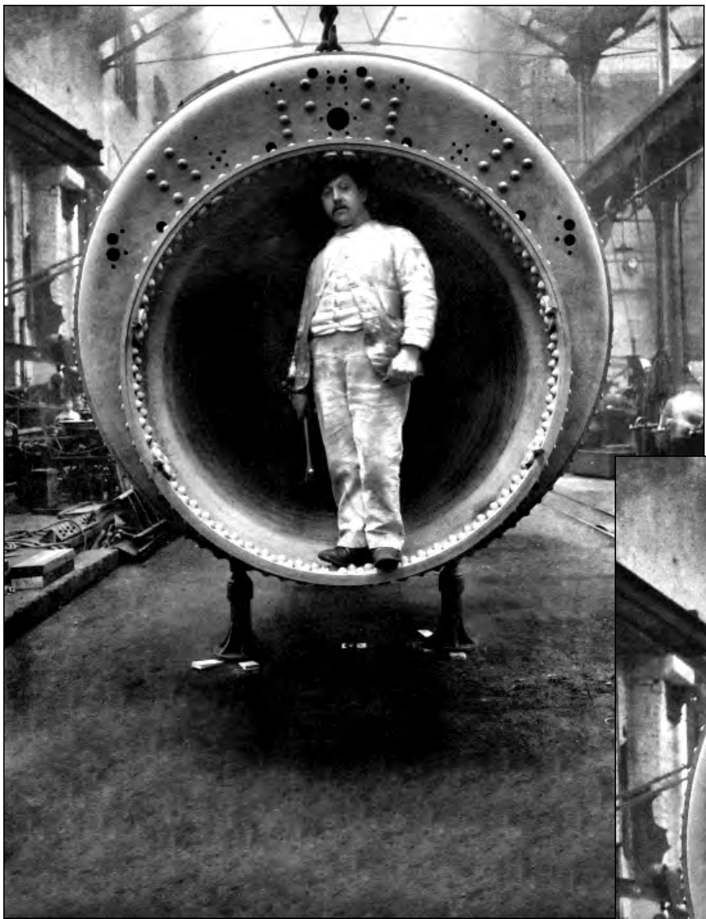




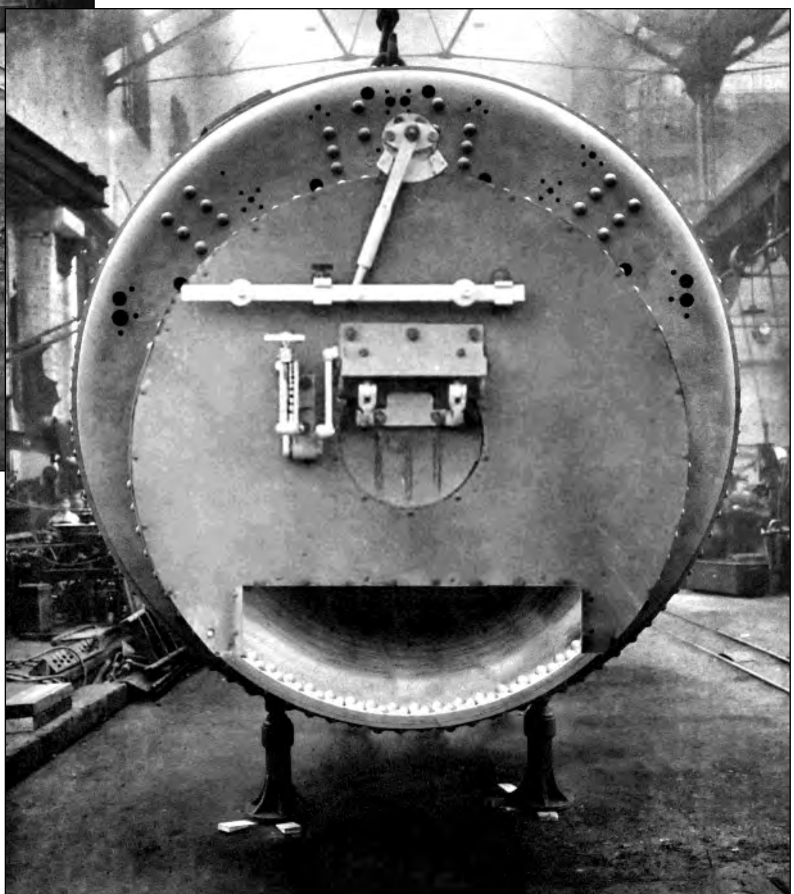


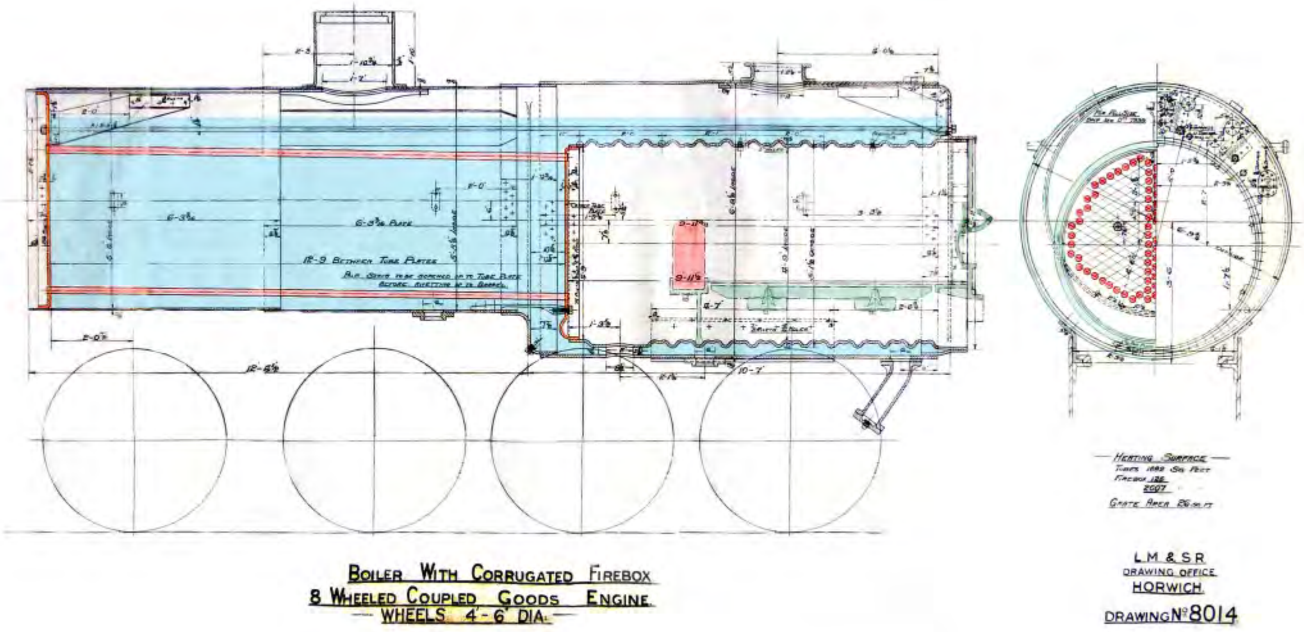
18 inch gauge BP drawings, and photo of complete 18 inch loco



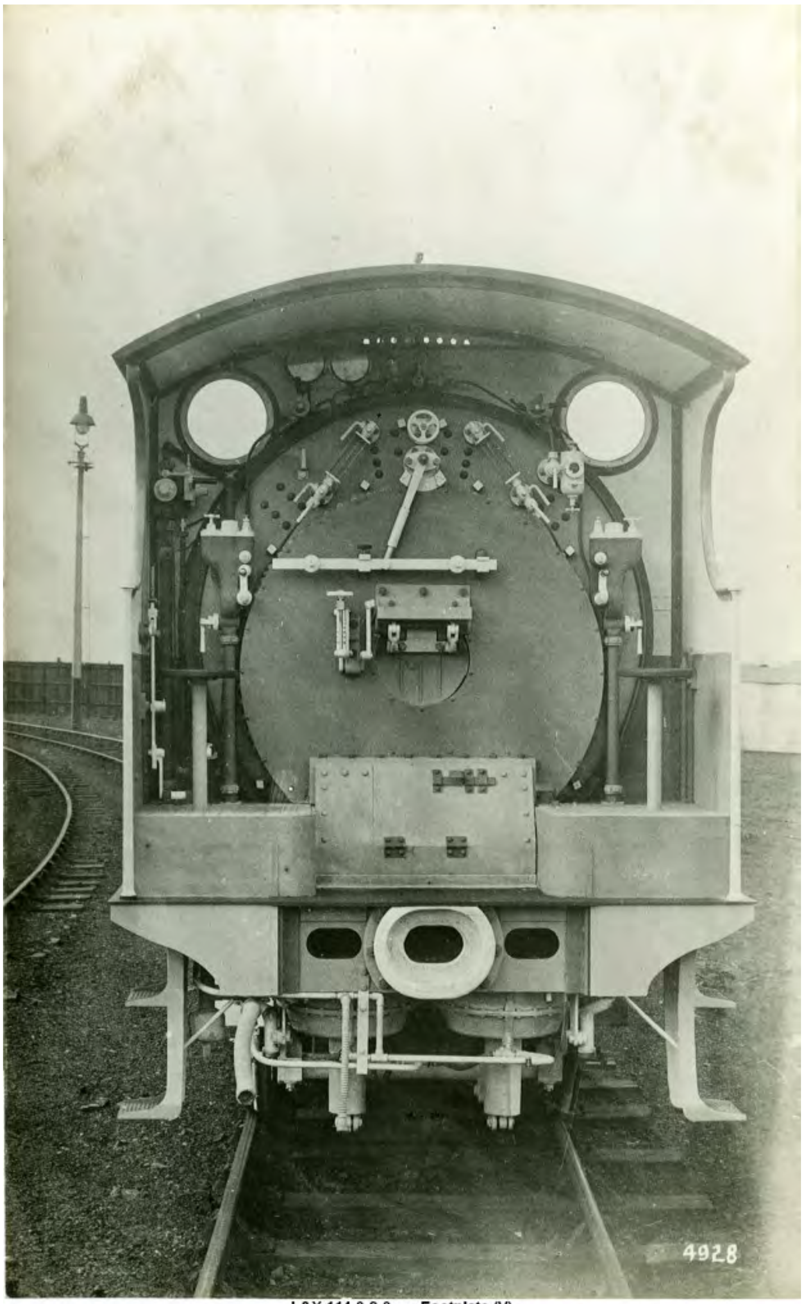


LYR corrugated firebox section drawing,
under construction,
and with mock up back plate.





LYR 0-8-0 with corrugated firebox



L&Y 114 0-8-0 · · · Footplate (V)
· · LPC #4928
John Alsop collection A 40912

Builder	Type	Shell (in.)	Flue (in.)	Ratio
Beyer Peacock	0-4-0T	27	18	0.667
L&YR	0-8-0	81.75	61.5	0.752

Doing Things Differently

Building a boiler with the flue, or firebox, vertically aligned with the boiler was not the only way of doing things. There is published evidence that Messrs. Marshalls of Gainsborough and Davey Paxman both made boilers with channels for water that were unequal, as tested by George Hughes in his model boiler, and most likely for the same reason – to create circulation. And in 1924, at a time when the Cornish boiler was past the peak of its popularity, Regnauld says in "Modern Power Engineering" (see Vol.4, p.2) of this type: "This furnace is usually placed with its centre line on one side of the centre line of the main drum in order to assist the circulation of the water in the boiler", before proceeding to illustrate just such "an improved form of Cornish boiler" made by Messrs. Galloways Ltd. of Manchester.

Functionally Asymmetric Boilers

There is no point in looking up this term in early 20th century engineering literature, because you won't find it. Given that the three manufacturers mentioned above, and possibly others, produced boilers with cylindrical fireboxes which had channels of unequal width between the furnace and the boiler barrel or outer firebox, I considered that it would be helpful to have a generic term for such boilers, as they need to be thought about separately from other 'launch-type' boilers.

From drawings of stationary boilers made by Marshall's of Gainsborough and Galloways, and a patent by Davey Paxman's for their Ultranomic boiler, as well as the tests carried out by the L&YR, it is clear that there are two ways of creating unequal sized channels for the passage of water.

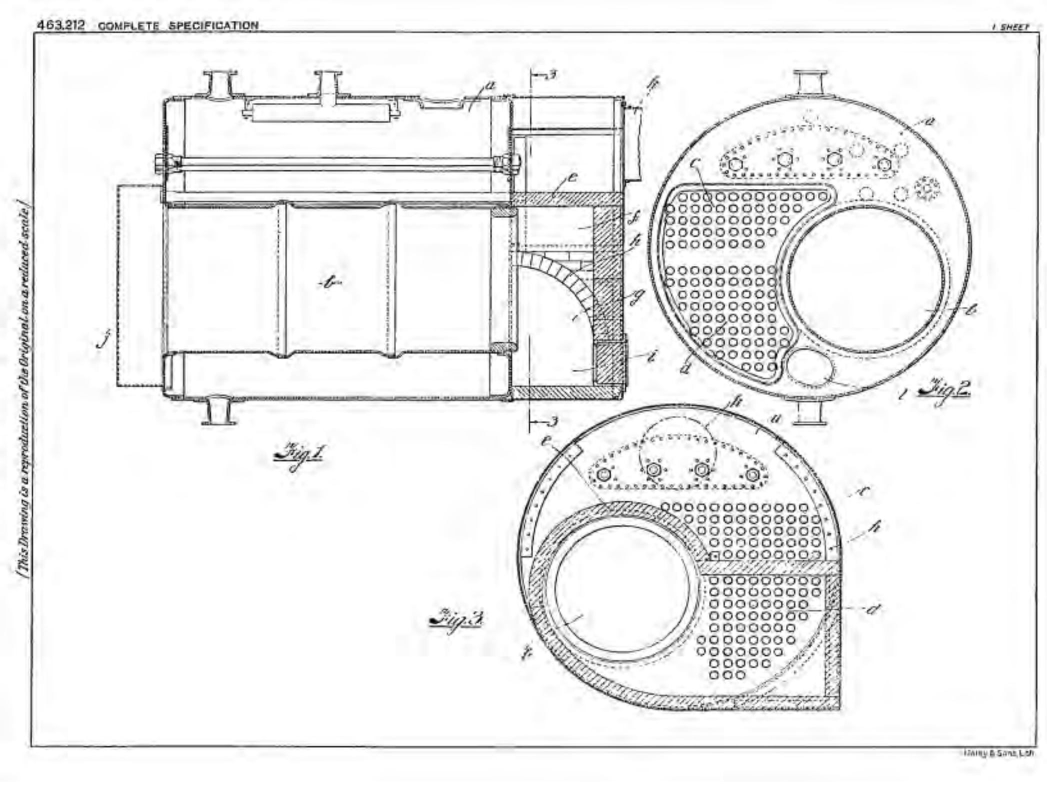
1) By rotating a normal Cornish, or similar, boiler to a different position.

2) By keeping the boiler in its normal position, with the centre of the firebox and wrapper vertically aligned, but inserting a diaphragm to create the unequal channels.

In both cases the boiler has all its internal forces balanced and dealt with in the normal manner, but inserting a diaphragm, or rotating the boiler, to provide channels of different sizes for the passage of water means that the boiler functions in an asymmetric way, hence the introduction of the term.

Davey Paxman 'Ultranomic Boiler'

Unfortunately, the only evidence I have in this case is the patent specification (see separate attachment) taken out by Hazell in 1936 and granted in 1937 (GB46312A). This not only mentions in the text improvements to circulation, but the drawings clearly show the offset firebox which is referred to in the text of the application. However, the banks of tubes to one side of the furnace are clearly very different from anything that would be appropriate for a boiler on a steam locomotive. Even so, it is evidence of a major boiler manufacturer using an off-centre furnace.



Adamson Boiler

I am afraid that the patent specification (GB637281A) (see separate attachment) is all I have on this design.

637,281 COMPLETE SPECIFICATION — 1 SHEET

[This Drawing is a reproduction of the Original on a reduced scale.]

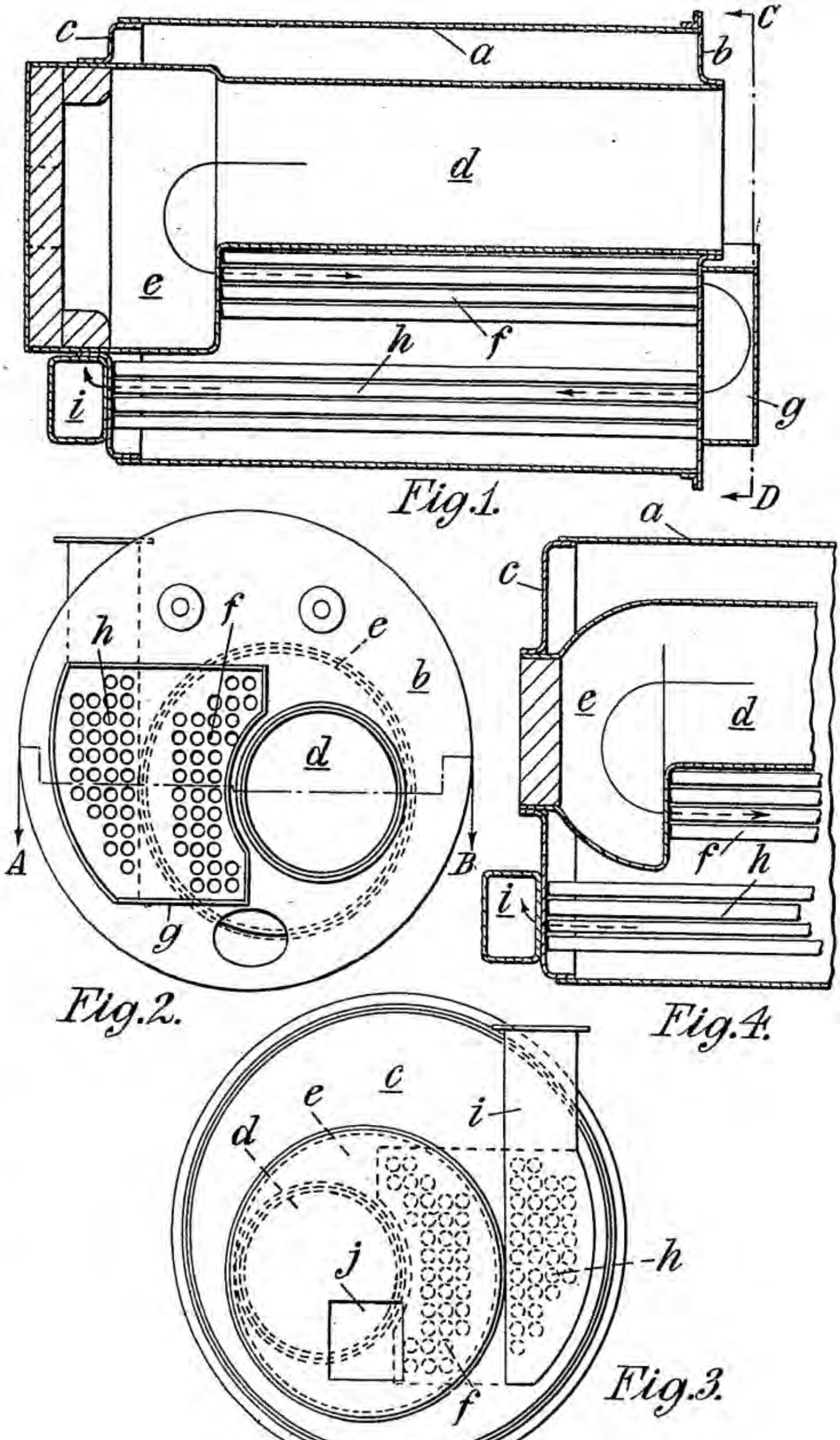


Fig.1. Fig.2. Fig.3. Fig.4.

H.M.S.O.(Ty.P.)

Marshall Boiler

Some useful information about the Marshall's 'multi-tubular Cornish' boiler can be found in Ernest Pull's "Modern Steam Boilers", along with a table of the sizes of boiler made. From the images below it is clear that there was a significant difference in the size of the channels for water and steam on either side of the cylindrical furnace. The text does not mention any improvement in circulation arising from this arrangement, but it would be hard to think of another reason for the claimed 15% improvement as compared to any normal Cornish or Lancashire boiler.

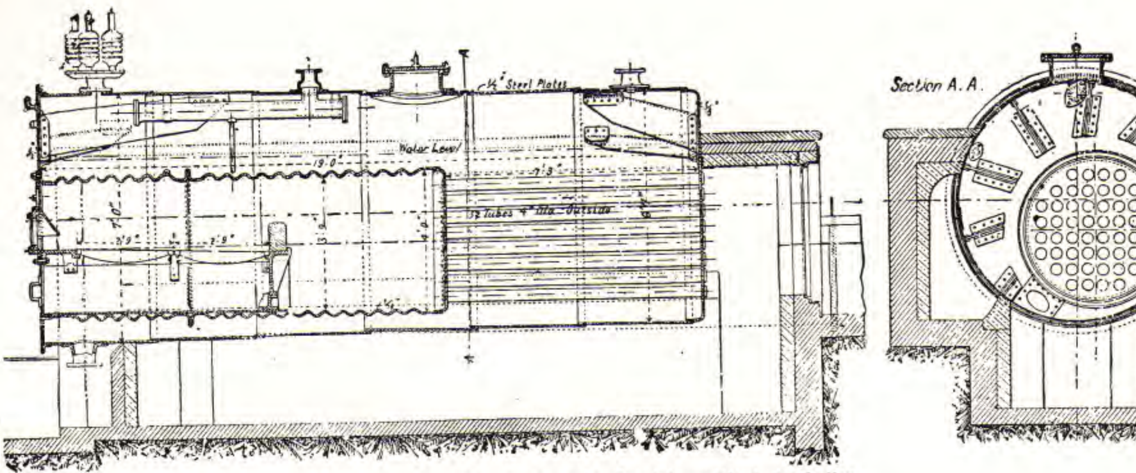


FIG. 85.—Section of a Cornish multitubular boiler.

TABLE 12.—STANDARD SIZES OF EXTERNALLY FIRED CYLINDRICAL MULTITUBULAR BOILERS.

Shell.				Tubes.		Steam Drum.				Grate Area in Square Feet.	Total Heating Surface in Square Feet.
Length.		Diameter.		No.	Diameter.	Diameter.		Length.			
Ft.	In.	Ft.	In.		In.	Ft.	In.	Ft.	In.		
9	9¾	3	0	17	3	1	6	3	0	6·6	180
13	0¾	3	0	17	3	1	6	3	0	8·4	236
12	0	3	5½	24	3	1	6	3	0	10	290
12	0	3	11	28	3	1	6	3	0	11·6	342
14	0	4	4	24	3½	1	6	3	0	13·1	395
14	0	4	8	26	3½	2	0	5	0	14·5	440
14	0	5	0	34	3½	2	0	5	0	17·5	540
14	0	5	3	43	3½	2	0	5	0	21	665
14	0	5	9	45	4	2	0	5	0	24·2	780
14	0	6	2	60	4	2	0	5	0	30	1020
14	0	6	8	77	4	2	0	5	0	38	1250

Cornish Multitubular Boiler.

This type of boiler is supplied in large numbers in connection with fixed engines, and is intended to work in localities where fuel is expensive. They are thoroughly reliable boilers when used with good soft water, and effect a saving of about 15 per cent. in fuel compared with the Cornish or Lancashire boiler. Fig. 85 shows a sectional illustration of a cylindrical multitubular boiler 19 feet long by 7 feet in diameter at the front end. The rings of the shell are arranged telescopically, so that the diameter at the rear end of the shell is 6 feet 7 inches. As will be seen from the end view, the furnace is placed on one side of the centre line, this arrangement giving readier access to the lower part of the boiler for cleaning. The boiler is set with side flues like an ordinary Lancashire boiler; the gases after leaving the tubes, pass first under the boiler, and then divide and traverse the side flues to the chimney.

The large amount of heating surface in the tubes allows of the external dimensions of these boilers being considerably less than those of the Lancashire or Cornish types of a similar evaporative capacity; they are consequently less bulky and heavy, which is often a matter of considerable importance where freight and transport have to be considered. The following are the particulars of standard sizes made by Messrs. Marshall:—

TABLE 13.—STANDARD SIZES OF CORNISH MULTITUBULAR BOILERS.

Shell.				Flue.				Tubes.		Grate Area in Square Feet.	Total Heating Surface in Square Feet.
Length.		Diameter.		Length.		Diameter.		No.	Diameter.		
Ft.	In.	Ft.	In.	Ft.	In.	Ft.	In.		In.		
8	6	3	10	5	10	2	0	16	3	4·6	121
10	0	3	10	5	9	2	0	21	3	6·6	163
10	6	4	5	6	10	2	4	29	3	8·4	204
11	6	4	5	7	0	2	4	34	3	10	243
12	6	4	9	8	0	2	7	37	3	11·6	273
13	9	5	0	9	8	2	9	32	3½	13·1	309
14	3	5	4	9	6	2	9	32	3½	14·5	356
16	3	5	9	11	0	3	0	44	3½	17·5	429
18	6	6	2	12	6	3	3	36	4	21	526
19	0	6	6	12	6	3	6	42	4	24·2	615
21	0	6	9	14	0	3	10	44	4	27·2	707
22	3	7	0	15	6	4	0	52	4	30	800
28	0	7	0	18	2	4	0	52	4	30	1143

The principal results of a seven-hour test of the boiler illustrated are:—

Combustion.

Pounds of coal burnt per sq. foot of grate surface per hour 21·5 lbs.
" " " " " " " " heating " " " 0·534 lb.

Evaporation.

Pounds of water evaporated per lb. of coal from feed temperature 9·36 lbs.
" " " " " " " " " and at 212° Fah. 10·15 "

Compound Cornish Boiler.

Fig. 86 illustrates a compound Cornish boiler constructed by the Cradley Boiler Co., Cradley Hall, Staffordshire. This type of boiler is designed for a working pressure of from 80 to 160 lbs. per sq. inch. The illustration explains the meaning of the name "Compound Cornish," for the reason that the shell is similar in every respect to the Cornish boiler, but the furnace tube terminates behind the bridge in a circular tube plate, and lap welded tubes are continued from this to the back end tube plate of the boiler, where the smoke-box is formed, and fitted with doors, for conveniently examining and cleaning the small tubes when necessary.

The gases, after leaving the tubes, are conducted by side openings in the smoke-box under the bottom or along the sides of the boiler by brick flues, in the ordinary manner, and thence to the chimney.

The brick setting is similar to that provided for the Lancashire and Cornish type boiler.

MARSHALL

Shell and Flue Ratios

Length	Shell (in.)	Flue (in.)	Ratio
8' 6"	46	24	0.522
10' 0"	46	24	0.522
10' 6"	53	28	0.528
11' 6"	53	28	0.528
12' 6"	57	31	0.544
13' 9"	60	33	0.550
14' 3"	64	33	0.516
16' 3"	69	36	0.522
18' 6"	74	39	0.527
19' 0"	78	42	0.538
21' 0"	81	46	0.568
22' 3"	84	48	0.571
28' 0"	84	48	0.571
Average Ratio			0.539

Resolving the Problems?

Investigations Needed

In order to see how far the problems with cylindrical fireboxes can be at least ameliorated or, even better, overcome, several design studies are needed, starting with a geometrical one. There are several questions that need to be investigated. This could, hopefully, be done more easily with computer graphics than would have been the case with graph paper and pencil 120 years ago.

Are there some proportions that clearly work better than others? At present no-one knows, and whether there is any surviving evidence from earlier research is an open question, hence the need for these investigations.

1) Geometrical

a) As outlined above, the available evidence suggests that the ratio of the flue or furnace to the boiler diameter could vary from 51.6% to 62.5% for stationary boilers, and go as high as 75.2% for a locomotive boiler. It would, therefore, seem sensible to consider the various changes arising from such variations. This could be done in a series of diagrams in 5% steps over the range of firebox to boiler from 50% to 75%.

b) This situation is complicated by the fact that no study appears to have been made to determine the associated impact of the distance between the bottom of the firebox or flue and the barrel in relation to the size of the firebox and barrel. Given that any improvement in circulation in a boiler with a cylindrical firebox or flue would need water to flow under the firebox, getting this wrong would have a sizable impact on water movement and the steaming properties of the boiler, so establishing some sort of minimum figure or ratio would be very helpful.

c) The third factor that has to be considered is the depth of water covering the top of the firebox or flue, and how much space that leaves for steam. Lack of adequate steam and water space has been a problem with many designs of cylindrical firebox, according to Wikipedia, but an analysis of the options should help to mitigate the problem.

As can be seen from the table below, If the height of the steam space available is less than 20% of the diameter of the firebox wrapper or boiler, then the length of the water line across the firebox cannot be more than 80% of the diameter, and reduces markedly as the height for steam reduces. This reduction inevitably reduces the area for steam bubbles to disengage from the water in the boiler, and risks increasing the wetness of the steam produced and, ultimately, priming.

Water line length above firebox crown as proportion of boiler diameter

	Diameter	Radius	Water line length	Increment
Steam space height	0.10	0.20	0.6000	
	0.12	0.24	0.6499	0.0499
	0.14	0.28	0.6940	0.0441
	0.16	0.32	0.7332	0.0392
	0.17	0.34	0.7513	0.0181
	0.18	0.36	0.7684	0.0171
	0.19	0.38	0.7846	0.0162
	0.20	0.40	0.8000	0.0154
	0.21	0.42	0.8146	0.0146
	0.22	0.44	0.8285	0.0139
	0.23	0.46	0.8417	0.0132
	0.24	0.48	0.8542	0.0125
	0.25	0.50	0.8660	0.0119

d) Fitting a boiler with an offset firebox to a locomotive would run the risk of creating an imbalance in the transverse load on the axleboxes, similar to what happened with Bulleid's 'Leader'. On that basis, the only practical way of making a locomotive boiler functionally asymmetric would be by inserting a diaphragm on one side, and bracketing it to the boiler so that there is no possibility of any difference of pressure on either side of it, while any imbalance in weight would be insignificant.

e) As diaphragms appear never to have been fitted to locomotive boilers, a study is needed on the effect of diaphragms on the width of channels to achieve a variety of ratios for the channels. Some very preliminary work has been done on this in relation to the L&YR test boiler to try and understand the changes that Hughes made, but more clearly needs to be done.

2) Practical Tests

It is not clear if the geometrical investigations will clearly indicate which of a wide range of possibilities can be safely ignored for future designs, or if all of that range has to be tested on a practical basis. While Computational Fluid Dynamics (CFD) may be of great assistance in dealing with the problems being discussed in this paper, I am afraid that my knowledge of mathematics has never been at the level needed for CFD, and I would welcome input from anyone who reads this who is familiar with it and can assist in reaching sensible conclusions. Failing that, tests will be needed to investigate and, hopefully, solve problems.

Given that the practical tests required cannot be known at present, I think that it is better to consider the outcome desired from such tests.

1) A range of options for new construction, covering the points previously made, and needing clarification.

2) Options for improving the circulation and steaming of existing boilers with cylindrical fireboxes (flues).

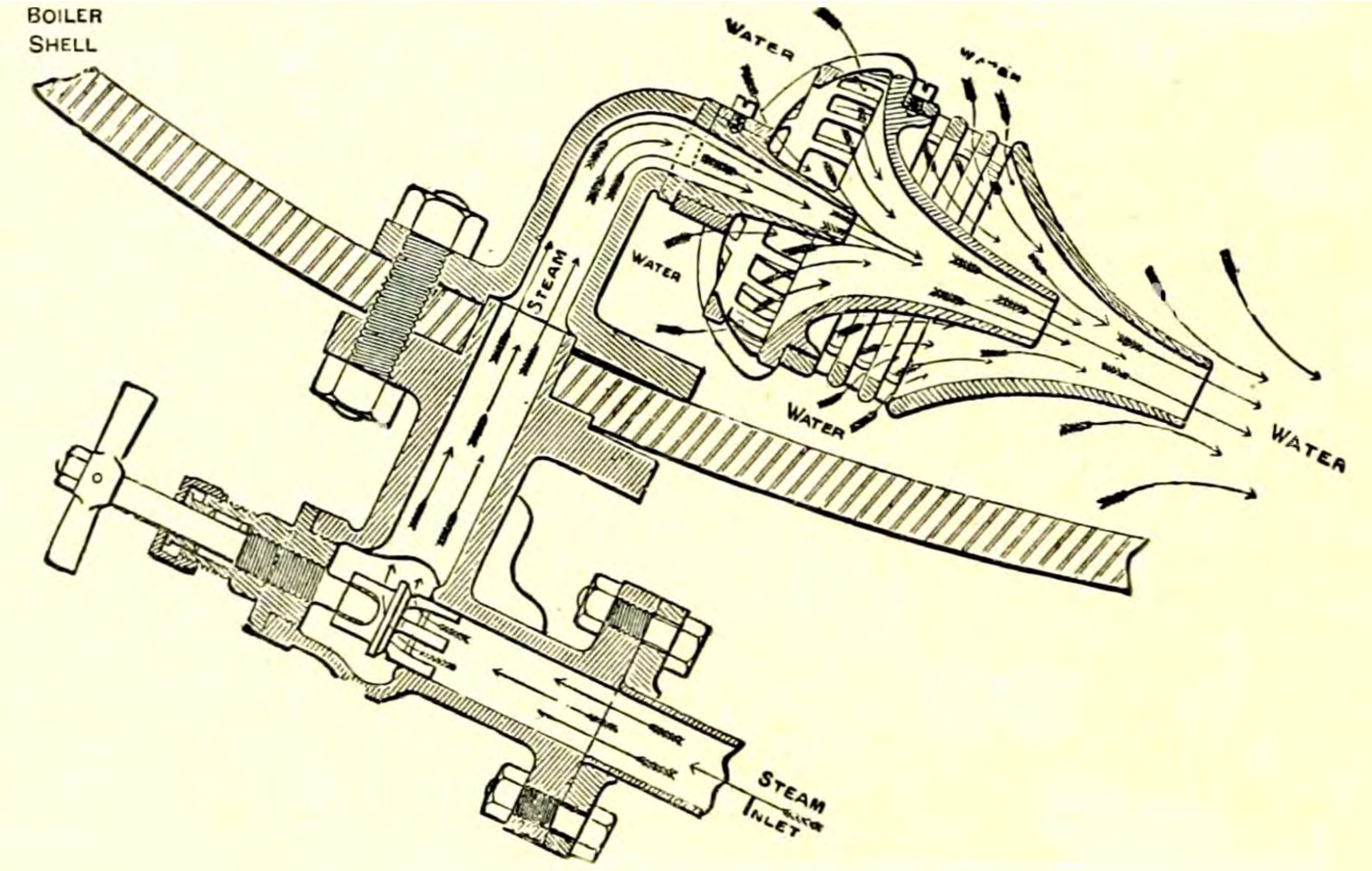
And Now For Something Completely Different

Creating Circulation

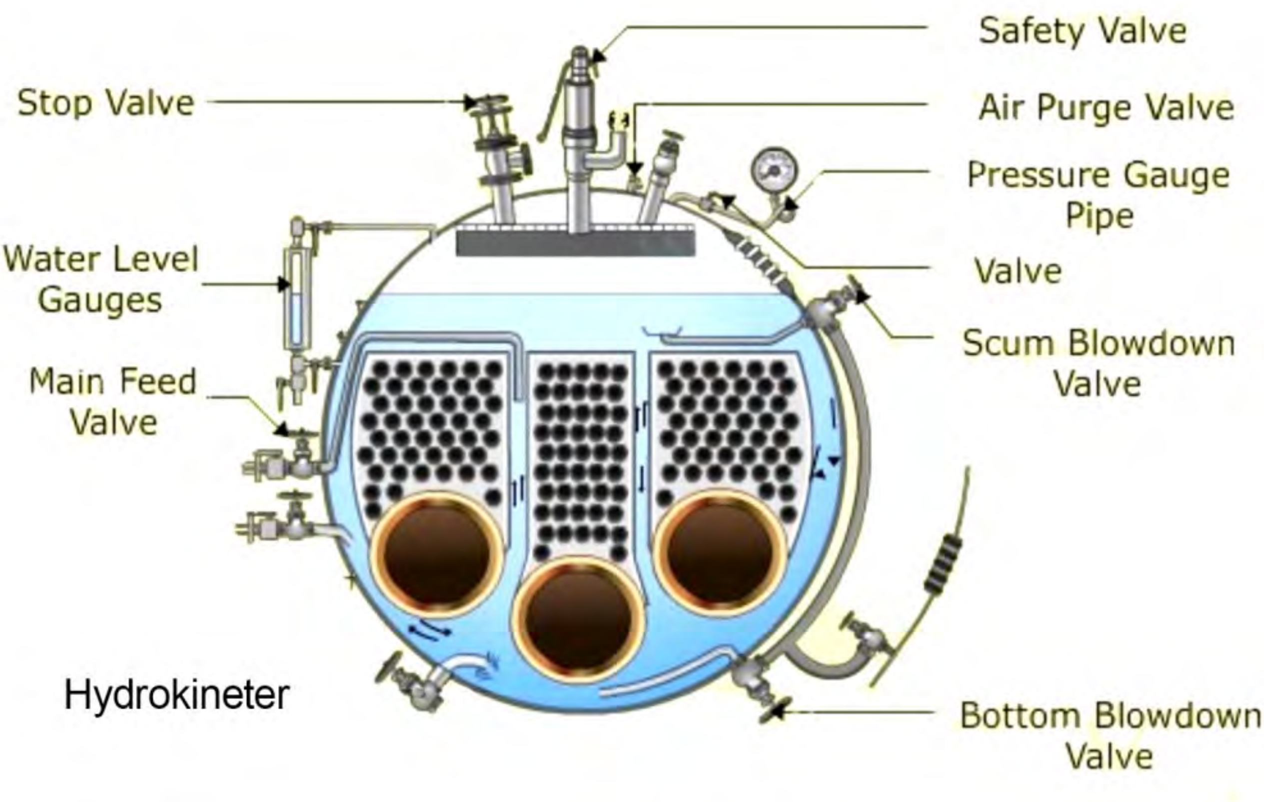
Unless you have at least a modest interest in marine steam engineering (or have been approached by me on the topic) you are unlikely to have come across a circulation device that was widely used on Scotch boilers for decades - the "Hydrokineter" - patented by Weir Brothers in Glasgow in 1875. Unfortunately, 19th century engineers appear to have operated in self-contained groups, with little or no communication between them and, as far as I can make out, this device was never used on any land-based stationary boilers, and certainly not on any locomotive ones, despite its success at sea.

The Patent Specification and diagrams show that this could work in either of two ways. The first of these, and one which was used widely, was for a steam jet similar to that of an injector to create a moving jet of water underneath the fireboxes of the Scotch boiler, with the circulation created reducing the time taken to raise steam to full pressure from cold by effectively 30%. As most ships had a donkey boiler as well as several boilers for driving the engines, steam from the donkey boiler was used to start one boiler and, once steam had been raised, this could then supply steam for the jet circulator on the others.

The second way described in the Patent was for an impeller to be used, feeding the moving water through a tube inside the boiler and under the firebox/flue. While the illustration shows an impeller that is hand cranked, or could be driven by a cord on a pulley, it would be much better these days to use a detachable electric motor to drive the spindle of the impeller. Unfortunately, I have not been able to trace any reports on the effectiveness of the impeller version of the Hydrokineter, and how it compared with the steam jet version, but if it should prove to be even half as effective, it would be worth testing on a full size locomotive. It looks as if a practical test of some sort is going to be necessary on a scale model boiler to start with.



Hydrokineter in close up (above), and attached to the boiler (below)



Combining the Two Approaches

In principle there seems to be no reason why a boiler with a cylindrical firebox or flue, and fitted with a diaphragm, should not also be fitted with one or other form of the Hydrokineter. In the form of that device relying on a steam jet, there was a shut-off valve (mentioned in the patent) so that steam was not admitted to the jet once a certain pressure had been reached, usually that of the donkey boiler starting the process. In the case of the impeller form of the device no information is available. However, once steam bubbles form, and there is a difference in density in the asymmetric channels for steam and water sufficient to create circulation, the need for it would clearly be reduced, though at what point after that the impeller should be turned off could only be established by a practical test.

Other Possibilities

1) Assuming for a moment that inserting a diaphragm into a boiler with a cylindrical firebox/flue is not effective or cost effective (though this seems not just unlikely but also improbable), it would still be possible to improve the performance of such a boiler with one or other form of the Hydrokineter, and reducing the time taken to raise steam to full pressure would have obvious benefits to availability, if nothing else.

2) In an exchange of emails I had with Alan Haigh in 2019 it became clear that he had never come across the Hydrokineter during his career at Green's in Wakefield, which surprised me. At the time I only knew of the steam jet form of this, and when I asked how it could be fitted to a normal locomotive boiler, he suggested that it that the best place would be at the bottom of the barrel, and directing the jet of water towards the firebox tubeplate. Though Alan Haigh did not suggest this, I wonder if removing about half a dozen tubes at the bottom of the barrel would give a much freer flow of water all the way from the smokebox tubeplate to the firebox tubeplate, and up through the hottest end of the tubes and superheater flues. The loss of heating surface would be minimal in such a case, but the improved flow of water could minimise the possibility of film boiling on the tubes and flues, thereby extending their life.

Conclusion

I hope I have provided sufficient evidence to show that the use of a cylindrical firebox or flue in a steam locomotive boiler is not automatically the failure that is assumed, but presents problems that can be researched and dealt with, at least to the extent of making such a boiler useful for short journeys on heritage railways, thereby reducing the cost of providing new designs to satisfy any interest in steam traction that passengers on such lines may have.

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Davey Paxman

Richard Carr's Paxman History Pages - HOME Page

Richard Garrett & Sons

The Long Shop Museum

File:Im20110715LS-i075a.jpg - Graces Guide Type CCS IV (Two other suitable photos available on GG website)

Marshalls of Gainsborough

File:Im19231115PE-Marshall.jpg - Graces Guide

Locomobiles @ The Marshall Club – Please note that this is not a Locomobile as illustrated by Graces Guide, but a large Portable with cylindrical firebox and boiler with axial taper.

THE PHYSIOLOGY OF THE LOCOMOTIVE BOILER - ANOTHER PEEP!

By Adrian Tester

Preface

At our 2022 conference in Darlington, Adrian Tester presented a paper titled "The Physiology of the Locomotive Boiler - Another Peep!". *The Physiology of the Locomotive Boiler* is the title of a book that Adrian will shortly be publishing (in two parts). The phrase "Another Peep" presumably alludes back to an earlier paper that he presented to ASTT's 2017 conference held in Bury when he was in an earlier stage of writing the book.

In Darlington, Adrian accompanied his slides by reading verbatim from several pages of handwritten notes. His slides can be found on ASTT's website, but his spoken words were lost because none of the presentations were recorded.

After the conference, at my request, Adrian kindly gave me his notes so that I might transcribe them. It's no small challenge interpreting Adrian's handwriting so I'm grateful that he has checked through my transcription which can be found in the following pages.

I have taken the liberty of adding several explanatory footnotes, each of them quoting, or being based on, clarification and explanations that have been given to me by Adrian.

It is hoped that those who attended the conference will be glad of the opportunity to refresh their memories of Adrian's words, and that those members who were unable to attend will appreciate the opportunity to read the words behind one of the many interesting presentations at the conference.

Adrian is in the process of publishing the first volume of his new book which will share the same title as his paper. It is hoped that he will allow ASTS to market it under the same arrangement that it has enjoyed for selling two of his other books – viz:

An Introduction to Large-Lap Valves & Their Use on the LMS, and

A Defence of the MR/LMS Class 4 0-6-0.

Both are recommended reading for anyone who is interested in what goes on "under the bonnet" of steam locomotives (and not just the MR/LMS Class 4 0-6-0s!) Both books can be purchased through the "Books for Sale" page of ASTT's website.

Chris Newman (Transcriber)

Slide 1—Introduction (Slide not included)

At first glance, the boiler is a simple beast containing a fire surrounded by water which it turns into steam – superheated if desired.

In fact, to arrange for a boiler to produce a stipulated quantity of steam at certain temperature and fuel consumption is a very difficult problem to solve. Its successful design demands an understanding of combustion, heat transfer, the applications of fluid flow, etc.

Tragically, for much of the life of the steam locomotive, not much was known about these items, with the partial exception of combustion. Then once fluid flow and heat flow began to be solved in the first half of the 20th century, too much of this new-found knowledge seems to have been ignored by locomotive engineers! Instead, they remained wedded to outdated concepts and thinking until the end of BR steam.

To explain this, and to explore how a boiler works, demands lots of time, long explanations and some meaty sums. This, doubtless you will all be relieved to hear, will not be attempted this morning!

Instead, this is a revised version of a talk that I gave a few years ago. For those of you with long memories, I apologise but hopefully there is sufficient new material. While for those of you who desire the full explanation, the heat transfer volume is currently being printed.[1]

[1] The boiler book is planned to be in two parts, the first one considers heat transfer.

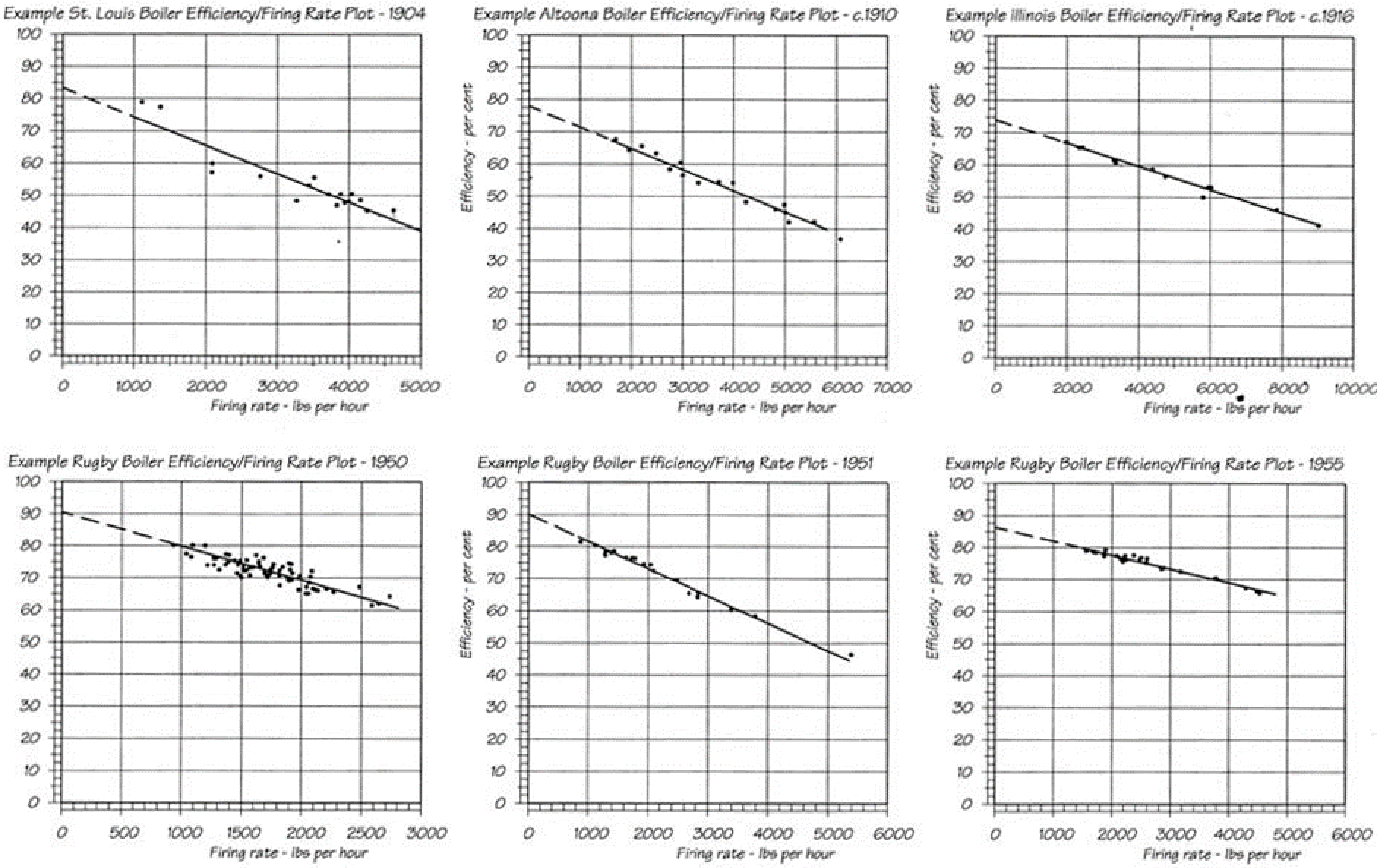

Fig. II.5 - The improvement in the accuracy of boiler efficiency tests with experience

Top Left	Top Centre	Top Right
St Louis Exposition PRR test plant new and in its temporary home	Plant transferred to Altona and demonstrably greater experience	The Illinois plant - not used much but capable of good results in skilled hands.
Bottom three examples are from Rugby.		

Slide 2

One of the earliest proponents of stationary locomotive testing, Professor Goss of Purdue University influenced by William Rankin, adopted a shallow curved line relationship to represent boiler efficiency.

This curve, if extended, would become almost horizontal at high firing rates, implying a more or less constant efficiency. This is erroneous but it remained a common belief in the UK, at least until the mid-1930s – examples being Phillipson in his design data book[2], the Great Western and the LMS.

Initially Fry used it, but around the time of the Great War the belief that a straight line was a better representation became more common in the USA and Europe. Fry almost certainly did not originate the straight-line theory, but he undoubtedly greatly encouraged its adoption with his book 'A Study of the Locomotive Boiler' of 1924.

Interestingly, other bodies responsible for testing non-locomotive coal-fired boilers also adopted this straight-line relationship, seemingly independently[3].

The six graphs, using the original plotting points – even in the case of older examples – demonstrate that boiler efficiency is we'll represented by a straight line and that the more carefully and accurately the tests were carried out, the easier it is to draw a straight line through them.

[2] Phillipson refers to Rankine's formula albeit he does not acknowledge it – see Steam Locomotive Design Data and Formulae by E.A. Phillipson (1936) page 49, republished by Camden Miniature Steam 2004.

[3] This was a boiler efficiency curve produced by an official UK research body. The locomotive boiler is fundamentally very similar to other multi-tubular internally fired coal burning boilers differing primarily in the use of a stronger draught.

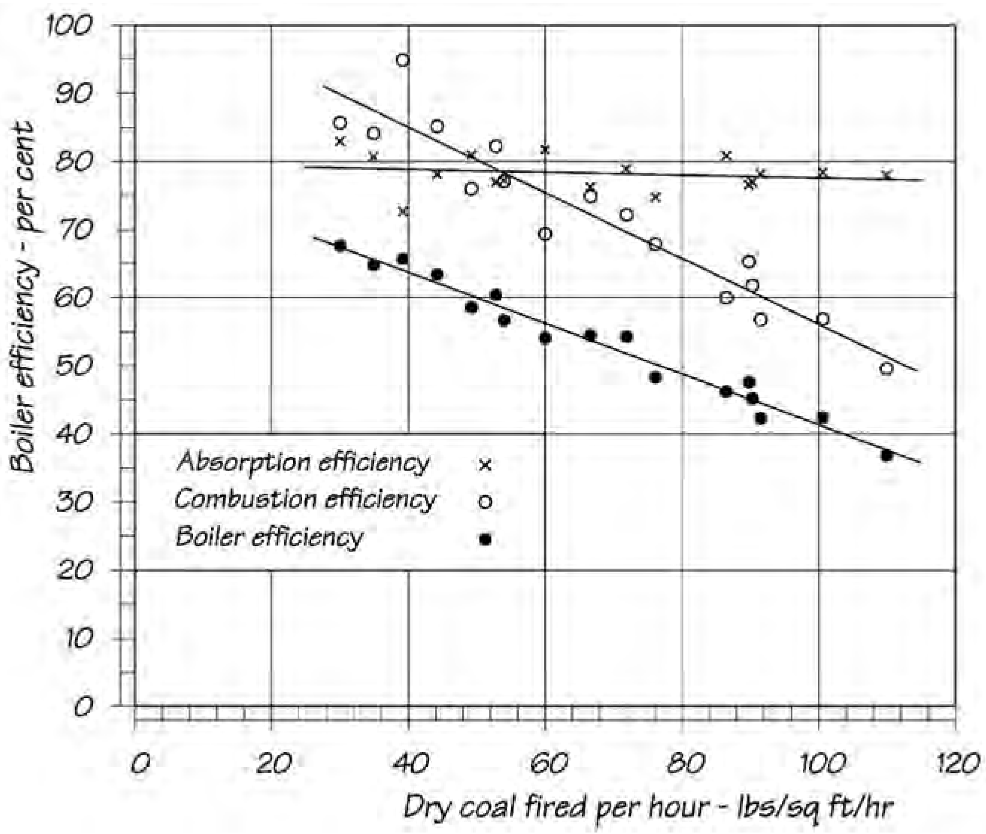

Fig. II.4 – Boiler efficiency curves for Pennsylvania Railroad E2a 4-4-2 N° 5266 tested at Altoona in 1905 – analysis by Lawford Fry

Slide 3

It had been known from the earliest days of the locomotive that boiler efficiency comprised two components: absorption efficiency and combustion efficiency. Unfortunately, the pioneers had no means of quantifying the efficiencies mathematically.

P.K. Clark, based on some shaky evidence provided by Richard Peacock, advised that combustion efficiency remained high in a locomotive boiler until very high firing rates, so therefore the drop in boiler efficiency was due to the loss in absorption efficiency - the associated rise in exiting gas temperature with output, being cited as proof. This theory prompted the practice of making the heating surface area larger than the grate area by a certain factor[4]. This worship of the value of heating surface, especially indirect, attained its zenith in Germany under Wagner. It is a load of nonsense!

In their analysis of the test results from testing stations, the early experimenters had two unknowns: the standing loss and the combustion efficiency. Methods for establishing the latter were based on contemporary stationary boiler practice modified to account for the higher unburned coal loss.

To do this, the early testing stations adopted elaborate means for collecting the sparks and cinders ejected from the chimney. But despite all their efforts, they were hopelessly inaccurate, producing very variable results.

In any series of runs, the calculated standing loss – the assumed unknown value – varied widely. It might range from 1% to 10% or more.

Lawford Fry had the brilliant idea that since the standing loss could only be very small, if a small value was assumed for it, then the combustion efficiency became the remaining unknown. This could now be solved without the need to try to capture the unburned material. Initially he used 5% as the standing loss but later reduced its value.

The three curves are the result of much analysis. The absorption efficiency is seen to be a straight line and to fall only a small amount with rise in output. The combustion efficiency is the factor largely determining boiler efficiency, and it falls significantly.

The lowest curve, which is the boiler efficiency, is the product of the other two reduced by 5%.

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[4] Heating surface ratio and free gas area, although related, are considered here as different factors. The former has little impact on boiler efficiency once a certain (smallish) amount is present while the latter has a big impact on boiler resistance and thence steam output. Not very confident (or good) designers liked to use 'optimum' ratios in the hope of success. It was not a recipe for progress.

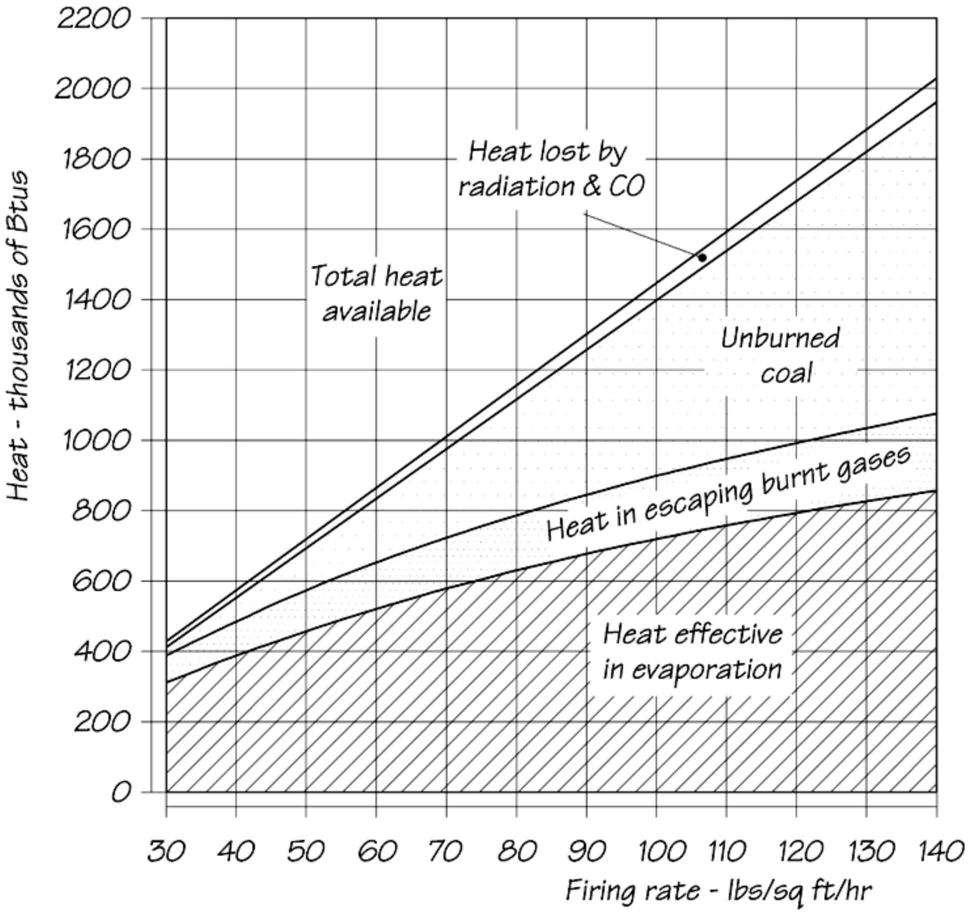


Fig. IV.10 - Approximate heat utilized and lost per square foot of grate per hour at different firing rates - St Louis Test Series 200 - Lake Shore & Michigan Southern Railway 2-8-0 N° 734

Slide 4

This diagram, which is based on one produced by Messrs Trevithick and Cowan in 1913, contrasts the heat utilized with that lost as boiler output rose. The diagram was derived from one of the test series running curves produced by Lawford Fry in 1905 for a IMechE paper. They were based on data obtained from the testing station during its temporary home in the St Louis Exposition.

This diagram caused consternation amongst the engineers present when the paper was delivered. William Rowland and George Churchward refused to believe it could be true! The latter's refusal to accept the findings undoubtedly affected the accuracy of GWR testing in the 1920s, and probably into the 1930s. What the disbelievers overlooked was that the vast majority of the ejected particles were very small in size and black, so they could not be seen! The loss due to poor combustion is by far the largest, but following the now well-known work carried out by Porta and others, much has been written on the subject of improving combustion. Hence, with one exception, we will concentrate for the remainder of this talk, on heat absorption.

Transcriber's Note: *This figure is a way of demonstrating the heat balance, but serving to emphasize the losses and the useful heat absorbed in evaporation relative to the heat in the fuel fired. The first appearance of this format seems to have been in the paper "Some Effects of Superheating and Feed-Water Heating on Locomotive Working" by Messrs Trevithick and Cowan in 1913, which in a slightly modified form appears here. The heat input, represented by the specific firing rate follows a simple linear relationship formed from the product of firing rate and calorific value. The second linear curve represents the available heat after the reduction for the standing loss. It also includes the carbon monoxide loss. The proportion of the total heat utilized in steam production is seen to fall as the rate of firing increased. Due to the remaining two curves being parabolas, the proportion of the total heat in the coal fired which escaped unburned increased ever more rapidly until the grate limit was reached. This prompted consternation and disbelief amongst some of the engineers present - as doubtless was intended. William Rowland thought the diagrams unreliable and the unburned loss could not be anything like what was being suggested - a point taken up by George Churchward.......*

Slide 5

This series of graphs summarise the boiler performance of an unidentified South African Railways locomotive. It was produced by Dr M.M. Loubser and formed part of a paper he wrote on railway mechanical engineering for the Institution of Loco Engineers.

The boiler portion of the paper was extracted and re-written by Cox to result in their “joint” paper. In its rewritten form it devoted a great deal of attention to LMS boiler experience particularly in respect to the Jubilees.

This diagram, which did not appear in the revised paper, is firmly based on Fry analysis – something which the LMS had seemingly only just accepted.

The two combustion curves appearing in the lowest graph are complementary. The curve recording the coal burned Gb approximates to a parabola as one would anticipate from the linear relationship describing combustion efficiency. The other curve approximating to another parabolic curve but to the opposite hand as it records the increase in unburned coal with rise in boiler output.

If we mentally extend these two curves forward with increase in firing rate, we may readily see how they will meet one another at some higher output, and hence the maximum output of the boiler is attained.

Whether this can happen in fact depends in practice on the capacity of the draughting system, but the important thing to appreciate is that for any given boiler there is a maximum steam output possible.

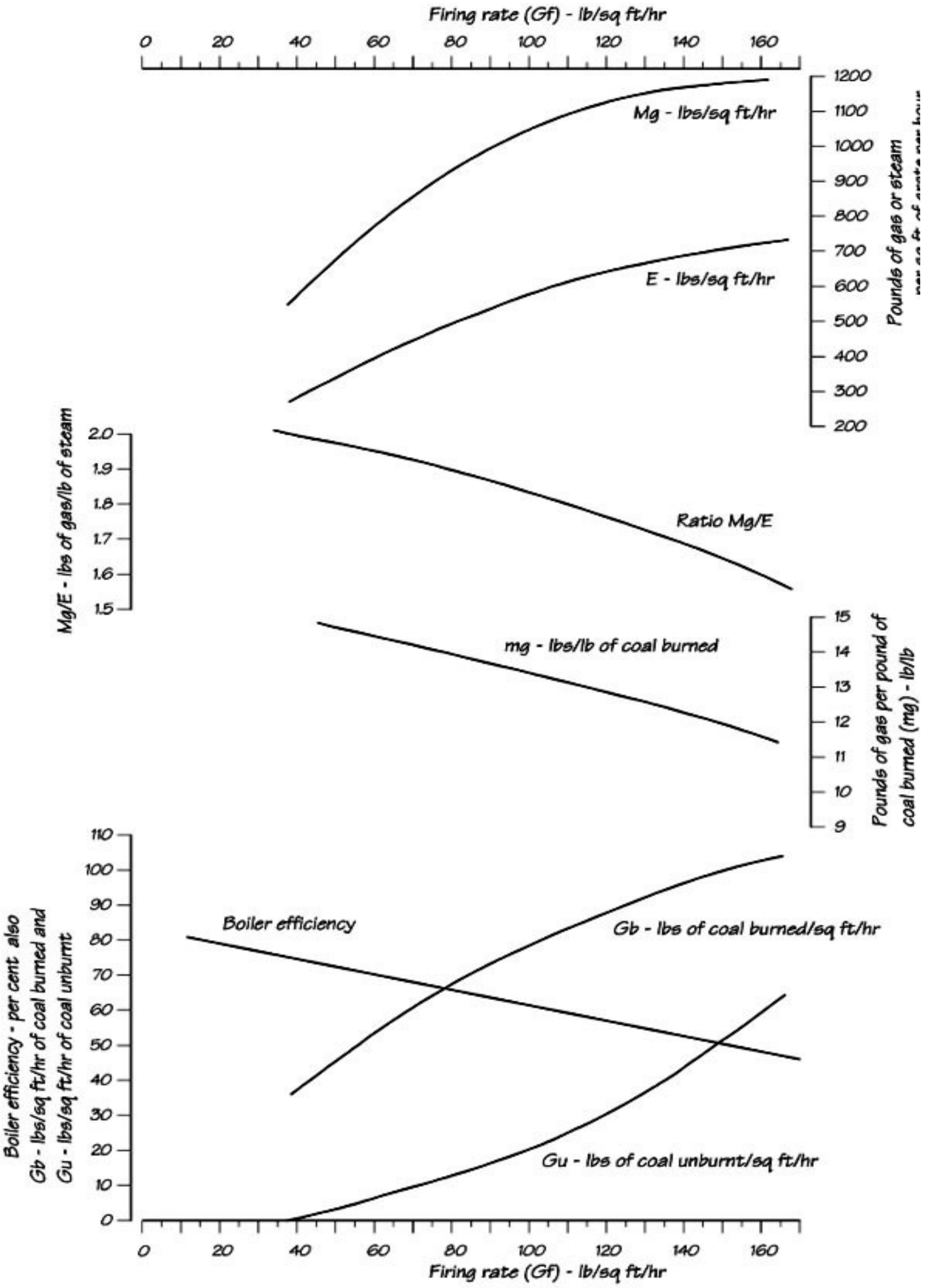
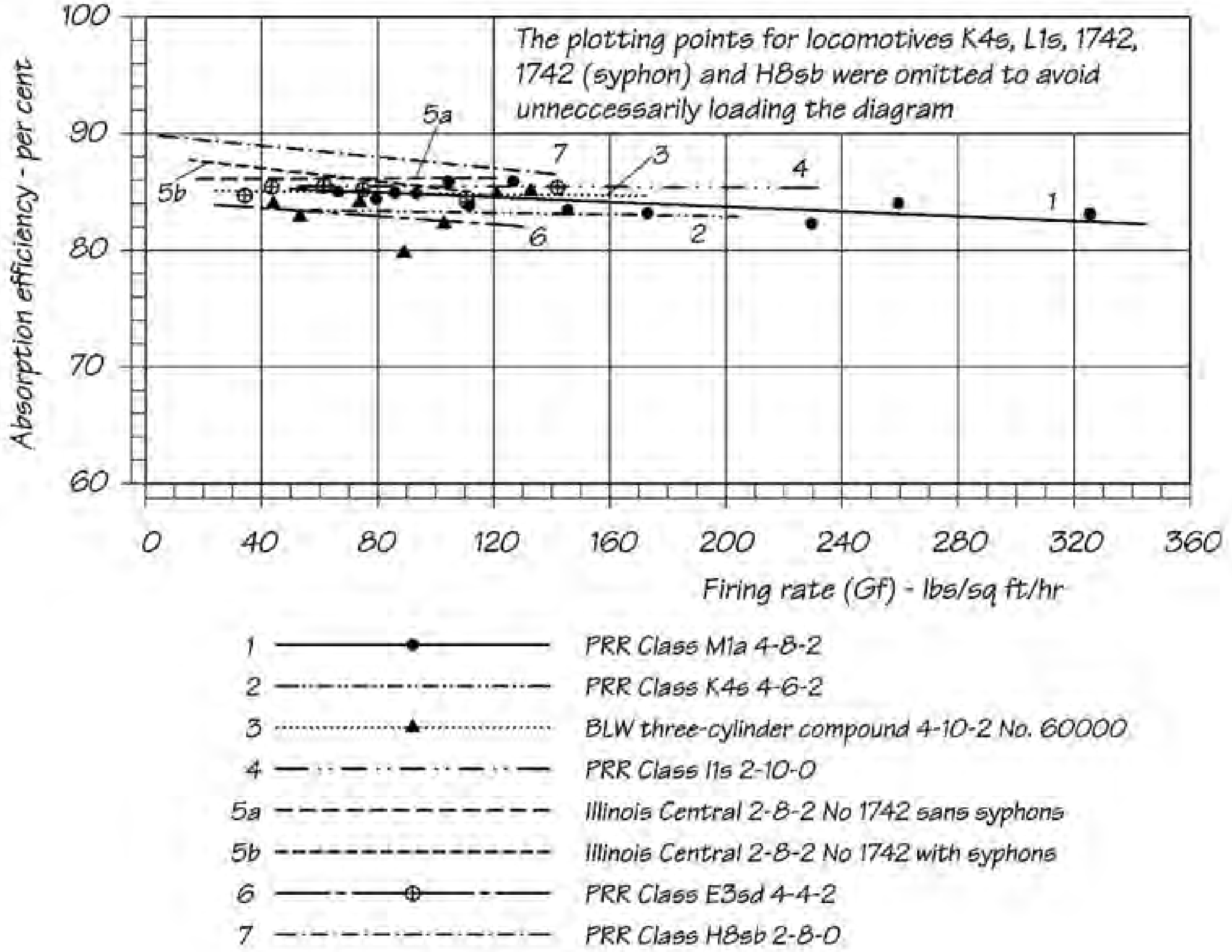


Fig IV.24 - Boiler performance chart for unidentified SAR locomotive - Dr M M Loubser’s paper “Some Aspects of Railway Mechanical Engineering”

Transcriber’s Note: *The bottom pair Gb and Gu record the coal burned as opposed to the coal escaping unburned when plotted against the specific firing rate Gf – lbs/sq ft/hr of grate.*

The diagram shews another way of displaying boiler performance as opposed to, say, the BR four-quadrant version used in Rugby Bulletins. Boiler efficiency follows the familiar linear relationship against firing rate. The topmost pair of curves, approximating to the anticipated parabolic, describe the evaporation and the flue gas produced per square foot of grate. The middle two curves both exhibit distinct curvature, though this is less pronounced in the case of specific gas production mg per pound of coal fired compared to the ratio Mg/E pounds of gas per pound of steam. Lawford Fry considered the first relationship to be linear, giving several examples in ***A Study of the Locomotive Boiler*** *with engines draughted nearly to the grate limit. Mr Ell advised when the flue gas Mg is plotted against steam production E, the resulting curve is linear; but plotting against firing rate results in a curve.*



Slide 6

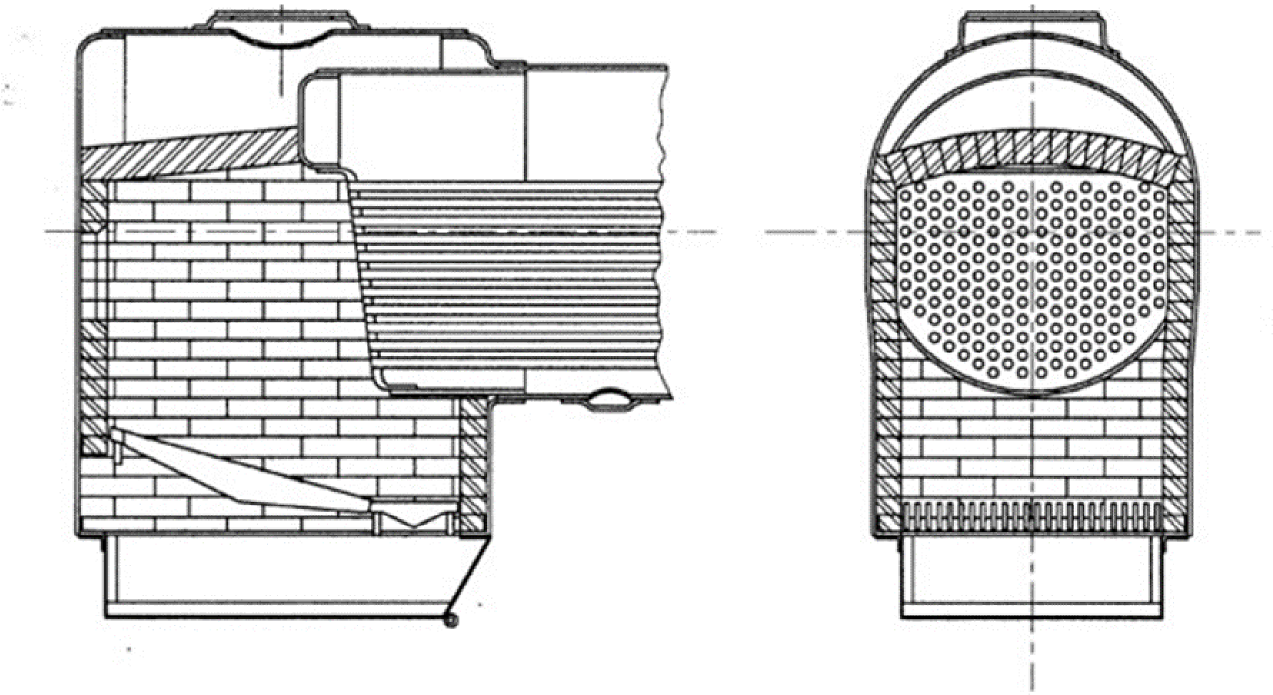
This diagram, extracted from Chapelon's *La Locomotives a Vapeur*, compares the absorption efficiency of a number of locomotives.

Inspection reveals that not only does the absorption efficiency follow a linear relationship, falling only slightly with rise in output – as may be confirmed by calculation – but also the differences in absorption efficiency between one boiler design and another are remarkably small. Over the operating range of these engine, their absorption efficiencies remain between 87% and 82%.

Note: These values are based on Fry's original assumption of a 5% standing loss. If a smaller, more appropriate value had been used, this would have resulted in a slight reduction[5] in these absorption values. If this is done then the revised values would be typical of a modern multi-tubular steam boiler not fitted with an economiser.

Only by introducing an economiser may absorption efficiency be increased to any degree, and then only by a few percent. The only comprehensively tested application in Britain of an economiser was of a Crosti boiler to a 9F 2-10-0. This was a failure primarily because it was applied to a large locomotive that was steamed at too low a rate resulting in the potential heat reclaim from waste gases being compromised. The low temperature and associated low flow rates prevented the feedwater heater from having much effect. Had Jarvis's suggestion of applying a Crosti to a 4-6-0 been adopted then a different conclusion might have been drawn.

[5] Before the heat can be lost it has to be first absorbed. So, if the standing loss is reduced, then the absorbed heat is reduced.



Locomotive	Line	Distance - km	Load - tons	Ton-km	Coal consumption Total - kg	Coal consumption per ton-km	Water consumption Total - kg	Water consumption kg/kg
19	Budapest - S Tarjau	123.4	380.0	46892	2,471	0.0527	11,550	4.67
104	- do -	123.4	352.8	435.4	2,425	0.0557	11,490	4.73
19	Budapest - Miskolcz	182.6	301.8	550.0	3,360	0.0611	15,410	4.59
104	- do -	182.6	292.5	534.4	3,428	0.0641	15,140	4.41
19	**Averaged results of the above**	**153.0**	**340.9**	**521.6**	**2,916**	**0.0559**	**13,380**	**4.62**
104	**- do -**	**153.0**	**322.7**	**493.7**	**2,927**	**0.0593**	**13,320**	**4.55**

Fuel - brown coal from S. Tarjau mines
Six-coupled goods engine No 19 was fitted with an ordinary firebox, No 104 was fitted with a Verderber firebox - both engines were class III.

Fig. VI.29 - Boiler having no direct heating surface - Verderber boiler Hungarian State Railways

Slide 7

The tubular heating surface was very effective at absorbing heat. Thus, almost any combination of tube number, diameter and length will result in a satisfactory value for the absorption efficiency – once the amount of tubular surface provided exceeded a surprisingly small minimum.

Indeed, as the drawing of a Verderber boiler fitted to a Hungarian State Railways goods engine demonstrates, it could also cope with the absence of a conventional firebox. The boiler efficiency was more or less identical to that of a conventional boiler fitted with a firebox.

There are several important reasons for retaining a firebox and also for making it as large as possible and thus providing plenty of direct heating surface[6]; but improving absorption efficiency is not one of them. Likewise, fitting siphons, security circulators, arch tubes, cross tubes, etc, has no real effect on improving heat absorption because the downstream tubular surface is so effective. Where such devices may have a positive, or negative, effect is on water movement around the firebox walls, not the combustion efficiency within it.

[6] In addition to increasing the direct heating surface, enlarging the firebox helps alleviate thermal stresses.

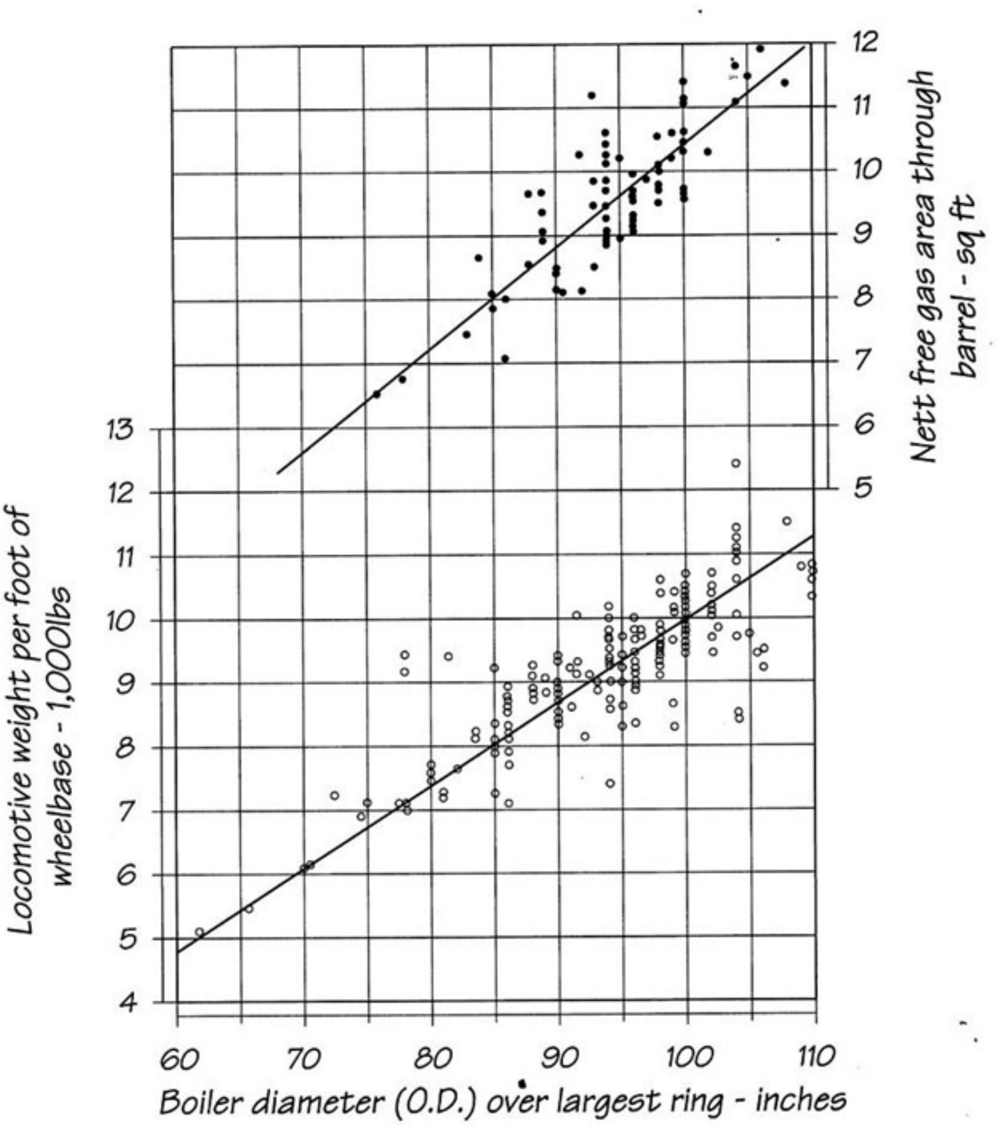


Fig IV.27 - Relation of boiler diameter to total weight of locomotive per foot run of wheelbase and also the available free gas area provided

Slide 8

The other important function of the tubes is that they serve as conduits for the waste gases. Whilst a small area might suffice for effective heat transfer – indeed it is improved by restrictive channels – this is not necessarily the case for gas flow. Tubular surface was very effective at consuming draught with the result that the latter might be so attenuated that the boiler could not supply the desired steam flow with the extant design – the boiler had too much resistance.

Increasing the free gas area was a way of reducing barrel resistance but if it was at the expense of a larger diameter barrel, it increased weight - as can be seen in this American diagram. Not a good idea for something that was intended to move! Crowding more tubes into the same diameter barrel could create tubeplate problems as could increasing tube diameter. Too large a free gas area can turn the engine into a coal thrower.

Boiler design is a balance between draughting efficiency, the steam output required and boiler size. This is part of the reason why an A/S ratio of 1/400 coupled with 15% free gas area[7] were considered crucial by certain LMS engineers - not because these values had particular merit per se, but rather adopting them would result in a boiler barrel resistance which was within the capability of the draughting system, and thus hopefully give the desired steam output.

Fit a better draughting system, a higher resistance barrel can be used – which is what Chapelon did. Now, a higher resistance barrel will not result in much of an increase in absorption efficiency, but it will act as a "snubber" so reducing the effect of the exhaust pulsations on the fire as well as helping to improve the evenness of the draught distribution over the grate.

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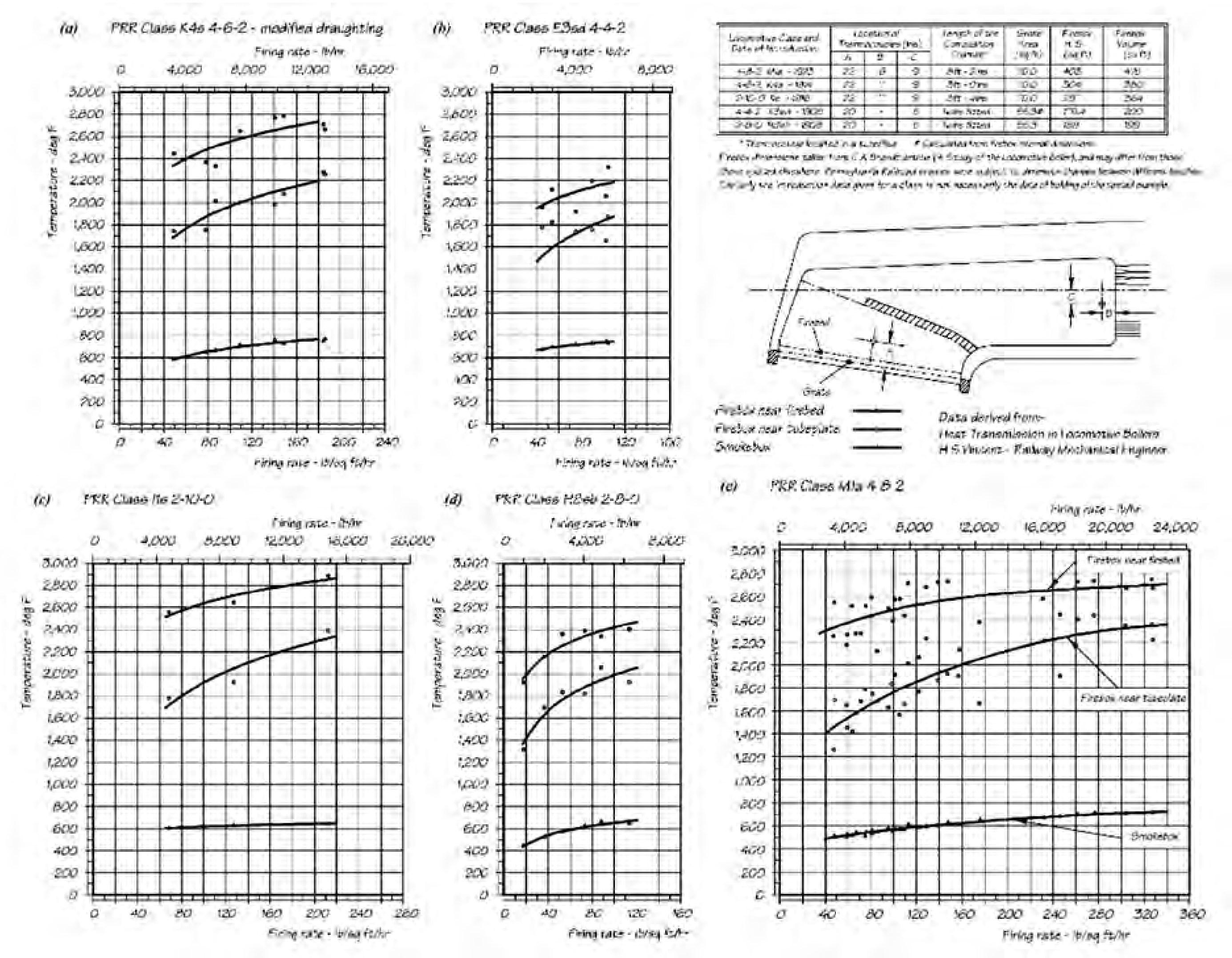
[7] "15% free gas area" means that the unobstructed area of flues and tubes divided by grate area ≈ 15%.

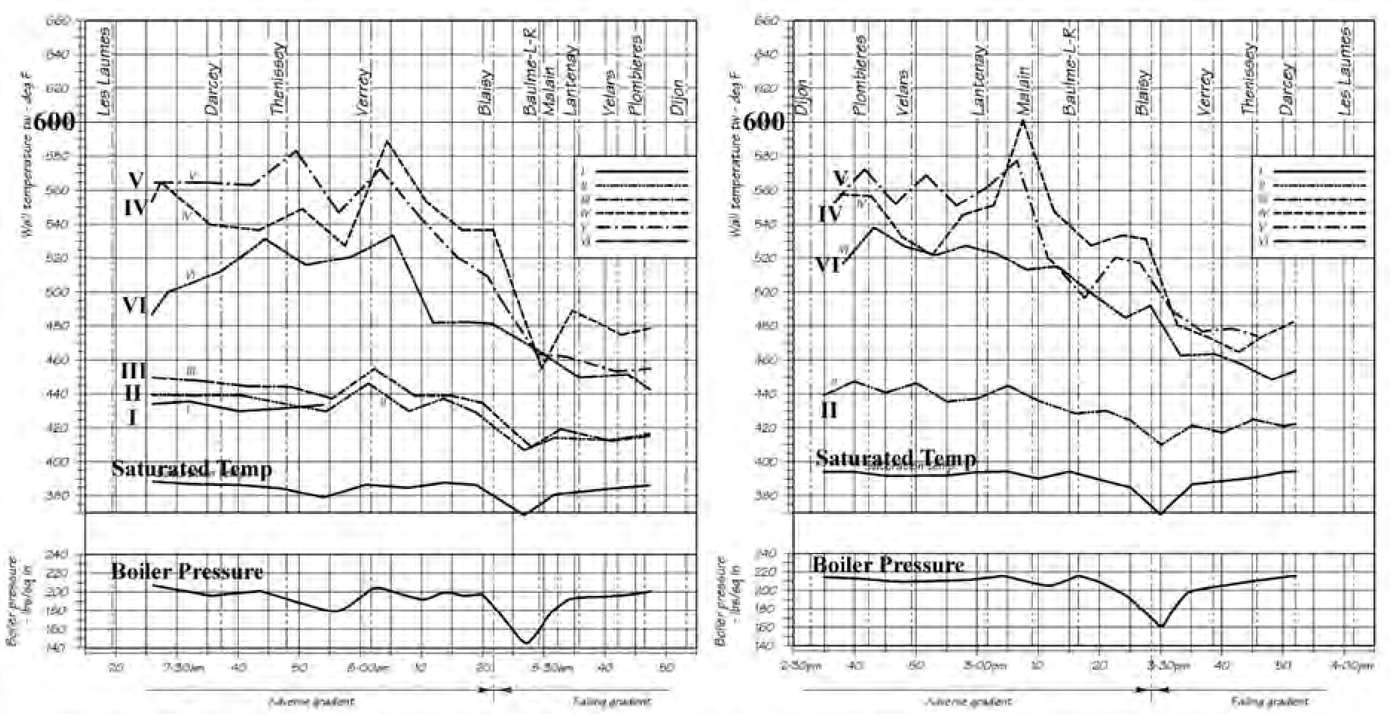


Slide 9

Here we see how the firebox and smokebox temperatures for a selection of locomotives rise with increase in steam output.

Two firebox temperatures are provided for each machine, one measured just above the firebed and the other near the tubeplate. The difference between these curves records the temperature drop as the gases negotiated the firebox.

Despite the considerable scatter – a reflection of the difficulty in measuring high temperatures – we may see as the output rose the temperature difference fell from around 1000 degrees at low output to roughly one third or a bit more at the highest output. This reduced drop indicates a fall in heat transfer effectiveness of the firebox. In contrast, the increase in the difference between the tubeplate temperatures and smokebox temperatures – despite a rise in the latter – indicates a gain in the effectiveness of the tubular surfaces.



Copper firebox - locomotive No. 141-C-448; left hand diagram running from Les Laumes to Dijon hauling train M.15 (902 tons) on 29th February 1940; right hand diagram running from Dijon to Les Laumes hauling train M.52 (894 tons) on 1st March 1940.

Fig. VII.30 - Comparison in wall temperatures in copper and steel fireboxes - South-western region of the SNCF 1940

Slide 11

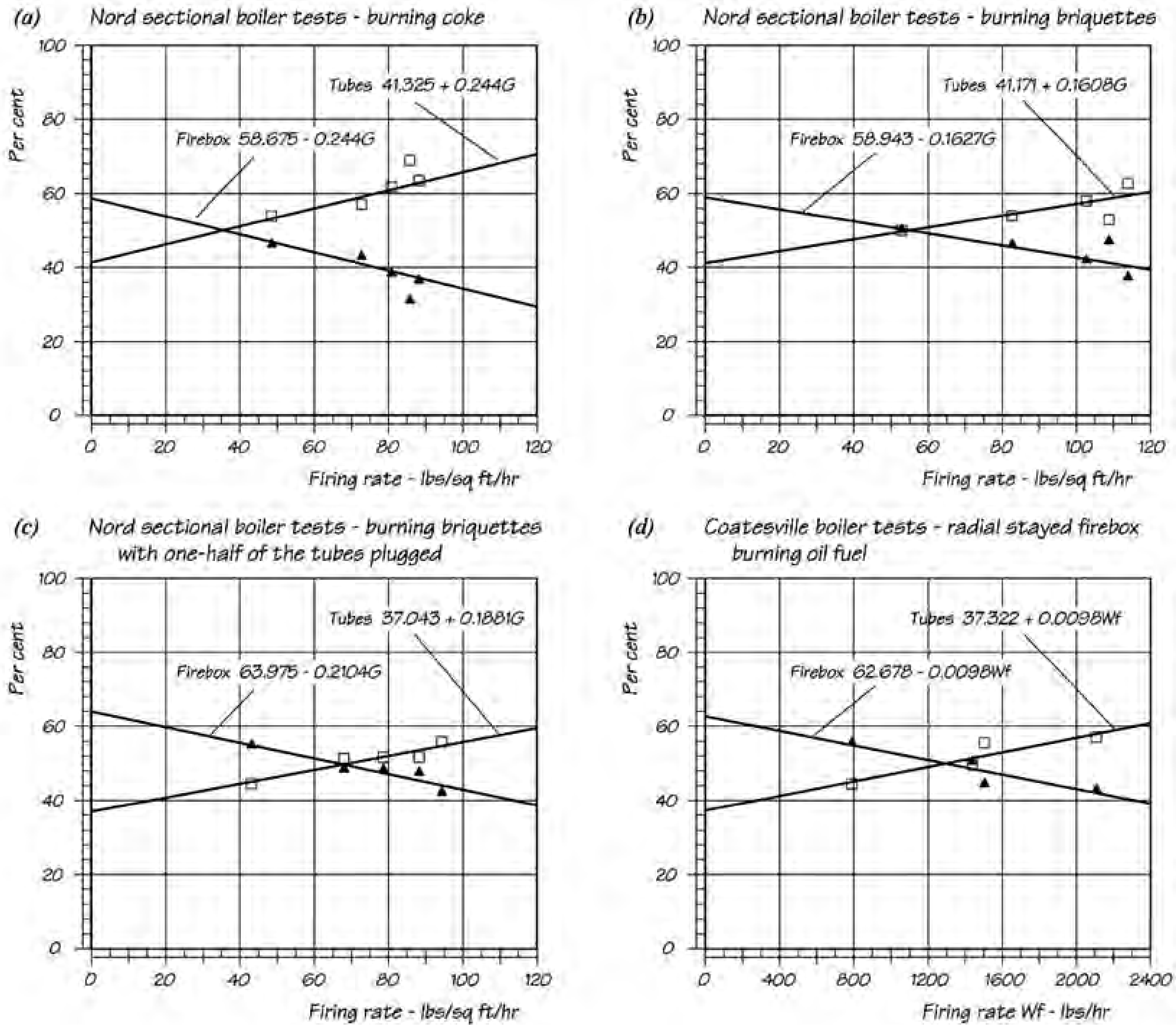
An earlier slide demonstrated that there is a significant drop in the temperature of the gases and flames as they negotiated the firebox. Perhaps it is only to be expected therefore that firebox temperature varied over the whole of its surface – differing both through its position and in its response to load.

These hot gases served to drive the heat through the firebox walls but in doing so increased the temperature of the wall above that of the saturation temperature. How hot the wall temperature became was a function of the intensity of the heat transfer, the waterside cleanliness and the conductivity of the wall metal.

As the wall temperature varied greatly with position, it was now possible to encounter localised heat transfer rates (heat fluxes) which were high enough to result in physical damage – to the stay and tube ends as well as the plates.

The above pair of diagrams record the firebox wall temperatures for a pair of runs made by a French 2-8-2 fitted with a copper firebox and especially provided with six thermocouples. Three of these, I, II and III, were positioned in effect high up out of the gas stream in the rear corner of the firebox. Thermocouple IV was located near the throat plate under the brick arch. V was positioned in the side wall just above the outer end of the brick arch. Finally, VI was in the tubeplate on the centreline of the boiler. Compared to the steel firebox tests conducted with a similar engine under nominally identical conditions, the wall temperatures were lower in the copper firebox seen here. Further, the differences in temperature extant between locations IV, V and VI were less pronounced, being a reflection of the superior thermal conductivity of copper.

In this example, the maximum temperature of 600oF was sufficiently low and of such short duration not to have initiated stay leakage.



Slide 10

The Nord and Coatesville tests involved boilers which had been specially modified so that the evaporation from the firebox could be measured directly and independently from that of the barrel.

In both boilers we see confirmation of our previous findings – namely, as the firing rate increases, the heat absorption distribution alters – a fact that the early pioneers such as Robert Stephenson were fully aware. Thus the total evaporation from the firebox, despite still increasing with rise in output, nevertheless assumed a smaller fraction of the total evaporation.

In effect, with rise in output, the heat is carried further into the boiler before it is absorbed. This has an impact on superheat performance. It can also have an impact on firebox performance. While the fraction of the total heat absorbed in the indirect surface increases, its area is many times larger than the firebox area that can be provided. Consequently, especially in large boilers endowed with large indirect surface – perhaps to overcome a poor draughting system[8] – may return a disappointingly low specific evaporation – i.e. the total evaporation divided by the total evaporative area.

If however an attempt is made to increase it, or as Wagner did to stipulate a universal figure for the specific evaporation that every boiler had to reach irrespective of its direct/indirect heating surface ratio, then it could result in an overloaded firebox and a maintenance headache – the complete opposite of what Wagner intended!

One or two of Ell's redraughted engines just fell into that category – e.g. the 4F 0-6-0.

Ell, it seems, followed the Germans in having the size of the desired upgraded output the extent of the total heating surface and thereby risked overloading the fireboxes in less favourably proportioned boilers.

Conversely, if after redraughting, the firebox loading remains unduly light then there is scope for further improvement.

A large direct surface relative to the indirect surface, increases the capacity of the boiler[9].

[8] The tubes along with being good at absorbing heat are equally effective at absorbing draught. Large boilers tend to be longer which resulted in higher frictional resistance, which coupled with shorter (less effective) chimneys meant outputs not commensurate with boiler size. Adding more tubes lowered tubular resistance (higher fga) so raising steam output from the same front end.

[9] There nevertheless has to be a certain amount of indirect surface for absorption efficiency and for superheating.

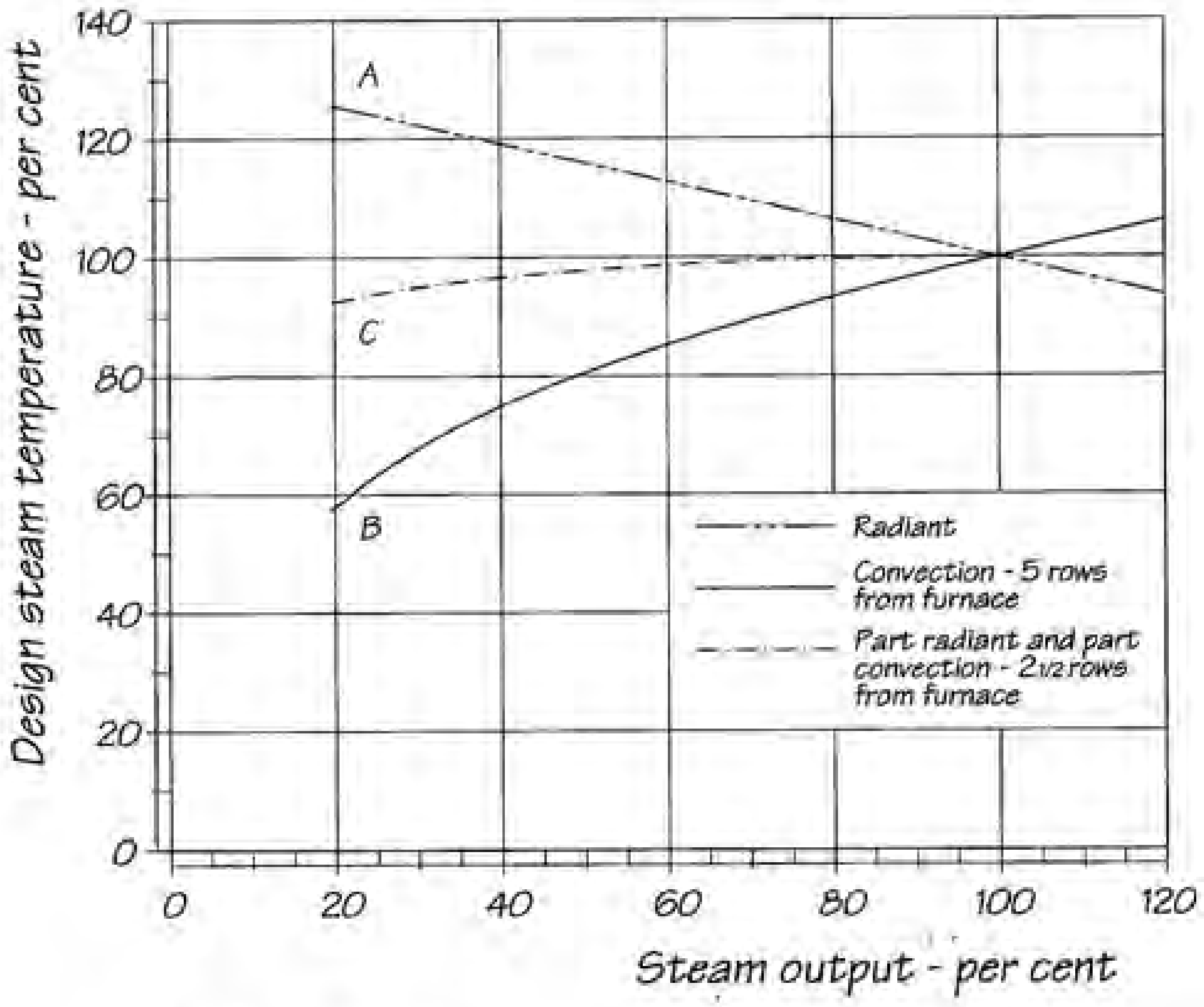


Fig. IX.4 - Superheater characteristics or the differences between radiant and convective superheaters

Slide 12

Superheaters fall into one or other of two fundamental forms depending on their locations in the boiler. Whilst this diagram has been derived from marine water-tube practice, the physical characteristics apply, making it relevant to a locomotive boiler.

We may see that due to the loss in firebox effectiveness with rise in output, a radiant superheater (A) has a falling characteristic delivering colder steam with rise in output[10].

A convective superheater such as (B) and applicable to a locomotive boiler has a rising characteristic. It does not keep rising with steam output however. Eventually it assumes an asymptotic value albeit usually at some impossibly high steam rate.

Curve (C) is the most interesting for it implies that if a degree of radiant heat can be introduced into a superheater, then while the maximum steam temperature will be reduced, its value at lower steam rates will be enhanced. Furthermore, it offers less variation in steam temperature in an otherwise uncontrolled superheater.

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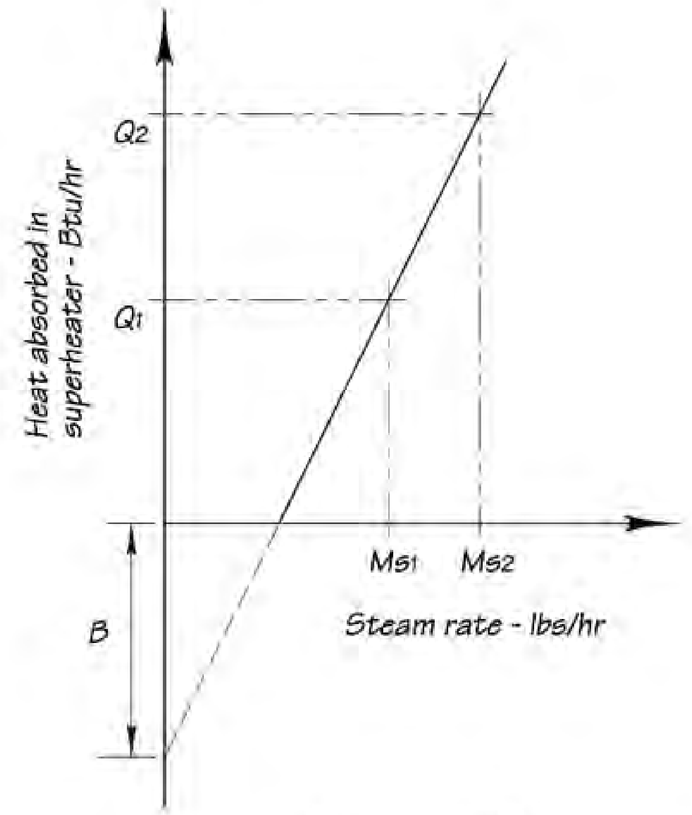
[10] The graph is from a marine boiler but the characteristics are universal. The boiler was intended to deliver a specified quantity of steam at a certain temperature - concepts seemingly largely alien to most loco engineers! There is a limit as to how hot the fire may become - vide slide 9 - and thus the heat that can be radiated into the steam results in a falling characteristic with rise in steam output.



	A	B	R^2
LMS Class 4 2-6-0	208	478,530	0.9998
BR Class 4 4-6-0	188.95	606,204	0.9999

Slide 13

If the amount of heat absorbed in a superheater is plotted against the steam rate, the result is a straight line, the gradient of the slope being an indication of the rate of gain in heat.

If this line is extended back, it will cross the X-axis at some intermediate steam rate thereby demonstrating that there is a minimum flow which must be passing through the engine before superheat appears.

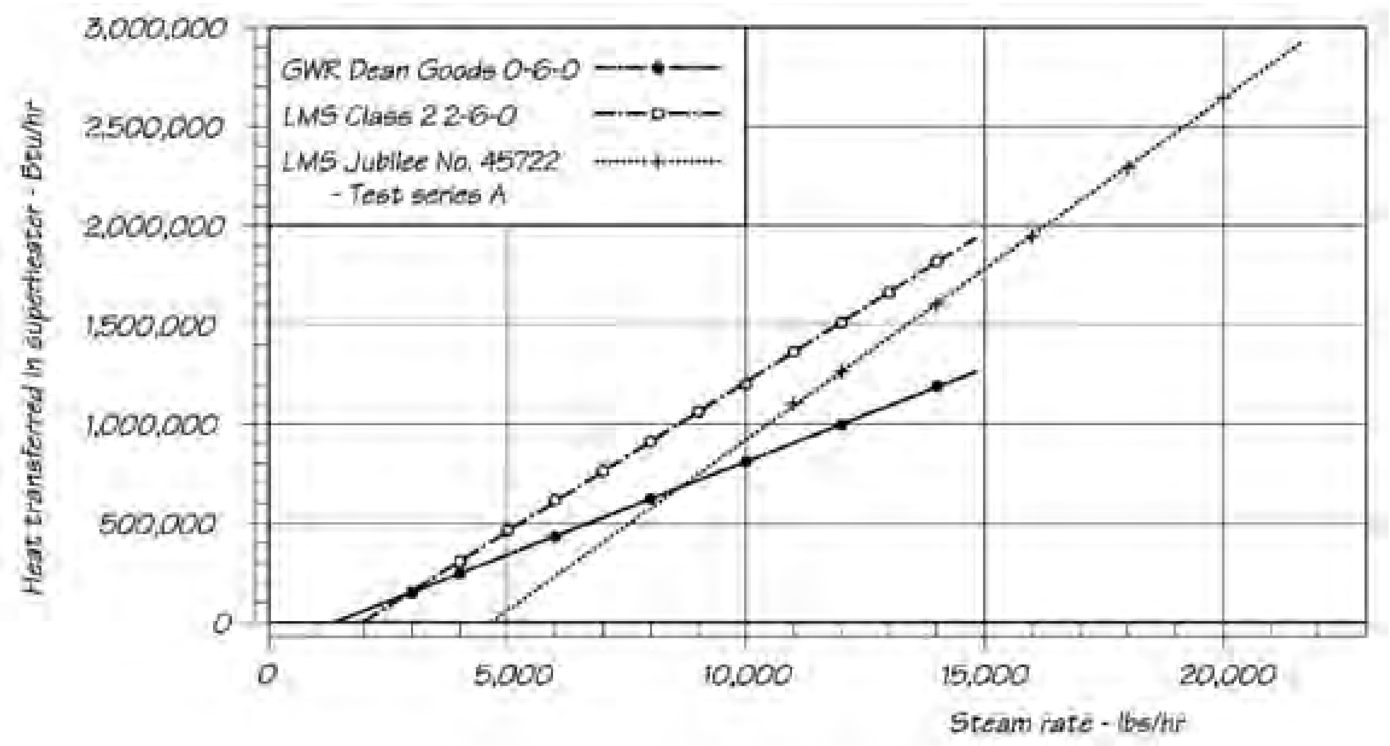
In order to establish the linear curve, a value has to be assumed for the steam quality (dryness fraction) of the steam entering the superheater. The actual value chosen has only a very small modifying effect but 0.98 – 0.99 would be normal.

Transcriber's Note: *Usually appearing in locomotive test data is a curve or table recording the steam temperature obtained at certain steam rates. By assuming a constant dryness fraction and boiler pressure we may estimate the amount of heat the steam absorbed during its passage through the elements. When a series of these heat absorptions is plotted against steam rate, they will be found to lie very close to a straight line. This linear curve is of the form:- Q = (A × Ms) – B. In this relationship A is the gradient of the curve obtained from:*

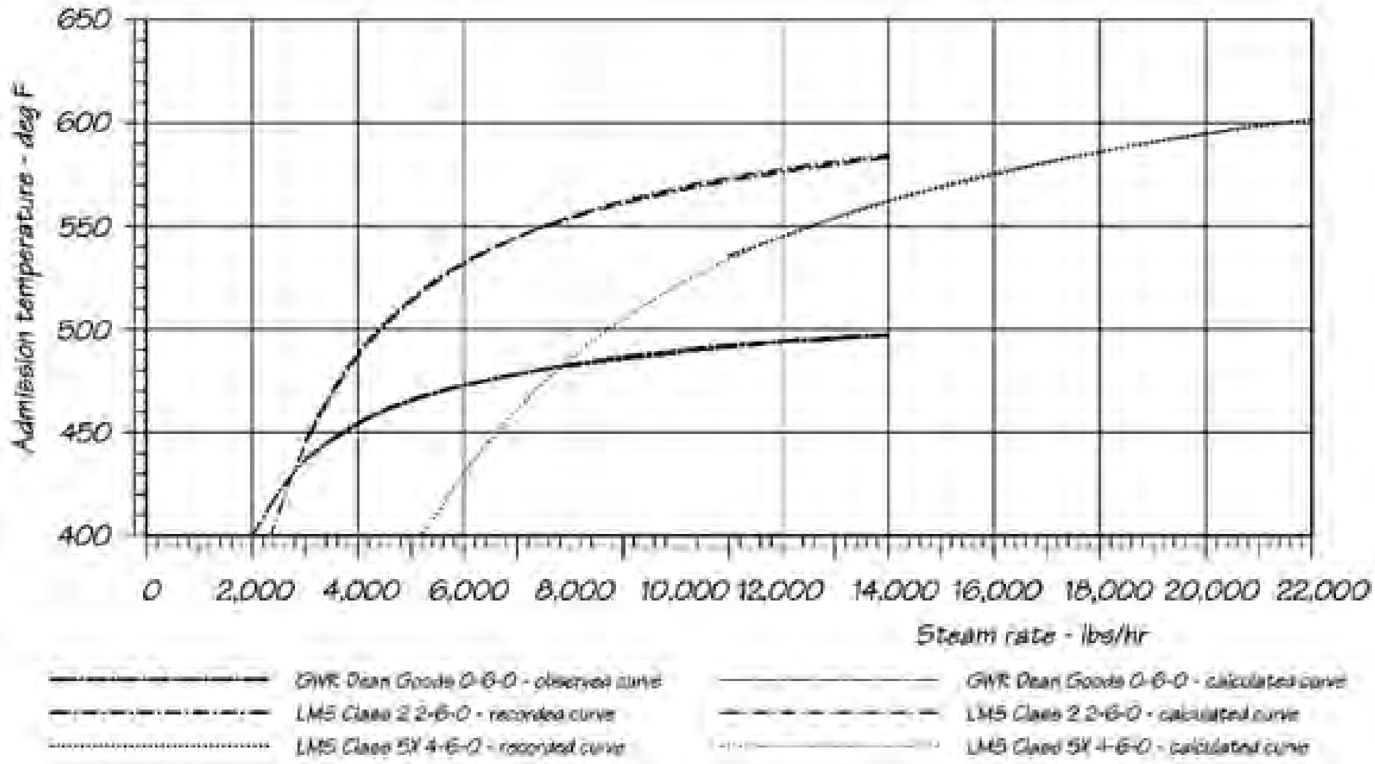
$$A = \frac{Q_2 - Q_1}{M_{S2} - M_{S1}}$$

It indicates the rate at which the superheater is absorbing heat per pound of steam passing through it. From this we may appreciate the larger the value assumed by constant A, the steeper the curve and therefore the hotter the steam for any selected steam rate. Conversely, constant B tells us when superheat will start to appear. This is as soon as Q reaches a zero value. Dividing B by A gives us the steam rate that has to be passing through the engine before any superheat starts to appear. This rate is influenced by several factors but the two most influential ones are the size of the firebox and the relative gas-side resistances of the flues and tubes. Large direct surfaces depress the firebox gas exit temperature while if the flues present a higher resistance than the tubes then the gas will favour the latter. The entering dryness fraction of the steam has often been held to be a cause of delayed or low superheat but test plant evidence, supported by calculations confirm the water content of the steam was consistently very low in well managed boilers.

[R is the correlation - statistical relationship confirming the 'accuracy' of the assumed relationship.]



Rate of heat absorption in three superheater designs assuming a constant saturated steam inlet condition of 0.98 dryness



Comparison between the recorded steam temperature and its calculated value for the same three engines based on a constant saturated steam inlet condition (0.98 dry)

Slide 14 (Previous page)

The upper diagram illustrates the rate of heat gain in a superheater fitted to three different locomotives - Dean Goods 0-6-0, LMS Class 2 2-6-0 and LMS Jubilee 4-6-0.

In the lower diagram, following the application of a little mathematics, the predicted superheater characteristic (thin line) is compared with the Bulletin observed steam temperature curve (thick line). Once again, we see there is a minimum steam flow required before superheat appears, its value being affected by the design of the superheater, its relative size, and also by how much gas can be encouraged to go through the flues (rather than the tubes).

Despite the Dean Goods having only a small superheater, it delivered superior superheater performance up to nearly 3000 lb/hr which, while not sounding much, is nearly 20% of the 2-6-0's maximum steam output. In the case of the Jubilee, superheat does not appear until around 5,200 lb/hr, which was 25% of the original steam output of 20,760 lb/hr.

The secret of obtaining a good superheat performance is to match it to the duties of the engine. In most cases, on preserved lines at least, it seems desirable for enhanced superheat to be obtained at low steam outputs. If this is the case, then rather than redesigning the tubular surface completely, one approach would be to increase the A/S ratio of the flues by modifying the elements to reduce their resistance relative to that of the tubes.

Adopting smaller diameter elements would have negligible impact on the steam pressure drops between the header and the cylinders (chances are the regulator will be only partially open!) while improving the steam distribution through the superheater[11]. Half-return elements[12] have their best heat collecting surfaces located where the gases are hottest while again presenting a lower gas-side resistance than the full-return type. There may even be some scope for judicious repositioning of the return bends closer to the firebox tubeplate[13].

Such action would encourage an enhanced gas flow over the elements especially at low steam outputs to exploit as far as possible the 'radiant' effect[14] even though it will result in a lower steam temperature at the highest outputs.

Finally, although not explored here, there will be considerable scope for locomotives used on heritage lines subject to the Light Railway Order, to plug a large proportion of the small tubes, again to encourage more gas to pass through the flues. Conventional draughting systems have significant capacity at low steam rates as is demonstrated by the high excess air values – vide BR Test Bulletin curves.

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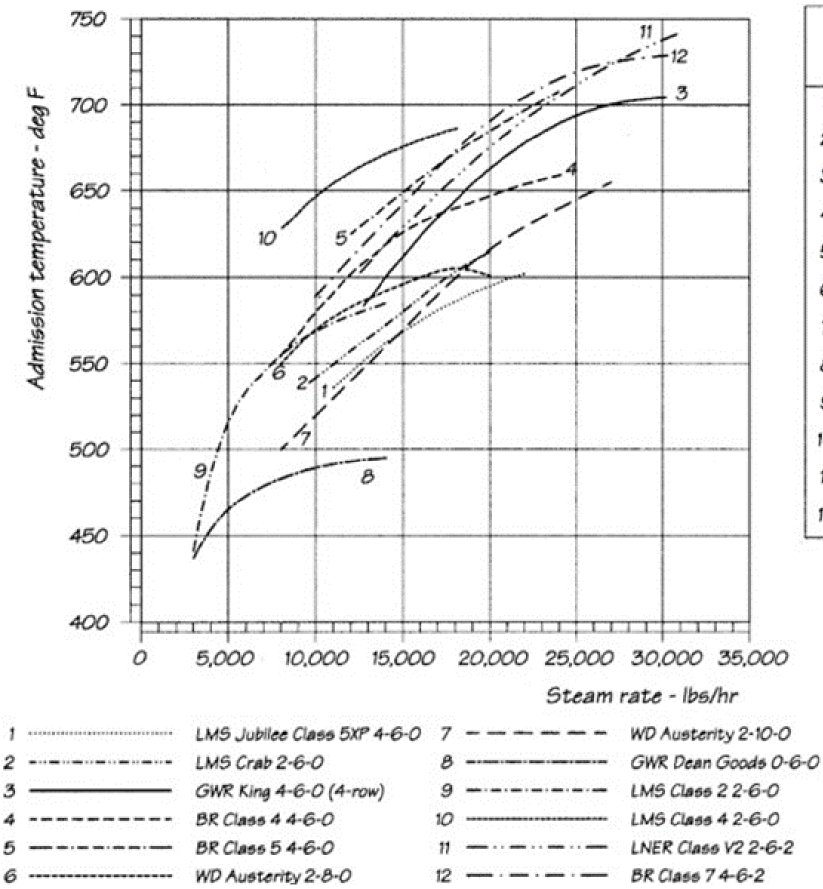
[11] The elements present a restriction to steam flow whose size varies with flow rate. At low flows this lack of resistance might result in some elements carrying negligible steam giving high metal temperatures. Smaller diameter elements encourage the steam flow to become more equitable.

[12] Half-return elements have four steam passes in the hot end with two passes in the front half of the flue. Conventional full return flues have four steam passes the full length of the flue.

[13] Lengthening the element potentially increases resistance but a gain in superheat might reduce the steam flow while obtaining the same power as formerly. On preserved lines this tends to be academic as the regulator is throttled!

[14] Certain gases can emit radiation even at 'black' heat.



	Locomotive Class	S/A Ratio - tubes	S/A ratio - flues
1	LMS Jubilee Class 5XP 4-6-0	392	378
2	LMS Crab 2-6-0		
3	GWR King 4-6-0 (4-row)		
4	BR Class 4 4-6-0	405	368
5	BR Class 5 4-6-0	392	383
6	WD Austerity 2-8-0		
7	WD Austerity 2-10-0		
8	GWR Dean Goods 0-6-0		
9	LMS Class 2 2-6-0	374	303
10	LMS Class 4 2-6-0	374	301
11	LNER V2 2-6-2	417	439
12	BR Class 7 4-6-2	435	420

NB:- Curves largely taken from diagram appearing the paper on the British Standard classes presented by E S Cox to the I Loco E. It is known that in some instances the curves he presented differed from the equivalent ones appearing in the relevant Bulletin.

Fig. IX.2 - Admission temperatures and S/A ratios for a selection of British locomotives

Slide 15

Curve 10 should therefore be of interest because it is a complete outlier, possessing high superheat at low steam rates, a situation created by the use of reduced diameter small tubes and a superheater having an A/S ratio very far from 1/400.