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The ASTT/ASTS were very lucky to be able to hold the 2020 AGM at Crewe Heritage Centre on Saturday 2 March when the COVID-19 restrictions we are experiencing today had yet to be brought in. The virus did however lead to elbow bumping in place of the normal greetings and one speaker, very understandably, choosing not to make the journey to Crewe. The day started with a meeting of the committee followed by the AGM and the afternoon spent being updated on and discussing the progress of the REVOLUTION Project. A tasty buffet lunch had been brought to the meeting by our Chairman who had ordered it from his local village tea room.

One of the pleasures of attending the AGM and the Autumn Conference is the ability to discuss advanced steam matters with the other members and I know that for some this is the "best bit" of the meetings. For those members who are unable to attend and for those wanting to continue the discussions we do have have a couple of methods. The first is writing a letter or article for publication in the Newsletter, these have been noticeable by their absence up to now. The second and more immediate way is to post on the forum which can be found on the website. I know that several people who have written articles to provoke discussion have been very disappointed not to receive any comments about their work. The ASTT should be a vehicle for the advancement of steam traction and as with any other scientific endeavours there needs to be ideas put forward that are discussed, in a respectful way, by other members.

At the meeting it was decided that during the year I will take on the Secretary's duties and hand over the Editors duties, so we are looking for a new Editor, to take over progressively before I fully take on the Secretary's role. Thank you to all those who have supported the Newsletter.

The Press date for the next issue is 3rd July 2020.

Chairman's Piece – John Hind

How the world has changed since the last newsletter. Those changes have rippled down into small organisations like ourselves and have affected the 2020 Conference, Revolution project and BioCoal trials. Updates follow in the newsletter.

In a way, it was fortunate that we held the AGM on the first weekend of March before the true implications of Covid 19 became apparent and the lockdown put in place. We did not know anything about social distancing though we did greet each other with elbow bumps rather than handshakes! I hope everyone who came along did not catch Covid 19 as a result of coming along.

Grant Soden and Alex Powell have joined the committee as ex-officio members, with Grant taking over running the website from Chris Newman and Alex taking on the Young Members/Student brief.

Grant lives in Farnborough and works for South West Western Railway in Rolling Stock Maintenance Management and is a member of South Western Railway team taking part in the Institution of Mechanical Engineers Railway Challenge (which is held at the Stapleford Miniature Railway). He is a member of Young Rail Professionals. He has been a member of the Great Cockrow Railway (7 ¼") since 2003 and is their Deputy Chief Mechanical Engineer involved in overhauls and building new locomotives.

Alex is in his second year at Newcastle University studying mechanical engineering and says;-

I have been a member of the AST since 2016. Since 2008 I have been a member of the Malden and District Society of Model Engineers (MDSME) in Thames Ditton, London. At MDSME I am a regular driver of one of the 5 inch gauge locomotives and am currently working on building a 3.5 inch tank engine and a 5 inch gauge A4. More recently I have joined the Merchant Navy Locomotive Preservation Society (MNLPS) the group that looks after Clan line. Just before the lockdown I was about to take the PTS course to enable me to join the support crew. Outside steam locos and engineering I am a keen rower, cyclist and cricket fan.

Cedric Lodge is stepping down as Secretary at the end of this year and David Nicholson will be taking over from Cedric and shadowing him this year before taking over. In turn that leaves a vacancy for the Editor of our newsletter. If anyone would like to take this on please contact me on john.hind@advanced-steam.org .

Cedric has been a great supporter of ASTT over the years as both Newsletter Editor and Secretary and on behalf of us all I'd like to thank him for his contribution and look forward to seeing him at future events.

Finally, I hope everyone comes through this and we see you at an ASTT event. It will be good to shake hands enjoy a coffee and generally re-engage those personal relationships we so often take for granted.

Conference 2020 – John Hind

We have put the 2020 conference on hold and have cancelled arrangements that we were making with venues, visits and speakers.

At the end of June, we will decide whether to hold the conference in 2020 or postpone till 2021. Most bookings are in the last 3 months before the conference but there remains the problem of booking venues, visits and speakers, which we like to have confirmed by April.

Come the end of June we will see whether we are still in lockdown and evaluate what is possible (or before if the lockdown finishes sooner). The target remains to hold a conference during the weekend of 3rd/4th October.

Revolution Update

Grant Soden has taken on board the design of the Tender.

Richard Coleby, Jamie Keyte and Mike Horne gave presentations at the AGM. These took up the afternoon. Richard's and Jamie's focussed on technical issues and generated several questions. Mike's posed some questions about organisation, ownership and fundraising, which are issues that are as important as the technical issues. Copies of the presentations are on the website.

We were planning to carry out Leakage Tests on one of the engines at the Stapleford Miniature Railway. These have been deferred for the foreseeable future. As soon as the Covid 19 situation clears, we will be looking to get this underway – they are key to helping us understand steam usage on miniature locomotives. There is information on standard gauge locomotives but nothing for miniature locomotives, so we will be pioneering and learning something new.

Jamie Keyte, Alex Powell, Chris Newman and myself have been experimenting with internet communications and we've had some success using Zoom for video conferencing and reviewing CAD data in real-time. Similar tools are in use industry, but the ones that we are using are free, with some restrictions on use that do not affect how we use them. A first-time 21st Century application use of these types of tools for steam engine design?

Biocoal – John Hind

In the January newsletter we were awaiting word from the HRA on the list of railways interested in carrying out trials.

Following the HRA Spring Meeting, we were given a list of railways that are interested in trying biocoal – North Yorkshire Moors, Bodmin and Wenford, Middleton Railway, Isle of Wight Steam Railway, Romney Hythe and Dymchurch and the Perrygrove. We are now waiting for names of contacts on the railways, so that we can start a dialogue with the railways about test plans, etc.

There is a degree of healthy scepticism, however, if the Heritage Movement is to survive, it does need to demonstrate that it is aware of the climate change issue and is not ignoring it. There will be the inevitable - 'why don't you use bio coal?', however, the results from coordinated trials will give the appropriate answers.

However, given the national lockdown and the effect on Heritage Railways, biocoal testing is not the top of their priorities, so I am not expecting much progress this year.

Our 5th. AGM -- Cedric Lodge.

Following the pattern of last year, Crewe Heritage Centre was again the venue for our 5th. AGM. The date was Sat. 7 March. By this time, the effects of the corona virus were starting to impact on meetings such as ours.

Twenty members attended. The formal business went smoothly with no hitches.

Thirty-four members had tendered apologies for absence, the most significant of whom was Joe Cliffe. Joe was scheduled to give a presentation after the formal proceedings, but having to cross London to get to Crewe was deemed too much of a risk in the in the present circumstances. Members sympathised with Joe. The Chairman informed the meeting that as an alternative, there would be presentations and a discussion on the Revolution Project.

The Chairman delivered the report of the Committee, highlighted with Powerpoint illustrations.

Of particular significance was the appointment of Alex Powell and Grant Soden as ex-officio members of the Committee. This was intended to enlarge the Committee and an opportunity to include younger Members.

Paul Hibberd went through the Accounts, and informed Members that our financial position had improved by £3,000 due, in the main, to book sales.

The Accounts confirmed a satisfactory financial position, and were approved.

Our main sources of income are subscriptions and the book sales, handled very capably by Chris Newman.

Paul recommended the development of other income streams in the future.

Our Auditor-Richard Coleby, complimented Chris for his book keeping, which had made his job easy. However, he did inform the Meeting that he wished to step down from auditing our accounts, but was willing to serve for another year. We are now actively seeking a new Auditor to take over next year.

This year, it was the turn of our Chairman-John Hind, and Mike Horne to retire by rotation. David Nicholson retired as a co-opted member.

In deference to good practice, the Election was conducted by the Secretary-Cedric Lodge.

Cedric informed the meeting that John and Mike who retired by rotation, offered themselves for re-election. At this stage in the meeting, David Nicholson was not offering himself for election, as it was the intention of the Committee to co-opt him after the AGM. As there were no other nominations, he announced that John Hind and Mike Horne were re-elected to the Committee.

Subscription Rates is an obligatory item on the agenda of the AGM. Members resolved to keep the Rates unchanged for another year. The Chairman invited those with Associate Membership to upgrade to Full Membership.

It was resolved that the next meeting be held on the Saturday of the first weekend in March next year. The Chairman informed Members that the cost of the Crewe venue was £130, which he considered good value; it was commodious and convenient. The meeting approved using the same venue next year.

Before closing the meeting, the Chairman informed Members that, subject to the lifting of Covid 19 travels restrictions, our Conference would be held during the first weekend of October, in Darlington. It was hoped arrangements would be made for us to visit Hopetown Works to see the P2 under construction.

After the meeting, a delightful buffet lunch was served, curtesy of an excellent caterer known to our Chairman.

In the afternoon, there were presentations by Richard Coleby, Jamie Keyte and Mike Horne on the Revolution Project. Some concern was expressed about how the Project would be publicised and funded. John informed Members that another meeting of the Committee would be arranged May/June, to concentrate more fully on these aspects of the Project.

On leaving the meeting room, Members were able to inspect a 15 in. gauge Crab loco under restoration in a secluded workshop.

At risk of leading Members into temptation, it is worth mentioning the shop of the Crewe Heritage Centre. This is well stocked with second hand books, and a large collection of British Rail manuals.

Membership – Chris Newman

1. Upgrades to Full Membership

The following five Associate Members have applied for, and been granted, Full Membership:

- **Alex Powell**, who has been a member since 2016, since when he has been a regular attendee at AST's AGMs and conferences. He is currently involved in the preliminary design of the boiler for the Revolution project.
- **Hendrik Kaptein**, who has been a member since 2017, since when he has been a regular attendee at AST's conferences, giving a paper at the 2019 conference.

- **Vyvyan Vickers**, who has been a member since 2017, since when he has been a regular attendee at AST's AGMs and conferences.
- **Les Turner**, who has been a member since 2017, since when he has been a regular attendee at AST's AGMs and conferences.

Hugh Odom, who has been a member since 2017. Hugh lives in the US so is unable to attend AST events, however he has a long-standing involvement with modern steam, acting as webmaster for the renown Ultimate Steam Page. Hugh also contributed to Chris Newman's 2008 report on the Indonesian Coal Railway project.

Associate Members who are interested in taking a more active role in the Trust are encouraged to apply for Full Membership by notifying the Membership Secretary at memsec@advanced-steam.org.

The main advantage of upgrading is that Full Members are entitled to a vote at general meetings, giving them the opportunity to participate in decision-making.

2. Membership Numbers

No new members have joined since January. Two past members have withdrawn their membership, and four have failed to renew such that their membership must now be considered "lapsed".

This brings the current total to 78, summarised as follows:

Full Members:	18	UK members:	57
Associate Members:	54	EU:	12
Student Members:	6	USA	3 (as before)
		Australasia:	5
Total Membership:	78	China:	1 (as before)

Book Sales – Chris Newman

Book sales since January have been disappointing. They are listed as follows:

- **Volume 1 of Porta's Papers – Tribology and Lubrication** (pub. ASTT): 9 sold including 7 to Camden Miniature Steam (total sales 92);
- **Volume 2 of Porta's Papers – Adhesion, Compounding and the Tornado Proposal** (pub. ASTT): 6 sold including 5 to Camden Miniature Steam (total sales 88);
- **Steam Locomotive Design Specifications and Calculations for New Build Baldwyn 2-4-2T 'Lyn'** (pub. ASTT): 2 sold including to Camden Miniature Steam (total sales 69);
- **Defence of the MR/LMS Class 4 0-6-0 by Adrian Tester** (pub. Crimson Lake): 1 sold (total sales 18);

Members are reminded that they can claim a 20% discount on titles published by ASTT, and 10% on other titles.

New Titles -- Chris Newman

Work is continuing with transcribing Porta's manuscripts for future publication, however the publication of a third volume of papers may be delayed whilst consideration is given to which papers to include in it. Since the content of Volume 3 was first promulgated some time ago, several additional papers have come into ASTT's hands, several being kindly donated by Martyn Bane, while a number of others were discovered by Chris Newman in an uncatalogued archive at the NRM in York. As a consequence, whereas Volume 3 was originally planned to contain the (then) 11 remaining transcriptions, we now have some 42 papers to choose from which would fill another three volumes if they could all be transcribed. In fact, 17 of these have already been transcribed (11 by Chris and 6 by others), leaving 13 handwritten manuscripts (497 pages), and 6 typescripts (271 pages).

The 13 handwritten manuscripts that have yet to be transcribed are as follows:

Title	Year	Pages
Steam Locomotive Power Advances made during the Last 30 Years. The Future	1991	182
Application Of The Gas Producer Combustion System to the 141R - An Exercise	1998	48
Locomotive Type Boiler for Bagasse Peat and Wood Refuse Burning	1992	26
A preliminary scheme for modernizing ex-Baldwin 2-6-2T locos in Australia	1995	25
Note on combustion efficiency of the Gas Producer Combustion System	1980	7
GPCS as an answer to coal-derived pollution from steam locomotives	1990	40
Note On burnout heat transfer	1980	5
A note on the Gas Producer Combustion System with fluidized bed conditions	1980	18
Notes on Responsiveness to Quick Load Changes of a Boiler Burning Wood	1978	40
Proposed Feedwater Heater for 3450	1980	15
An essay on steam locomotive boiler tubeplate birdnesting (Ash fouling)	1983	25
Note on the present status of the grate design in connection with the GPCS	1993	33
Some Aspects of the LVM800 Locomotive Design	1998	33

The 6 typewritten manuscripts that have yet to be transcribed are as follows:

Title	Year	Pages
Third Generation Steam - Facing the Energy Crisis	1978	72
Fundamental Principles of Steam Locomotive - Barcelona 1998	1988	9
21st Century Steam - Barcelona 1998	1998	12
Surface Feedwater Heater for 3450 1980 by LDP	?	9
On the Possibility of Manufacturing the 800HP LVM 800 Steam Locos in Cuba	1998	39
A List of Some 500 Possible Improvements to Existing Steam Loco Power – ACE	1983	121

Transcribing of typescripts can usually be accomplished relatively quickly with the assistance of OCR (optical character recognition) software, while transcribing handwritten manuscripts is much more laborious. However, it is possible that the use of "Speech to Text" software could speed up the process. (Initial attempts are being made to do this.)

Should any members be willing to assist in the transcription task, can they please contact Chris Newman at info@advanced-steam.org.

Transcribed Papers:

The following 17 papers have been transcribed, however additional work is required to proof-read some of them. In addition, some of them need to be reformatted to bring them into conformity with ASTT's "house style".

Title	Year	Pages
Lempor Theory	?	16
Steam Locomotive: Running with Closed Regulator	1977	5
Improvements to Superheater Element Joints for Advanced Steam Locos	1982	11
A New Superheater-Economizer Element For Advanced Steam Loco Technology	1980	8
Locomotive Boiler Combustion Calculations – The Heat Balance - A Criticism of the Lawson-Fry Method	1974	15
The Thermo-mechanical behaviour of the Steam Locomotive Boiler Firebox - An Overall View	1984	23
Notes on the Flat Plate Stayed Firebox Construction for Locomotive Boilers working at 30 and 60 ate. Steam Pressure	1977	16
On the Hudson-Orrok Heat Transfer Eqn as applied to Locomotive Boiler	1977	4
The Thermo-Mechanical Behaviour of the Steam Loco Boiler Firebox	1984	24
Boiler Water Circulation	1981	26
What Steam Pressure for Old Locomotive Boilers (published by CSR)	1999	*10*
Hand Firing in connection with The Gas Producer Combustion System	1976	30
The Fischer Knuckle Pin in Advanced Steam Locomotive Engineering	1986	6
Cario - An Advanced Axlebox Scheme for XXIst Century Steam Locomotives	2000	*9*
The Case of a Better American Steam Locomotive (published by CSR)	?	13
The Contribution of a New Steam Motive Power to an Oilless World	1987	26
Steam Locomotive Power for the 21st Century	2001	9

Porta's Water Treatment paper has been omitted from the above lists. This is because consideration is being given to combining it with other papers on the same topic to create a separate volume dedicated to the subject.

In the same way, consideration is being given to grouping other papers into separate subject matters, so that they can be published together in "themed" volumes. For instance, volumes might focus on "boilers and superheaters" and/or "combustion and fireboxes", and/or "modern steam – general". However, whether this will be practicable or not will depend on how quickly the remaining manuscripts can be transcribed, because there is an urgent need for a new volume to be published in order to boost flagging sales. Hence members' participation in transcribing, or proof-reading, texts - or both - will be most welcome. Such work is an excellent way to fill these days of enforced isolation!

Website – John Hind & Grant Soden

Chris Newman is grateful to Grant Soden who has taken over the role of webmaster. Grant will report here on current website activity.

ASTT Website Usage – John Hind with information from Grant Soden

Just a few statistics on how often our website advanced-steam.org is viewed and what are the most popular pages.

During March there were 3586 views with a daily average of 116. At the time of writing (just halfway through April) we already have 2289 views and the daily average is up to 134 views, with the highest of 266 views on April the 6th – perhaps a reflection of the lockdown? Our best daily figure was 538 on April 11th, 2018.

Top 5 most viewed pages 18 March 2020 -16 April 2020 were: -

Page Title	Page Views
Mollier Diagrams	386
Equivalent Evaporation	361
Specific Steam Consumption	232
What is the difference between "Gas Oil" and Diesel?	78
The Red Devil	59

Top 5 most viewed pages of all time are: -

Page Title	Page Views
Equivalent Evaporation	9644
Specific Steam Consumption	7193
What is the difference between "Gas Oil" and Diesel?	5835
Mollier Diagrams	3326
The Red Devil	1835

The Red Devil continues to attract interest, but not as much as interest in Equivalent Evaporation, Mollier Diagrams, Specific Steam Consumption and the differences between Gas Oil and Diesel – perhaps a reflection that our viewers are in interested in the more technical points of steam locomotives.

Articles from Stephen and Martin

In this issue, we have the first in a series of articles by our member Stephen Whithead. Stephen would like his articles to prompt debate amongst the membership and would be happy to see your thoughts about his suggestions, either agreeing or disagreeing, via the Forum on the ASTT Website.

We also have the second part of the article on boiler design from Martin Johnson in which he also invites debate on the subject on the ASTT Website forum, the link to which is at the end of his article.

Basic Design Proposal for a Compounded Locomotive

Basic Design Proposal for a Compounded Locomotive with Re-Superheat for the Low Pressure Pistons

My proposal for a compounded 2-8-0, 3 Cylinder Express Steam Locomotive number 38001, with British Caprotti Valve Gear Water Tube Firebox running on Bio Gas-Oil or Hemp Oil that might fit in the British rail system.

A 2-8-0 locomotive numbered 38001, inspired by Urie & Maunsell and their locomotives, the King Arthur and Lord Nelson classes of the Southern Railway, also the W.M. Smith system used on the Midland Compound; along with Chapelon, L. V. Porta, 5AT and others; with all steel welded boiler and all steel welded water tube firebox; looking like a Maunsell engine with that type of smoke deflector, with almost completely enclosed slightly longer cab, with long sliding side windows, wedge shaped front windows to improve drivers view and side sheets folded in behind the drivers seat on the left, and also on the right to reduce draft.

The idea of this engine is to run a 10 coach train at 75 miles an hour on the flat and up most inclines without loss of speed and with as much economy of fuel and water as possible. Maximum speed to be 90MPH. The insulation should be such that the engine left during the afternoon will have sufficient pressure left in the boiler the next morning to be conveniently fired up for continued usage and an absolutely minimal day to day maintenance.

Wheel arrangement to be 2-8-0 with 6ft drivers with lateral motion device on the first pair of drivers and bissel bogie with 3ft diameter wheels and lateral control geared rollers, wheel base 27ft 8in; Distance between bogie wheel and first driver 8ft 11ins, then 6ft 3in to the second, third and fourth drivers.

The axle box's to have 'Franklin' spring loaded wedges to take up wear and play, the driven axles and crank axle to have spherical (self aligning) roller bearings, the welded on horn guides to have pedestal braces. The wheels of 6ft diameter of the Bulleid-Firth-Brown type, the driving wheels to have clasp brakes. The connecting rods of lightweight alloy and profile from the rear (4^{th}) pair of wheels to have tapered roller-bearings to control the uprightness of the connecting rod to the third pair which should be spherical roller bearings, spherical from the third pair of wheels to tapered roller bearings for the driven (2^{nd}) pair of wheels, tapered roller bearings from the driven to first pair of wheels with spherical roller bearings. The axle loadings to be 19.5 ton's each, with equalised springing, giving an adhesive weight of 78 tons and a total weight of 87.75 tons.

The three pistons to drive the second wheel axle with the cranks set at 120°, the coupled wheels to be balanced without the driving rods from the pistons to give zero hammer blow as per Holcroft,

British Caprotti Valve Gear to be used with the centre valve gear driven by Cardan shaft from an adjacent outside valve gear, the steam supply for lifting the poppet valves to be controlled by a valve operated by the driver before opening the regulator. The inlet port to be 25% of the piston area and the outlet port to be 35% of piston area; for 24 inch piston
$\pi r^2 = \pi 12^2 = 452.4 \div 4 = 113$ sq in for the inlet port and $452.4 \div 2.86 = 148.89$ sq in for the $r^2 =$ $\pi r^2 = \pi 12^2 = 452.4 \div 4 = 113$ sq in for the inlet port and $452.4 \div 2.86 = 148.89$ sq in for the 12^2 $= 452.4 \div 4 = 113$ sq in for the inlet port and $452.4 \div 2.86 = 148.89$ sq in for the outlet port, but these could be made bigger if possible and realistic.

The pistons with tail rod to be 28 inch stroke, HP 17 inch bore, LP 19 inch bore; to have multiple narrow piston rings and multi ring glands for the piston rod for steam tightness. The driving rods of machined lightweight alloy and profile to have taper roller bearings at the outside cross-head and spherical roller bearings at the second pair of driving wheels. The inside driving rod to have needle roller bearing with nitrided steel for the crank journal and the big end housing in two halves with the joint vertical, spherical roller bearings at the cross head.

Tractive Effort;- (high pressure cylinder)$17^2 \times 28 \times 150 \times 85 \div (72 \times 100 \times 2) = 7164.79$ +(low pressure cylinders) $19^2 \times 28 \times 150 \times 85 \div (72 \times 100) = 17899.58 + 8032.5 = 25{,}932$TE

In the compound system developed by W.M. Smith and built by Johnson on the Midland Railway, the tractive effort at starting was aided by feeding steam direct from the boiler to the low pressure cylinders as well as the high pressure cylinder, the tractive effort will be the same as above.

Indicated Horsepower $(25932 \times 75) \div 375 = 5186.4$ divide by 2 = 2,593IHP at 75MPH therefore at least 2 thousand (DBH) horsepower should be available at the drawbar.

With ideal cams and events for the valve gear plus large air-smoothed steam pipes the continuous DBH could be higher.

Direct pumped lubrication with steam oil containing 0.2% colloidal graphite (for friction and wear reduction) to the pistons in the centre of the cylinder timed to feed into the piston rings at the centre of the stroke every 40 strokes or so. The fabricated stainless steel (for its low thermal conductivity) cylinders and end caps to be insulated and the liners to be pearlitic chromium cast iron and have multi ring glands to the piston rods for the least amount of steam leakage possible and minimal clearance volume.

The pistons to be light weight of steel with 7 narrow rings, five of cast iron one grade harder than the piston liners, the second and sixth rings to be of bronze, the bronzing of the liner reduces friction. The outlets from the cylinder drain cocks should be routed to the front centre to avoid depositing oil on the rails.

The boiler to be All welded steel round topped boiler to be used tapering up at the top from 5ft 4in at the smoke-box to 6ft at the fire-box, the barrel to be 12ft long, an economiser plate one fifth of barrel length away from the smoke-box tube plate to isolate the feed water; three safety valves to be mounted radially in front of the fire-box and the dome for steam collection in front of the valves. The boiler pressure to be 310lb per sq in to run at 300lb per sq in to avoid the safety valves blowing off. There are to be 70 or so large tubes (flue's) for the super heater (no small tubes) and two much larger tubes for the resuperheat. The boiler smoke-box and fire-box to be surrounded with the best type of insulation. Use automatic pumped water feed with Porta feed water treatment.

The all welded steel fire-box to be of the water tube type. The length of the combustion (grate) area to be 10ft 6in the grate to be 30sq. ft. The round top fire-box tapering down from the barrel to the cab, and the sides tapering in by 6 to 8 inches to the cab, this will give better forward view.

Feed-water heated by two closed circuit surface type feed water heaters, one fed from the front and one from the rear exhaust outlet and placed each side of the barrel just behind the chimney, being inside the hot smoke-box the pipes will not need to be insulated, as they will feed up directly into the feed water heaters above, which will be placed directly onto the barrel and the insulation placed over the top of the feed water heaters and the barrel of the boiler. The feed water pump to be on the right hand side above the running plate, also on the right hand side, the two clack valves below the boiler waterline feeding into the economiser, the second clack valve for a standard live steam injector.

Type E super-heaters to be used, each bifurcated element occupying two flue's (or four?); a 12 inch diameter pipe from the centre of the super-heater chest feeding the steam chest for the high pressure piston. The steam should be heated to 450°C. The three regulator valves, small – larger – big, to the same design as the Caprotti poppet valves, to be after the super-heater, these valves to be steam operated (held open and held shut by reversing the steam pressure from bottom plunger to top plunger) from the three position regulator, their resting (no steam) position being closed, there should also be a fourth valve of the same type to admit steam to the low pressure pistons for starting.

The steam from the high pressure piston to the low pressure pistons should be re-superheated (Porta used two very large flue tubes to contain the paralleled multi-folded resuperheat tubes, which were of larger diameter than the high pressure ones, one set for each of the low pressure cylinders).

The smoke-box to be sufficiently long, around 6 ft, to allow the blast pipes to be positioned low down in front of the inside piston. Use double (or single?) Lempor blast pipe, each with coaxial Kordina under to isolate the exhausts from their respective cylinders, with a take off between the Kordina and the Lempor blast pipe of each for the two surface feed water heaters. The blast pipes should be to one side of the tail rod of the offset centre piston and at the lowest possible point.

The tender to have the rear part of the cab built into it and be of the double bogie water cart type holding 10,000 gallons of water and 2500 gallons of Bio-Gas oil fuel, to look outwardly like a Bulleid tender with corridor to support wagon. The wheels to have clasp brakes.

CAB INDICATORS drivers side (left), Speed indicator showing miles per hour, Indicator to show variation of speed between bogie and driving wheels so that any tendency to wheel slip is visible to the driver, high and low steam chest pressure gauges, water gauge, in cab signalling, vacuum and pressure brake gauges and any others deemed necessary.

CAB CONTROLS drivers side, three position regulator giving cracked open for coasting, small opening for engine movements, then full open, the steam valve for lifting the poppet valves should be so placed that the regulator cannot be moved until it is open, the steam valve for starting should be interconnected with the regulator so that it can only be moved when the regulator is in the second and third positions; cylinder cock lever, reverser making one full turn in each direction from mid-gear for easy full regulator driving control, the steam brake levers each conveniently placed for the drivers hands from his seated position and a foot operated brake for the driving wheels only to control wheel slip. Not forgetting a lever for the whistle so that the driver (or fireman) can announce Yeah-Yeah-Here-We-Come.

CAB INDICATORS fireman's side (right), boiler steam pressure gauge, superheat steam temperature gauge, water gauge next to which should be the gauge with floating ball to control the feed water pump, lubricating oil pressure gauges.

CAB CONTROLS fireman's side, water injector controls, fire-box air damper controls and a handy place for the shovel.

DAY to DAY Maintenance, none other than inspection, ensuring that oil reservoirs are topped up, and polishing the paintwork! Also of course seeing that there is enough fuel and water in the tender.

The entire forgoing basic concept is based on the writings and works of David Wardale, Livio Dante Porta, Alan J. Haigh, Andre Chapelon, O.V.S bulleid, R.E.L. Maunsell, Harry Holcroft, E.S. Cox and others.

This is a work in progress, as yet unfinished.

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17. The Three Cylinder Compound Locomotives – B.A. Oliver.
18. The Last Steam Locomotive Engineer; R.A. Riddles – Colonel H.C.B. Rodgers.19 Handbook for Railway Steam Locomotive Enginemen – B. T. C. (Ian Allan) 20. British Locomotive Types – Railway Publishing Co. Ltd.
21. The Fire Burns Much Better – J.J.G. Koopmans.
22. The Book of the King Arthur 4-6-0's – Richard Derry.
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Stephen Whitehead.
20th January 2020.

Designing Model Boilers
The Truth about Model Boilers
What Really Happens in Model Boilers
By Martin Johnson

Part 2

THE "CONSTANTS"

There are several "constants" that the program requires to calculate boiler performance.

a. Grate Loading

> …..we work our fires harder than normal in big engines, and that we waste a greater proportion of the heat generated. – E.C. Martin ME 3423

Without some reasonable estimate of a realistic grate loading no calculations of heat production, gas flow, flow regime or heat transfer can commence.

The power output from a boiler clearly depends on how much fuel is put into it. However, with coal firing there is a limit to how much coal can be fed into a given grate; in full size rail practice it was reckoned that 100 to 120 lbs/sq.ft/hr could be fed before clinkering was likely to take place on express engines, approximately 50 lbs/sq.ft/hr was a more usual figure for freight work or shunting. There are no equivalent values for miniature practice, where one might expect a much thinner fire, and hence lower grate loading. Busbridge (Ref. h) estimated that his tests could have been continued to higher grate loadings.

I tabulated results of various locomotive efficiency trials from pre 1967 to 2007 on engines from 3.5" to 7 ¼" gauge. Some 360 individual runs have been tabulated. I also collated as many design details as possible for some 30 locomotives across the same range of gauges. I was then able to calculate grate loading data for 114 runs on various designs.

I tried analysing this data set of 114 runs in various ways. Initially, I looked at a cumulative frequency graph which showed a reasonable Gaussian distribution with a median result of 33.4 lbs./sq.ft/hr., and an upper quartile value of 45.5 lbs./sq.ft/hr. However, the data is heavily skewed toward 5" gauge locomotives, mostly of main line outline.

One might expect a rising trend in grate loading with size, which is true as Figure 1 shows. There is, of course an extra point not shown on the graph at 56.5" gauge and 100 – 120 lbs/sq.ft/hr, which indicates that a curve of grate loading against gauge must flatten off from the trend shown in Figure 1. Clearly, we do not work our miniature engines harder than full size!

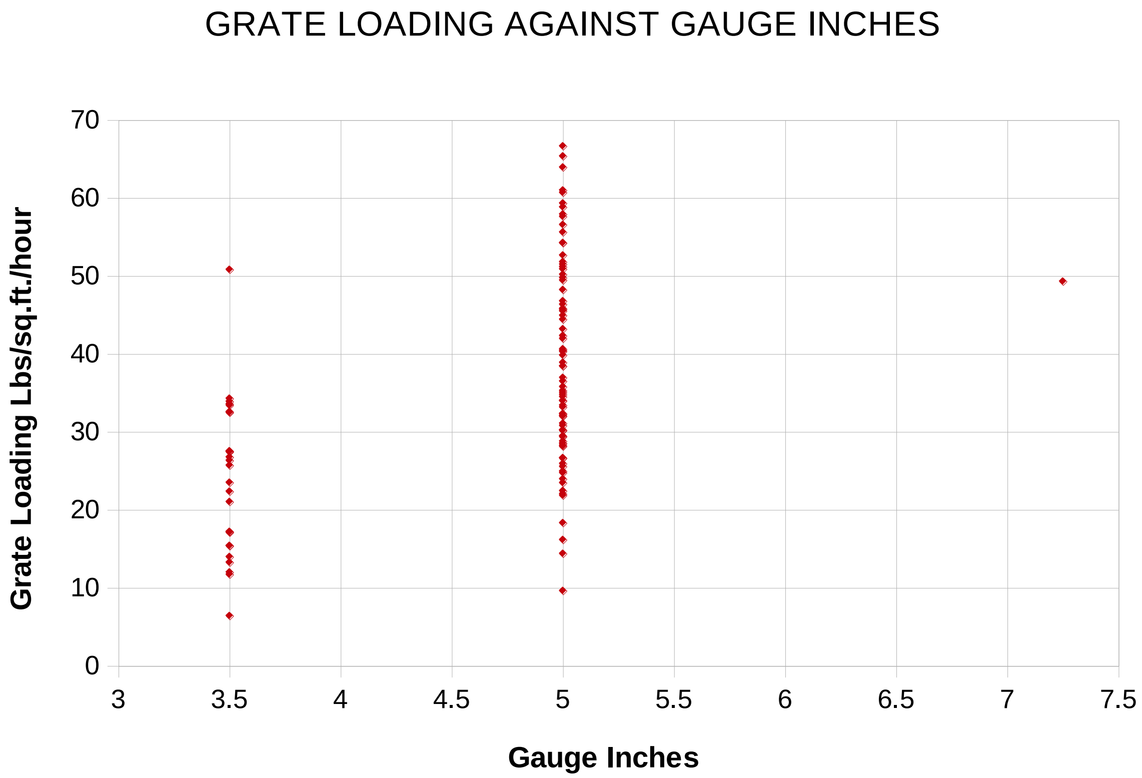

Figure 1 - Grate Loading Vs. gauge of engine

Some additional data can be found in Ref. n, in which a "Rob Roy" was worked at 42.2 lb/sq.ft/hour. The author reported that 3 adults were being hauled and estimated the fire was being worked too hard. 3.75 lb of water was evaporated in this test. J. Busbridge (Ref. h) estimated that during his tests on a 3.5" gauge Brittania boiler, the peak recorded firing rate of 20 lb/sq.ft/hour could have been easily exceeded.

For road use, a test on a full size Sentinel S6 steam waggon was reported in The Commercial Motor on 6th Jan. 1950. This used the usual Sentinel test route over Horseshoe Pass. 62.5 miles were covered in 3hrs. 2 mins. Using 4cwt. 13lb of coal on a grate of 3.28 sq. ft, hauling a gross load of 23tons 12cwt. This gives an average grate loading of 46.3 lbs./sq.ft/hr; as outlined above, one might expect a higher loading on a thicker full size fire, and a considerably higher value at peak loading.

I also tried a more sophisticated analysis of grate loading against Draw Bar Horse Power developed per unit area of grate (a measure of how hard the locomotive is working compared to it's grate size). However, the data showed very poor correlation, which was probably masked by the important variable "driver skill". This is not surprising given a 20:1 spread in measured locomotive efficiency over the various trials.

Several factors might affect the value for grate loading deduced from the above analysis:

1. The data is based on a competition where the object is to use as little coal as possible.
2. According to the competition rules, coal used during "waiting time" before the competitive run is debited against the competitors allowance.
3. Coal may be lost "overboard" during the excitement of competition.
4. The power output and hence boiler demand on a miniature locomotive is limited by the adhesive weight of the miniature, which in turn would limit the required grate loading. This would not apply to a road vehicle, where wheel slip is virtually unknown.

Based on all the above considerations, I estimate that a design loading of 40 lbs./sq.ft/hr would be reasonable for 5" and 7 ¼" gauge, and perhaps 20 to 25 lbs./sq.ft/hr as a more conservative value for smaller miniatures. There is not sufficient data to conclude much about larger boilers, except that the grate loading can be higher than 40 lbs./sq.ft/hr. For my own project of a boiler with a grate of just over 1 square foot, a value of 50 lbs./sq.ft/hr looks reasonable. So, grate loadings in miniatures are not proportional to Scale^3, mainly due to overscale fire thickness, but the grate loading per unit area is significantly less than full size.

It may be that grate loading should really be related to cubic volume of firebed, so that lbs/hour/**cubic foot** of firebed would be quasi constant. In full size practice, I have seen fire thickness typically from 4" to 8". So based on full size values 100 lbs./sq.ft/hr would be equivalent to about 200 lbs./cubic ft/hr. In miniatures, we have a much thinner fire – a maximum of 1 ½" – 2" in a 3.5" gauge Brittania, for example. So based on 200 lbs/**cubic ft**/hr and 1 1/1" to 2" fire thickness, grate loading would be 25 - 33 lbs./sq.ft/hr on a 3.5" gauge Brittania and around 50 lbs./sq.ft/hr on a deep firebox 5" gauge miniature. Figure 1 shows these estimates to be reasonable but somewhat higher than observed figures, which might be expected as they are based on the thickest possible fire.

A Plea for More Data

If anybody can help with leading boiler dimensions, particularly grate size, for the following designs I would be very grateful:

5" gauge Brittania, 3.5" & 5" gauge Maid of Kent, 5" gauge Merchant Navy, 5" & 3.5" gauge Netta, 5" gauge Springbok, or indeed any locomotive for which IMLEC style data exists.

Replies to Martin Johnson 1 on the Model Engineer forum, please.

b. Coal Lost Before Combustion

A remarkably large amount of coal is lost before combustion. This is probably due to the fierce draught through the fire carrying small coal particles away. The upward velocity through a full size fire is some 7 m/s (15 m.p.h.) which is quite a stiff wind, and able to carry small coal particles away. The velocity through a miniature firebed is around 1/3 of the full size value.

I have been able to infer values for fuel loss from Busbridges tests, I also have S.O. Ell's summary of testing on a King class locomotive and a Professor Nicholson published a formula for predicting fuel loss in full size. All of these results and prediction methods are shown in Figure 6, which shows:

- Professor Nicholson's method gives a proportional rise in percentage fuel loss with grate loading. However, Nicholson's constants predict a much higher rate of fuel loss than Ell measured.

- I have fitted a Nicholson type law to Ell's data, which is shown as “Full Size MJ Model”, which roughly approximates to the measured data.

- The fuel loss for a miniature is much higher in proportion to grate loading than for full size, as shown by the much steeper line through fuel losses inferred from Busbridge's data.

I have also tried plotting fuel loss against air flow through the grate, reasoning that it is air flow that carries the fuel away. However, it seems there are other factors affecting fuel loss and I am investigating effects of coal particle size and retention time within the boiler. In the interim, the following relationship which I have used to correlate Busbridge's data is an approximation to performance in miniatures:

$$Coal\ Lost = Grate\ Loading\ x\ 445$$

Where:

Coal Lost = Coal lost before combustion as percentage of total coal fired. [%]

Grate Loading = Coal fired per unit area per unit time [kg/m2/s]

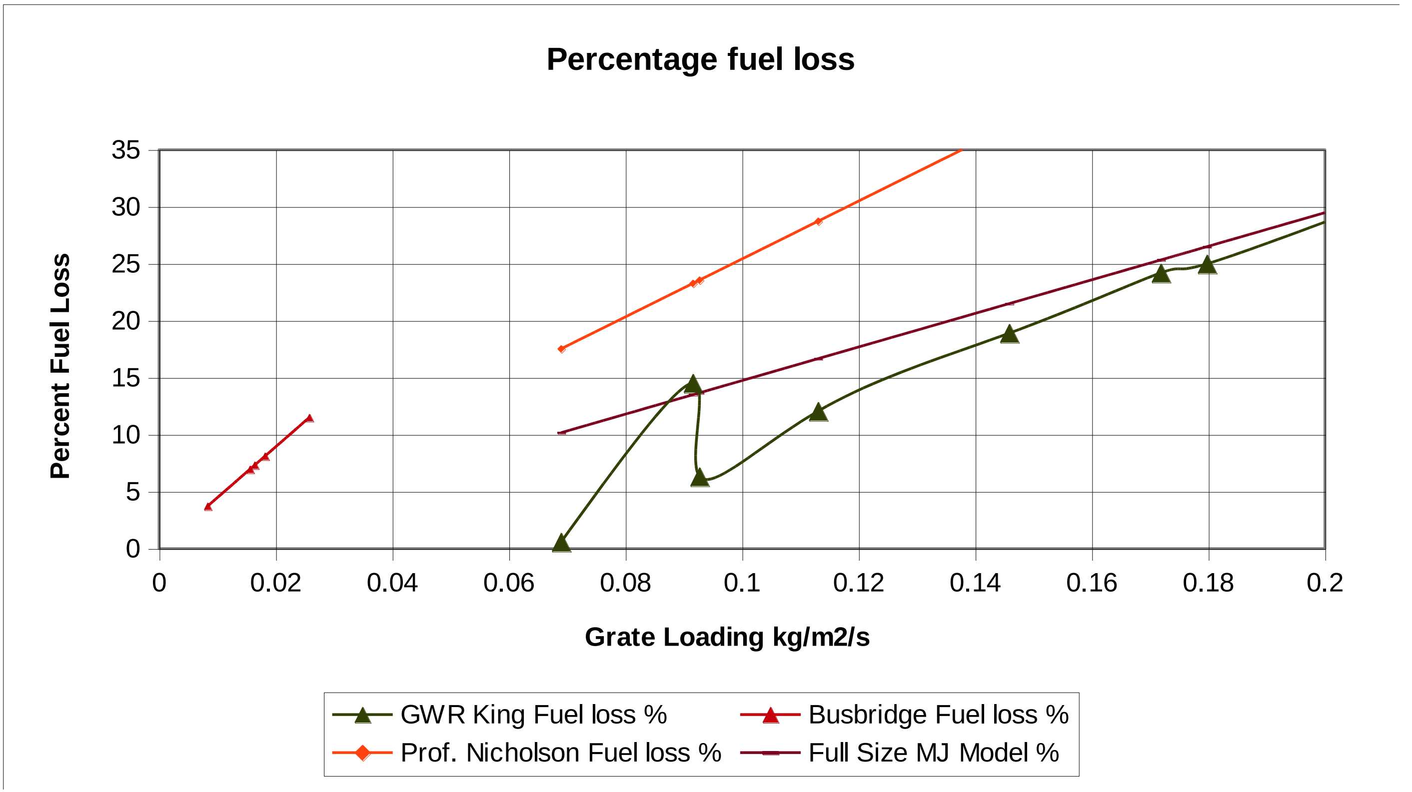

Figure 2: Percentage of fuel lost before combustion against grate loading.

c. Air Ratio

> Might closing the dampers when pulling hard reduce the excess air and still be sufficient to maintain the fire? Don Broadley ME 4558

The amount of air reaching the fire governs combustion temperature, quantity of flue gas passing through the boiler and the quantity of heat ejected with the flue gas and is therefore an important variable in boiler performance. The quantity of air drawn in can be directly deduced from analyses of flue gas as made by Busbridge, Ewins and many workers in full size. Figure 7 shows miniature data from Busbridge and Ewins along with full size data from a GWR King by Ell. The grate loading on the x axis is based on coal burned, not total coal fired.

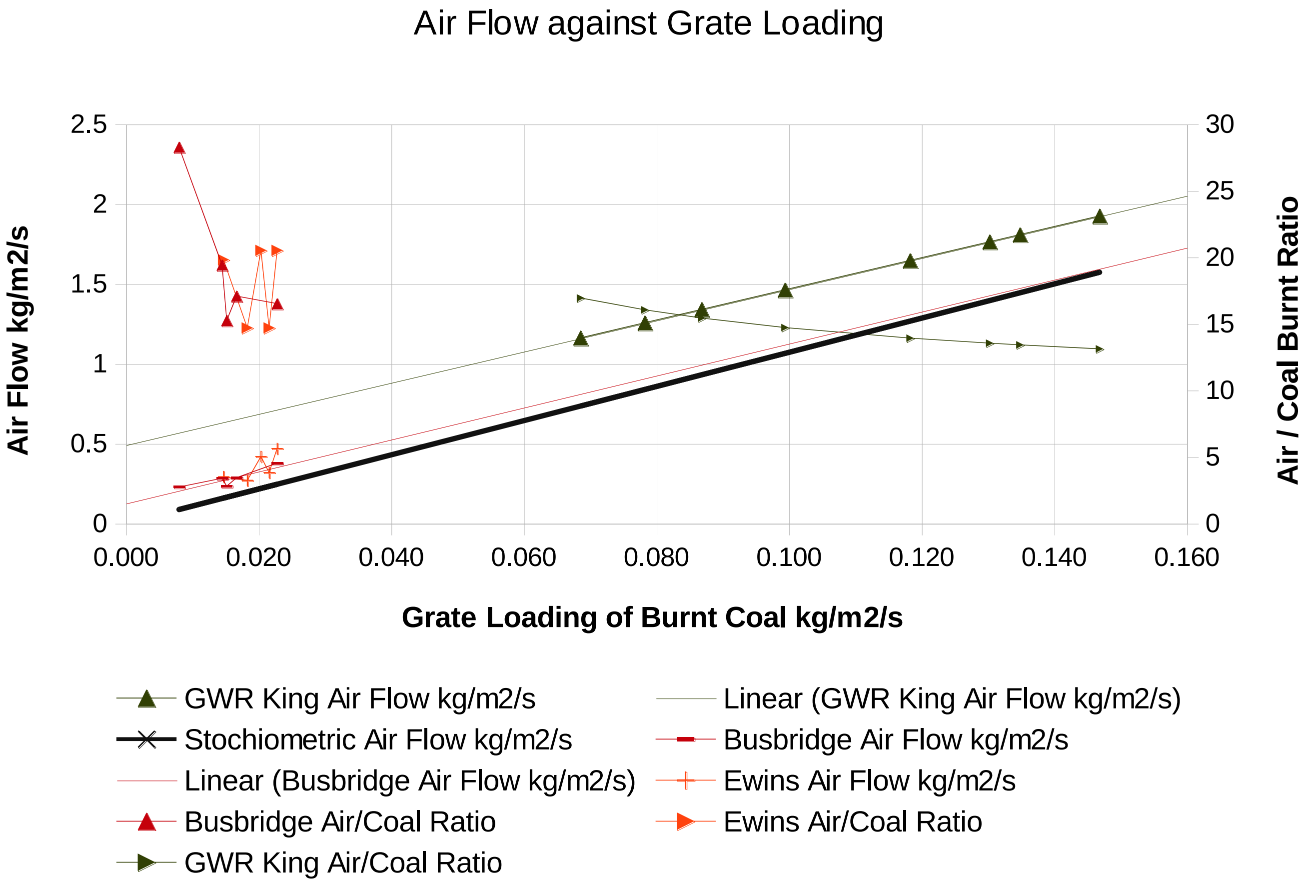

Figure 3: Air flow through coal fired grates

The full size data shows a linear relationship between air demand and grate loading. The gradient of the relationship is about 90% of the stoichiometric air demand, but there is a significant air demand at zero grate loading. I have shown the line of stoichiometric air flow, based on a good quality anthracite, which is a rather steeper line than that calculated by Ell; I do not know the reason for the discrepancy.

This zero intercept means that the air ratio (mass of air / mass of coal) tends asymptotically toward the stoichiometric value at high grate loadings. So changing the air flow must imply a change in combustion rate, so fiddling with dampers is not an option – at least in terms of steady state operation.

The miniature data shows similar trends, with the gradient of Busbridges air flow data being about 93% of the stoichiometric value but the zero grate load air demand is only a quarter of the full size value. This means that the miniature air to coal ratio tends toward a stoichiometric air ratio at much lower grate loadings. Ewins' results show much scatter but fall in a similar area.

Once again, there are clearly other effects that distinguish full size and miniature work; particle size, fire depth, grate geometry, edge effects around the grate are all possible influences. I intend to undertake further theoretical investigations to see if a universal law for miniatures and full size can be developed. In the meantime, the air flow through a miniature grate can be calculated from:

$$Air\ Flow = 0.126 + 0.93\ x\ Stochiometric\ Air\ Ratio\ x\ Coal\ Burned\ on\ Grate$$

Where:

Air Flow = Air flow through grate [kg/m2/s]

Stoichiometric Air Ratio = Theoretical mass of air to burn unit mass of coal

Coal Burned on Grate = Total coal fired less coal not burnt [kg/m2/s]

Areas are the grate area.

d. Other "Constants"

There are several other constants in the program that need some explanation or justification:

Fuel burnt above the grate – Not all the fuel is burnt on the grate, volatiles and light coal particles will burn in the firebox volume. In the analyses I have allowed for 10% of the burned fuel to be burned above the grate within the firebox. Combustion is assumed to cease in the tubes or flues.

Combustion Efficiency - From analysis of published tests, typical combustion efficiencies of 97.5% are appropriate to miniatures, giving carbon dioxide values between 0.5 and 1 %.

Absorption Coefficient of Flue Gas – This is a measure of the thickness of the "fog" of combustion gases and determines the ability of infra red radiation to pass through or be absorbed by the combustion gas. Work on combustion gases from forest fires and house fires has given absorption coefficients of around 0.8. I have found that 0.9 seems to work well for both miniatures and full size practice; the higher value is probably due to greater solids content in the flame of a forced draught coal fire.

Dryness fraction of steam – Analysis of Busbridge's superheater test results show that the steam must have been dry before entering the superheater. It may well be that a boiler can generate virtually dry steam, but Busbridge's boiler was set up on a bench. I would expect significant water carry over from a boiler travelling over imperfect track. For consistency, I have used a dryness fraction leaving the boiler of 99.9%. The program is then calculating a maximum estimate of superheat temperature.

analysing a typical locomotive design

> Most of the entire heat of the fire is transferred via the firebox…………….In a model of 1/10 full size, firebox heating surface decreases to 1/100 but firing rate in lbs/hour decreases to 1/1000, so that the model firebox presents nearly 10 times as many square feet per pound of fuel fired………. – H.S. Gowan ME 3416

In order to illustrate some of what has been learnt about miniature boilers, I will show some results for the "Speedy" boiler as designed by Curly Lawrence (LBSC). Unless stated otherwise, the results are for a grate loading of 40 lbs/sq.ft./hour and a stoichiometric air ratio of 16.5, boiler pressure 5.4 Bar (80 psi) and 4 Bar after the regulator. The output from the program is in the form of a summary table, See Table 1, plus histograms and graphs of energy balance and temperature profiles.

Most of Table 1 is self evident, but the results will be discussed in subsequent sections which should explain everything.

FLUE GAS FLOW	7.115E-03 kg/s
DRAUGHT(MIN ESTIMATE)	3.61 mm H2O
DRAUGHT(MAX ESTIMATE)	3.73 mm H2O
TOTAL HEAT IN COAL	18.891 kW
MAX. TEMP IN FIREBOX	1190 Deg. C
INLET TEMP TO FLUES	1034 Deg. C
EXIT TEMP FROM FIRETUBES	279 Deg. C
EXIT TEMP FROM SUPERHEATER FLUES	233 Deg. C
AVERAGE SMOKEBOX TEMPERATURE	265 Deg. C
EVAPORATION RATE (MIN ESTIMATE)	3.584E-03 kg/s
EVAPORATION RATE (MAX ESTIMATE)	3.610E-03 kg/s
CALCULATED EVAPORATION RATE	6.385 Ratio
ENERGY IN STEAM PRODUCED	11.054 kW
BOILER EFFICIENCY	58.52 %
SUPERHEATED STEAM TEMPERATURE	236 Deg. C
SUPERHEAT	84 Deg. C
SUPERHEAT PRESSURE DROP	3320 Pa
AVAILABLE VOLUME FOR POWER	1.260E-03 m3/second

Table 1: Sample of summary output produced.

The heat balance for the fuel is distributed as shown in Figure **4**4 and illustrates the surprisingly large amount of fuel lost before combustion. This quantity can be inferred from Busbridge's test results and has been assumed to vary linearly with grate loading. From fuel with over 18.9 kW of energy, only some 13.5 kW is burned in the boiler – remember this next time you are removing coal particles from your eye on a Sunday afternoon at the track.

Of the 10.8 kW heat absorbed by the boiler, Figure 5 shows that less than half of the heat is absorbed in the firebox, most of the heat transfer being within the tube bank, including the superheaters. The superheater flues provide a significant proportion of the evaporative capacity of the boiler, but only a small proportion of the heat goes toward superheating the steam. Finally about 3% of the evaporative capacity is lost due to heat loss from the boiler casing.

The heat transfer within the firebox is rather less than might be expected; the heat transfer mechanism is fundamentally different between full size and miniature and reflects the opposing views that:

1. The firebed radiates directly to the firebox walls. Flames have no effect.

2. The flames act as radiators and radiate heat to the firebox walls. The firebed has no effect on the firebox walls.

If we consider a firebox as a perfect cube, it will be seen that proposition 2 will radiate 5 times as much heat as proposition 1, assuming flame and firebed temperatures are equal. Thus the choice of radiation model is important. In fact, neither proposition is quite true and there is a mixture of both effects but the first proposition dominates in a miniature, whereas the second one dominates in full size. I am pleased to acknowledge Duncan Webster's help in developing this important line of thought.

The explanation for this effect lays in the absorptivity of flue gases and the beam length between fire and firebox. Think of the flue gas as fog. Seeing short distances in a fog is easy, but the fog obscures long distances. In a full size firebox, the fog obscures the fire and absorbs energy from the fire, then the fog emits the energy again to the firebox walls. In a miniature, the firebox walls can "see" through the fog and receive radiation directly from the fire. In flue gas, the main constituent of the "fog" is soot particles and fly ash plus a small effect from carbon dioxide and water vapour; the oxygen and nitrogen have virtually no effect. This subject has been closely studied in relation to propagation of forest fires and building fires and I have used some of the methodology and measurements of smoke absorptivity

I have put together a relatively simple algorithm that takes account of these effects, but it is an area of the program I would like to improve, but a rigorous approach would require a finite element method and would mean the program would not be viable on a spreadsheet platform.

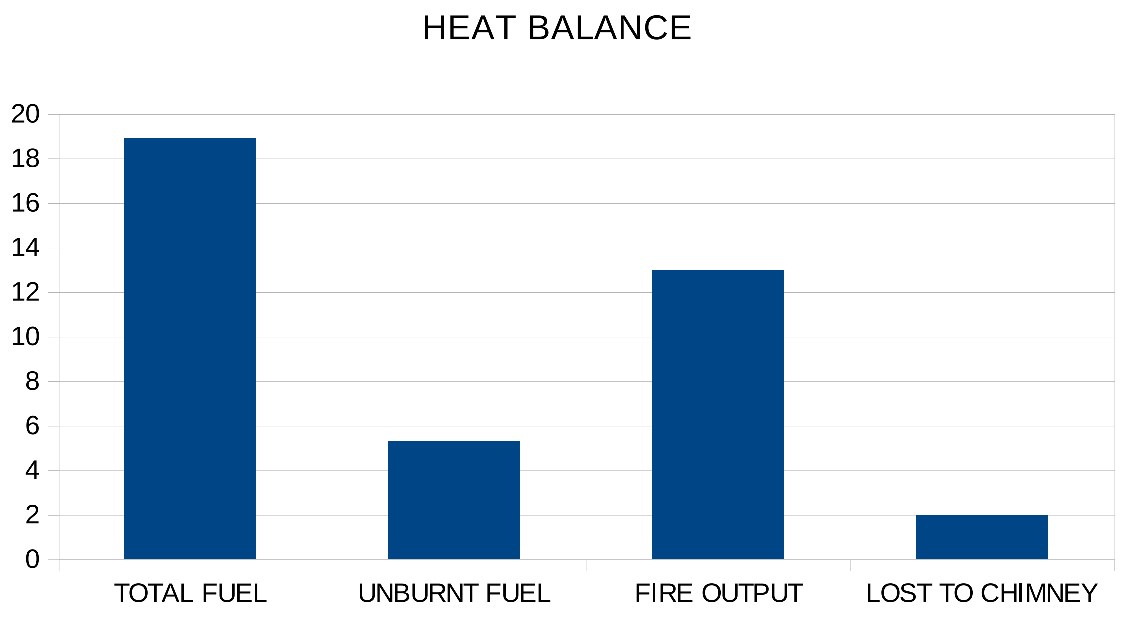

Figure 4 - Heat balance

Figure 8 shows that fuel with a heat value of some 18 kilowatts is consumed, but of that around 5 kilowatts are not burned. There are then further losses due to heating of water vapour and incomplete combustion to carbon monoxide, leaving about 11 kilowatts available for transfer into the boiler. Some 2 kilowatts are lost in chimney gases.

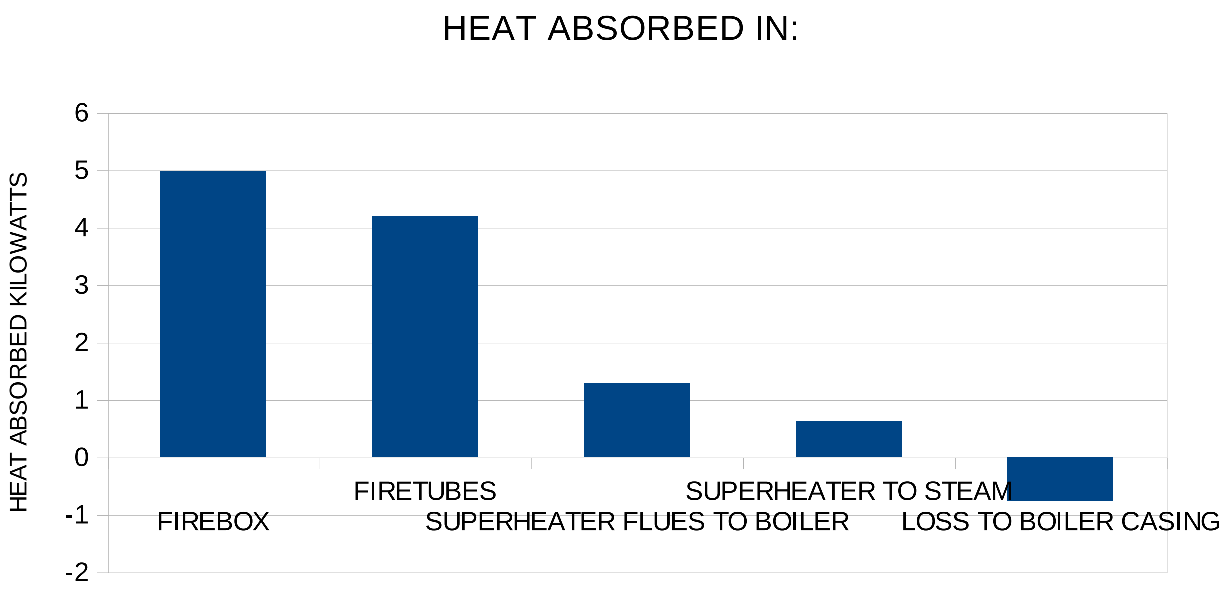

Figure 5 - Heat consumption in boiler

Figure 5 shows that of the heat absorbed in the tubes, some 20% is transferred from the superheater flues. The quantity of heat absorbed in superheating the steam is relatively minor compared to the heat transferred for boiling water.

> ……only about the first third of the tube length in the model effectively receives heat – D.E. Lawrence ME 3417
>
> In fact, the last 1/3rd contributes practically nothing and the last inch or two only just about makes up the the loss by radiation of the exterior of the boiler barrel surrounding this part of the tube bank. – J. Ewins as reported by M. Evans "Model Boilers"

The temperature profile and heat transferred along the length of the firetube is shown in Figure 6 where the firebox is at the left of the diagram. It will be seen the gas temperature drops rapidly at first, then tends toward an asymptote of the boiler water temperature. It also shows that most heat transfer takes place toward the tube entrance; half the available heat transfer takes place in the first 10% of tube length, and 75 % of the heat transfer takes place in the first 1/3 of length.

The last two inches of tube and superheater flues contribute about 225 watts to boiling water. The heat loss from a boiler would be some 85 watts over the same length, so even the last two inches are making a significant contribution.

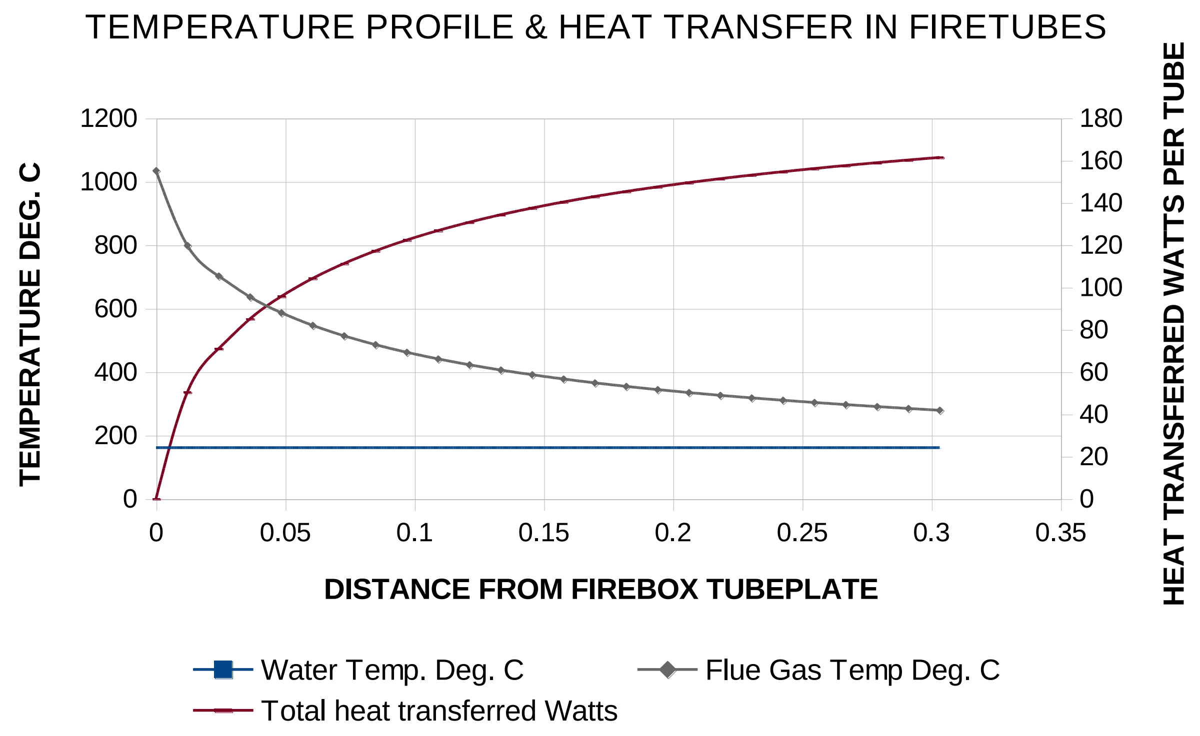

Figure 6 - Temperature profile and heat transfer in firetubes.

The draught required to produce flow across the tube bank of "Speedy" is just 3.7 mm water gauge, but to this must be added draught loss through the ashpan, grate and fire to arrive at the vacuum in the smokebox.

The boiler produces some 3.6 x 10-3 kg/s of steam, that is 6.4 times the fuel mass flow.

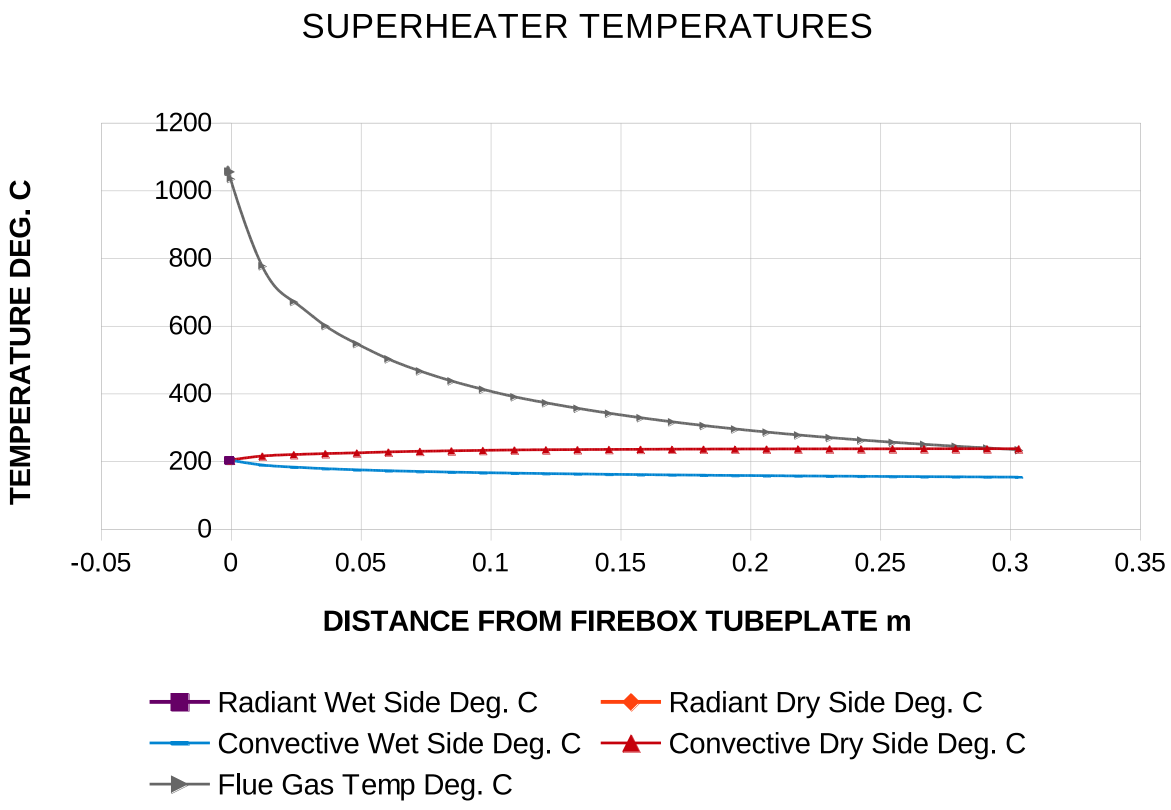

Figure 7 - Temperature profile in superheater flues

The predicted temperature profiles in the superheater flues are shown in Figure 7. The program is capable of calculating the temperature rise in radiant superheaters, but these were not specified by LBSC. I have assumed the steam is reduced to 4 Bar in the regulator for the calculation, which produces a very small amount of superheat. The steam temperature after superheat would be 236 Deg. C or 84 Deg. C of superheat (compared to the saturation point at 4 Bar). As with the firetubes, the flue gas temperature falls rapidly away from the firebox and the steam temperature rises most rapidly near the firebox.

The analysis shows the superheater flues and firetubes are not very well matched; the exhaust temperature of the firetubes is 279 Deg. C, whereas the exhaust temperature from the superheater flue is only 233 Deg. C. This seems to be because the flow resistance of the superheater flues is high compared to the firetubes, which is reflected in the flow velocity. Velocity at entrance to the firetubes is 9.4 m/s but only 8.3 m/s for the superheaters, showing that the flue gas finds it easier to escape through the firetubes. Steam flow velocity inside the superheaters varies from 19 to 24 m/s, giving a steam pressure drop across the superheaters of just 0.49 p.s.i. at maximum grate loading.

Overall, an impressive performance from a very popular design. However, the real power of the mathematical model lays in being able to assess the influence of design changes in an instant and to show just what changes take place as working conditions are varied.

assessing design changes

e. Firetubes

….the tubes need to be just long enough for the exit temperature of the gas at the smokebox tubeplate to be not lower than the boiler temperature (otherwise heat may transfer back to the gas) and not over long so the tube resistance will be unduly high. – D.E. Lawrence ME 3417

Well exactly so, but how do we find that magic situation?

Anyway, whether this reasoning be correct or not, the ratio of L/d^2 = 50 to 70 seems to be correct for any loco. boiler plain tube. – C.M. Keiller ME May 26th 1938

If a variation between 50 and 70 [of Keiller's tube factor] is remarkably constant, then the meaning of the term has changed since I went to school. – D.F. Holland ME 3423

I am indebted to Duncan Webster for finding Keiller's original article (Ref.k) so that the following comments can be based on the **original** source. I agree with D.F. Holland, that Keiller's formula lacks a theoretical basis and a spread of 40% is hardly a design guide, more of a "serving suggestion". The issue is further complicated since not all users of the formula apply it to the tube INSIDE diameter.

Keiller proposed that the ratio of gas volume in a tube to heating surface (which reduces to L/d2) should be between 50 and 70. He based this assertion on correlation of 6 full size locomotive designs and 6 miniature designs. Subsequent workers have proposed different numerical ranges for the constant.

To test whether Keiller's proposition is correct, I ran a number of alternative design options for "Speedy" through my program. I looked at a boiler without superheaters, otherwise the interaction of superheater flue and firetubes completely masks any conclusions that might be drawn. The possible changes I examined are summarised in Figure 8 where the tube size is varied from ¼" to 5/8" diameter, but all with the "LBSC designed" tube length. In all cases I assumed the tube wall was 22 gauge (0.028") and the tube ligament was 3/32" as used by LBSC. The outer circle in the various cases represents the inside of the smokebox tubeplate flange.

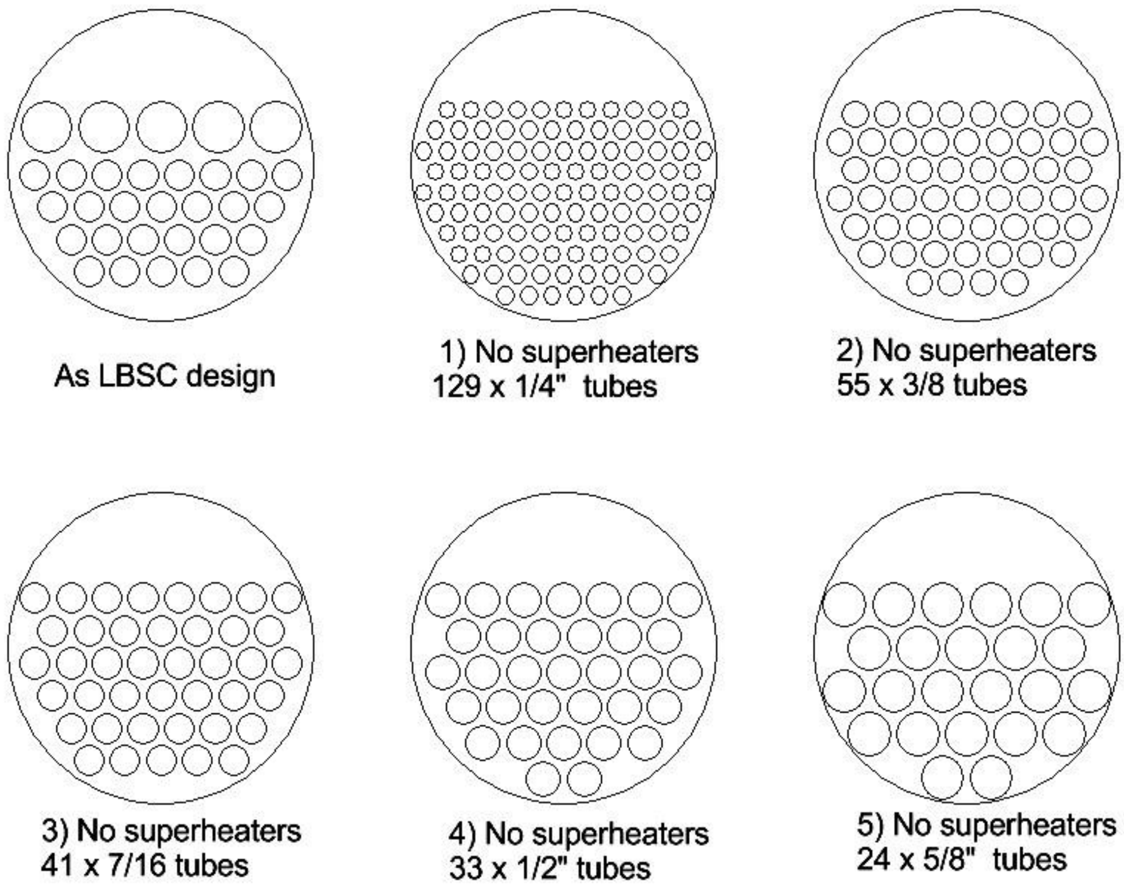

Figure 8 - Speedy design variations

The results of my calculations are shown in graphical form in Figure 9 and in tabular form below:

Tube Layout - No superheaters

Tube Length 11.93 Inches Grate area 0.01029 m2

Number	Size	Draught	Evaporation	Keiller factor	L/ID	Gas flow area	Heat exchange area	Gas Area/Grate Area
	Inch OD	mm H2O	g/s	L/ID^2 per inch		m2	m2	
129	0.25	7.6	4.112E+00	317	61	2.419E-03	4.691E-03	0.235
55	0.375	3.8	3.938E+00	117	37	2.789E-03	7.713E-03	0.271
41	0.4375	2.9	3.833E+00	82	31	2.974E-03	9.224E-03	0.289
33	0.5	2.3	3.746E+00	61	27	3.242E-03	1.074E-02	0.315
24	0.625	1.4	3.611E+00	37	21	3.872E-03	1.376E-02	0.376

Table 2: Summary of saturated steam design options

From the table it will be seen that as the tube size and number change, the gas flow area (area the gas passes through) and heat exchange surface areas (area the gas sweeps past) change significantly, as does the ratio gas area / grate area and the Keiller tube factor.

The graph shows evaporation (grammes of water boiled per second) for the complete boiler including firebox. The change in tube size covers a range from well above Keiller's recommended value to well below, so one might expect a peak in performance for the 33 x ½” tube option if Keiller were correct.

No such peak exists as Figure 9 shows, in fact the best boiler would be with ¼” tubes, or possibly even smaller. Of course such a boiler would block up in very short order and is not practical, so the question really becomes – ‘What are the smallest practical tubes to prevent blockage?’. Keiller’s original article made no mention of tube blockage or draught requirements.

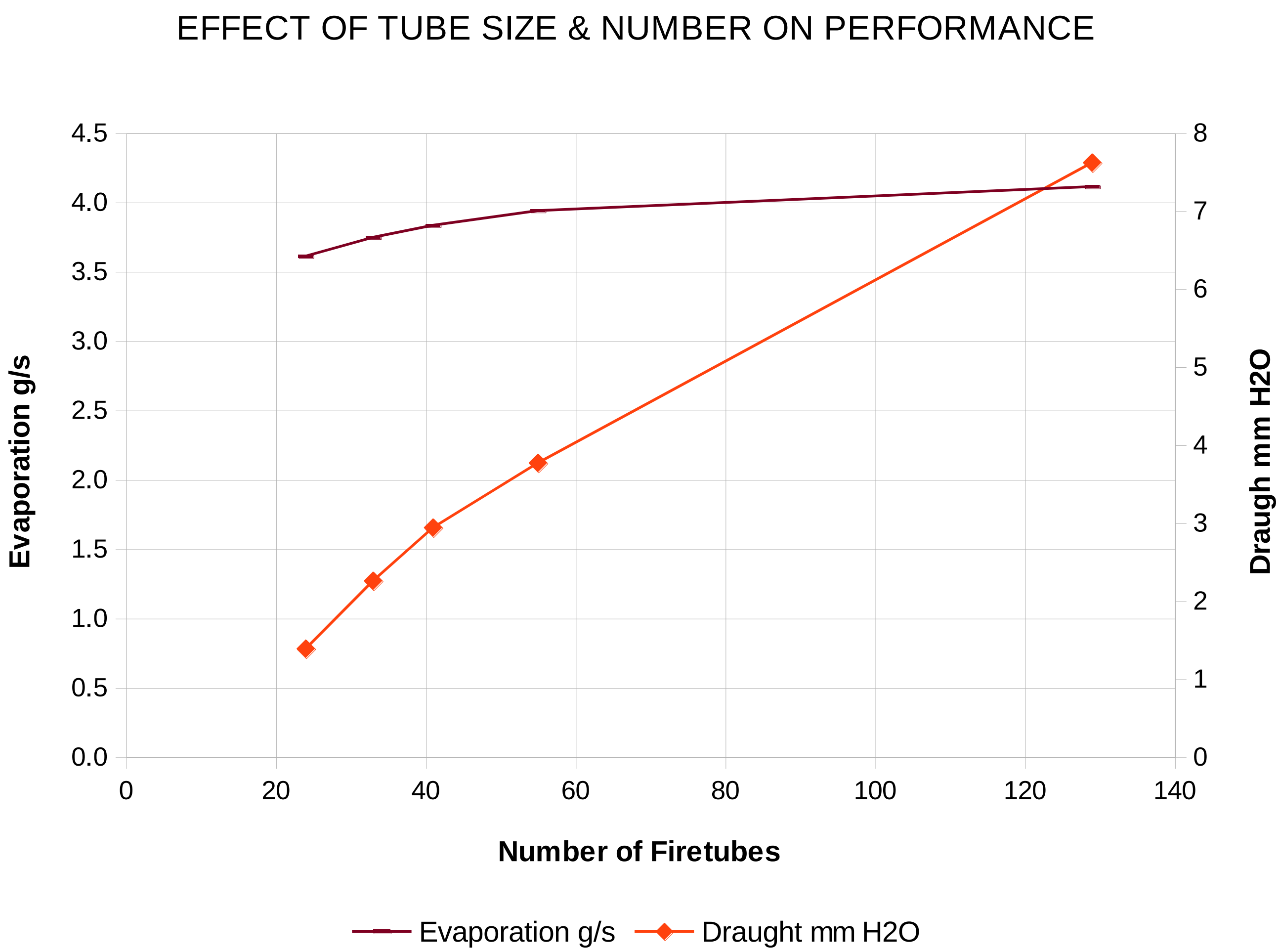

Figure 9 - Effect of tube number and size on Speedy performance

Supporters of Keiller will be pointing out that many successful boilers have been built to Keiller’s proportions. The reason for this can be seen in Figure 9 showing that the evaporation is a flat curve covering a 17% range for the wide range of tube size investigated, this range reduces to 6% over the more realistic 3/8” to ½” range. **In other words, it doesn’t make much difference what size you use!** So practical considerations of blockage become the dominant factor, followed by how much draught is required to pull smoke through the tube bank.

The graph also shows the draught (pressure drop through the tubes) on the right hand axis expressed in mm water gauge, which varies from 1.4 to 7.6 mm water gauge. Ewins proposed a boiler factor based on the proposition that long, small diameter tubes would place excessive restriction on the flow of flue gas (Ref. l). However, a more careful examination of the situation is needed, which we shall base on Ewins' own test results.

Ewins measured a total smokebox draught of 0.35" water for the higher grate loadings and a blast pipe back pressure of 0.22 p.s.i. (Ref. g) If we can change the draught by 6 mm by our choice of tube size, that is a 67% increase in total draught. That could be accommodated by restricting the blast pipe to give the extra draught. Assuming the efficiency of the the blast pipe and chimney assembly stays constant, there would need to be a corresponding increase in back pressure (approximately). That would take the back pressure up to around 0.36 p.s.i., an increase of 0.14 p.s.i. My analysis of IMLEC results showed that the average Brake Mean Effective Pressure (BMEP) is around 13 p.s.i.. Hence the extra load of a smaller blast nozzle to account for increased boiler draught would be around 1 % of the BMEP. So the performance of the engine drops by 1% - hardly noticeable, but the boiler performance has increased by 15% to provide more steam to overcome the slight drop in smokebox performance. Ewins' deduction that tube bank resistance has an effect on performance is obviously true, but when the quantities are checked the effect is insignificant. The same is not true in full size where draught values of up to 500 mm and blast pipe pressures of 8 p.s.i. are possible (Wardale as reported in The Red Devil), which takes blast pipe flow to (or very near) sonic velocity and choking.

Having severely criticised both Keiller's and Ewins' work on miniature boilers, it might reasonably be asked how I would design them. The answer is that I use my program to arrive at the best compromise of evaporation rate, superheat and tube bank resistance to maximise the **useable** steam at the engine (or conversely to minimise the grate loading for a given duty). In the next section I shall look at superheater performance which makes things even more complex and less suitable for analysis by such simple "rules of thumb".

f. Superheater Tubes

How to Assess Superheaters

> It appears from this mathematical analysis that, in our process conditions, superheat is largely a waste of time - D.A.G. Brown ME 4307

Well, if you look at the available energy drop for superheated and saturated steam between two pressure, that is true, but it does not reflect what happens in a reciprocating engine.

A boiler with superheater tubes is a compromise. Boilers, particularly on vehicles, are quite restricted in size by loading gauge or weight restrictions. Therefore, if we put in superheaters, we probably have to take out some other heat exchange surface. So a boiler without superheater tubes will always produce more steam from the greater surface area available for boiling water. However, it can be a compromise worth making when we consider how efficiently that steam can be used.

To understand the problem we must start from basic thermodynamics. Steam has a unique saturation temperature at a given pressure. At atmospheric pressure that temperature is 100 degrees C, any temperature less than that and steam condenses to water, anything more and steam is "superheated". That is of course why water condenses from the atmosphere on cold windows or bathroom mirrors. For comparison, steam at 4 Bar (60 p.s.i. approx) has a saturation temperature of 150 degrees C, but will condense on cooler surfaces in just the same way.

Now consider what happens in an un-superheated steam engine cylinder. It is fairly obvious that the temperature of the metal will settle to some temperature that is an average of the temperature across the whole working cycle from inlet to exhaust. Calculating the precise temperature is actually rather tricky, but from the above figures it could be somewhere around 125 Deg. C for example.

So our inlet steam rushes in at 4 Bar to find all the surfaces surrounding it are at considerably less than the 150 degrees C saturation temperature, so starts condensing on the cylinder walls. That is serious because the amount of steam condensed can be quite large. The amount of extra steam varies with the size of engine, R.P.M., valve type etc. but the engine would be using more steam than would be expected from the cylinder volume and cut-off setting; we can express the steam demand as a "steam ratio", being the actual steam demanded divided by the theoretical steam demand based on cylinder volume and cutoff. (Some older texts refer to the "missing quantity" which is actual steam demand minus theoretical demand.) To supply the extra steam we have to boil more water, and use more coal to do it. The engine can produce virtually no useful work from the condensed steam, since water does not expand.

Continuing with our steam cycle, as the steam expands and the pressure drops, the saturation pressure also drops until on the exhaust stroke, much of the condensate is boiled off by the cylinder walls which are now at a higher temperature than the saturation temperature of the low pressure steam. This helps to cool the cylinder walls ready for the whole sorry saga to begin again!

If we superheat the steam sufficiently, three things happen:

- The average metal temperature is increased, so steam is less likely to condense on it.
- The steam has a greater reserve of heat to be extracted before it can start condensing.
- The steam density is lower, so we use less mass of steam to fill a given engine cylinder.

The first two help to suppress condensation, and the third reduces demand on the boiler even if there is no condensation.

If we are going to assess how useful superheating might be, we need to know the value of the steam ratio. Fortunately, Professor Bill Hall undertook an extensive series of tests on an engine based on a “Speedy” cylinder block under conditions very similar to those found in miniature locomotives. His results are shown in Figure 10, where Hall’s data is shown in blue and my curve fit to his data is shown in pink. I have fitted his data to the formula:

Steam ratio = 1 + [1.7827*e$^{(-0.0197*\text{Superheat})}$]

Where:

Superheat [Deg. C] = Steam temperature – Saturation temperature (at relevant pressure)

e = Base of natural logarithm

The curve fit is not particularly good, and there may be other effects in Hall’s data to explain the rapid drop in steam ratio between 50 and 60 degrees of superheat, but the proposed smooth curve is sufficient for our purpose.

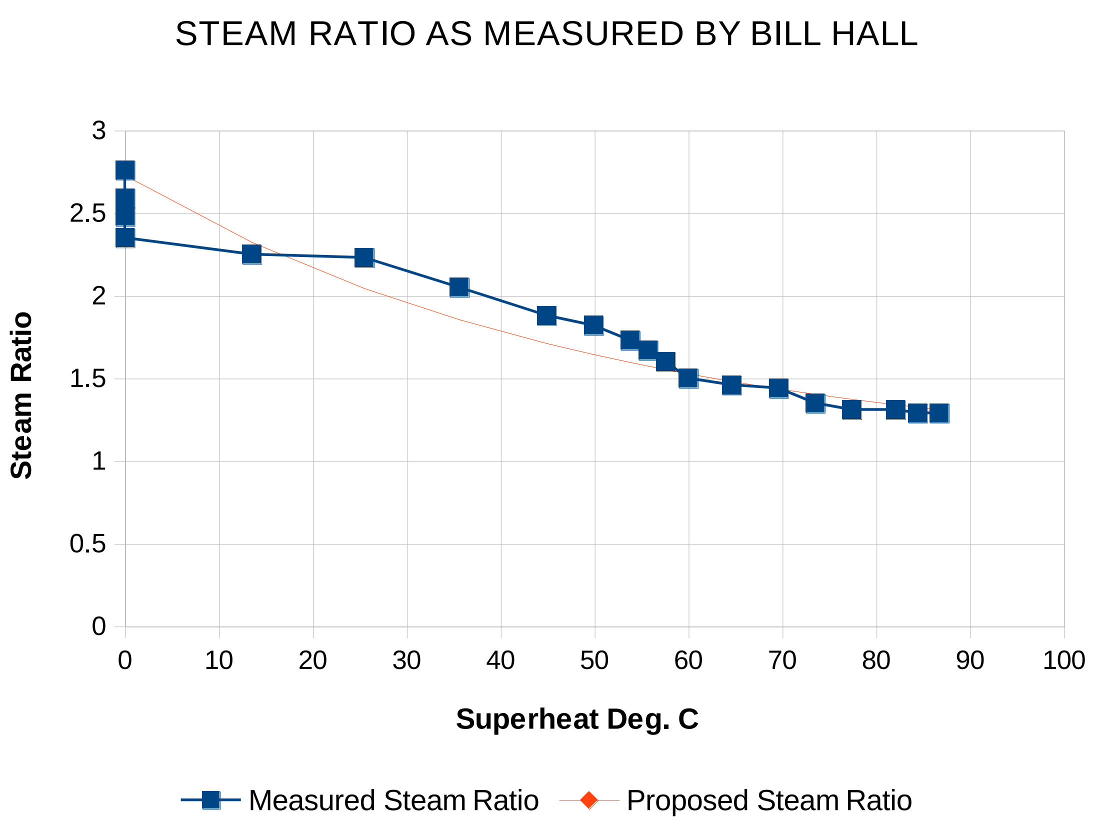

Figure 10 - Steam ratio as measured by Hall on a “Speedy” cylinder

Now, the engine needs a **volume** of steam to fill the cylinder, but we tend to talk in terms of **mass** of water boiled by a boiler. When comparing different superheat options, we must stick to volume and the effective volume available to the engine is:

Volume = Mass of steam m/density/Steam ratio

Where:

Volume = Effective volume of steam available to engine [m^3/second]

Mass of steam = Mass of steam generated by boiler [kg/second]

Density = Density of steam at relevant temperature and pressure [kg/m^3]

Different Superheat Options

A widely adopted technique of improving engine performance is to fit radiant superheaters and Figure 11 shows the predicted performance. There are several points to note here:

- Radiant superheaters are not magic. They offer a useful increase in heat transfer area, but the increase in steam temperature with distance along the superheater is not as great as at the flue entrance, because the absence of boundary layer at the firebox tubeplate end makes the convective heat transfer at that point very effective. The radiant section only receives heat on the side facing the fire because in miniatures the firebox radiant heat transfer is predominantly direct from the fire, so any areas in shadow are not heated.

- In this instance, the steam temperature in the dry leg does not change in the last 100 mm of length, because the steam temperature (280 C) exceeds the flue gas temperature (280 to 225 C). The reason for this is discussed in detail below. (In fact steam temperature drops very slightly)

- The analysis of the Speedy boiler showed that the exit temperature from the superheater flues was considerably lower than that from the firetubes, suggesting flow through the superheaters was "sluggish". I have also noticed that some recent miniature locomotive designs have 4 x 5/16" elements in a 1" flue which gives similar sluggish flow to the "Speedy" design. Apart from the heat transfer problem, it seems to me that allowing a lower velocity is a good way of helping to block the superheaters. Interestingly, L.D. Porta reports using reduced diameter ferrules in the firetubes to force more gas through the superheater tubes in his work in Argentina. Keiller even addressed this problem in his original article, proposing that circumference / area should be roughly the same for superheaters and firetubes – a point on which I heartily agree! **I believe that balancing flow in superheaters and firetubes is an important factor in effective boiler design.**

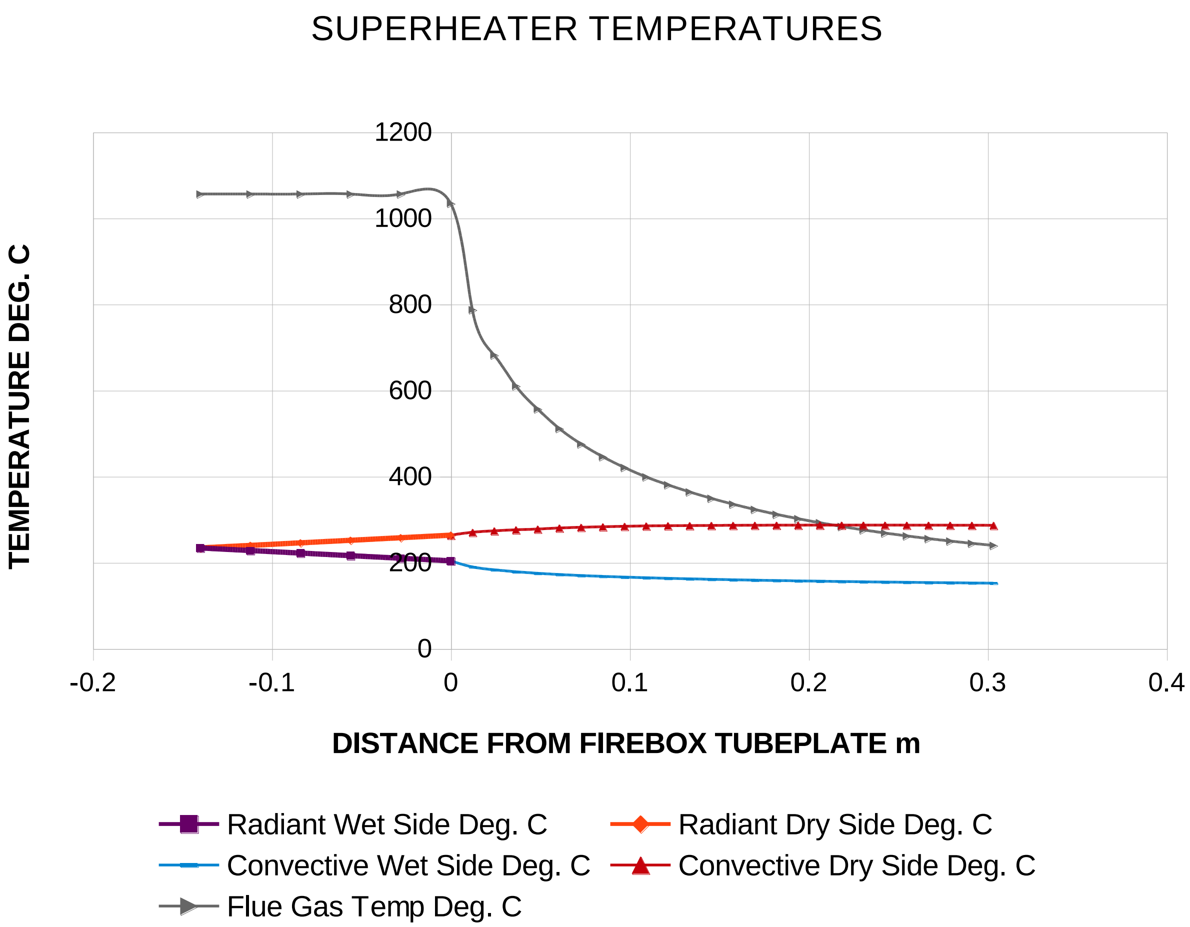

Figure 11 - Performance of Radiant Superheaters on "Speedy"

Roger Froud had designed a tube layout which allowed increased water space between tubes, using 18 firetubes and has developed his own technique for TIG welding 3/8” stainless superheater elements. His proposed boiler design used four 1” flues with 3/8” elements. I was able to use an early version of my program to predict how this might perform for him and the graph of the superheater performance is shown in Figure 16.

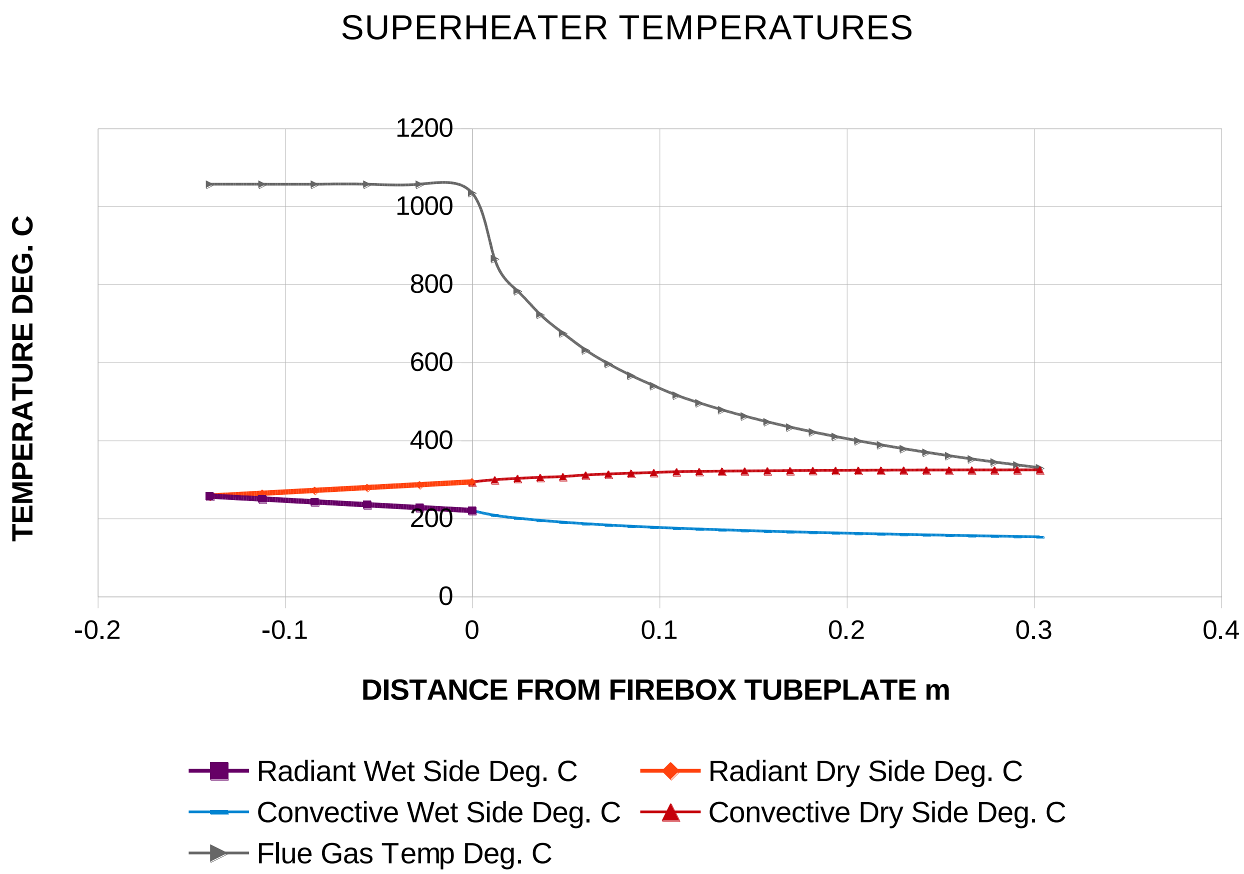

Figure 12: Superheater performance in Roger Froud's "Speedy" boiler design

In Roger Froud's design, the flows through the firetubes and superheater flues are in better balance, giving exit temperatures of 294 and 329 Deg. C respectively. Figure 16 Also shows that the dry side element continues providing temperature rise throughout it's length. The superheat temperature has been improved to 323 Deg. C from 286 Deg. C in the radiant version of LBSC's design. I was concerned the temperature might be too high, but it is easier to take out excess superheater surface than to put extra in!

I then investigated just how far the superheat could be increased; was there a maximum after which performance would drop away? **There was no such maximum**; the predicted performance rising until I had 8 radiant superheaters and just two firetubes! This arrangement gave a predicted steam temperature of 446 Deg.C. Not a very practical arrangement with a firebox full of radiant elements and a steam temperature that would rapidly degrade even the best superheated cylinder oil, and destroy PTFE and most rubbers. Front end plumbing would also become a significant problem with so many superheater elements to connect.

The required draught of 14 mm water gauge did not appeal either, but nevertheless it is an interesting numerical exercise – perhaps a glimpse of a future IMLEC winner?

My final design option was found by accident. Using LBSC's original tube spacing it is possible to get four 1” superheater flues and 24 x 7/16” firetubes into the tubeplate. If the superheater flues are fitted with two 1/4” diameter elements per flue with a radiant length in the firebox, then we get an excellent balance of evaporation AND superheat. This design gave the maximum effective volume of steam to the engine.

Contrary to what some have stated, the pressure drop across miniature superheaters is tiny. For all the geometries described here it is well below 1 p.s.i. In fact, the flow velocity through typical miniature superheaters could be usefully increased, which would need more but smaller elements.

The various superheater options are summarised in Figure 17 and the results of investigations are summarised in Table 3. There are several points to notice in this table:

- All the non superheated options produce effective steam volumes less than half the superheated options, despite producing a greater mass of steam.
- The design options produce a volume of useable steam covering a 380 % range, while the evaporated mass covers only a 136 % range. This demonstrates the importance of effective superheater design for miniature boilers.
- There is an advantage to further superheating beyond superheats of 150 Deg. C, due to increased steam volume and further small improvements in steam ratio. However, this might not be feasible with some cylinder materials and lubricants.
- An optimum boiler design balances flow through smokeflues and superheater flues.
- Radiant superheaters provide a useful extra heat exchange area, but the performance of the whole superheater system must be considered for effective design.

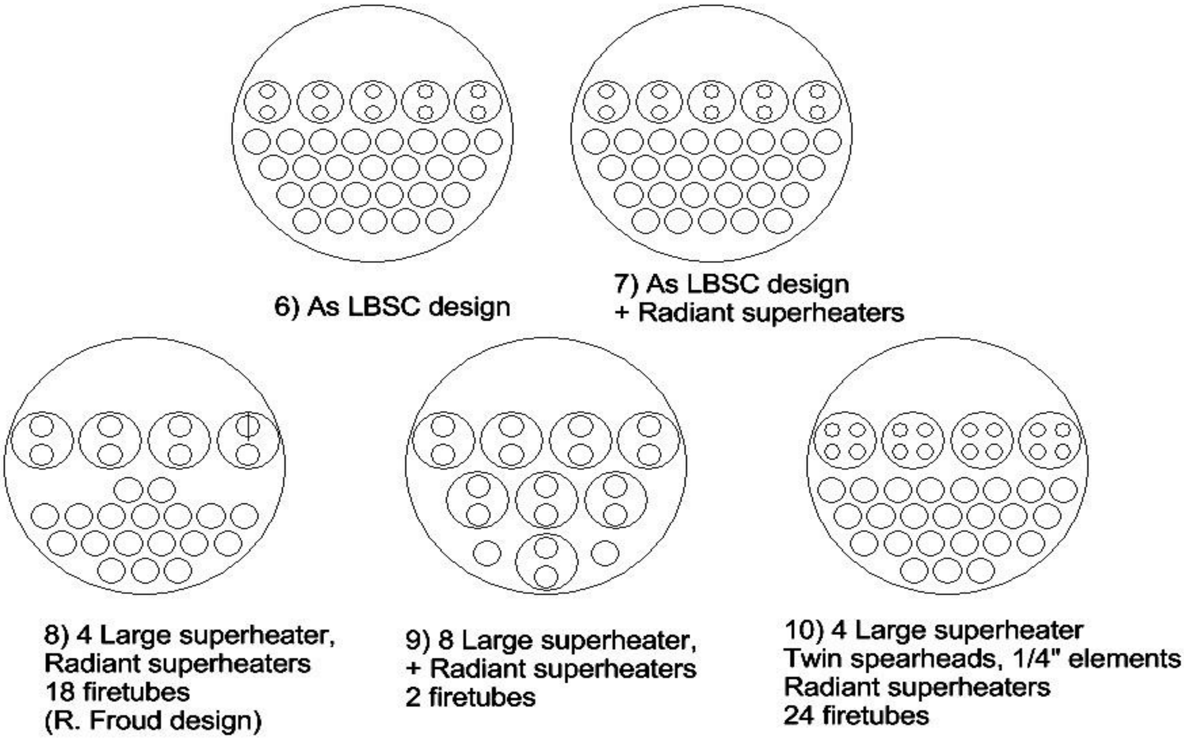

Figure 13: Summary of superheater options tried

OPTIONS (See Diagrams of tubeplate)		FLUES	S/HEATERS	RADIANT LENGTH	DRAUGHT	EVAP. RATE	STEAM TEMP	SUPERHEAT	AVAILABLE VOLUME FOR ENGINE
OPTION	DESCRIPTION			mm	mm H2O	g/s	Deg. C	Deg. C	m3/second
1	SPEEDY – 129 X 1/4" TUBES	129	0	1	7.62	4.11	152	1	5.69E-004
2	SPEEDY – 55 X 3/8" TUBES	55	0	1	3.77	3.94	152	1	5.45E-004
3	SPEEDY – 41 X 7/16" TUBES	41	0	1	2.94	3.83	152	1	5.30E-004
4	SPEEDY – 33 X 1/2" TUBES	33	0	1	2.26	3.75	152	1	5.18E-004
5	SPEEDY – 24 X 5/8" TUBES	24	0	1	1.39	3.61	152	1	4.99E-004
6	SPEEDY AS DESIGNED BY LBSC	26	5	1	3.73	3.61	236	84	1.26E-003
7	SPEEDY AS DESIGNED BY LBSC + RADIANT S/HEATER	26	5	140	3.74	3.54	286	135	1.64E-003
8	SPEEDY AS MODIFIED BY R.FROUD	18	4	140	4.47	3.35	323	172	1.70E-003
9	8 S/HEATER SPEEDY	2	8	140	13.91	3.02	448	297	1.89E-003
10	MAX. VOLUME SPEEDY	24	4	140	3.31	3.55	360	209	1.98E-003

Table 3: Summary of design options investigated.

And finally:

> One cannot discuss tube proportions without considering the boiler as a whole, and indeed the whole machine, and what it has to do. – D.E. Lawrence ME 3417

I could not have said it better.

CLOSING THOUGHTS

They can never be "conclusions" on a project as open ended as this one!

I am confident that performance of a miniature (or full size) boiler can be numerically predicted. There is still uncertainty about how such factors as grate loading, air ratio and fuel lost might vary with scale, but the calculations will give a good indication of how design changes compare to each other.

I am not aware of an analysis of grate loadings in small engines that has been done before and certainly not published. A design value of specific fuel flow rate of around 40 lbs/sq.ft/hour is appropriate to miniature boilers in the 5" to 7 ¼" gauge range and 20 to 25 lbs./sq.ft/hr as a more conservative value for smaller miniatures.

I hope I have demonstrated that really effective design of miniature boilers cannot be accomplished using simple rules such as Keiller's formula or Ewins' boiler factor, especially when the interaction of superheater and firetube flues has been shown to have such a significant effect of boiler performance.

I continue to work on refining prediction methods for air ratio and fuel lost before combustion, plus fine tuning the calculation and researching a better method of firebox heat transfer modelling;

The program is available from : https://www.rmweb.co.uk/community/index.php?/topic/143493-eim-may-2019-%E2%80%93-martin-johnsons-7-inch-fowler-wagon-build-project/&_fromLogin=1 However, if members would like a copy of the latest version, please contact me via. the ASTT website members forum. Similarly if anyone would like to discuss the findings, methods and analysis, I have started a thread on the ASTT forum as a place-holder.

https://www.advanced-steam.org/forums/topic/boiler-thermal-design-calculations/

I am working on other related topics such as condensation prediction, picking up from Bill Hall's work again, but trying to extend it beyond 5" gauge locomotive cylinders. I also get out in the workshop occasionally.

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