The Physiology of the Locomotive Boiler

- A Peep!

ASTT Conference, Bury

Adrian Tester - 30th September 2017

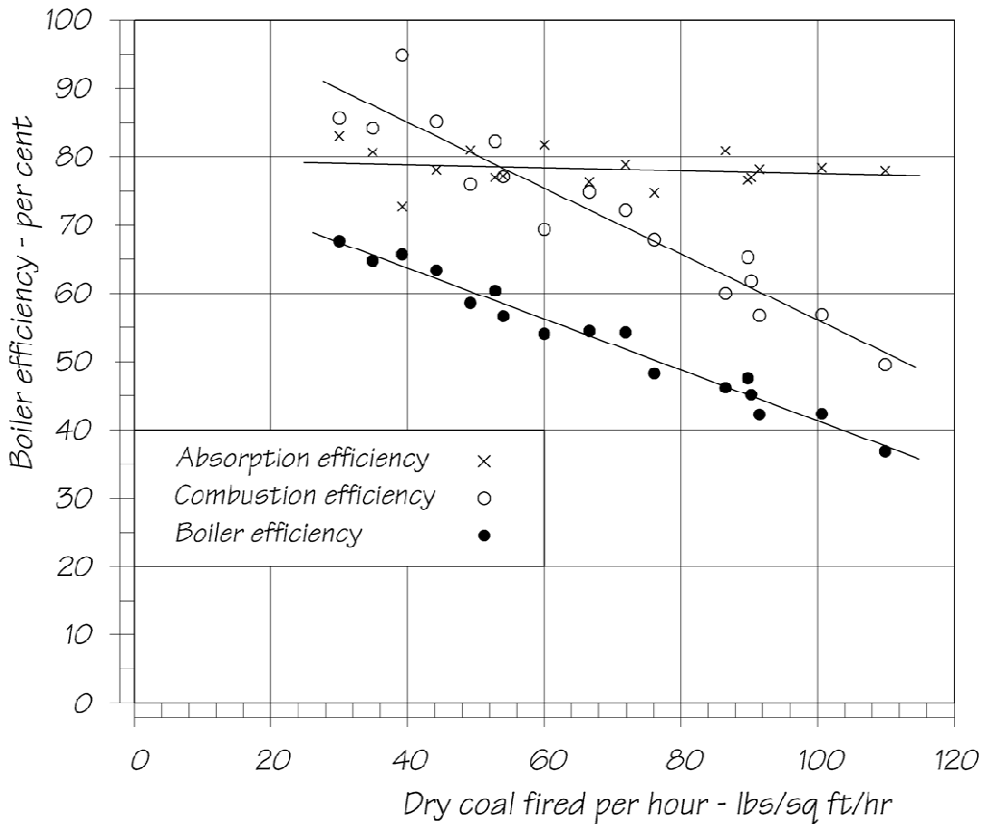


Fig. II.4 – Boiler efficiency curves for Pennsylvania Railroad E2a 4-4-2 N° 5266 tested at Altoona in 1905 – analysis by Lawford Fry

$$\text{Boiler efficiency} = \frac{\text{Absorption efficiency} \times \text{combustion efficiency}}{100 \times n}$$

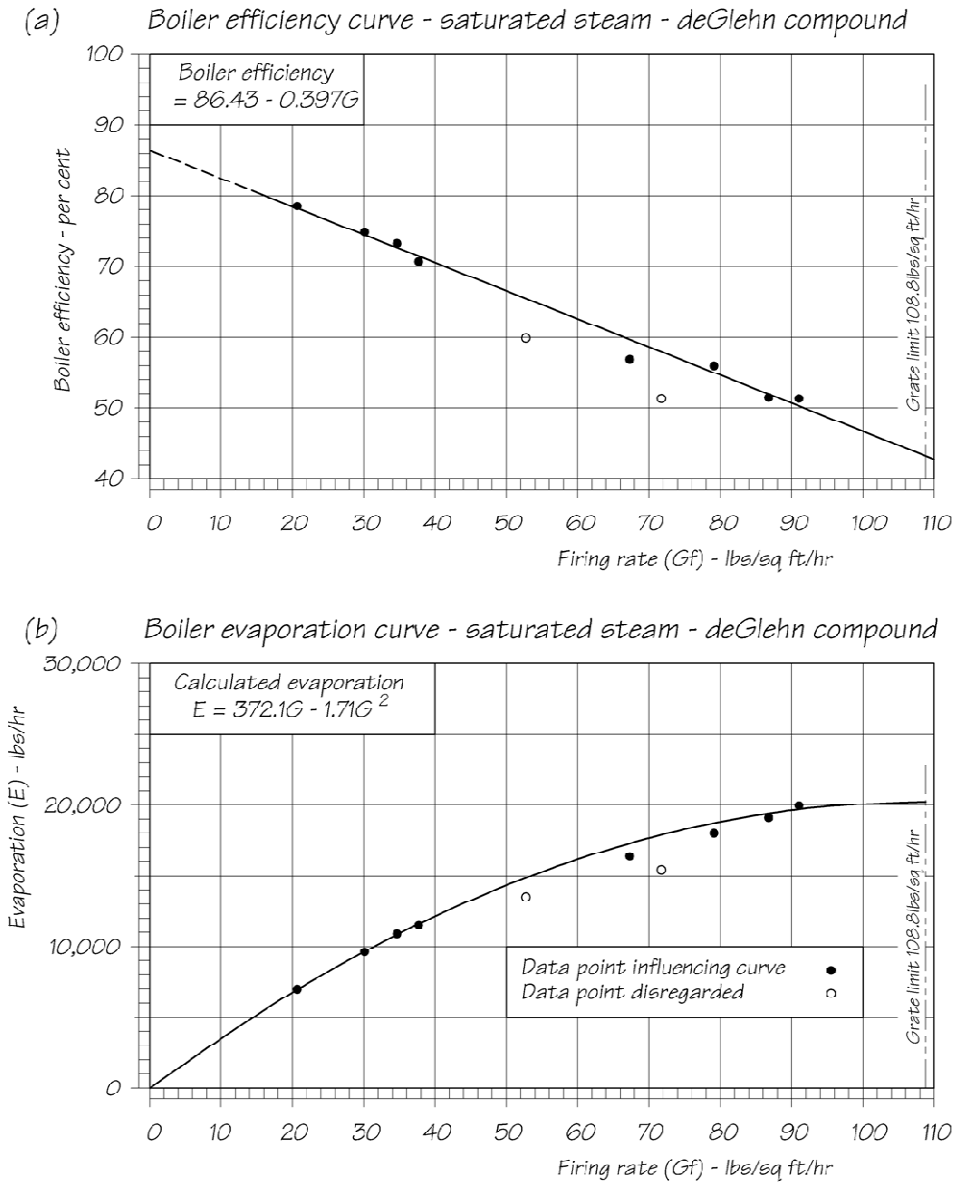
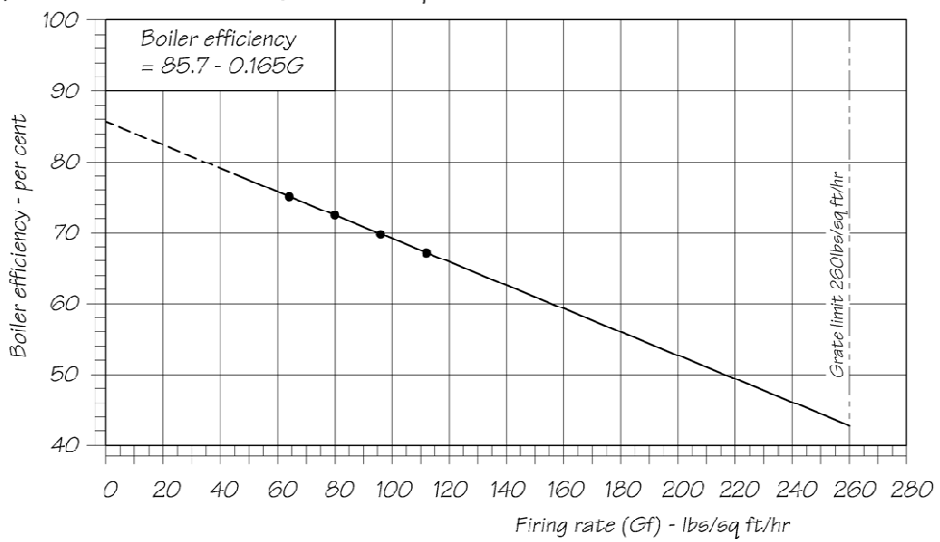


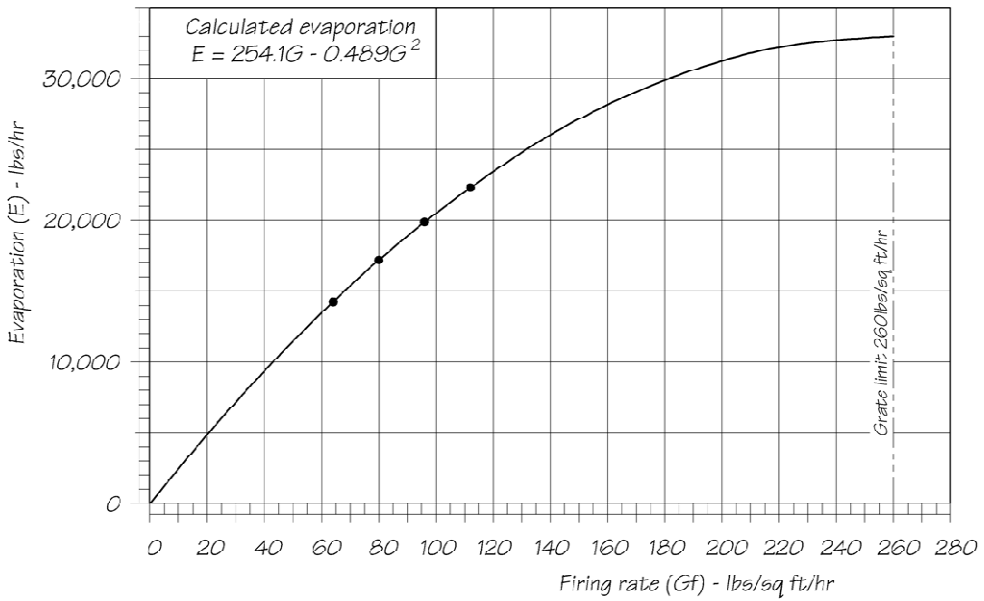
Fig. III.1 - Applying Mr Fry's method for determining the evaporation to a saturated boiler

Firing rate Wf - lbs/hr	*Firing rate Gf - lbs/sq ft/hr*	*Recorded steam rate - lbs/hr*	*Calculated evaporation E - lbs/hr*
690	*20.66*	*6,990*	*6,958*
1005	*30.10*	*9,634*	*9,651*
1157	*34.65*	*10,889*	*10,840*
1259	*37.71*	*11,514*	*11,600*
2247	*67.30*	*16,382*	*17,297*
3038	*91.00*	*19,977*	*19,701*
*1761**	*52.74*	*13,508*	*14,868*
*2395**	*71.73*	*15,430*	*17,892*
2641	*79.10*	*18,003*	*18,734*
2897	*86.76*	*19,115*	*19,412*

(c) Boiler efficiency curve - superheated steam LMS 2A boiler

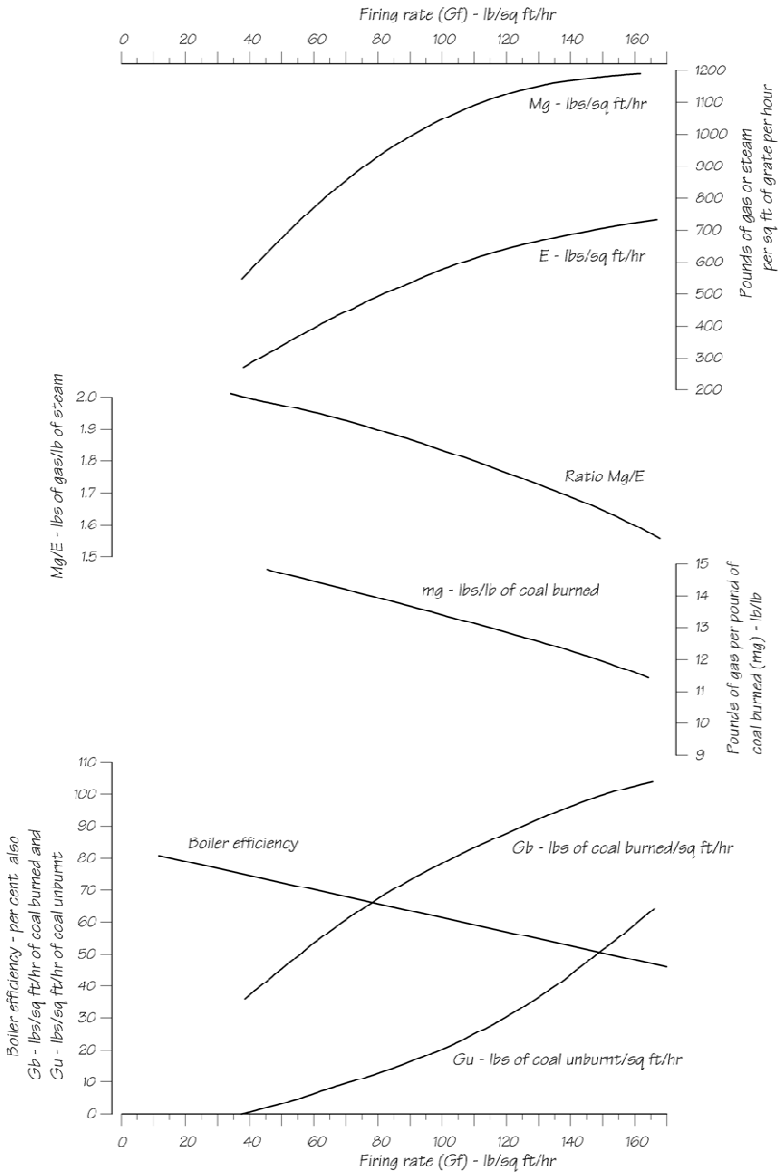


(d) Boiler evaporation curve - superheated steam LMS 2A boiler



Firing rate lbs/hr *Wf*	Firing rate lbs/sq ft/hr *Gf*	Measured steam rate lbs/hr	Calculated evaporation lbs/hr
2000	64	14,820	14,260
2500	80	17,445	17,200
3000	96	20,030	19,890
3500	112	22,370	22,330

Fig. III.1 – Applying Mr Fry's method for determining the evaporation to a superheated boiler



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. 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***

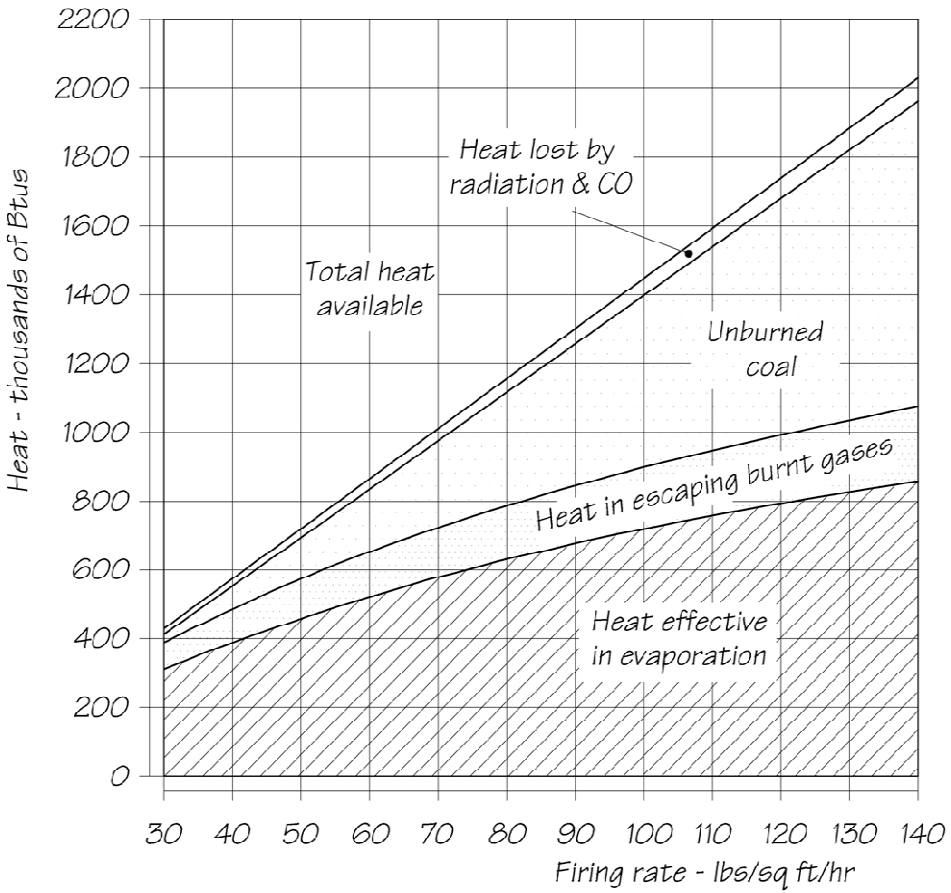


Fig. IV.10 - Approximate heat utilized and lost per square foot of grate per hour at different firing rates - St Louis Test Series 200 - Lake Shore & Michigan Southern Railway 2-8-0 N° 734

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, and another diagram present in the paper - containing a curve equivalent to *Gu* (similar to previous slide) - 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.....

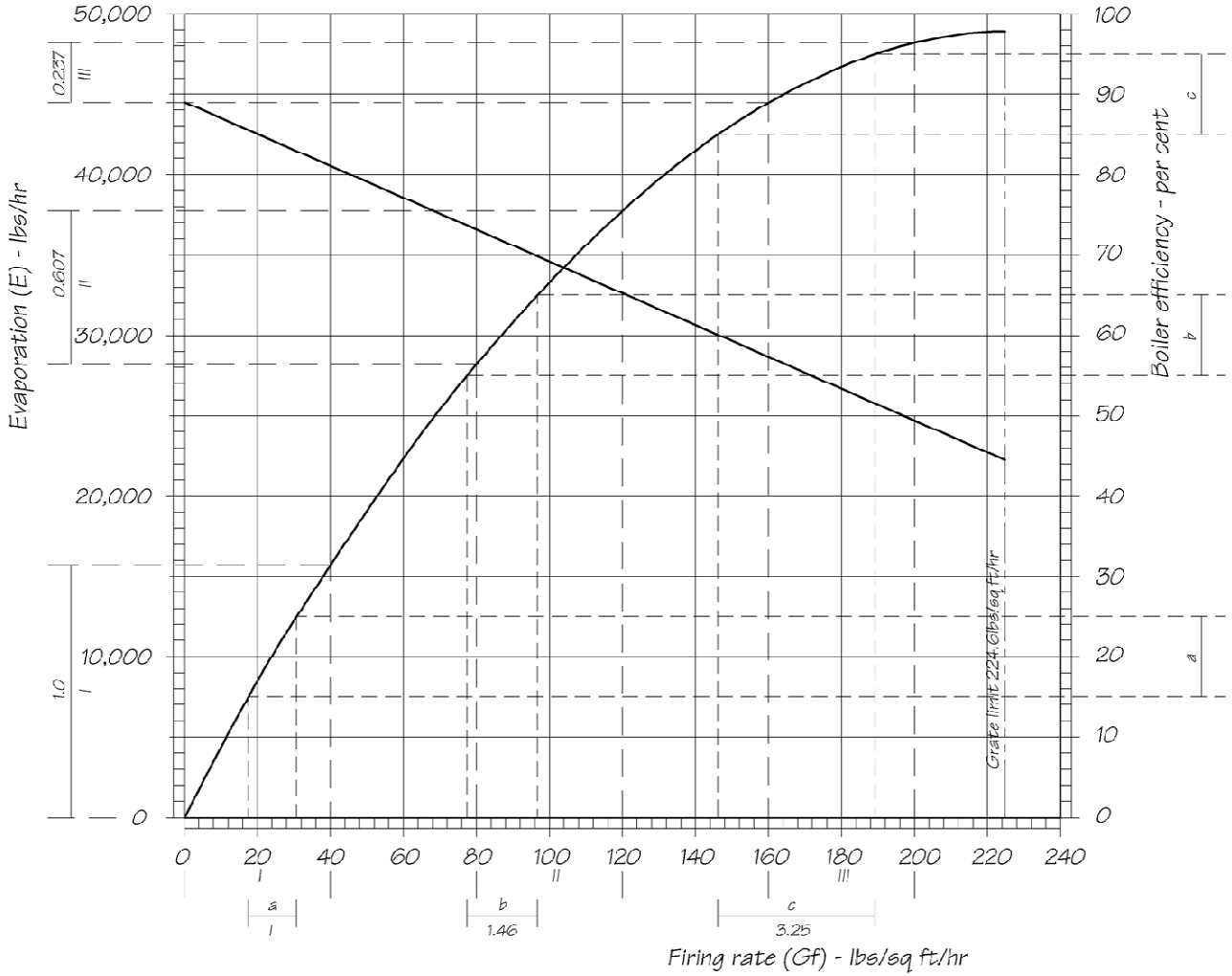


Fig. III.11 – Changes in evaporation and boiler efficiency as a function of the firing rate – PO 4-6-2 class N° 231-700

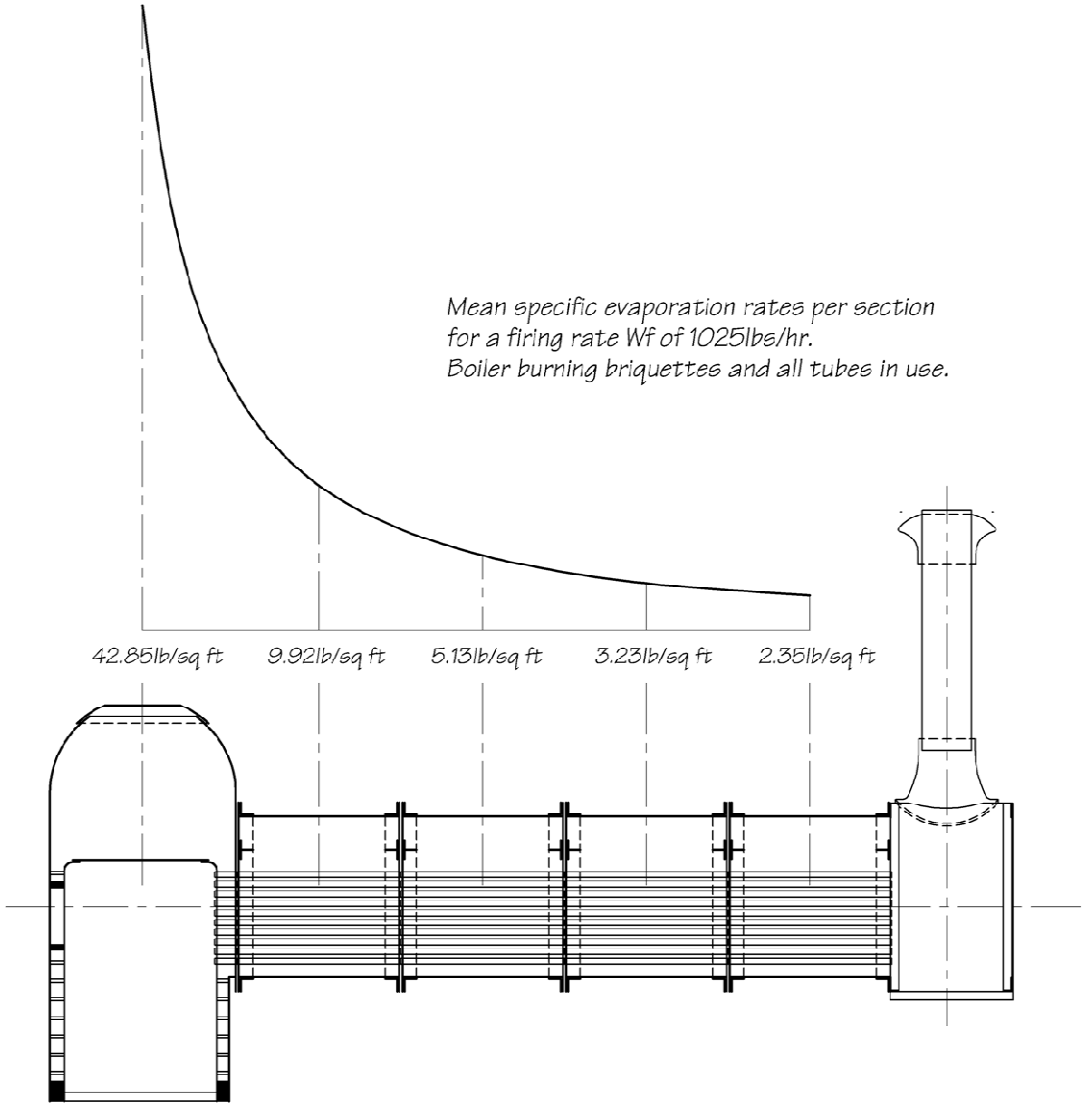


Fig. 1.7 – Experimental sectional boiler used on the Northern Railway of France

Table 1-7 - Progressive Evaporative Performance of Stephenson Goods Engine Boiler - Northern Railway of France

Fuel rate - lbs/hr W_f	Draught - ins of water	Firebox section - lbs/hr E_0	First tube section - lbs/hr E_1	Second tube section - lbs/hr E_2	Third tube section - lbs/hr E_3	Final tube section - lbs/hr E_4	Total evaporation - lbs/hr	Evaporation - lb/lb
Coke								
436.5	0.79	1530	996	430	228	128	3312	7.59
654.7	1.57	2018	1408	671	380	231	4708	7.19
727.5	2.36	2222	1879	931	528	337	5806	7.98
793.7	3.15	2229	1921	997	572	396	6115	7.71
771.6	3.94	1810	1892	1030	614	440	5786	7.50
Briquettes								
476.2	0.79	1806	964	445	240	147	3602	7.56
743.0	1.57	2356	1368	735	387	264	5110	6.88
923.7	2.36	2933	1969	1025	645	425	6997	7.58
1025.0	3.15	3291	1778	920	579	422	6990	6.82
978.8	3.94	2981	2499	1228	774	502	7984	8.16
Briquettes - half the tubes plugged at the firebox end								
388.0	0.79	1811	803	356	191	117	3278	8.45
610.7	1.57	2057	1138	550	308	187	4240	6.94
707.7	2.36	2710	1448	722	449	290	5619	7.94
793.6	3.15	2970	1624	845	475	334	6252	7.88
848.8	3.94	3058	1874	948	580	425	6885	8.11

NB:- The individual section evaporation figures do not always equal the total evaporation given in the eighth column. Figures taken from *The Steam Engine* by D K Clark

Professor Perry observed when the boiler was burning briquettes the efficiencies were all higher when half the tubes were stopped. He deduced a formula suggesting the heat take up is independent of the area of the heating surface and approximately the same fraction of the available heat is transferred from the gases into the water irrespective of the gas velocity:-

$\varepsilon = L \div (37.5d + L)$ Units may be feet or inches provided they are consistent

This was an influential and long-lived formula albeit after being subjected to modification by William Rowland.

Later in his book John Perry referred to the long known beneficial effect from placing artificial obstructions in boiler tubes. He admitted he had no real idea why fitting 'smoke retarders' were of benefit until ironically misled by a paper Thomas Stanton read to the Royal Society in 1897 and he deduced a formula for determining the efficiency of the tubular surfaces based on the *L/d* ratio. However, to make this formula applicable to duct sections other than a simple round tube of internal diameter *d*, he expressed it in terms of the hydraulic mean depth:-

$$\varepsilon = 1 - e^{-(cL/dm)}$$

where:-

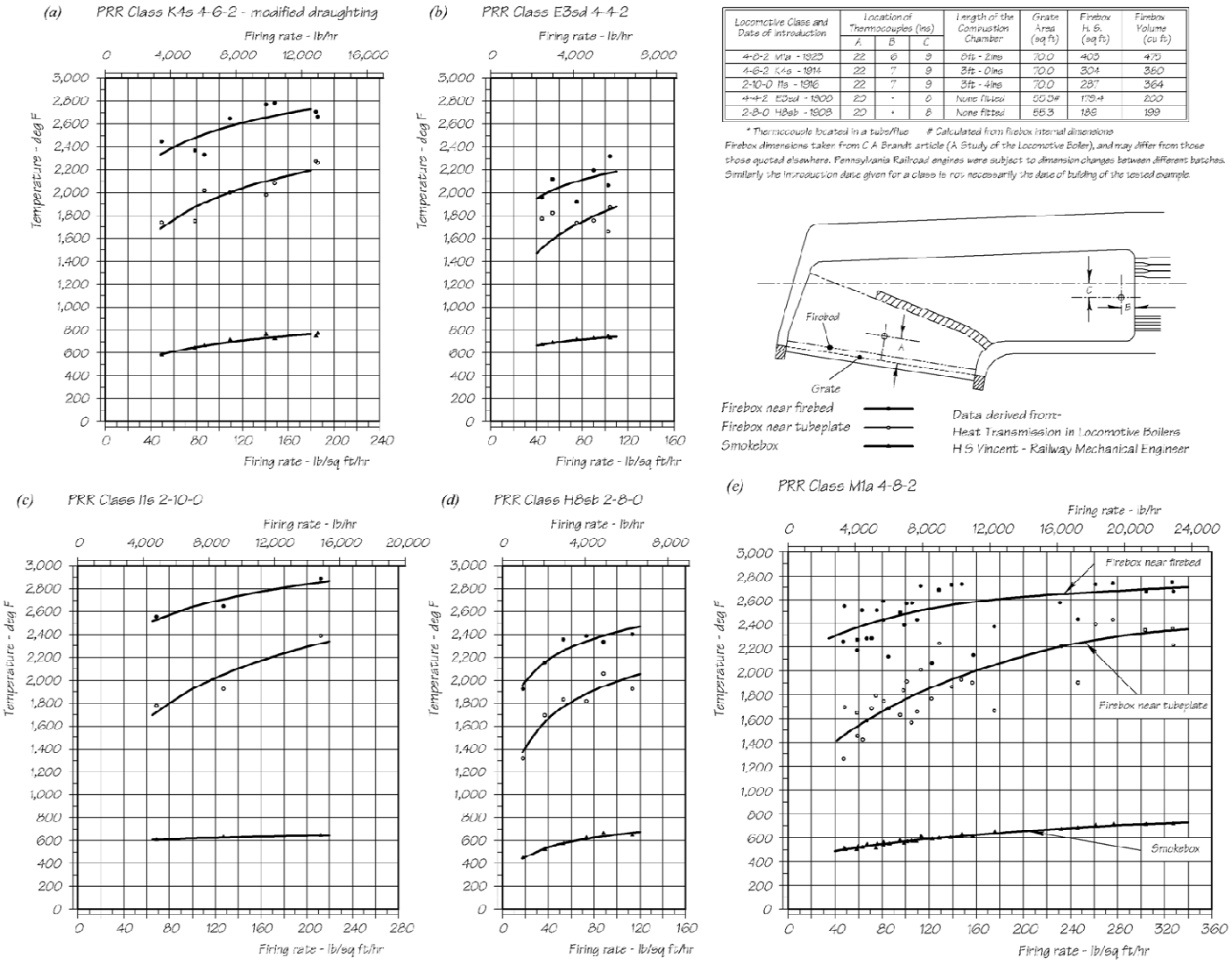
ε = tube efficiency or effectiveness

e = 2.71828 (*i.e.* base of Natural logarithms)

c = a constant

L = tube length - inches

dm = hydraulic mean depth - inches



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				
4-6-2 M1a - 1923	22	6	9	8ft - 2ins	70.0	405	470
4-6-2 K4s - 1914	22	7	9	8ft - 0ins	70.0	304	360
2-10-0 H1s - 1916	22	7	9	8ft - 4ins	70.0	287	364
4-4-2 E3sd - 1900	20	-	6	None fitted	66.3#	179.4	200
2-8-0 H8sb - 1908	20	*	8	None fitted	66.3	186	199

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

Firebox dimensions taker from C A Brandt article (A Study of the Locomotive Boiler), and may differ from those 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.

Fig. IV.1 – Example firebox and smokebox temperatures recorded in PRR classes tested at Altoona

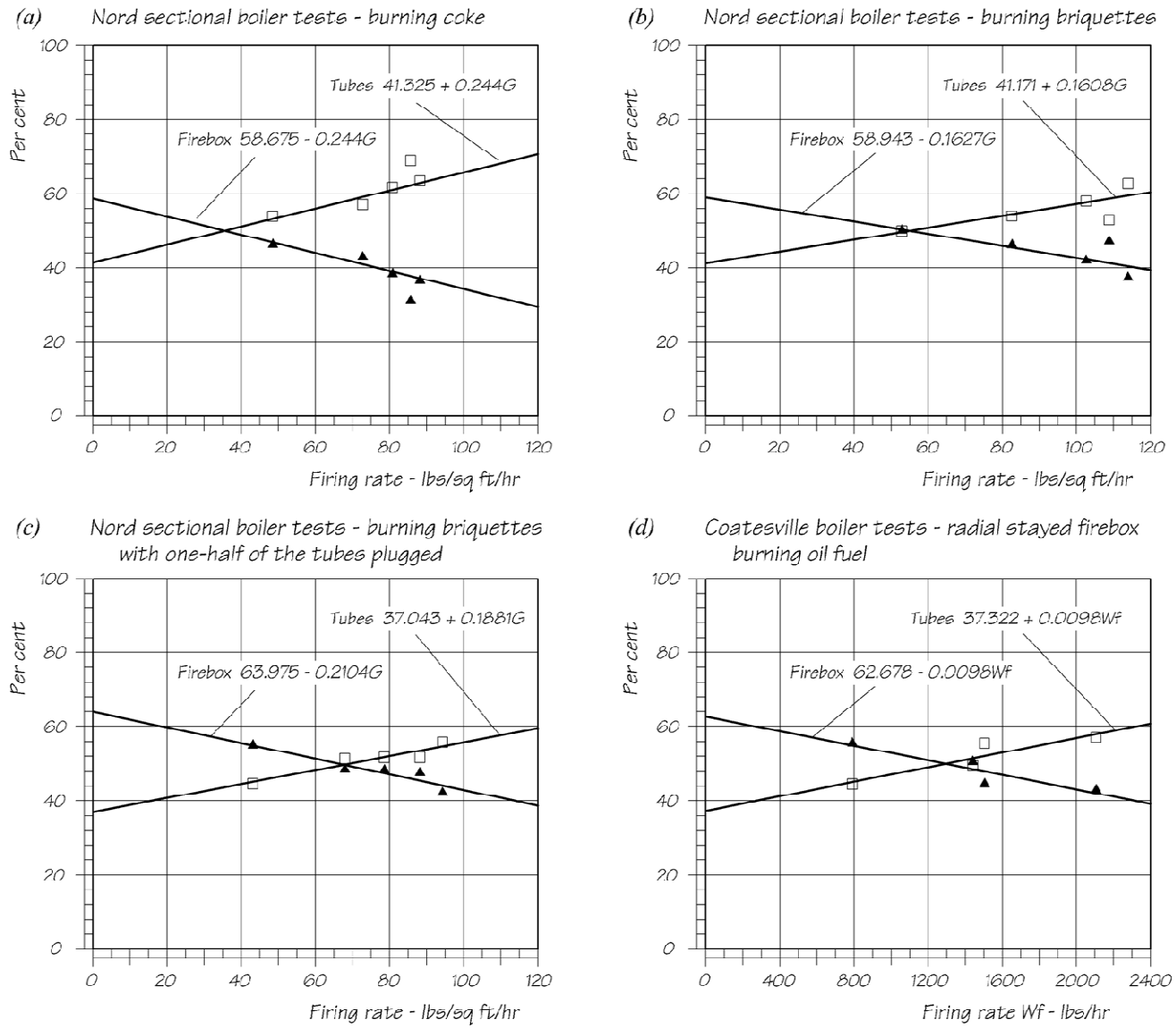


Fig. IV.3 – Observed change in heat apportionment between direct and indirect surfaces with rise in steam output – Nord and Coatesville boiler tests

Firebox: The quantity of heat absorbed in the firebox is the difference between the quantity of heat leaving the 'box and that supplied by the combustion of the fuel burned plus the quantity of heat present in any pre-heated combustion air. The latter was not usual in locomotives. The quantity of heat leaving the firebox is calculated from the product of the weight of the combustion gases, their specific heat and the firebox exit temperature. The portion of the total heat liberated in and absorbed by the firebox is sometimes referred to as the *furnace absorption efficiency*.

As there is no energy loss, *firebox effectiveness* is a more correct term.

$$\text{Firebox effectiveness} = \frac{\text{Heat absorbed in firebox}}{\text{Heat released}}$$

and since we are considering the firebox only this may be rewritten:-

Heat absorbed = Heat released - Heat escaping in exhaust gases

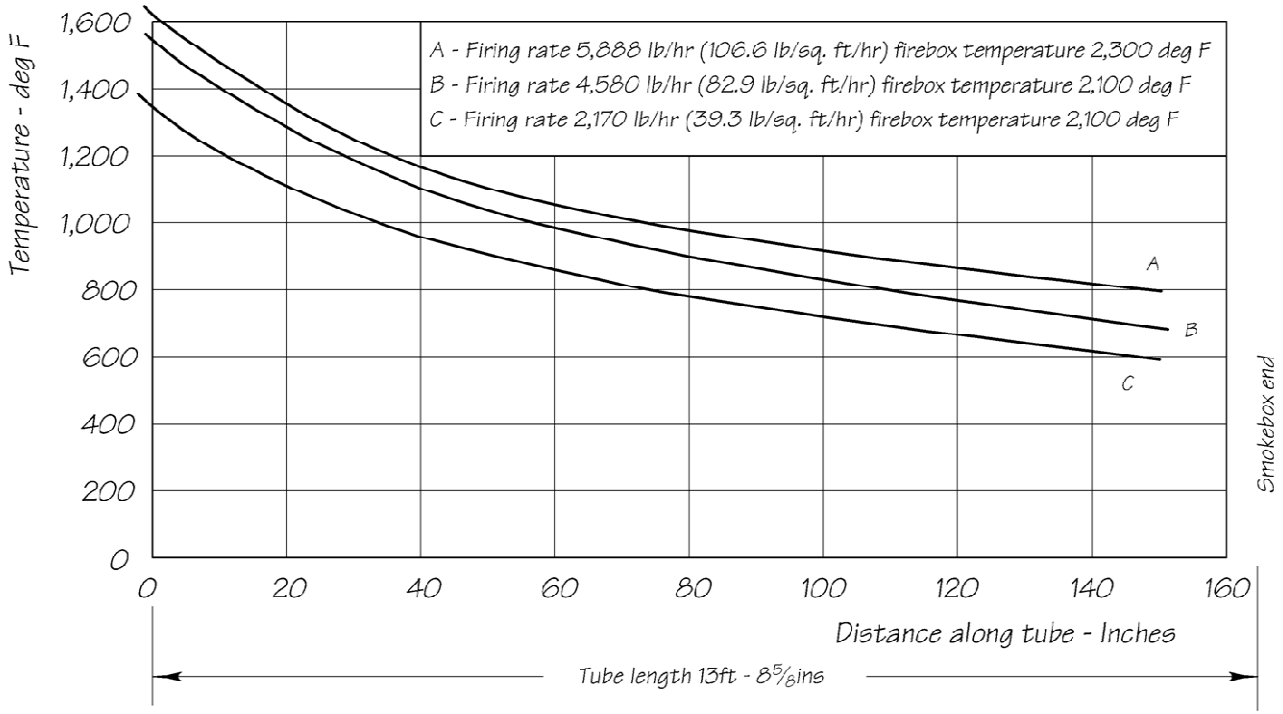
Barrel: The heat *Q*, absorbed within the boiler barrel is found from:-

$$Q = U \times A \times \Delta t$$

where *U*is the overall heat transfer coefficient between the gas and the water known as the *U*-value; *A* is the total surface area of the tubes and Δt is the temperature difference.

The *U*-describes how easily heat passes from the gas into the tube, then how easily it crosses the tube wall before finally how easily it enters the water. Of these three the dominant one is the gas-side transfer. It so dominant the other two may be largely ignored. The factor determining the *U*-value is the velocity of the gas raised to the power of 0.8, while for any boiler and gas flow – gas velocity is determined by tube diameter and tube number.

The drop in gas temperature Δt as it passes along the tubes is allowed for by using a relationship known as the logarithmic mean temperature difference (LMTD) – but we do not need to explore that here.



The temperature drop, in any given boiler, or tube system within it, varies with change in boiler output but the curves follow a similar profile. This is seen in the above diagram which records the drop in gas temperature as it negotiated the tubes of a PRR Atlantic. can deduce from this diagram that once the barrel has a reasonable tubular area, adding more has little real effect. lengthening the tubes from 10ft to 12ft - 6ins lowers the gas temperature by about 70-80°F - or roughly a gain of 18lbs/hr in steam production for every 1,000lbs/hr of gas passing through the boiler - but the barrel is 25 per cent heavier.

Fig. V.33 - Boiler tube gas temperature profiles for different firing rates - PRR E6s 4-4-2 N° 89

The heat transfer in both barrels is given by:-

$$Q = U \times A \times \Delta t$$

With 49 tubes plugged out of 221 the area *A* was reduced to become:-

$$(221 - 49) \div 221 = 0.778$$

However, reducing the number of tubes by 22.2% increased the gas velocity by:-

$$1 \div 0.778 = 1.285$$

Consequently the *U*-value increased due to the higher gas speed to become:-

$$1.285^{0.8} = 1.222$$

Hence the revised heat transfer $Q = 0.778 \times 1.222 \times \Delta t$

$$= 0.95 \times \Delta t$$

Thus the barrel steam production was 95 per cent of that from the standard number of tubes.

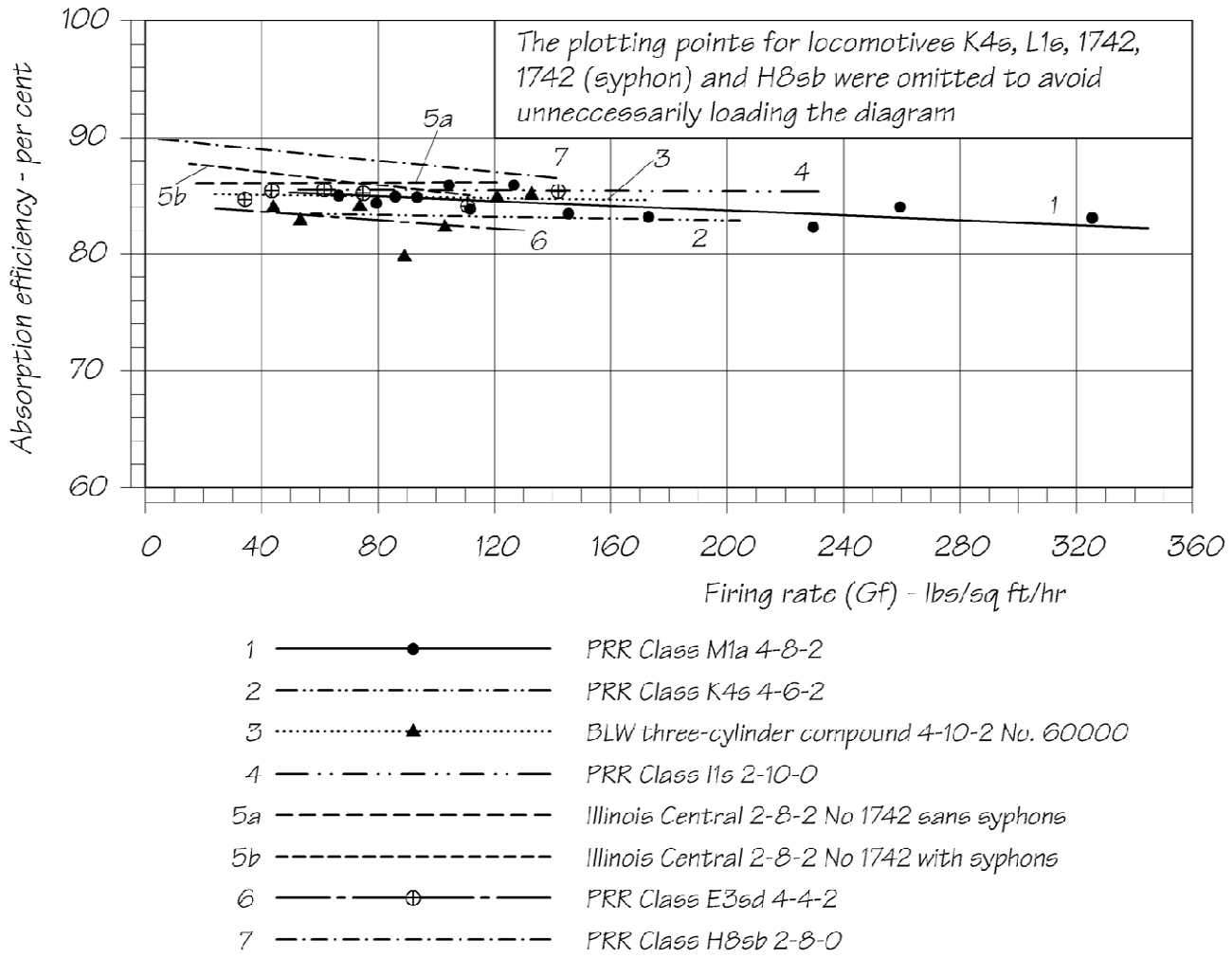
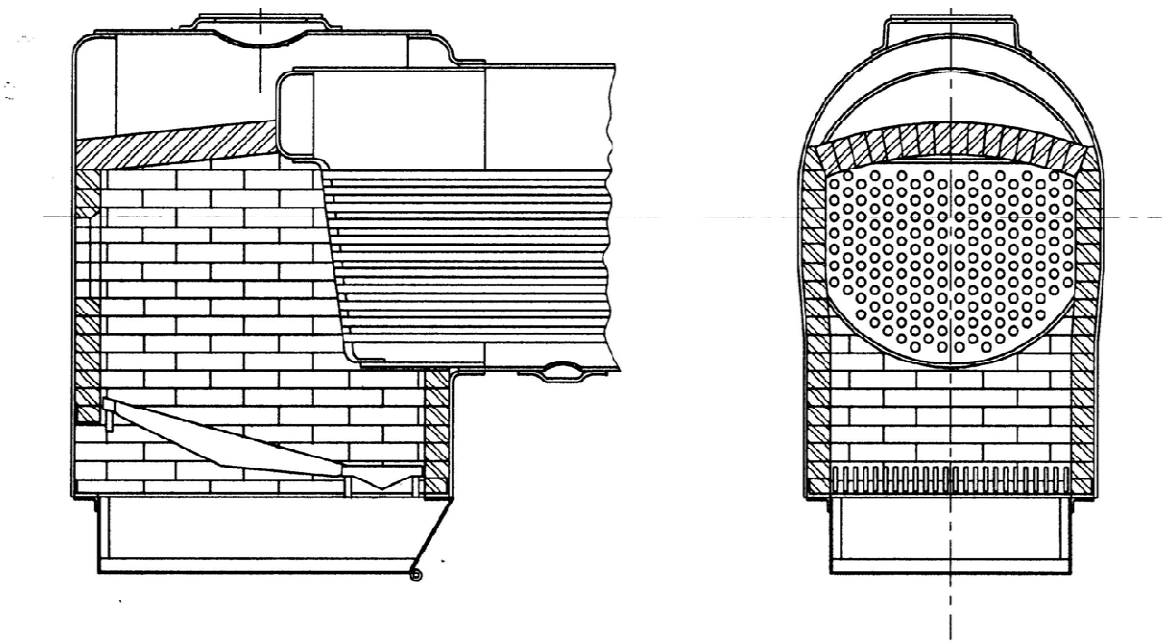


Fig. III.18 – Comparison in the absorption efficiency for a selection of American boilers – A Chapelon La Locomotive à Vapeur

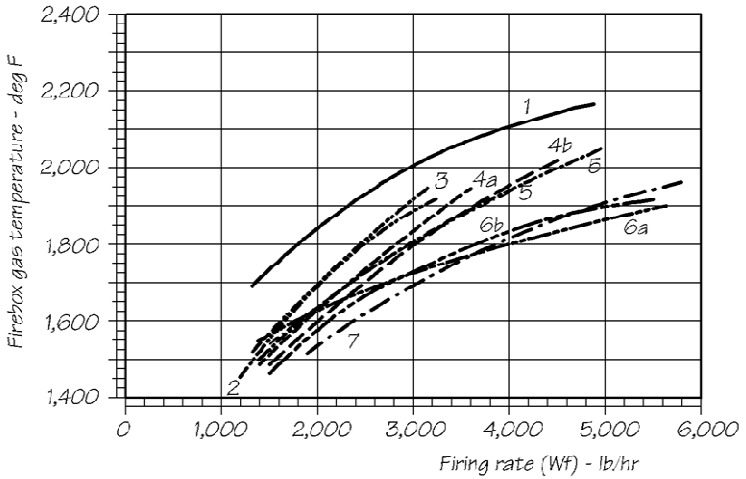


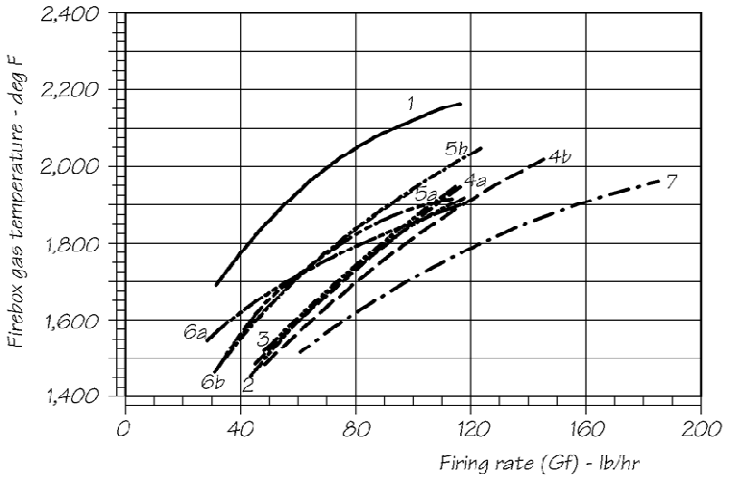
Locomotive	Line	Distance - km	Load - tons	Ton-km	Coal consumption		Water consumption	
					Total - kg	per ton-km	Total - kg	kg/kg
19	Budapest - S Tarjau	123.4	380.0	46892	2,471	0.0527	11,550	4.67
104	- do -	123.4	352.8	435.4	2,425	0.0557	11,490	4.73
19	Budapest - Miskolcz	182.6	301.8	550.0	3,360	0.0611	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

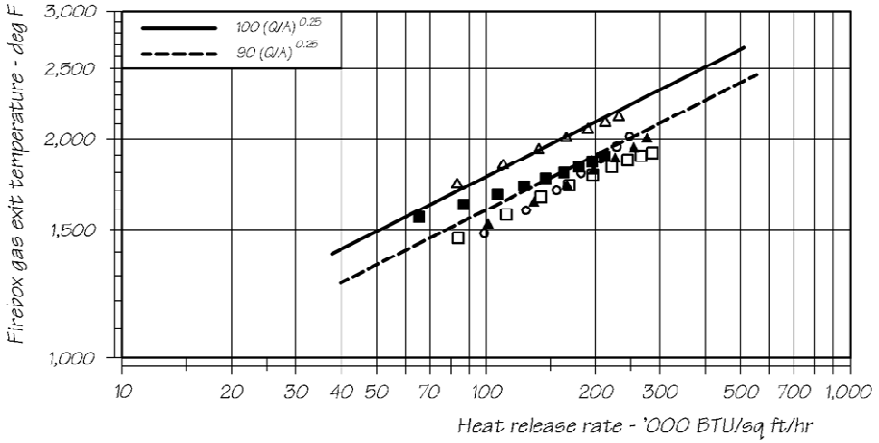
Fuel - brown coal from S. Tarjau mines

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







1 — BR Britannia Class 7 4-6-2

2 — LMS Crab 2-6-0 No. 42725

3 — LMS Crab 2-6-0 No. 42824

4a — LMS Jubilee Class 5XP 4-6-0 No. 45722 - original chimney

4b — LMS Jubilee Class 5XP 4-6-0 No. 45722 - modified single chimney

5a — BR Class 9 2-10-0 No. 92050 - plotted from tabulated results

5b — BR Class 9 2-10-0 No. 92050 - scanned from report curve

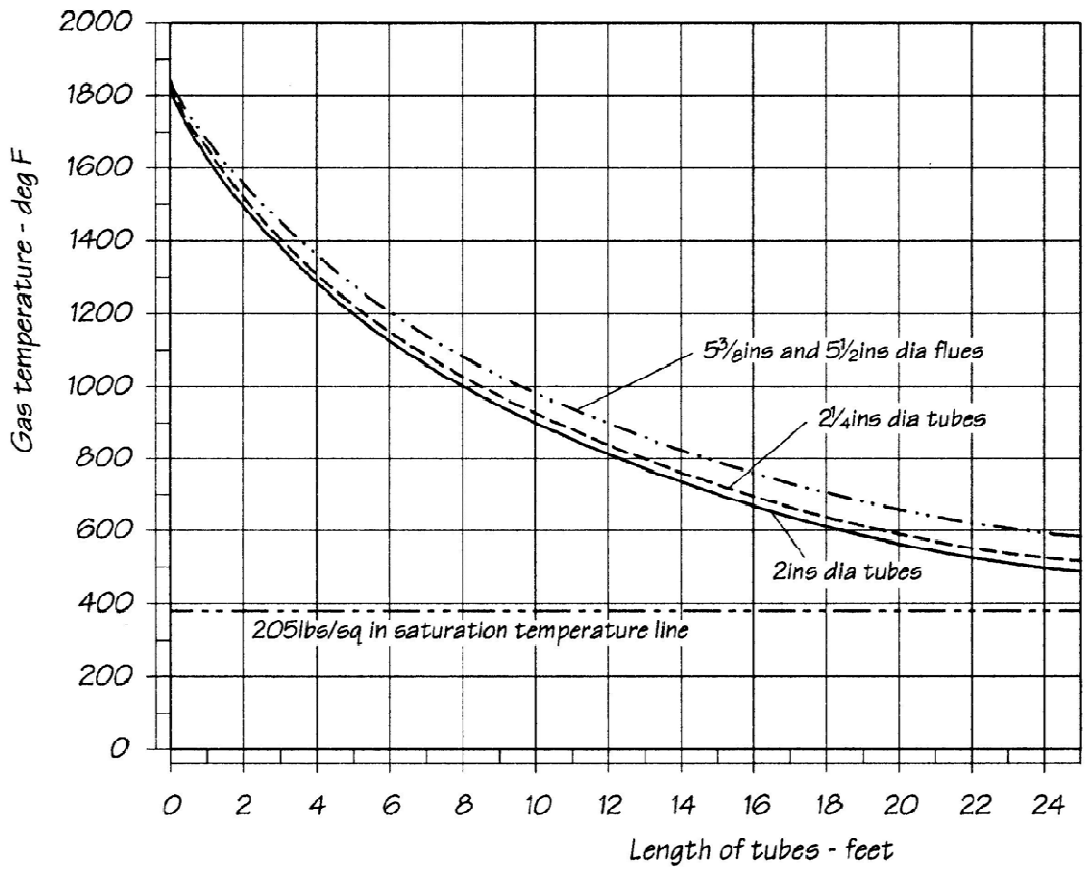
6a — SR Merchant Navy Class 8 4-6-2 No. 35022 - with syphons

6b — SR Merchant Navy Class 8 4-6-2 No. 35022 - without syphons

7 — LMS Royal Scot Class 6 4-6-0 No. 46165 revised double chimney 3¾ins caps

All locomotives burning Blidworth coal

Fig. V.19 - Firebox temperatures - Rugby Testing Station



Average temperature readings were plotted shewing the temperature drop of the gases in 2 and 2¼ins dia small tubes and 5⅜ and 5½ins flue when passing from firebox to smokebox. Rate of combustion - 100 to 115lbs/sq ft/hr of grate

Obtaining the value of the index for the 2¼ins tube:-

$t_f = 1810°F$, t_x at flue exit $= 520°F$ and $t_w = 389°F$

The plotted curve is converted into a straight line by using Natural logarithms, from which the gradient is obtained:-

$$\frac{dy}{dx} = \frac{\log_e (t_x - t_w) - \log_e (t_f - t_w)}{\text{length of tube x ft}} = k$$

$$= \frac{\log_e (520 - 389) - \log_e (1810 - 389)}{\text{length of tube x ft}} = k$$

$$= \frac{\log_e 131 - \log_e 1421}{25} = \frac{4.875 - 7.259}{25}$$

$$k = \frac{-2.384}{25} = -0.0954$$

$$t_x = t_w + [(t_f - t_w) \times e^{-kx}]$$

$$t_x = 389 + 1421e^{-0.0954x}$$

Fig. V.2 - Tube and flue temperature curves

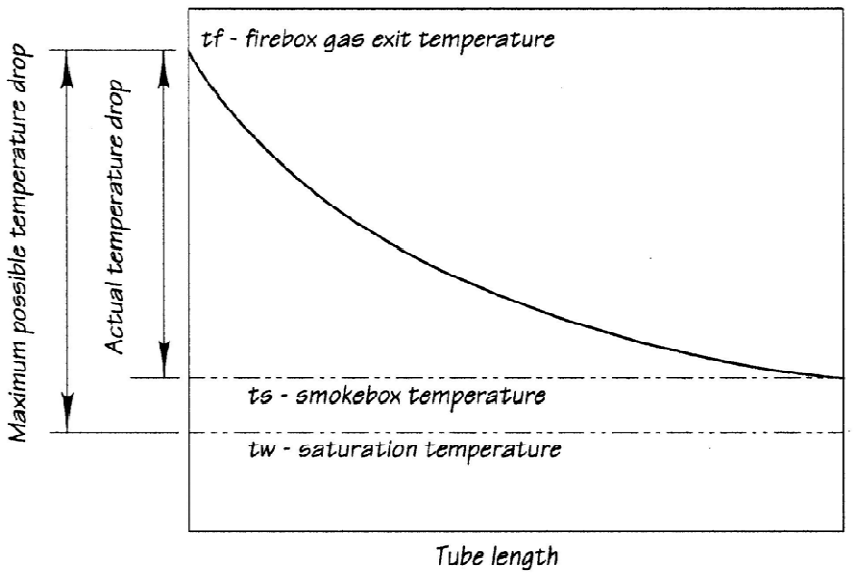


Fig. V.5 - Boiler tube effectiveness

For a gas of specific heat Cp Btu/lbs/°F, flowing at Mg lbs/hr, the maximum possible heat transfer of Q Btu/hr from the gas to the water is given by:-

$$Q = Mg \times Cp \times (tf - tw)$$

whereas, because the gas is not cooled to the saturation temperature the actual heat transfer:-

$$Q' = Mg \times Cp \times (tf - ts)$$

If ε is the tube effectiveness then $Q' = Q \times \varepsilon$

Hence substituting for Q' and cancelling:-

$$\frac{Q \times \varepsilon}{Q} = \frac{Mg \times Cp \times (tf - ts)}{Mg \times Cp \times (tf - tw)} \qquad \varepsilon = \frac{(tf - ts)}{(tf - tw)}$$

However tube effectiveness may be obtained without any temperatures:-

$$\varepsilon = \frac{(tf - ts)}{(tf - tw)} = \frac{[(tf - tw) - (ts - tw)]}{(tf - tw)}$$

$$= 1 - \frac{(ts - tw)}{(tf - tw)}$$

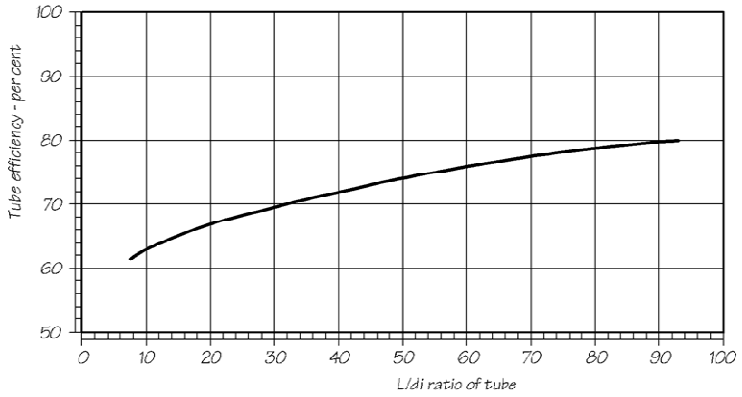
However from figure V.2, and considering ts = tx

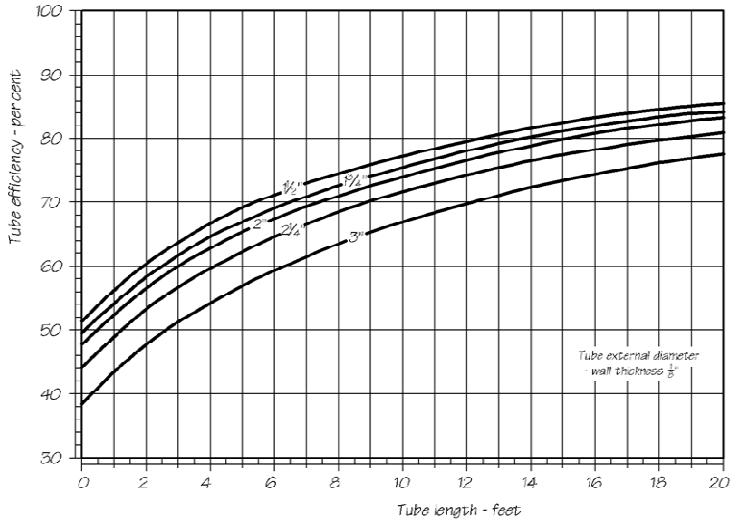
$$ts = t_w + [(t_f - t_w) \times e^{-kx}]$$

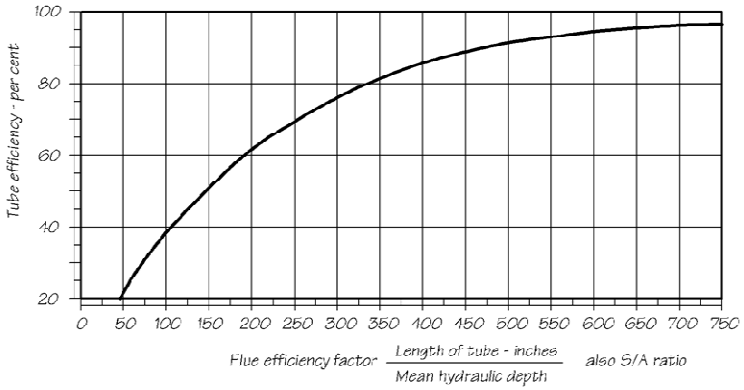
$$(ts - t_w) = [(t_f - t_w) \times e^{-kx}]$$

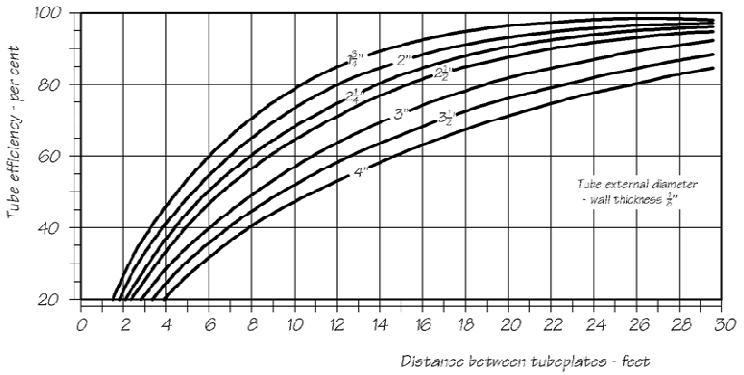
Substituting for (ts - tw) and cancelling:-

$$\varepsilon = 1 - e^{-kx}$$



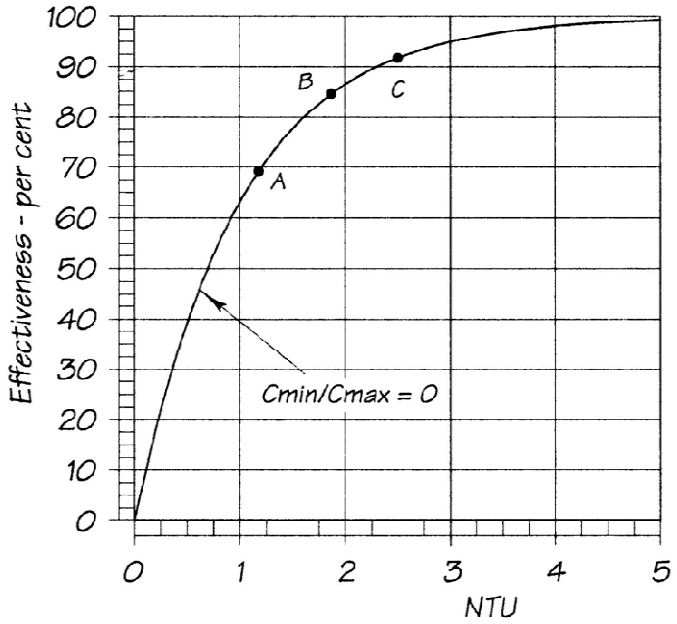






Left ***Fig. V.6 – Tube efficiency according to William Rowland – 1916***

Above ***Fig. V.7 – Tube efficiency from the paper – The Locomotive Boiler by C A Brandt – ASME 1940***



Tube proportions	Free gas area	L/d ratio	Tube dia.	Heating surface
A	7.5%	75	2"	528 sq. ft
B	15%	100	1¾"	1431 sq. ft
C	20%	125	1½"	2437 sq. ft

Tube proportions taken from table VIII-2

Fig. VIII.8 - NTU - Effectiveness comparison for three combinations of small tubes in a boiler barrel

Nowadays we use a concept known as the 'number of transfer units' or *NTU*. The larger the number of transfer units the more effective is the heat exchange. It is obtained by combining and rearranging the two earlier equations:-

$$Q = mg \times Cpg \times (tgi - tgo) = U \times A \times LMTD$$

Rearranging:-

$$NTU = \frac{(tgi - tgo)}{LMTD} = \frac{U \times A}{mg \times Cpg}$$

For a boiler tube:-

$$\varepsilon = \frac{(tgi - tgo)}{(tgi - tsat)} = 1 - e^{-NTU}$$

As the *NTU* factor is formed from the overall heat transfer coefficient, gas-swept tube area, gas flow rate and its specific heat, it means this formula may be used to *design* a tube system. Each of these four factors may be calculated for a proposed barrel layout boiler. Knowing the *NTU* then enables the tube effectiveness to be established and from that gas exit temperature may be predicted along with the evaporation.

Using these heat transfer formulae, we may obtain an insight into the thermal and frictional behaviour of different *L/d* ratios.

The graph gives three sample results obtained from a series of 54 simulations on a saturated boiler barrel based on a constant gas flow of 22,000lb/hr, and entering gas temperature of 2200°F. Boiler pressure taken as 175lbs/sq in with a grate area of 21sq ft. These simulations compared three tube diameters 1½, 1¾ and 2ins, for *L/d* ratios of 75, 100 and 125 (*S/A* ratios of 300, 400 and 500), and free gas areas of 7½% up to 20% of grate area in 2½% increments.

Table IX-1 - Fraction of the total heat in the steam gained in the superheater - for different steam temperatures - boiler pressure 180lbs/□ inch and entering steam 0.98 dry

Steam temperature °F	Enthalpy supheated steam Btu/lb	Enthalpy saturated steam Btu/lb	Enthalpy of feed Btu/lb	Enthalpy rise within boiler Btu/lb	Enthalpy rise within superheater Btu/lb	Total enthalpy rise Btu/lb	Fraction of total enthalpy rise occurring in superheater
450	1241	1182	30	1152	59	1211	4.9%
500	1267	1182	30	1152	85	1237	6.9%
550	1296	1182	30	1152	114	1266	9.0%
600	1322	1182	30	1152	140	1292	10.8%
650	1348	1182	30	1152	166	1318	12.6%
700	1374	1182	30	1152	192	1344	14.3%
750	1400	1182	30	1152	218	1370	15.9%
800	1426	1182	30	1152	244	1396	17.5%
850	1451	1182	30	1152	269	1421	18.9%

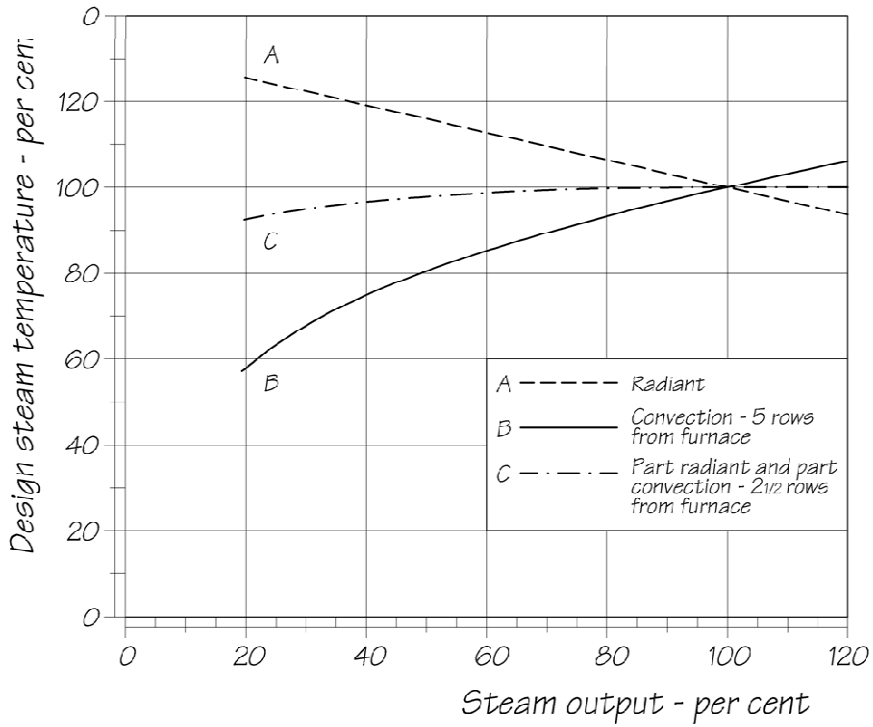
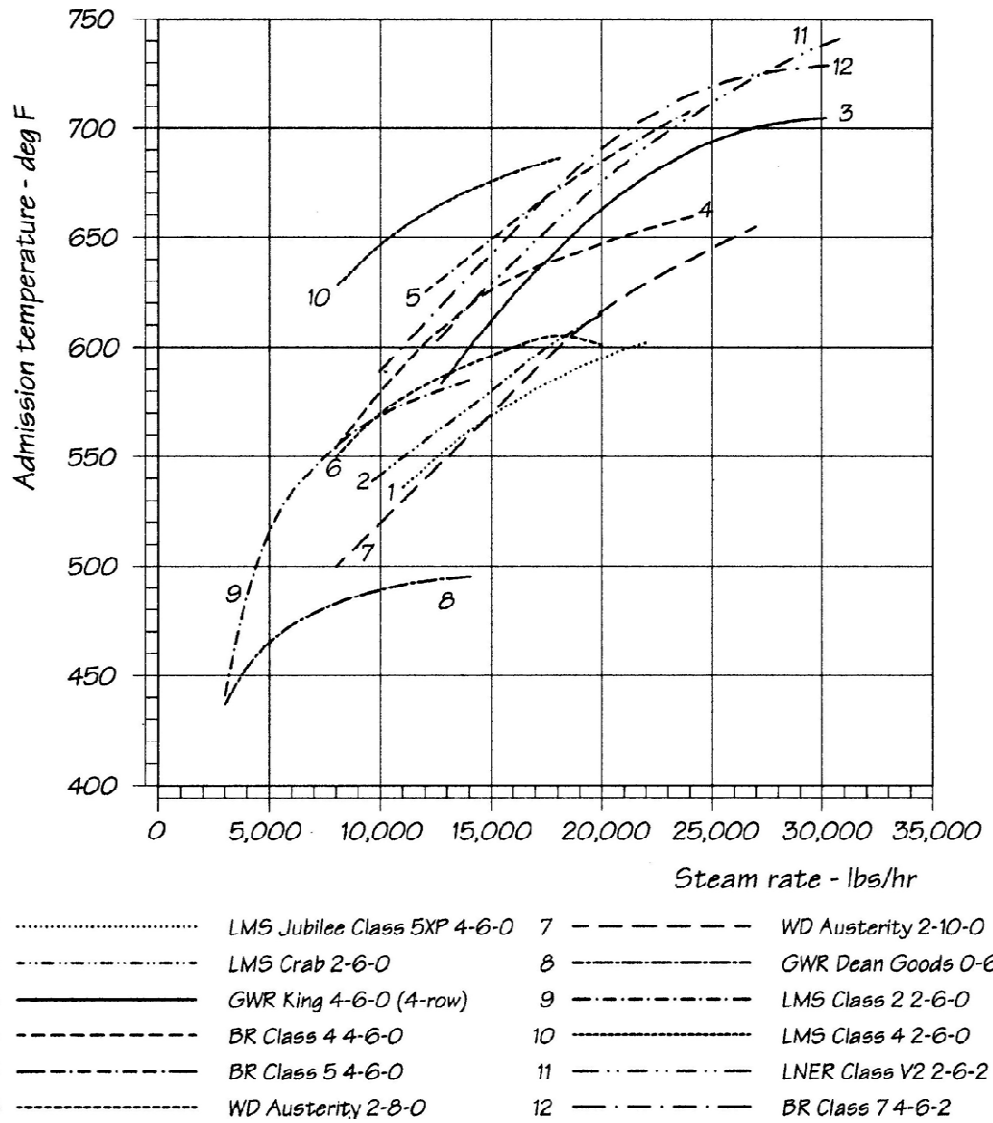


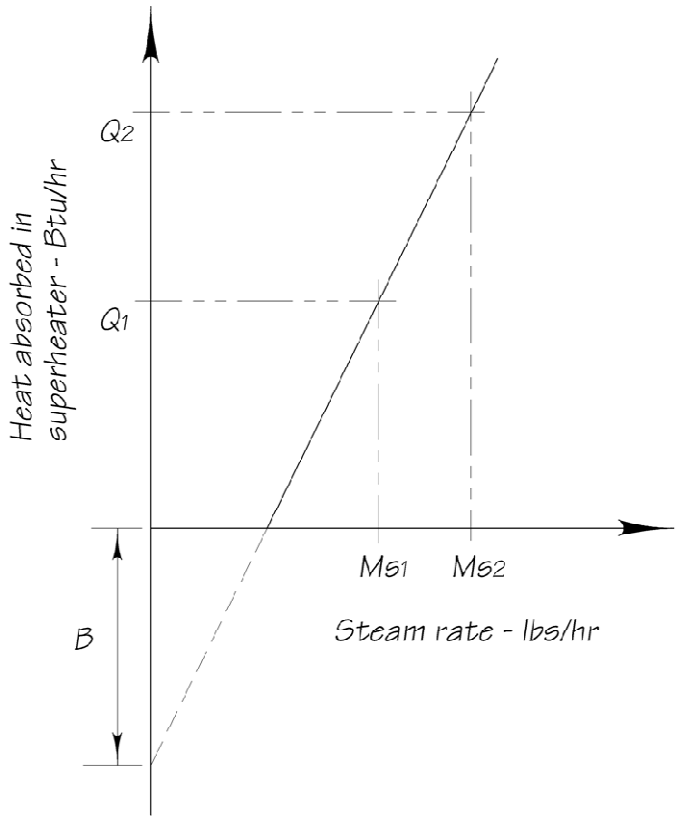
Fig. IX.4 – Superheater characteristics or the difference between radiant and convective superheaters



	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



	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

Fig. IX.30 – Linear relationship describing the heat absorbed in the superheater

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 energy 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:-

$$Q = (A \times Ms) - B$$

In this relationship *A* is the gradient of the curve obtained from:-

$$A = \frac{Q2 - Q1}{Ms2 - Ms1}$$

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 when superheat will start to appear. This is as soon as *Q* zero in 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.

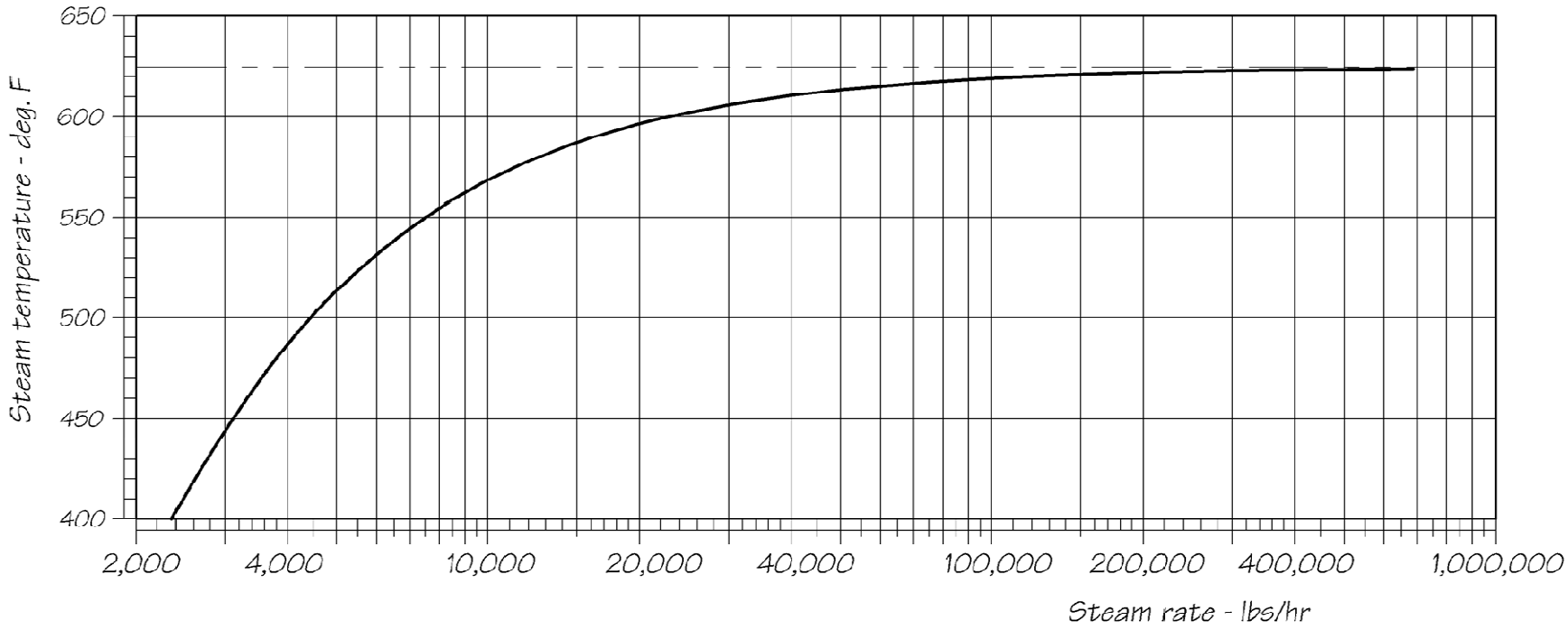
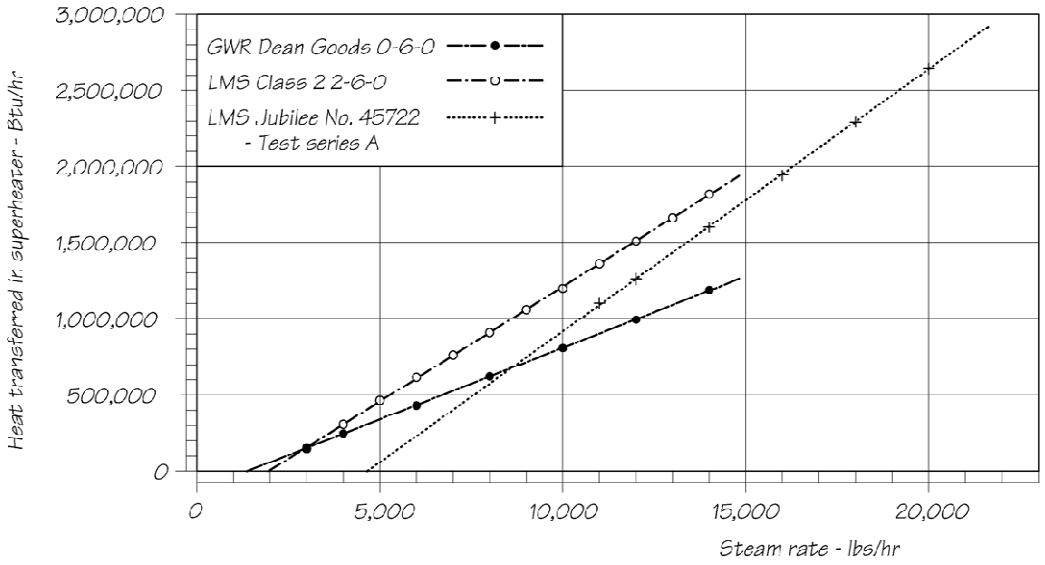
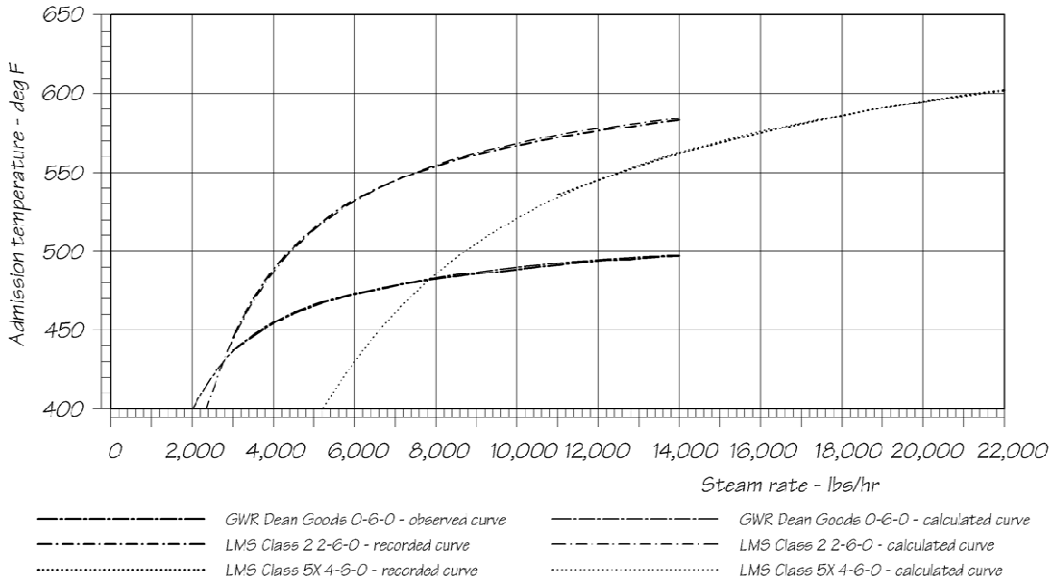


Fig. IX.32 – Superheater characteristic curve for a LMS/BR Class 2 2-6-0 superheater

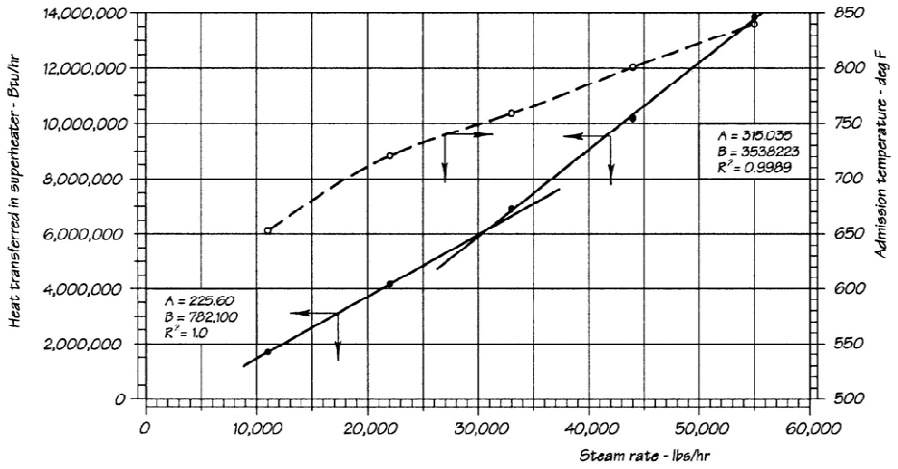


Upper – ***Fig. IX.34 – Rate of heat absorption in different superheater designs assuming a constant saturated steam inlet condition of 0.98 dry***

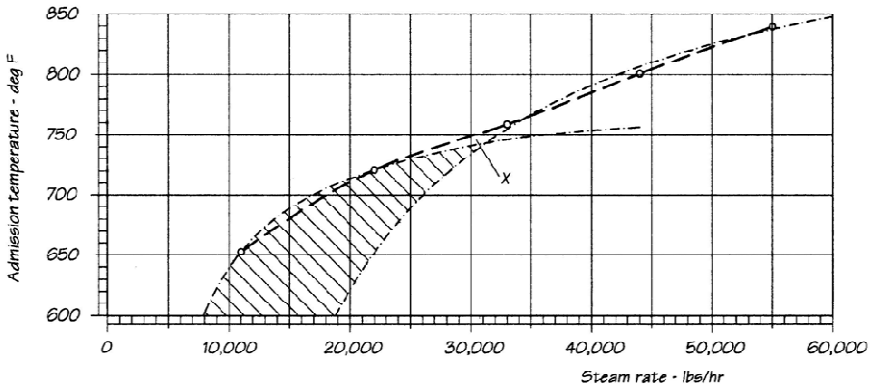


Lower – ***Fig. IX.35 – Comparison between the recorded steam temperature ants its calculated value for three engines in each case assuming a constant saturated steam inlet condition (0.98 dry)***

Steam rate - lbs/hr	Steam temperature - °F	Enthalpy - Btu/lb	Enthalpy rise - Btu/lb	Heat absorbed - Btu/hr
11,000	653	1347.3	154.51	1,699,610
22,000	721	1382.85	190.06	4,181,320
33,000	759	1402.61	209.82	6,924,060
44,000	801	1424.43	231.64	10,192,160
55,000	840	1444.7	251.91	13,855,050



Steam temperature profile curve and heat transfer, steam inlet conditions 0.99 dry - SAR Class 26



Steam temperature profile curve and heat transfer, steam inlet conditions 0.99 dry - SAR Class 26

Fig. IX.38 – Predicted steam temperature profile curves – SAR Class 26 4-8-4

Table XI-6 - Basic method - differences between calculated and observed temperatures - BR Standard class 9 2-10-0 tested at Rugby & BR Standard class 4MT 4-6-0 N° 75006 tested at Swindon

BR Class 9F 2-10-0								BR Class 4MT 4-6-0							
Mean excess air								Minimum excess air							
Steam rate	12,500	15,000	17,500	20,000	22,500	25,000	27,500	Steam rate	8,000	10,000	12,000	14,000	16,000	18,000	19,600
Gas rate - bulletin	23,517	27,707	31,898	36,089	40,280	44,471	48,662	Gas rate - bulletin	12,670	15,560	18,420	21,350	24,300	27,500	30,320
Gas rate - calculated	21,659	26,170	30,716	35,442	40,257	45,162	50,102	Gas rate - calculated	12,799	15,873	18,948	22,008	25,082	28,157	30,602
Firebox gas exit	1941	2044	2130	2205	2270	2329	2382	Firebox gas exit	1874	2000	2101	2184	2255	2317	2362
Gas apportion AL/(AL + AS) method (a) = 57%								Gas apportion AL/(AL + AS) method (a) = 46%							
	569	599	628	656	683	707	725		555	601	645	684	715	740	756
Small tubes	596	627	653	678	702	729	750	Small tubes	568	604	627	658	679	706	721
	4.7	4.7	4.0	3.4	2.8	3.1	3.4		2.3	0.5	-2.8	-3.8	-5.0	-4.6	-4.6
	627	650	671	689	705	718	729		557	589	608	620	631	639	643
Steam temp.	637	652	665	676	686	698	707	Steam temp.	560	571	579	587	594	601	605
	1.6	0.3	-0.9	-1.9	-2.7	-2.8	-3.0		0.5	-3.1	-4.8	-5.3	-5.9	-5.9	-5.9
	584	613	641	669	695	719	740		537	581	624	660	689	709	720
Large tubes	583	609	629	651	673	689	709	Large tubes	585	613	638	663	687	707	727
	-0.2	-0.7	-1.9	-2.7	-3.2	-4.2	-4.2		8.9	5.5	2.2	0.5	-0.3	-0.3	1.0
	12,500	15,000	17,500	20,000	22,500	25,000	27,500		8,000	10,000	12,000	14,000	16,000	18,000	19,600
Evaporation	12,072	14,772	17,414	20,108	22,780	25,469	28,113	Evaporation	7,744	9,897	12,030	14,089	16,109	18,097	19,676
	-3.4	-1.5	-0.5	0.5	1.2	1.9	2.2		-3.2	-1.0	0.3	0.6	0.7	0.5	0.4
Gas apportion hydraulic method (b) = 58%								Gas apportion hydraulic method (b) = 47%							
	569	599	628	656	683	707	725		555	601	645	684	715	740	756
Small tubes	595	632	649	674	702	723	748	Small tubes	570	604	634	656	678	702	723
	4.6	5.5	3.3	2.7	2.8	2.3	3.2		2.7	0.5	-1.7	-4.1	-5.2	-5.1	-4.4
	627	650	671	689	705	718	729		557	589	608	620	631	639	643
Steam temp.	642	656	670	681	692	701	712	Steam temp.	563	575	583	590	598	604	610
	2.4	0.9	-0.1	-1.2	-1.8	-2.4	-2.3		1.1	-2.4	-4.1	-4.8	-5.2	-5.5	-5.1
	584	613	641	669	695	719	740		537	581	624	660	689	709	720
Large tubes	584	611	632	655	673	695	709	Large tubes	586	616	642	669	693	714	729
	0	-0.3	-1.4	-2.1	-3.2	-3.3	-4.2		9.1	6.0	2.9	1.4	0.6	0.7	1.3
	12,500	15,000	17,500	20,000	22,500	25,000	27,500		8,000	10,000	12,000	14,000	16,000	18,000	19,600
Evaporation	12,046	14,731	17,379	20,064	22,727	25,434	28,069	Evaporation	7,730	9,875	11,989	14,064	16,070	18,070	19,624
	-3.6	-1.8	0.7	0.3	1.0	1.7	2.1		-3.4	-1.3	-0.1	0.5	0.4	0.4	0.1

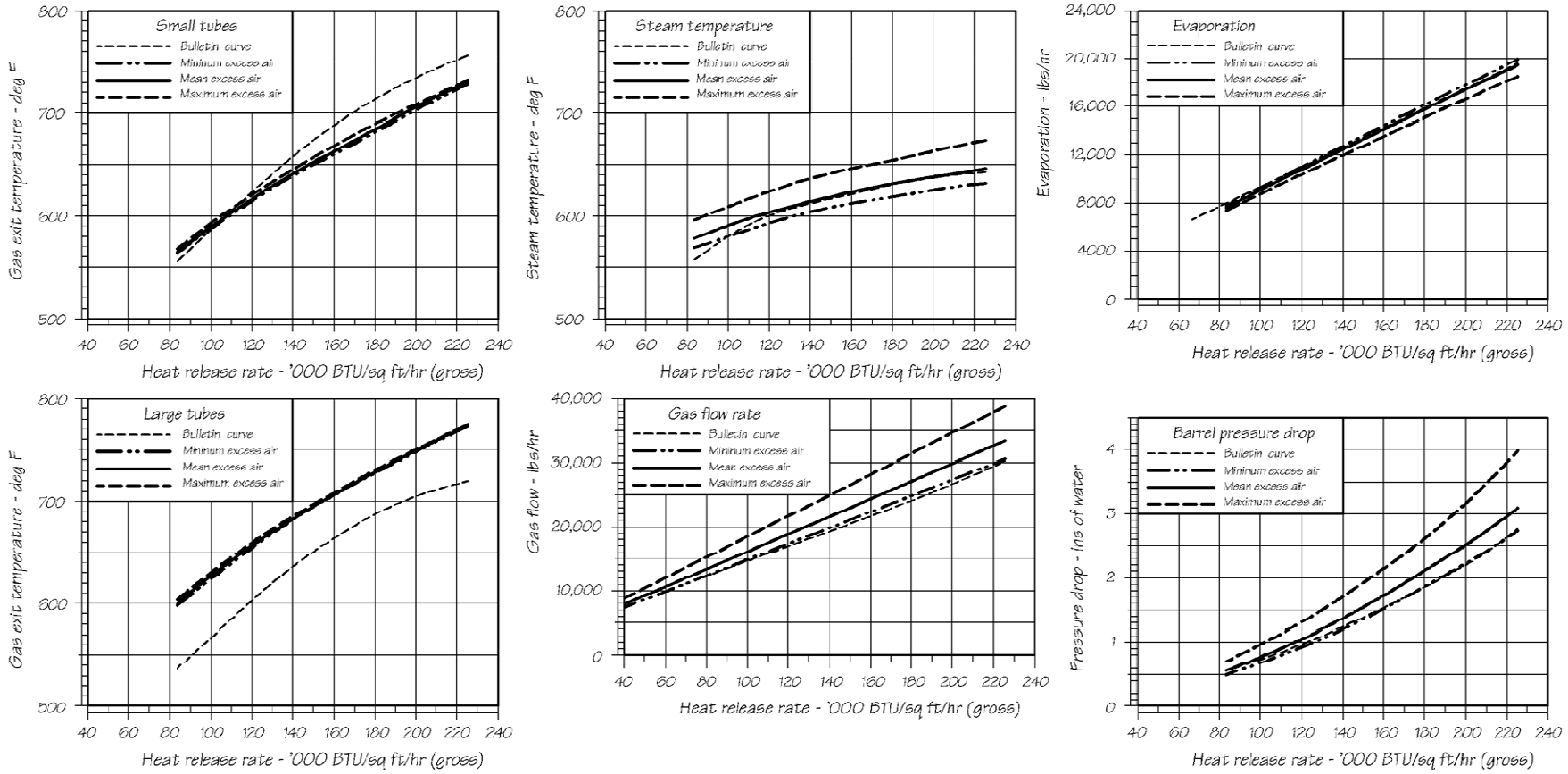


Fig. XI. ... - Comparison between predicted boiler performance and that recorded in its Bulletin – BR Class 4 4-6-0