Notes on the transcript of

Adrian Tester's paper titled

"The Physiology of the Locomotive Boiler - Another Peep!"

given at ASTT's October 2022 conference in Darlington

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

The Physiology of the Locomotive Boiler - Another Peep!

ASTT Conference, Darlington – 15/16 Oct 2022

by Adrian Tester

Introduction (Slide 1 – Cover slide *omitted from this transcription***)**

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]

Slide 2 – *see overleaf*

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 *(overleaf)*, 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.

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

[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.

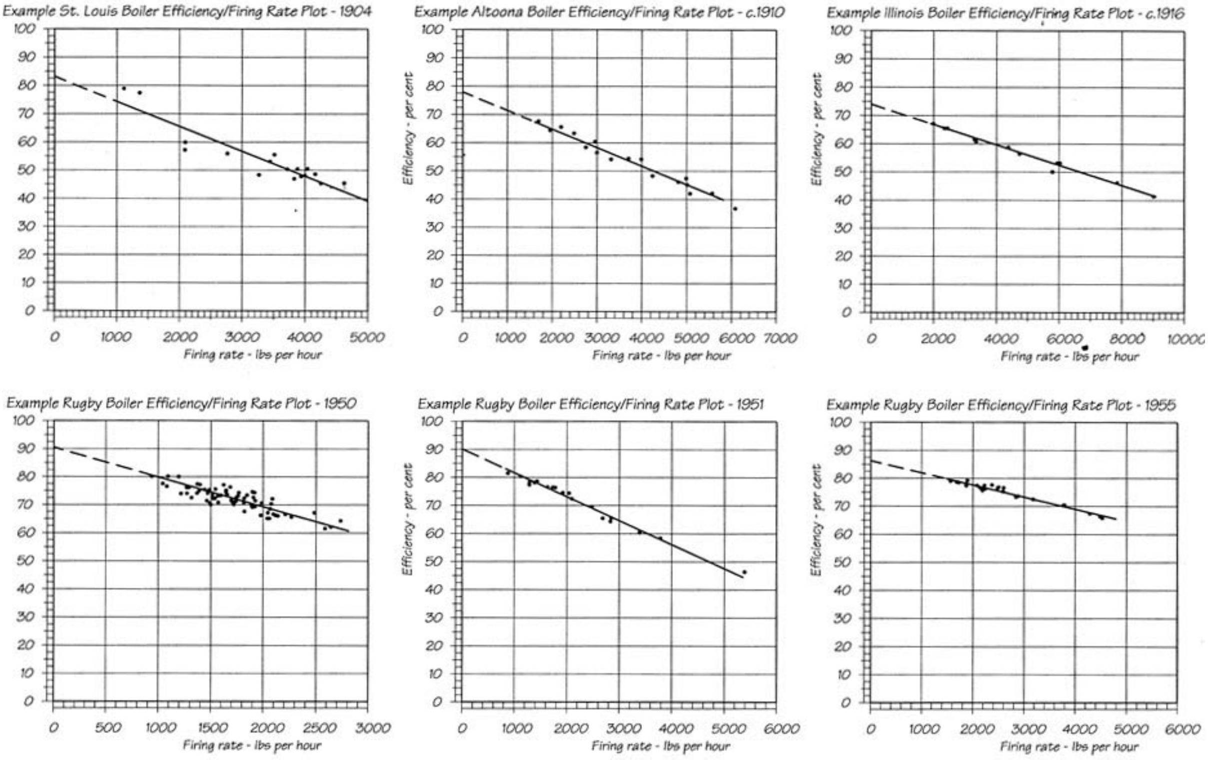

Fig. II.5 - The improvement in the accuracy of boiler efficiency tests with experience

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 3 – *see overleaf*

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 <u>very</u> 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.

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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.

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.

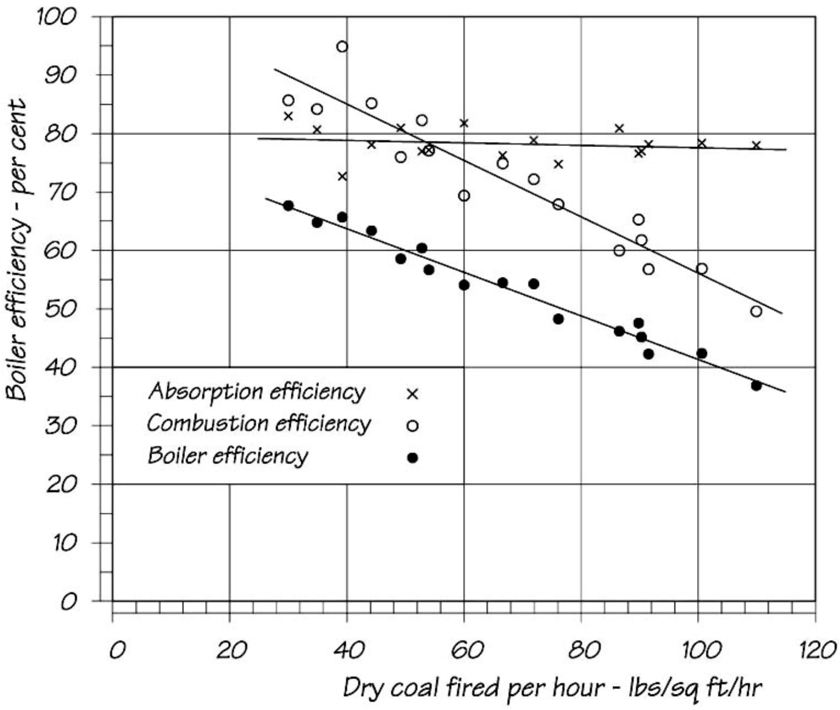


Fig. II.4 – Boiler efficiency curves for Pennsylvania Railroad E2a 4-4-2 N° 5266 tested at Altoona in 1905 – analysis by Lawford Fry

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%.

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.

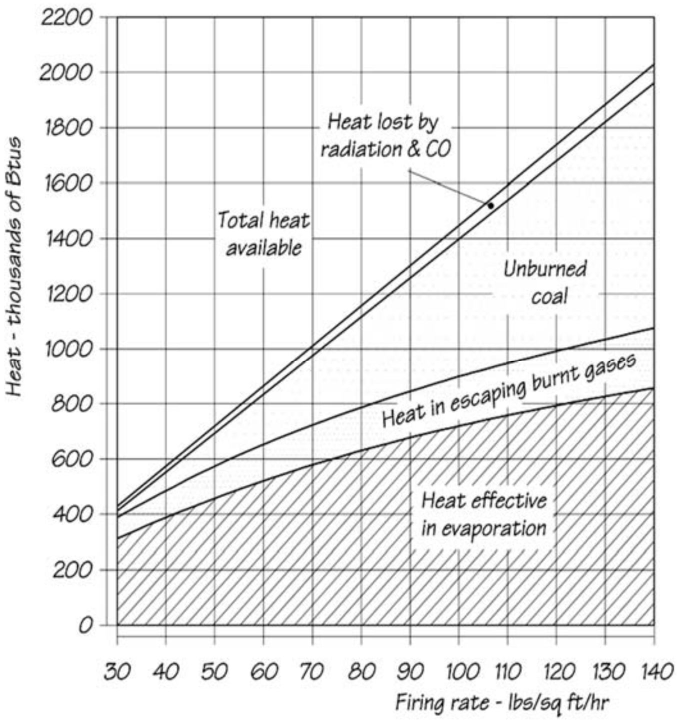


Fig. IV.10 - Approximate heat utilized and lost per square foot of grate per hour at different firing rates - St Louis Test Series 200 - Lake Shore & Michigan Southern Railway 2-8-0 N° 734

From PPT slide 4: *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 – *see overleaf*

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 G_b 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.

> ***From PPT slide 5****: 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.*

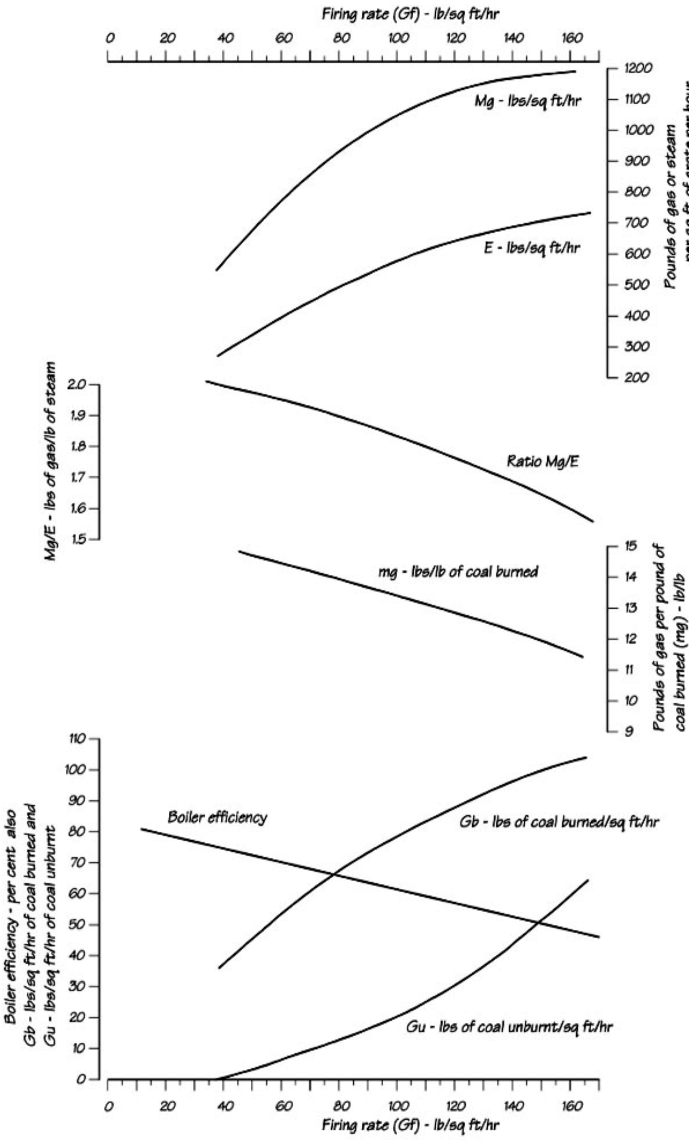
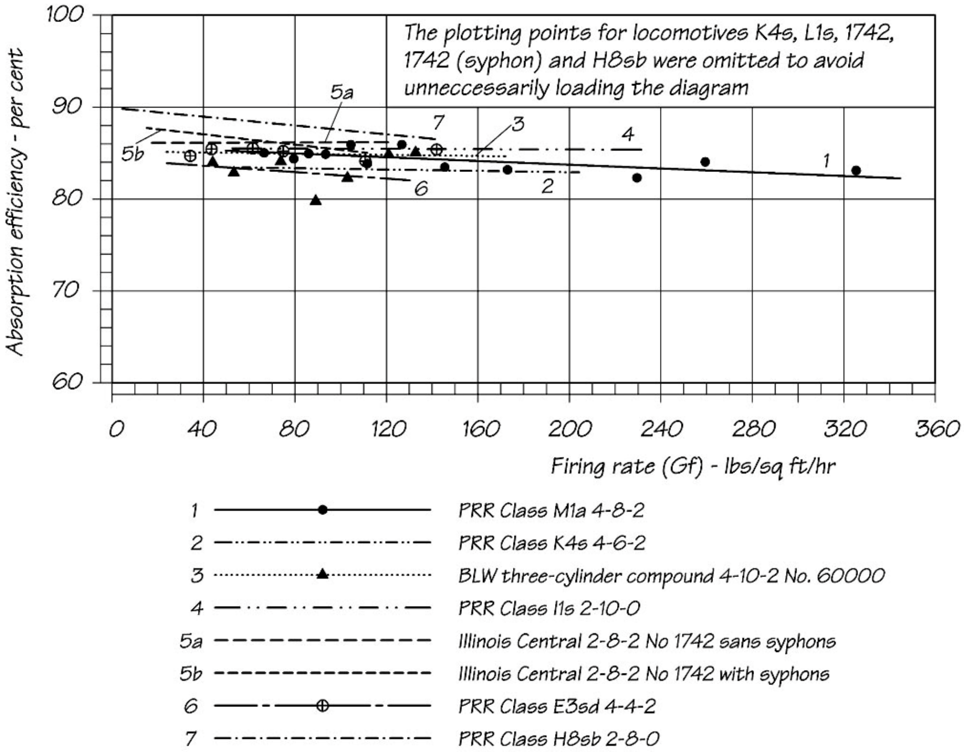


Fig IV.24 - Boiler performance chart for unidentified SAR locomotive - Dr M M Loubser's paper ***"****Some Aspects of Railway Mechanical Engineering"*

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.

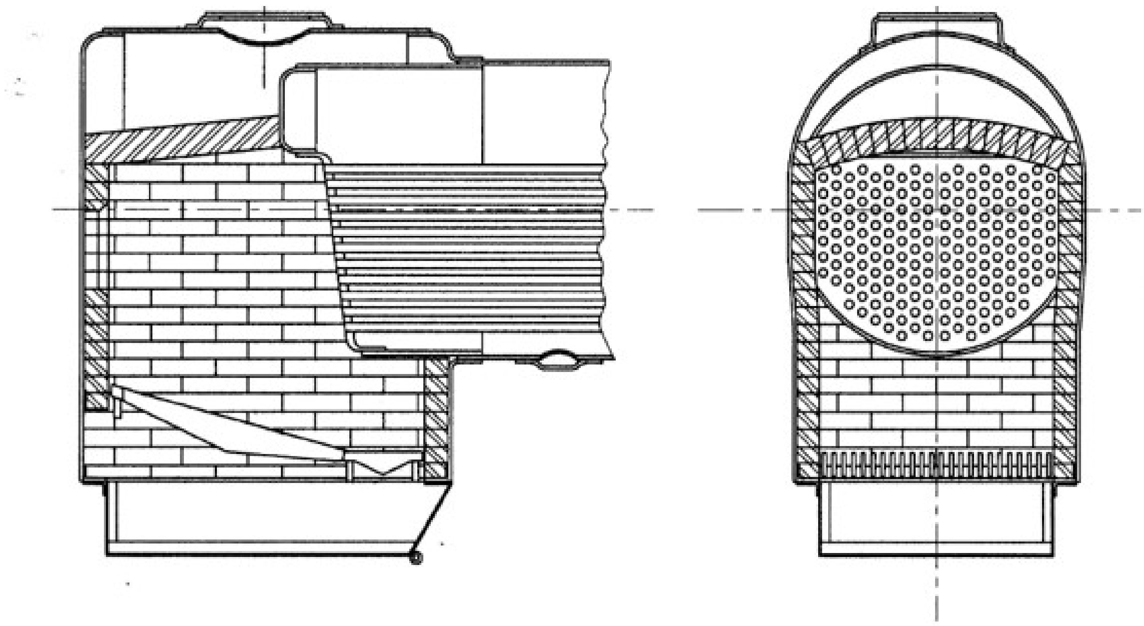
[5] Before the heat can be lost it has to be first absorbed. So, if the standing loss is reduced, then the absorbed heat is reduced.

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.



Locomotive	Line	Distance - km	Load - tons	Ton-km	Coal consumption		Water consumption	
					Total - kg	per ton-km	Total - kg	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

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

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.

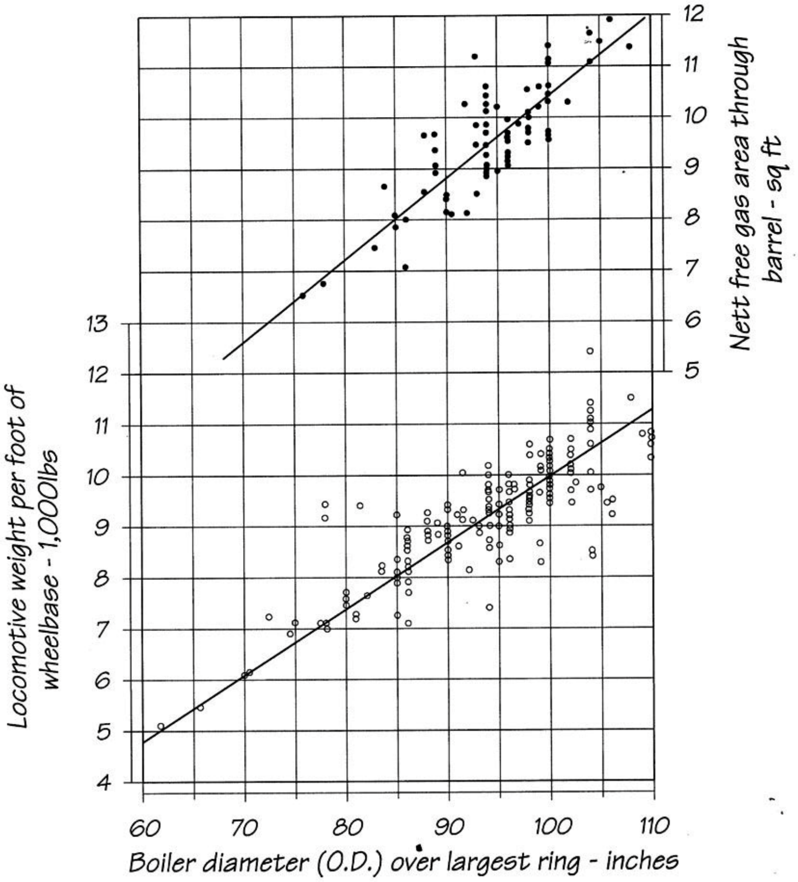


Fig IV.27 - Relation of boiler diameter to total weight of locomotive per foot run of wheelbase and also the available free gas area provided

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.

Slide 9 – *see overleaf*

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.

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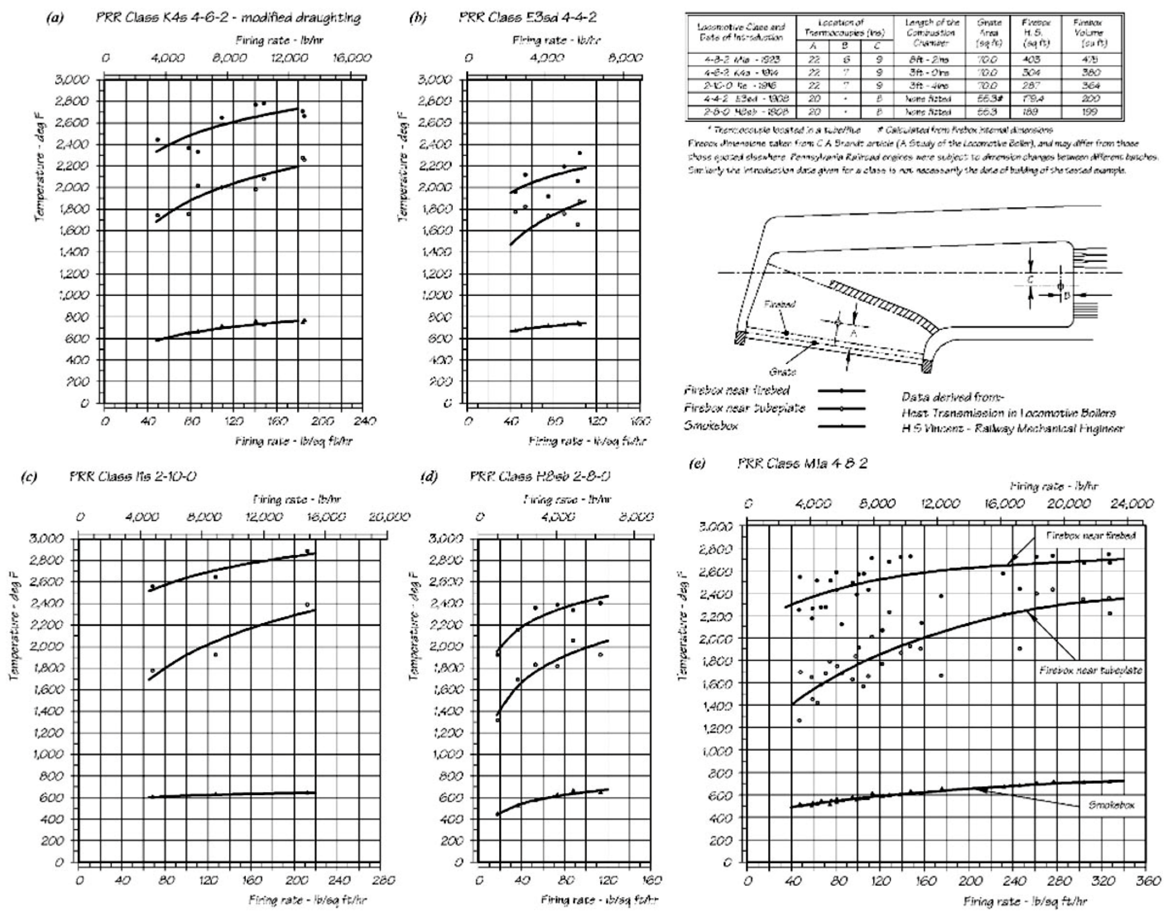
[7] "15% free gas area" means that the unobstructed area of flues and tubes divided by grate area ≈ 15%.



Locomotive Class and Date of Introduction	Location of Thermocouples (ins)			Length of the Combustion Chamber	Grate Area (sq ft)	Firebox H. S. (sq ft)	Firebox Volume (cu ft)
	A	B	C				
4-8-2 M1a - 1923	22	6	9	8ft - 2ins	70.0	403	475
4-6-2 K4s - 1914	22	7	9	3ft - 0ins	70.0	304	380
2-10-0 I1s - 1916	22	7	9	3ft - 4ins	70.0	287	364
4-4-2 E3sd - 1908	20	*	8	None fitted	55.3#	179.4	200
2-8-0 H8sb - 1908	20	*	8	None fitted	55.3	189	199

* Thermocouple located in a tube/flue # Calculated from firebox internal dimensions

Firebox dimensions taken from C A Brandt article (A Study of the Locomotive Boiler), and may differ from those quoted elsewhere. Pennsylvania Railroad engines were subject to dimension changes between different batches. Similarly the introduction date given for a class is not necessarily the date of building of the tested example.

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.

Slide 10 – *see overleaf*

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.

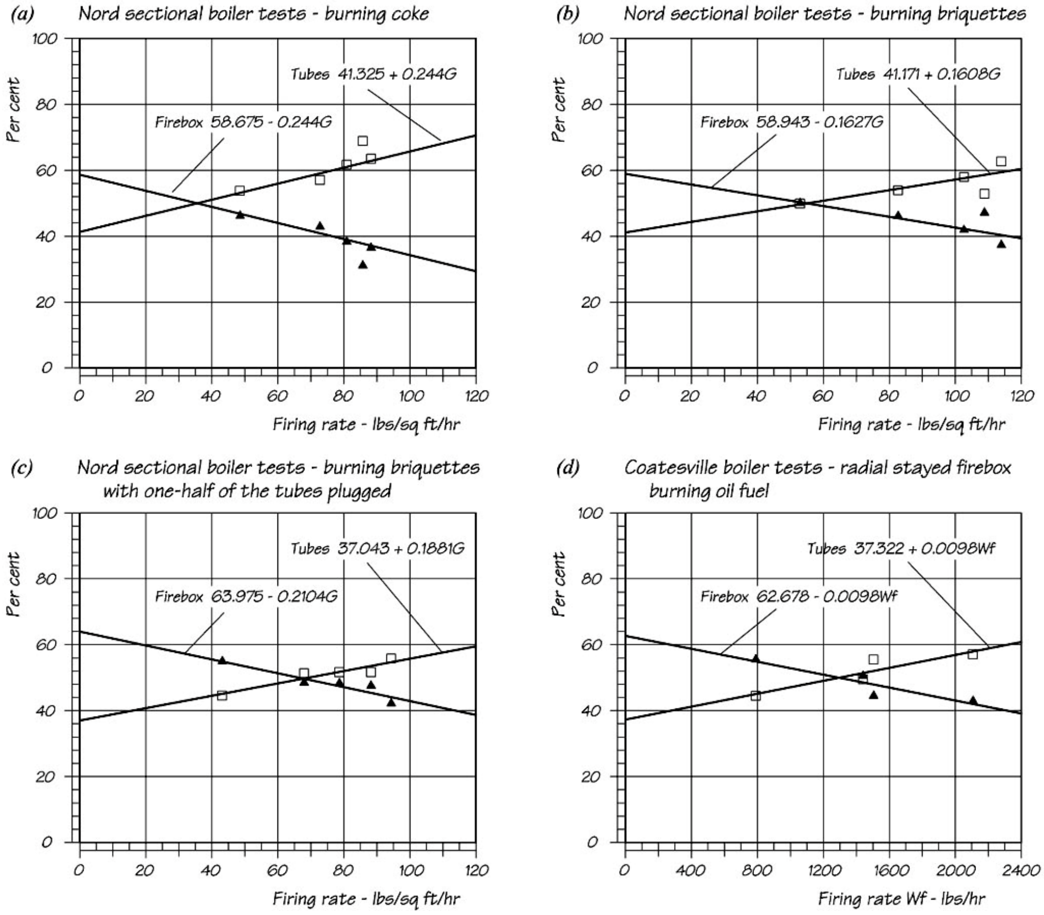
In both boilers we see confirmation of our previous findings – namely, as the firing rate increases, the heat absorption distribution alters – a fact that the early pioneers such as Robert Stephenson were fully aware. Thus the total evaporation from the firebox, despite still increasing with rise in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

[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.

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]

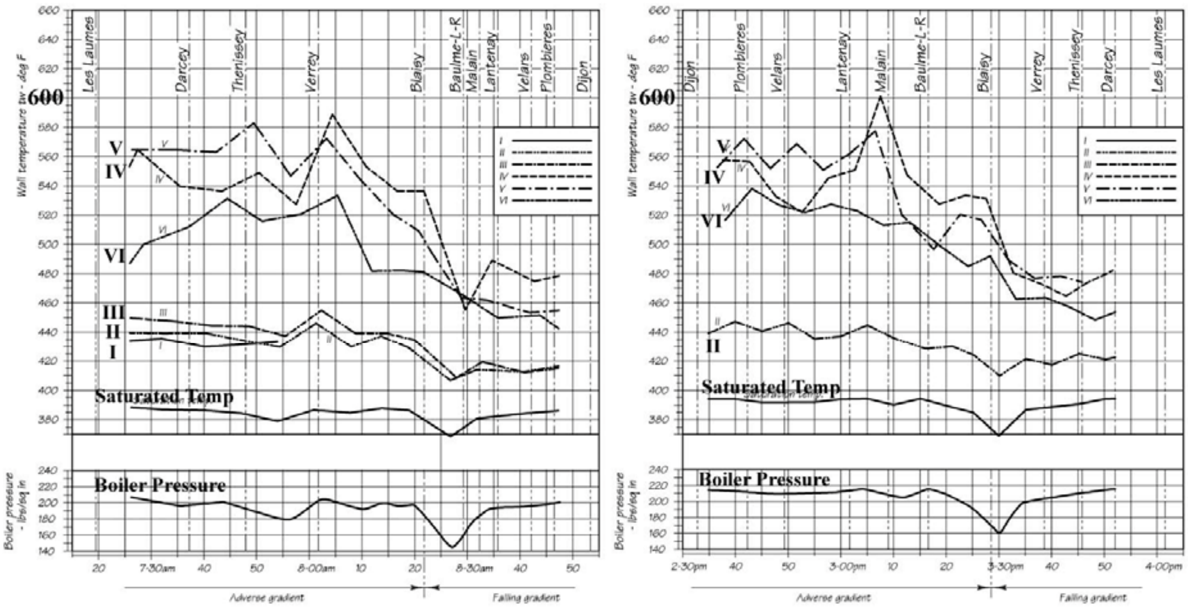
[9] There nevertheless has to be a certain amount of indirect surface for absorption efficiency and for superheating.

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.



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

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 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.

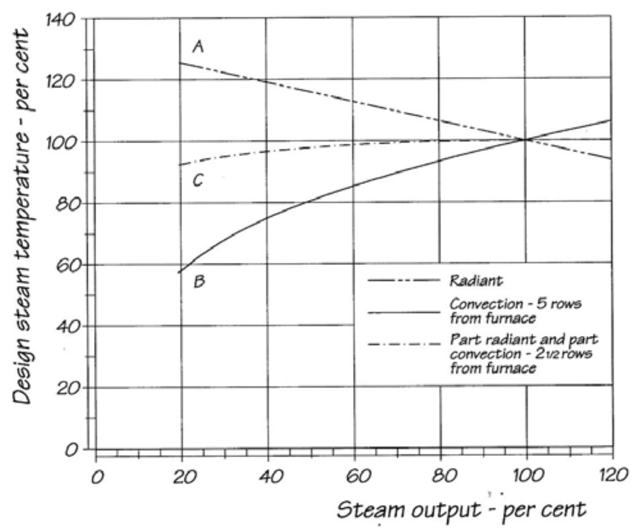


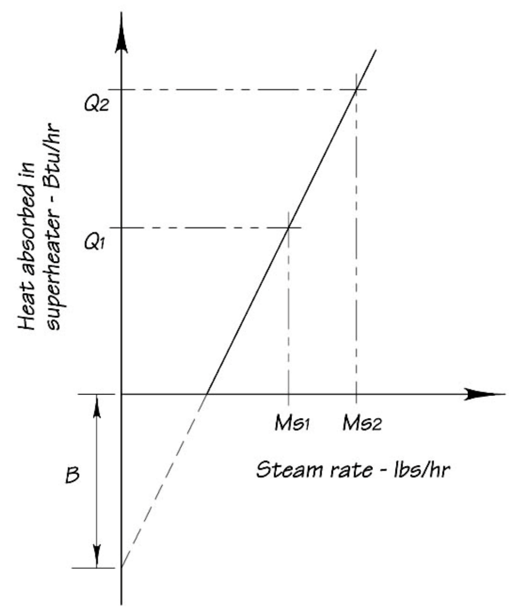
Fig. IX.4 - Superheater characteristics or the differences between radiant and convective superheaters

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.



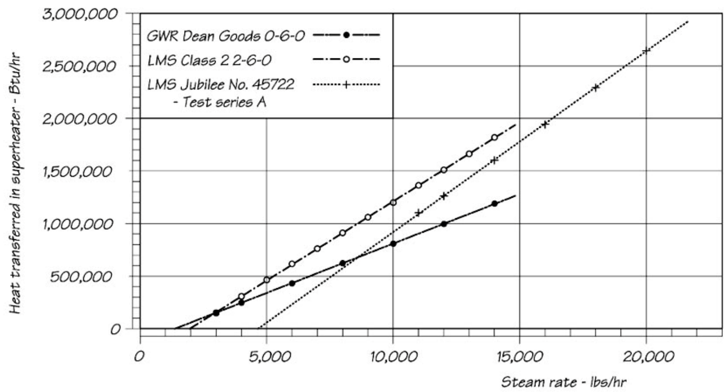
	A	B	R^2
LMS Class 4 2-6-0	208	478,530	0.9998
BR Class 4 4-6-0	188.95	606,204	0.9999

[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.

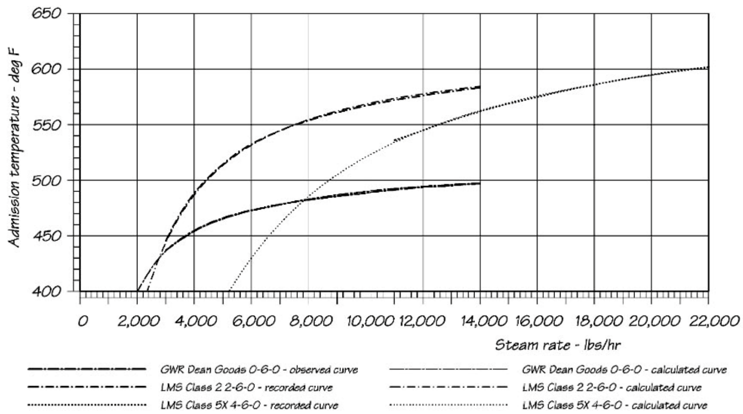
*F**rom PPT slide 13**: Usually appearing in locomotive test data is a curve or table recording the steam temperature obtained at certain steam rates. By assuming a constant dryness fraction and boiler pressure we may estimate the amount of heat the steam absorbed during its passage through the elements. 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.]

Slide 14



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)

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.

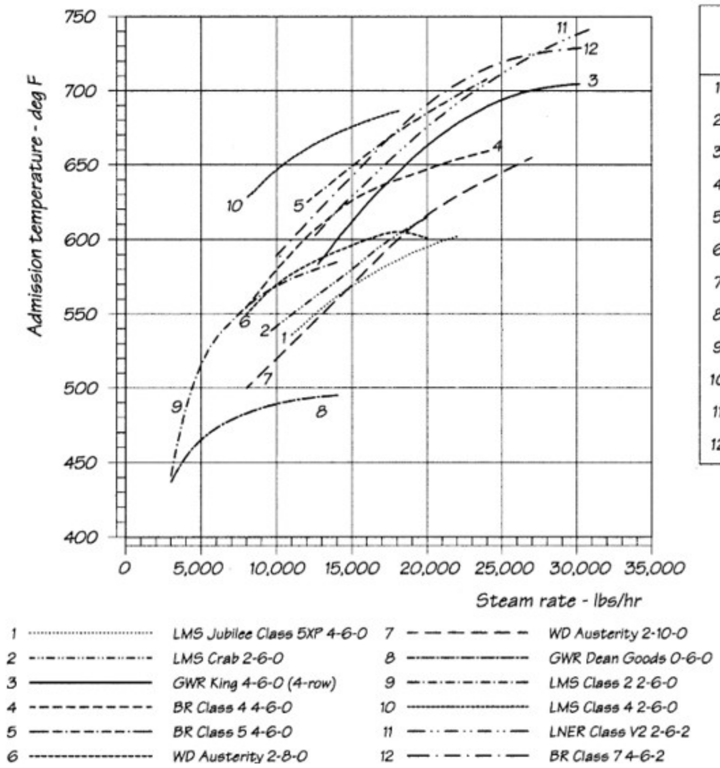
[12`] Half-return elements have four steam passes in the hot end with two passes in the front half of the flue. Conventional full return flues have four steam passes the full length of the flue.

[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.

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.



	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