Designing Model Boilers

The Truth about Model Boilers

What Really Happens in Model Boilers

1 INTRODUCTION

It is a strange coincidence that when I first started reading Model Engineer back in 1971, such illustrious figures as D. E. Lawrence, Martin Evans, E.C. Martin and other were debating just how to design model boilers. I have to admit that I did not understand what these people were discussing at the time. Here I am, some 46 years later discussing the same topic, but in the intervening years computers have taken over from slide rules and I have had an engineering career concerned mostly with fluid flows, so I now understand terms such as Reynolds number. I think we can do better than 46 years ago.

I am currently designing and building a coal fired boiler for my 7” scale miniature Fowler steam lorry, which is considerably larger than most 7 ¼” gauge locomotive boilers, but not as large as full size. It is a vertical firetube type with a relatively large firebox, a large number of short tubes and the prototype had a smokebox superheater coil. I found that the information available in the model press on proportioning boilers was very short of quantitative design rules and those that existed seemed dubious to say the least. Even much of the qualitative material seemed to be based on repeated assertion rather than fact. So the only way forward was working from first principles.

This article deals with the thermal design of coal fired boilers in the 3 1/2” to 7 1/4” gauge range. To avoid confusion with computer **modelling**, I have called them "miniature" boilers throughout this article.

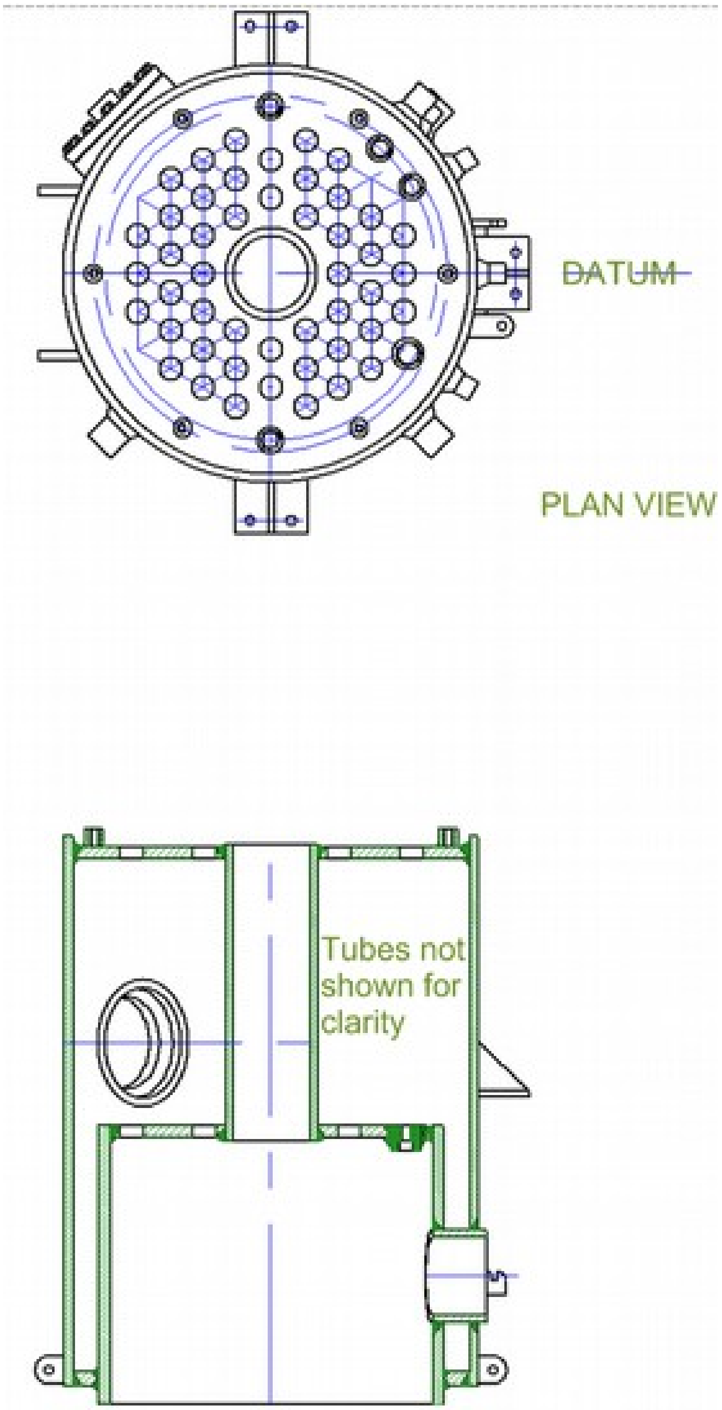

Illustration 1: Cross section and plan of the boiler for my steam lorry. The outer shell is 18" diameter, there are 47 vertical tubes and a coil superheater in the smokebox (not shown).

The ruling equations for heat transfer in fireboxes and tubes are well known and can be readily solved with the massive computing power available to us all nowadays. There have been computational approaches to boiler design before, notably from Bill Hall. However, I decided to develop my own method and this article summarises some of what I have found out along the way. It will hopefully be of use to other designers of “Stephenson” engines with small coal fired, fire tube boilers; and of interest to users of such small engines – whether road, rail, marine or stationary.

I would like to acknowledge generous help in my researches from Duncan Webster who provided copies of previously published material, encouragement and occasional critique; also Roger Froud, who provided locomotive data and a chance to exercise my computer program for real.

2 CONSTRUCTING A MATHEMATICAL MODEL OF A TYPICAL LOCOMOTIVE BOILER

> If anyone wishes to flog his brains on heat transfer in ….superheaters, L.H. Fry's paper…….refers. The complex mathematics would probably give 99.9% of model engineers rigor mortis at first sight! In any case, I think these matters need not apply to our small engines. – D.E. Lawrence ME 3417

I am not the first to "flog my brains out". Some of the current work follows Bill Hall's methods. However, my program is constructed the Excel spreadsheet platform (2002 Version) and will run on Open Office, whereas Hall's program was written in C making it a much less open platform for use and public development. Hall did not consider his boiler work "complete". A brief summary of the work was written up in an unpublished report (Ref.8.3).

I have spent over 2 years developing my calculation in which the geometry of a boiler can be entered along with the working conditions. The input data can be grouped as follows:

- Boiler Working Conditions - Working pressure, Pressure after regulator, Firing rate, Air Ratio, Fuel lost rate, Fuel burnt on grate, Combustion Efficiency, Dryness fraction of steam leaving boiler.

- Boiler Geometry - Grate area, Firebox wall area, Length of tube stack, Number of firetubes, Number of Superheater flues, Number of spearheads per superheater flue, Internal diameter of firetubes, Internal diameter of superheater flues, Inside diameter of superheater elements, Outside diameter of superheater elements, Length of radiant superheater section, Roughness of fire side surfaces, Roughness of steam side surfaces, Outside diameter of boiler barrel, Overall length of boiler barrel, Thickness of Insulation, Conductivity of Insulation.

- Coal Analysis - Carbon, Hydrogen, Sulphur, Water, Ash, Oxygen, Nitrogen contents.

- Constants - Firebed emissivity, Specific heat of flue gas, Absorbtion coefficient of flue gas.

The calculation involves some 14500 calculation steps, split roughly as follows:

Fire & Heat Release – 46

Firetube – 1900

Firebox heat transfer – 1800

Superheaters - 11000

The calculation produces the evaporative capacity of the boiler and superheat temperature of the steam. It also produces the temperature profiles along the length of the firetubes and superheater tubes. Typical results produced by the program will be discussed in relation to analysis of the boiler for LBSC's "Speedy" design.

I believe my work has advanced the state of the art in model boiler design in the following respects:

- More detailed analysis of the combustion process, in particular account of unburnt fuel, moisture content & coal analysis.
- A firebox radiant heat transfer model that includes the effect of absorbent / emissive flue gases.
- The option of considering wet steam generation.
- A section to calculate isenthalpic property changes across the regulator.
- Firetube performance calculation using well established correlations against Reynolds & Prandtl numbers.
- Calculation of superheater performance.

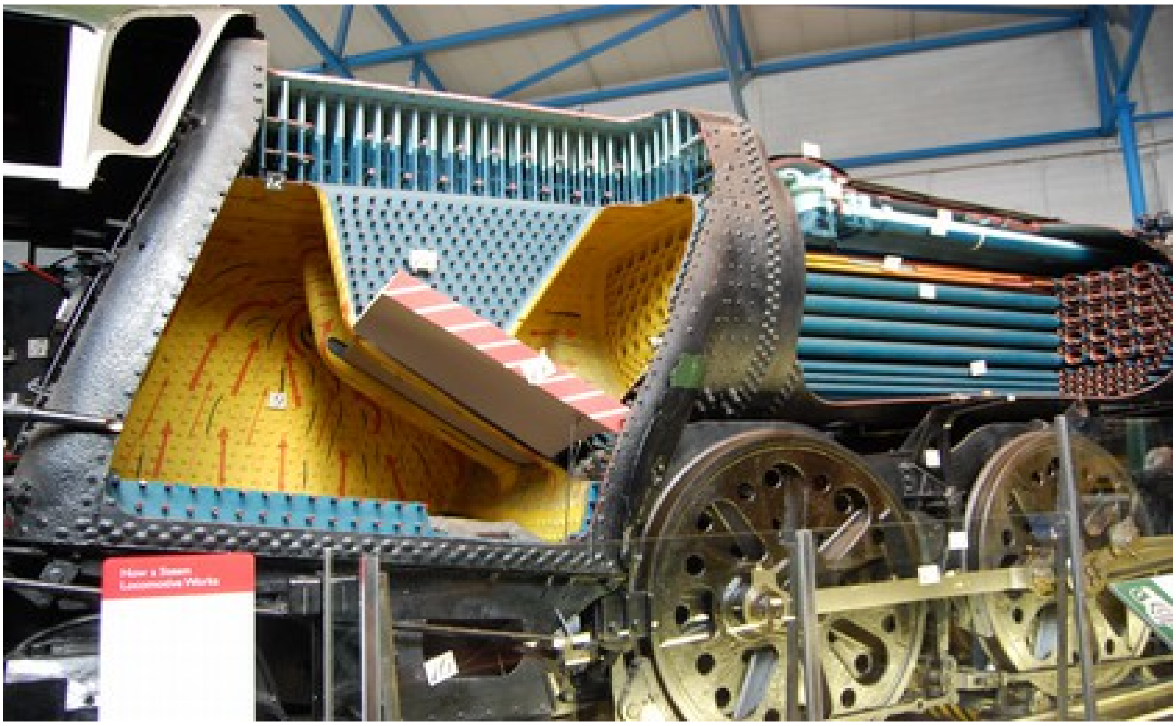

Illustration 2: A complex boiler, but just how do you calculate firebox size, tube size & number & superheater details? - Photo by P. Johnson

The calculation is based on well founded relationships for convective heat transfer in a tube, pressure drop along a tube, radiative heat transfer from a firebed to firebox walls, combustion process for coals, conservation of energy, etc. Empirical relationships, of which there are plenty, have been avoided because they usually apply to a limited range of full size practice. Readers (and the Editor) will be pleased that I do not intend to reproduce all the fundamental equations involved, most of which are covered in thermodynamics or heat transfer textbooks or the website http://thermopedia.com/. However, it is worth summarising some of the difficulties encountered in putting the mathematical model together.

The computer has to repeatedly calculate values of densities, viscosities, Prandtl number, enthalpy, entropy etc. over the range of temperatures and pressures likely to be encountered by wet steam, saturated steam, superheated steam and combustion gases. Therefore, a range of equations have been developed which will replicate published values of these data. Combustion gases are assumed to have the properties of dry air, which is the main constituent of combustion gas .

The combustion process is calculated using data contained in Ref. 8.10. This method will account for different quality coals. The calculation further accounts for:

- Some fuel being lost to ahspan, smokebox or chimney before being burned. This ratio is set by the user, and tests have been analysed to determine realistic values for this quantity.

- The quantity of air drawn in to burn the coal tends to vary with grate loading. The ratio of air to coal is set by the user and tests have been analysed to determine realistic values for this quantity.

- Some carbon burns to produce carbon monoxide, this ratio is user settable.

- Some heat is trapped as latent heat in high temperature water vapour generated from combustion of hydrogen and evaporation of moisture in the coal – this heat is debited from the total generated, since it cannot be recovered in a non-condensing boiler.

- The heat generation can take place mostly in the firebed and partly above the firebed, the ratio depending on what parameters are set in the program. Heat release above the firebed would typically be due to volatiles burning in the firebox volume.

The heat released in the fire is assumed to heat the combustion products to the same temperature as the firebed, and the combined heat increase in the gases plus the radiant heat emitted by the fire are equal to the energy released in the fire. The radiated energy from the firebed to the firebox is calculated using the Stephan Boltzmann equations. Some heat is radiated from the incandescent particles in the flames direct to the firebox surfaces. The ratio of “firebed to firebox” heat transfer to “flame to firebox” heat transfer depends on the emissivity and absorptivity of the firebox gas, plus the dimensions of the firebox. The calculation also accounts for heat transfer by convection within the firebox, which is relatively small but significant. As gas passes through the firebox, the calculation reduces the gas temperature accordingly as heat is transferred into the boiler.

The flow in the firetubes is split into 25 stages along the firetube length. In each stage, the convective heat transfer to the boiler is calculated at the mean flue gas temperature in that stage and then the flue gas temperature leaving that stage is calculated as the input to the subsequent stage. The calculations take account of the formation of a boundary layer starting at the tube entrance and increasing along the tube length, plus the thickness of boundary layer based on correlations against Reynolds number. Thus a reasonably accurate picture of the temperature profile along the tube is generated. An estimate of the superheater flue evaporation is made by assuming a similar temperature profile to the firetubes, the estimate being refined during the superheater calculation.

The regulator is usually placed before the superheater in a miniature, and the program calculates the change in steam temperature and dryness due to the adiabatic pressure drop across the regulator.

The temperature profile in a superheater flue is calculated in a similar manner to the smoke tube calculation. However, the steam temperature in the superheater at entry to the firebox end of the superheater flue is not known, therefore an iterative calculation is used to determine temperatures along the superheater. In addition, the evaporation from the superheaters is not known until the temperature profile has been calculated, so another iteration process is needed to get over that difficulty.

The computer model does not account for thermal resistance through the metal of the boiler, since it is actually very small compared to the resistance of the gas films. It would be a simple step to add this to the program, but there are more demanding areas of development.

The program has been constructed on a modular basis, so alternative configurations such as water tube boilers or smokebox coil superheaters can be constructed if needed.

For readers interested in learning more about the methods, the author can be contacted via the editor, and will be pleased to furnish the spreadsheet calculation, detailed equations, notes & references on request.

3 VALIDATING THE COMPUTER MODEL

> …..The gas flow in model boiler tubes, unlike full size, is streamline and the boundary layer characteristics inhibit heat transfer. I personally think it is semi-streamline, or if you like, semi-turbulent because of eddies set up by deposits in the tubes. – D.E. Lawrence ME 3417

A complex calculation technique such as that described above must be validated against test results. Two sets of miniature tests have been published (Refs. 8.1 & 8.2), those by Busbridge being the better tests, while those of Ewins reported by Evans were obtained using a convenient, but error prone, test method.

3.1 Busbridge's Test on 3.5" Gauge "Brittania" Boiler

Busbridge carried out a well controlled series of tests on a Brittania boiler to LBSC's design and published the results in Reference Busbridge, J. – Publ. Model Engineer 1st August 1964, pp 565-577 (1964). The tests looked at the boiler only and were conducted at the University of Cape Town.

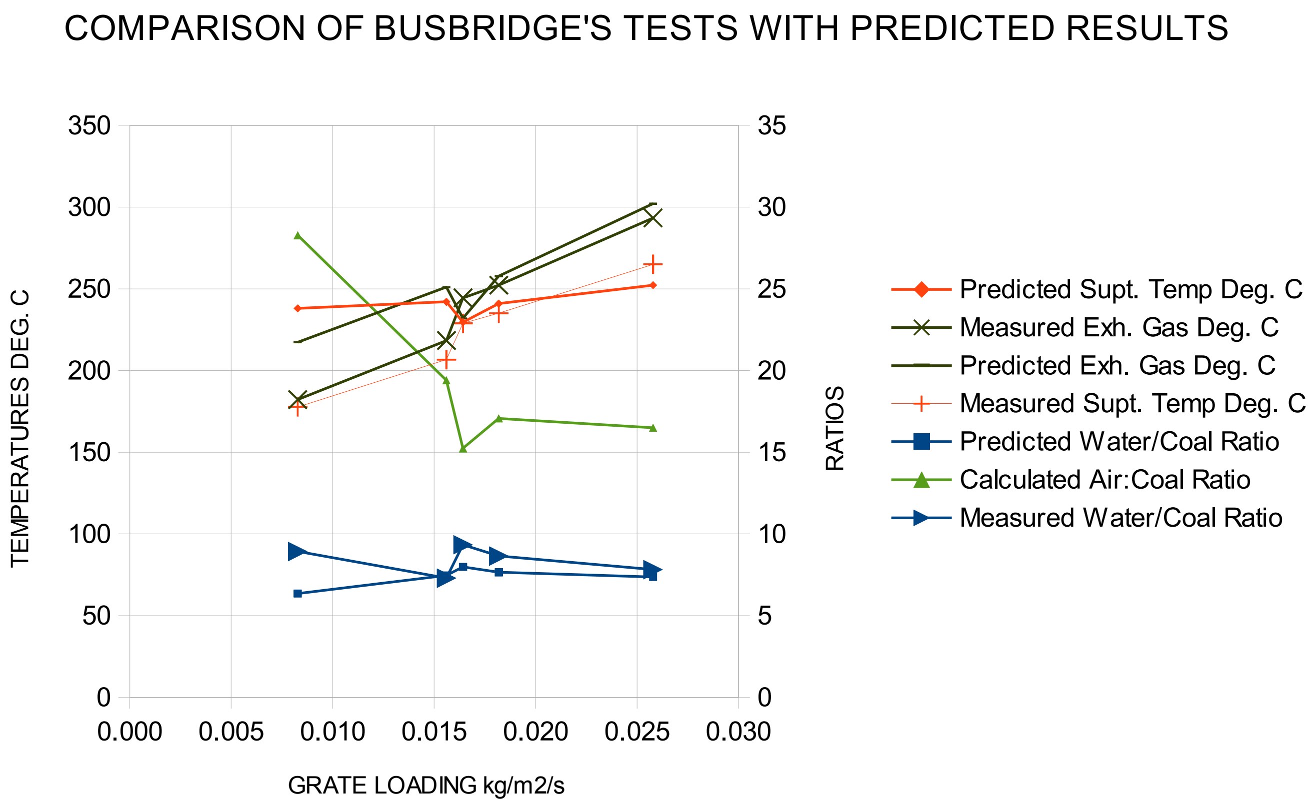

Figure 1 - Comparison of Busbridge's test results with predictions from program.

Figure 1 shows good agreement has been achieved to Busbridge's tests across a range of grate loadings. The calculated stoichiometric ratio has been derived from Busbridge's measurements of carbon dioxide in the flue gas. It can be seen that the general pattern is for this ratio to asymptotically approach a value of around 15 at high grate loadings. At low grate loadings the ratio climbs to values near 30 or perhaps more.

The quality of prediction becomes worse at lower grate loadings, partly because variables in the program have been chosen to give better agreement at high grate loadings. However, the results suggest that free convection, probably in firebox spaces, become more significant at lower grate loadings. Since the program is intended as a design tool, accurate predictions at maximum grate loading are more important.

The computer confirms that flow in the fire tubes is indeed streamline (also known as laminar flow). Busbridge's tests included careful measurements on the non-radiant superheater performance, which is fortunate as this gives an accurate indication of heat transfer within the flues. The validation process also confirms D.E. Lawrence's opinion that the flow is not fully settled laminar flow, leading to much better convective heat transfer than well accepted correlations would predict. The results suggest that normal correlations of convective heat transfer need uplifting by about 60%. Possible reasons for this are:

- Gas flow from the firebox to the tubes is turning through 90 degrees, causing major disturbance to the boundary layers in the flues and hence improving heat transfer.

- Gas flow is well into the laminar regime (Reynolds number 500 to 1000 in a miniature firetube) but within the lengths of a miniature boiler, flow is unlikely to be "settled", giving thinner boundary layers and hence improved heat transfer.

- I don't accept Lawrence's opinion that "deposits" would cause such an increase in heat transfer, as Busbridge's tests were on a new boiler.

3.2 Ewins' Tests on 5" Gauge "Mona" Locomotive

Correlations against Ewins' tests are somewhat frustrating. The tests are subject to significant experimental errors and inconsistencies as follows:

- Flue gas volume was measured by catching the gas in a large plastic bag and then measuring the volume by a "water displacement method" – details not given. This seems very prone to large errors.

- The flue gas analyses are not consistent with a predominantly carbon fuel burnt in natural atmosphere. The calculations relating to flue gas analysis can be found at Ref. 8.10. These analyses are used to infer coal consumption and grate loading. This results in experimental error in interpreting how hard the engine was working. If the CO_2 and CO results are assumed correct when burning a good quality coal, the O_2 contents would be as follows:

TEST NUMBER	EXPERIMENTAL O_2 BY VOLUME %	CALCULATED O_2 BY VOLUME %
1	9	10
2	3.5	6.2
3	10	10.9

- The grate loading can be inferred from Ewins' values for "coal consumption per DBHP hour" and "DBHP per sq. ft of grate". If this is done, the predicted flue gas production do not match his measurement of flue gas production. Grate loadings some 12 to 18% higher than those stated would be required to generate the stated gas volumes. Only some 4 % could be accounted for by ash content and the calculation was conducted assuming no fuel is lost before burning, which is an implicit assumption in Ewins' data analysis.

- The smokebox and firebox draught measurements are not consistent with the inferred changes in grate loading. I am indebted to Duncan Webster for pointing this out.

- The gas temperature measurements in the firebox will be significantly in error due to the radiant heating from the fire. BS 2790 Appendix C states this error can be up to 300 Deg. C. Taking accurate temperature measurements in a highly radiant environment needs very specialised (and expensive) equipment.

- Multiple IMLEC results suggest a maximum efficiency of around 2.5 % is feasible on miniatures, while Ewins claims an efficiency of 4.9 % in Test 3, which illustrates the probable magnitude of experimental errors in Ewins' work.

- Some sources indicate the engine tested had thermal siphons in the firebox, but Evans in Ref. 8.1, makes no mention of these. I have sought the original SMEE papers of Ewins' tests, but without success. If anyone is able to help with copies of these, I would be very grateful.

Nevertheless, I have attempted to compare some of Ewins' results with the numerical model. Grate loadings are based on assuming Ewins' readings of flue gas volume flow, CO_2 and CO readings are correct, but the grate loadings inferred from stated values of "coal consumption per DBHP hour" and "DBHP per sq. ft of grate" are also included. The following graph summarises the coal / air ratio, superheater and exhaust gas temperature results:

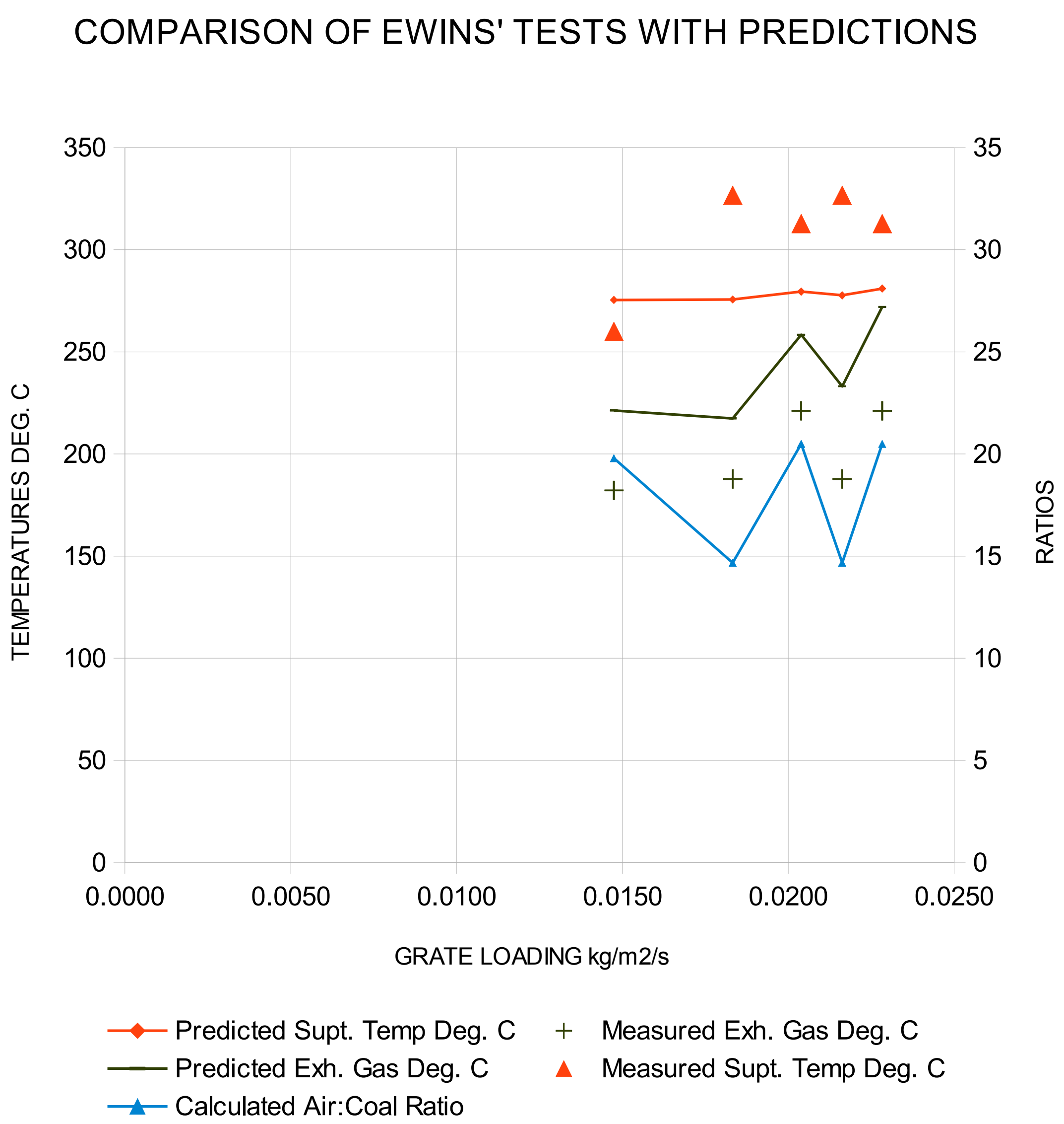

Figure 2 - Comparison Ewins' test results with predictions from program.

Figure 2 shows that Tests 2 & 3 were taken at very similar grate loadings, although the power produced by the engine varied over a 400 % range, confirming that the 5% claimed efficiency of Test 3, is not true and the boiler was not keeping up with steam demand. The air:coal ratio shows a drop from around 20 at low grate loading to 15 at moderate loading, which is consistent with Busbridge's result, but then a sharp increase; This suggest the fire may have had a hole in it, artificially increasing the air ratio.

The superheated steam temperature observed is some 45 to 50 deg. C higher than predicted, while the exhaust gas temperature is some 80 deg. C lower than predicted at the higher grate loading. The actual rate of evaporation cannot be inferred from Ewins' results.

I have also compared the predicted temperature profile in the firetube for Ewins' test No. 2, which appears to be the most reliable set of data. The results are shown in Figure 3, which shows the program predicts temperatures some 175 Deg. C higher than observed at the tube entrance, reducing to 45 Deg. C higher at the tube exit. Firetube exit temperatures observed by Busbridge were also considerably higher than those noted by Ewins.

Ewins' observed fire temperatures are far higher than my predictions (1600 Deg. C against 1150 Deg. C). If the fire were at the temperature Ewins claims the fire would be well beyond white heat. My calculation estimates the fire at yellow to white heat, which seems more logical. The experimental errors associated with measuring temperatures in highly radiant environments have already been noted. Therefore, it seems to me that Ewins' temperature readings must be treated with some suspicion, and consequently so must many of his other conclusions.

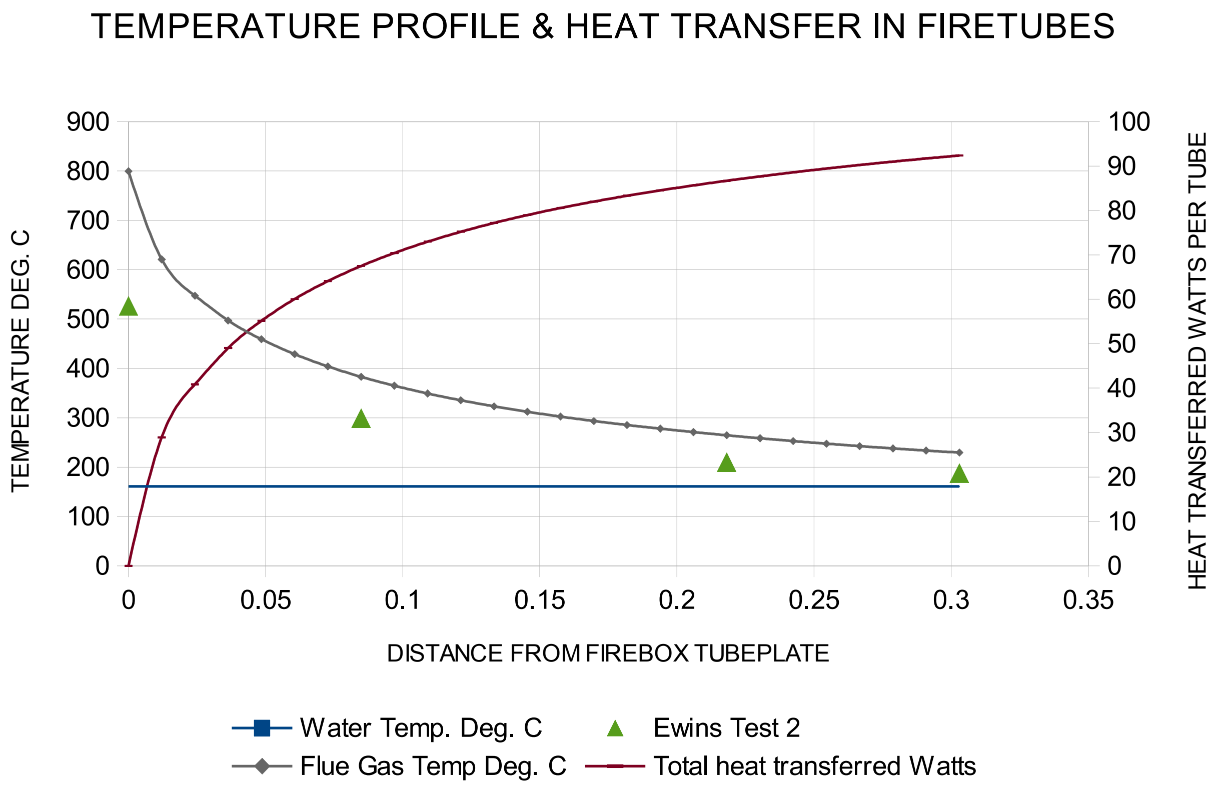

Figure 3 - Comparison of predicted and observed firetube temperature profile from Ewins.

3.3 Full Size Tests

In Ref. 8.4 I found a design chart for the temperature profile in boiler tubes. The chart, originally published in ALCO bulletin 2017, is based on a grate loading of 100 to 115 lbs./sq.ft./hour. Unfortunately, it does not state the relative areas of grate and tube gas flow area. However, I have tried reproducing the chart and have used boiler design details for the GWR Castle class, since I have that data to hand. This is probably not the best choice, since an American test was probably on a wide firebox boiler and certainly not burning best Welsh coal. The results of the comparison are shown in Figure 4, and considering the potential problems as noted above gives remarkably good agreement. Note particularly that the temperature leaving the firebox is very well predicted.

The flow in boiler tubes for full size loco boilers is turbulent, having a Reynolds numbers in the range 7500 at tube entry to 13000 at tube exit. By comparison, the equivalent numbers in a miniature boiler are typically 500 to 900 which indicate laminar flow. This means there are completely different flow patterns in full size and miniature work, and consequently the heat transfer rates are fundamentally different, which makes any attempt to draw parallels between full size and miniature dangerous; as LBSC said “you can’t scale nature”!

To accommodate the difference in flow pattern and heat transfer, the program uses different formulae for heat transfer depending on the Reynolds number, so the comparison in Figure 8 exercises a different part of the program to that used for model comparisons. One of the tasks on my “to do list” is to obtain some of the Rugby and Swindon test plant data now held at York Railway Museum and use this to check and improve that area of the program.

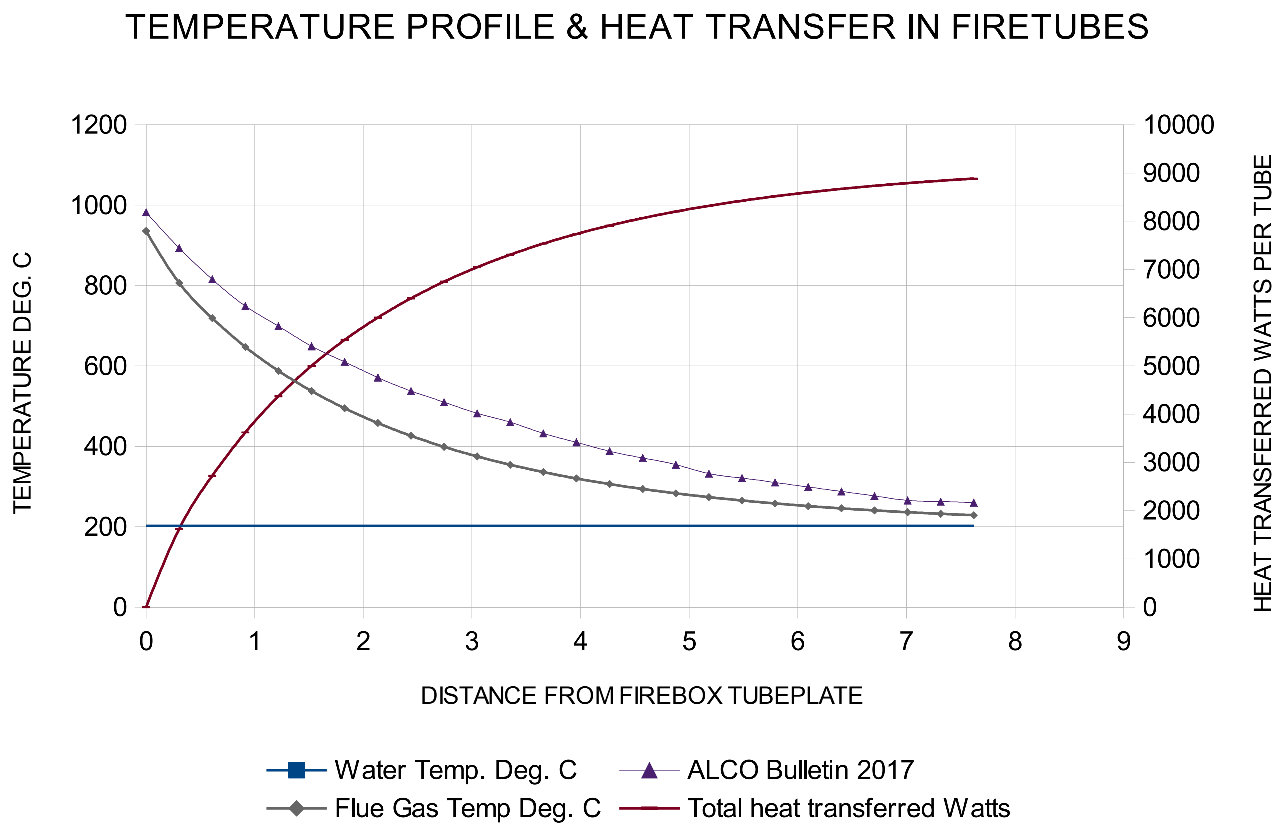

Figure 4 - Comparison of ALCO Bulletin 2017 design values and calculated temperatures

Ref. 8.4 also includes a design figure for evaporation due to firebox walls, which is quoted at 55 lbs evaporation per square foot. The program predicts evaporation of 64 lbs/ sq.ft.; It may be that the design is a conservative value to give some safety margin in design, and there is also the issue of whether my assumptions match the original test boiler.

If anybody can help with a copy of ALCO Bulletin No. 2017, the author would be very pleased to hear from them.

4 THE “CONSTANTS”

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

4.1 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. 8.2) 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 5 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 5. Clearly, we do not work our miniature engines harder than full size!

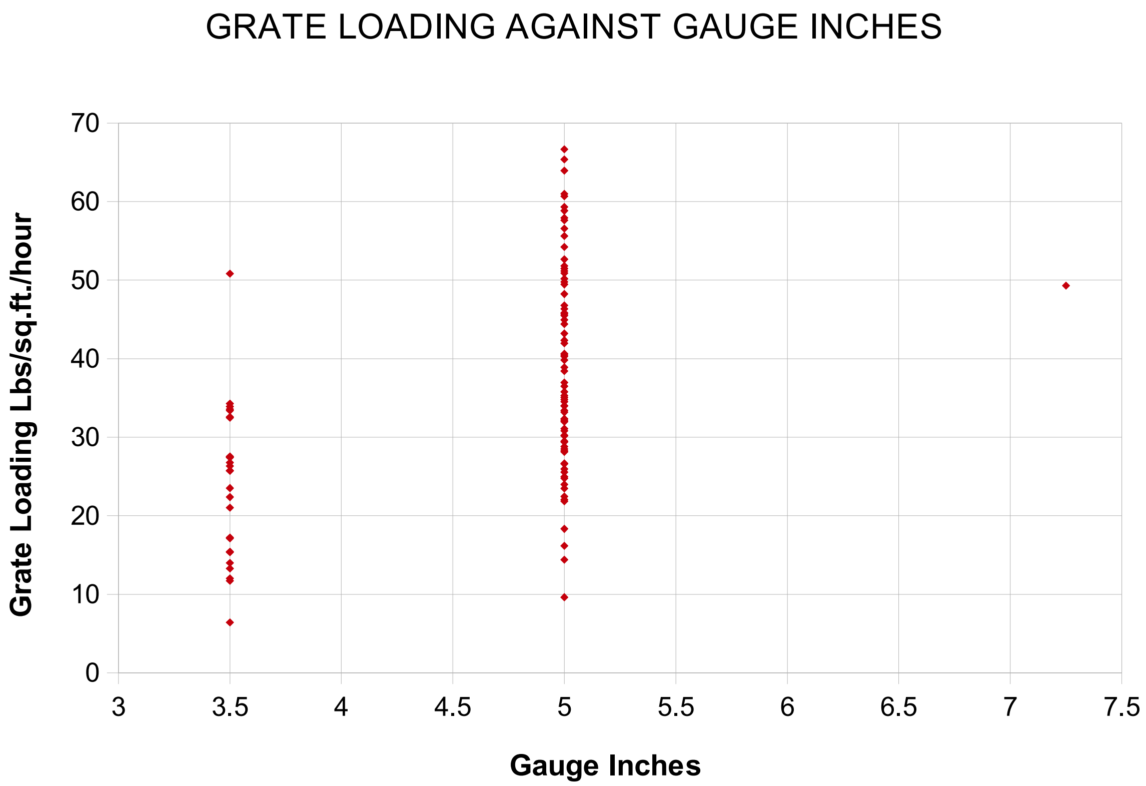

Figure 5 - Grate Loading Vs. gauge of engine

Some additional data can be found in Ref. 8.8, 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. 8.2) 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 5 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.

4.1.1 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.

4.2 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]

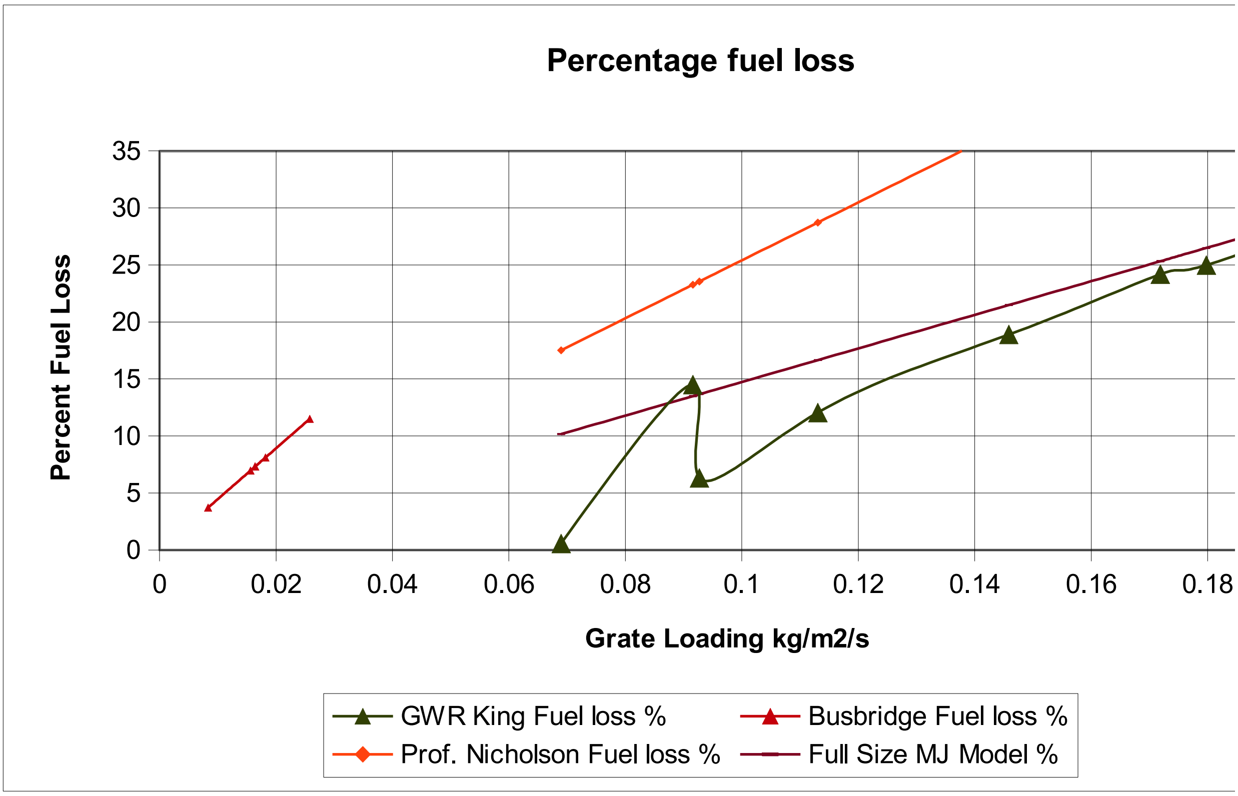

Figure 6: Percentage of fuel lost before combustion against grate loading.

4.3 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.

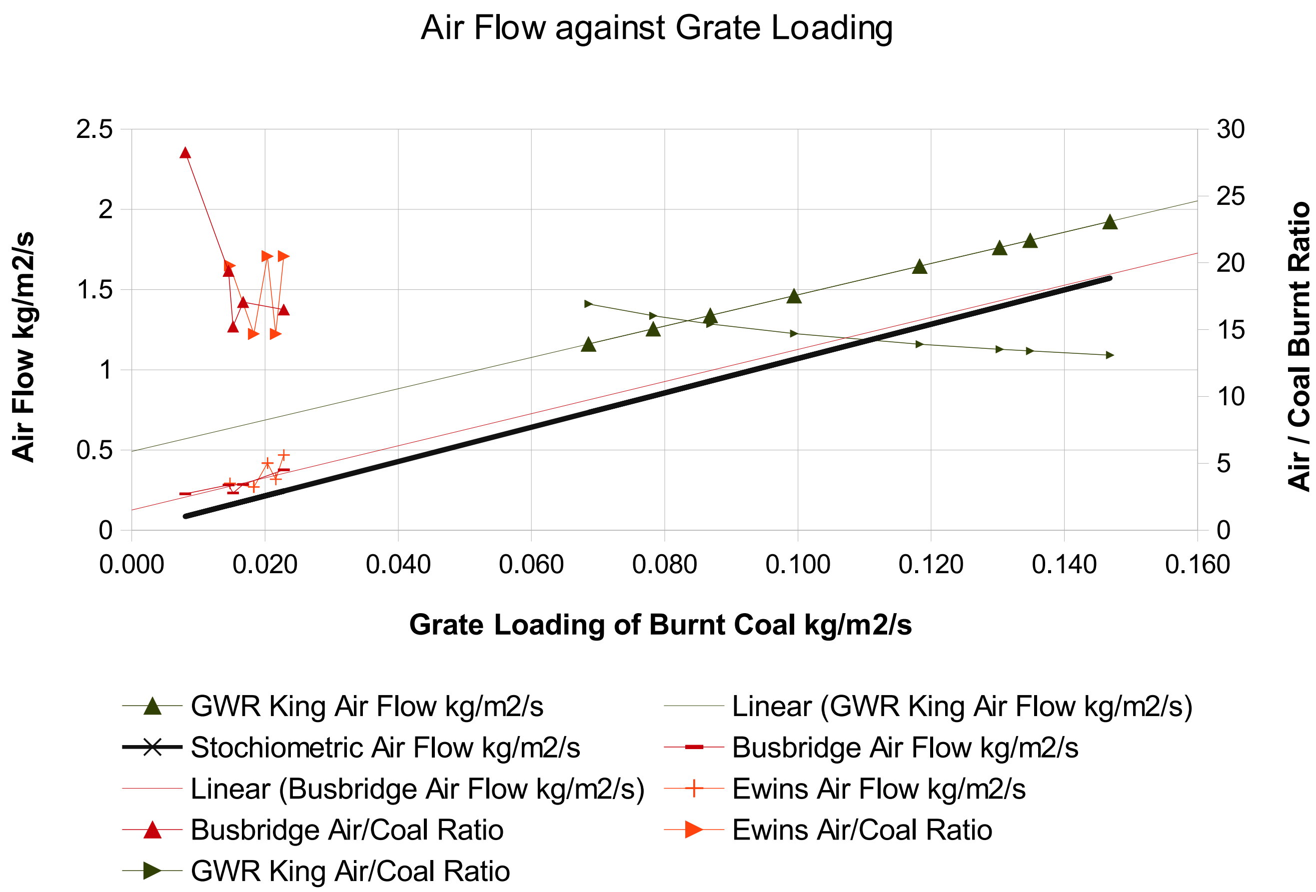

Figure 7: Air flow through coal fired grates

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.

4.4 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.

5 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 8 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 9 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.

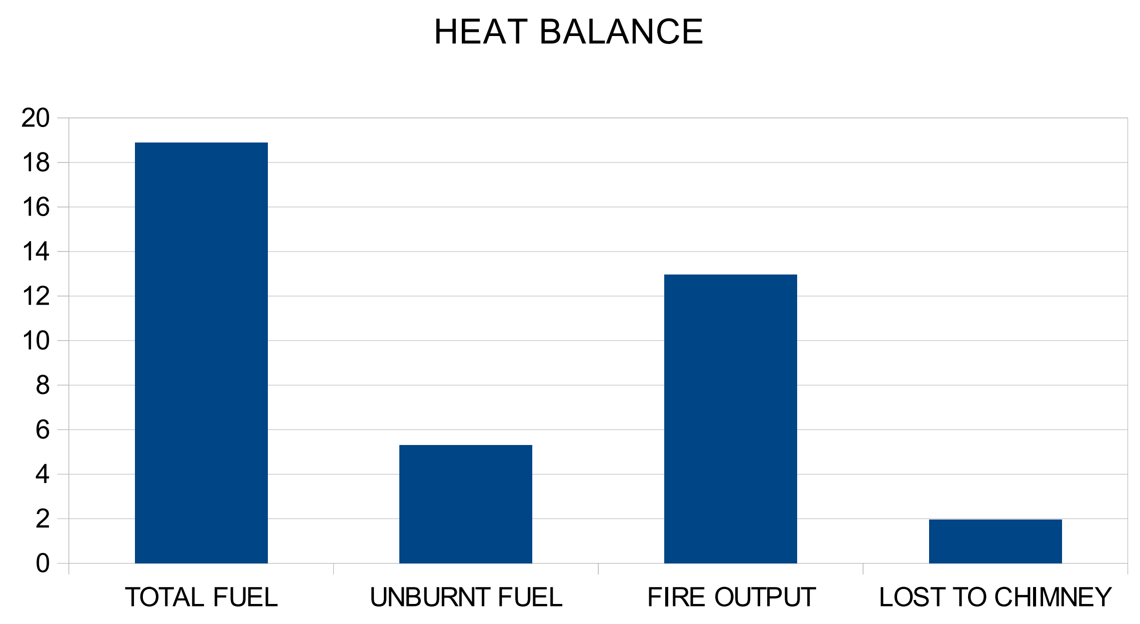

Figure 8 - Heat balance

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.

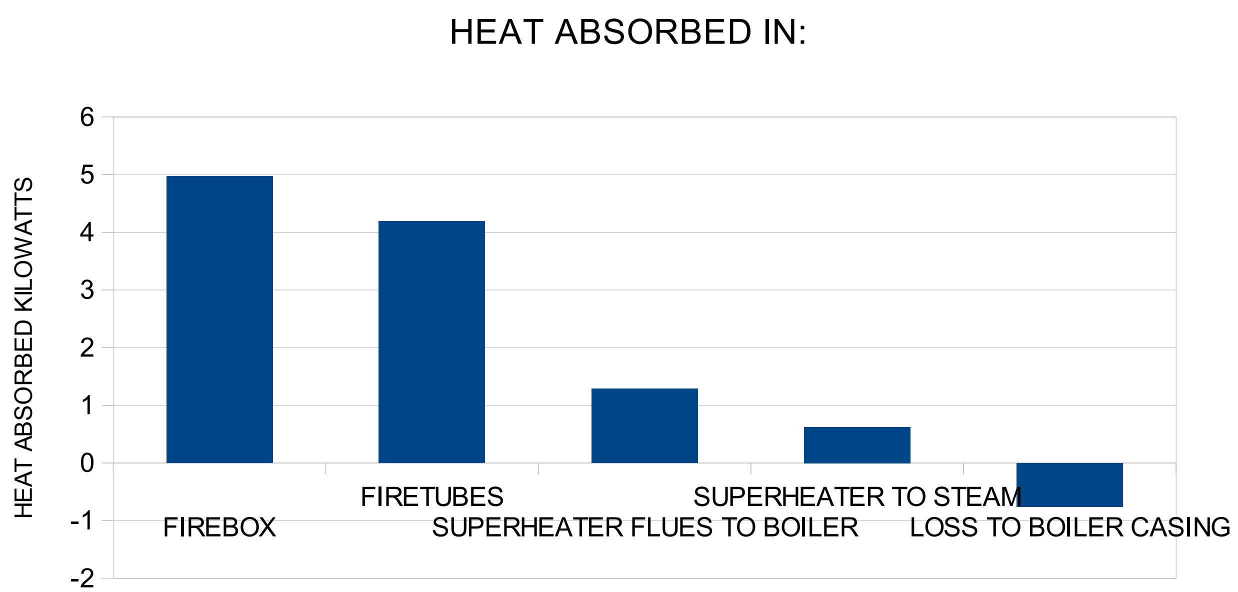

Figure 9 - Heat consumption in boiler

Figure 9 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 10 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.

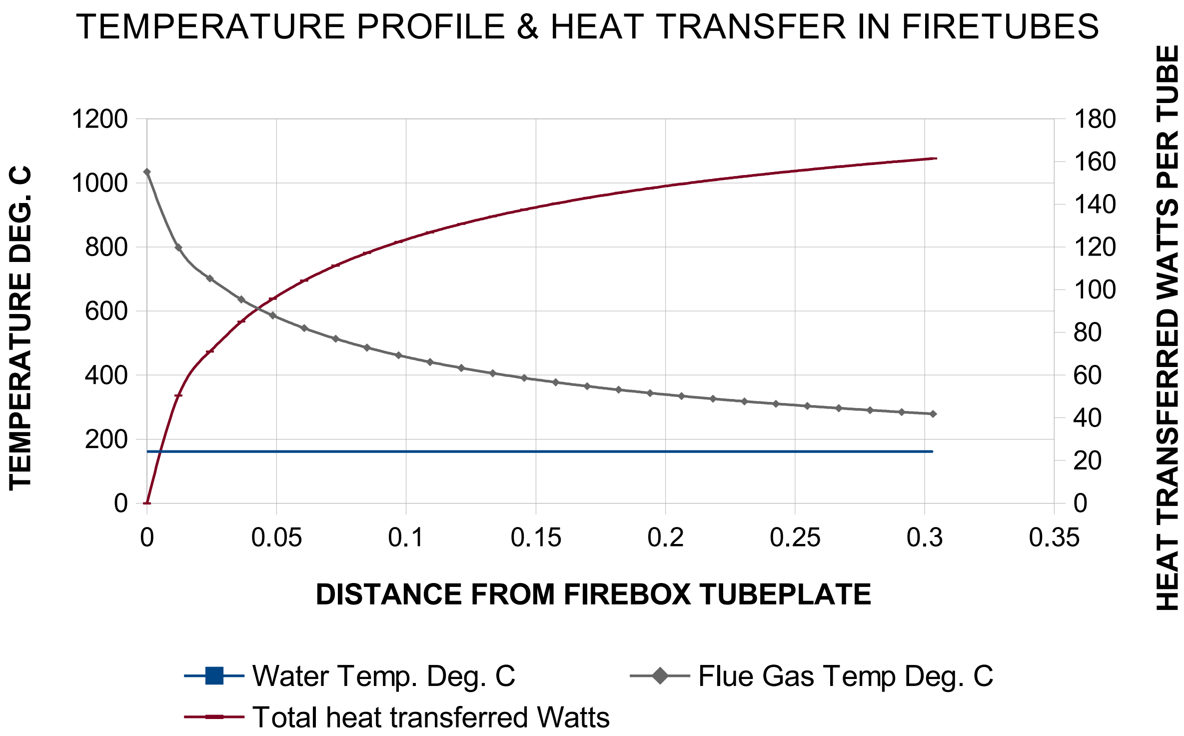

Figure 10 - Temperature profile and heat transfer in firetubes.

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.

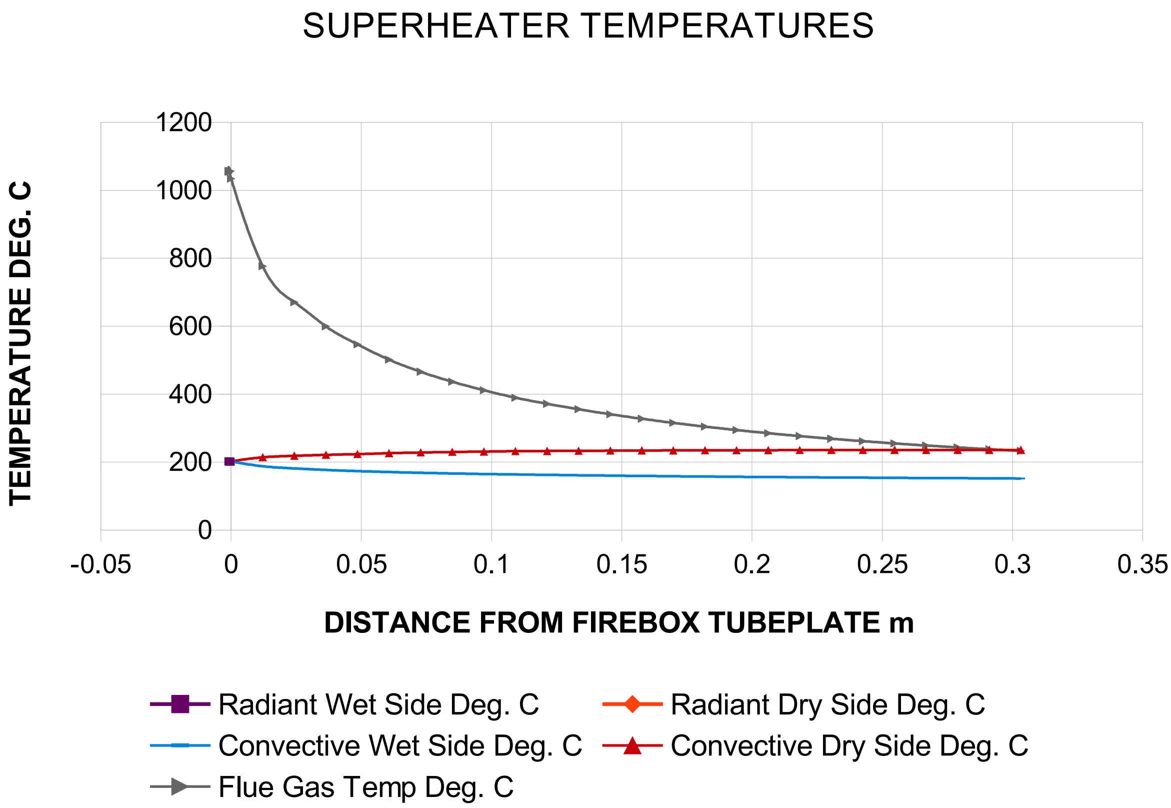

Figure 11 - Temperature profile in superheater flues

The predicted temperature profiles in the superheater flues are shown in Figure 11. 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.

6 ASSESSING DESIGN CHANGES

6.1 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.8.5) 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 12 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.

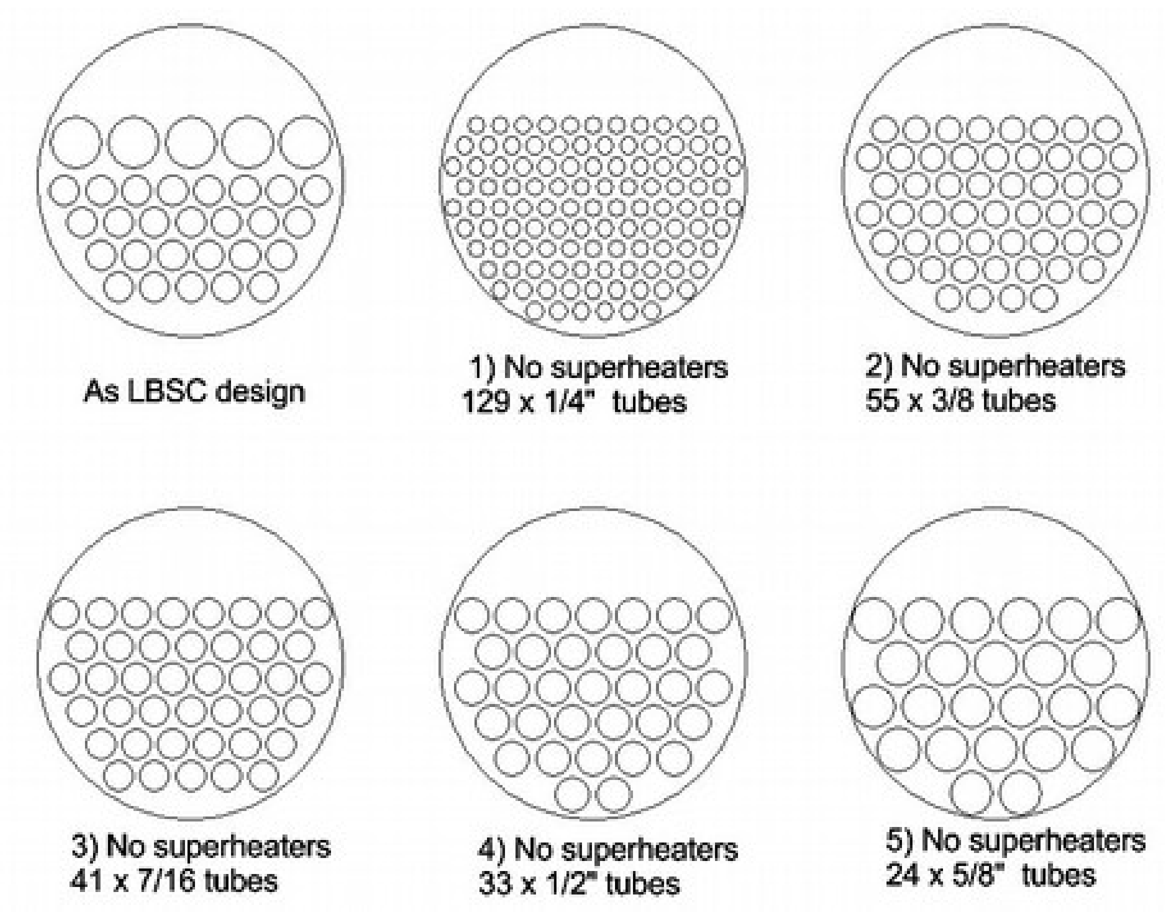

Figure 12 - Speedy design variations

The results of my calculations are shown in graphical form in Figure 13 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^2 per inch	L/ID	Gas flow area	Heat exchange area	Gas Area/Grate Area
	Inch OD	mm H2O	g/s			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 13 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.

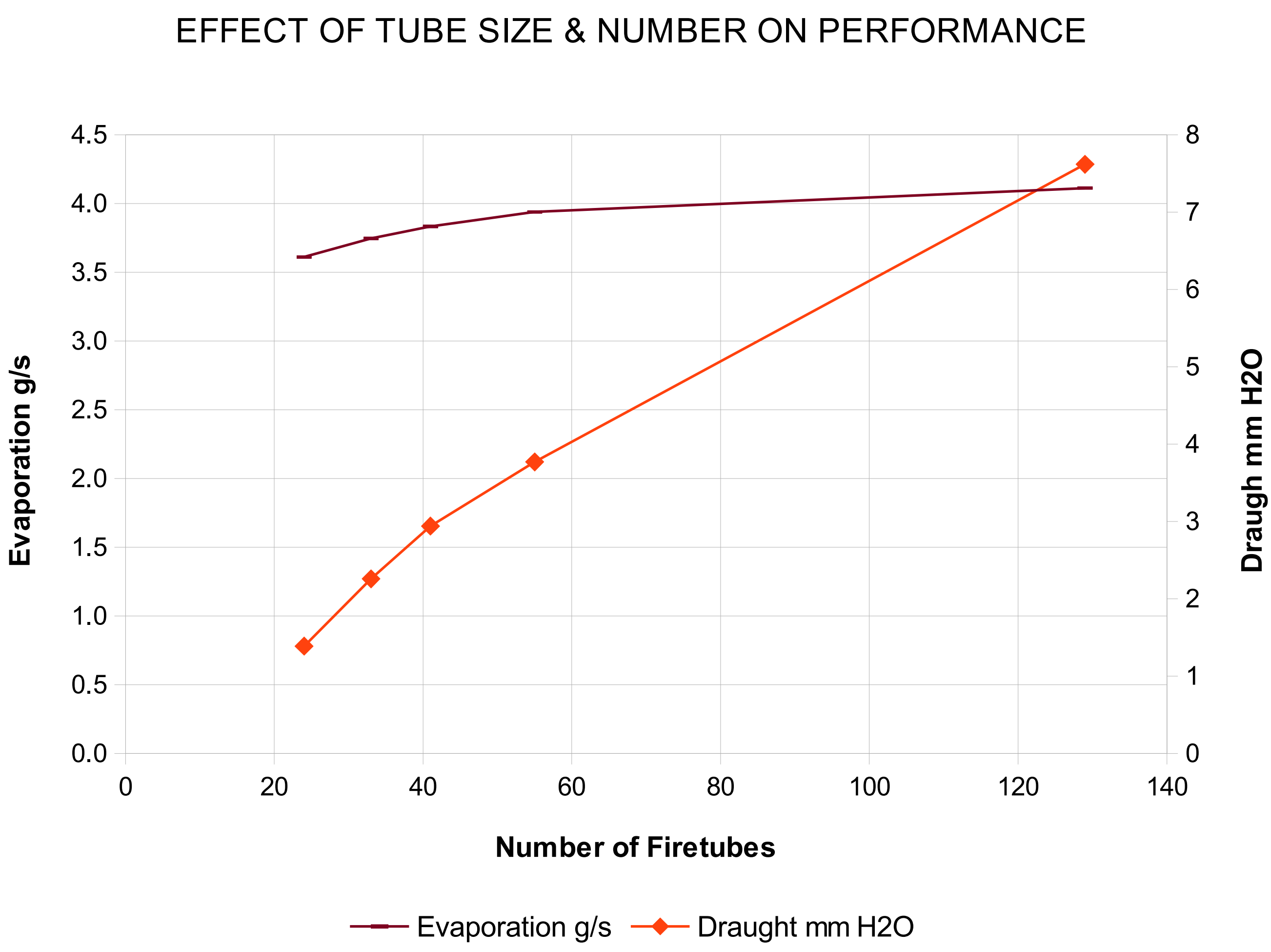

Figure 13 - Effect of tube number and size on Speedy performance

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 13 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. 8.6). 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. 8.1) 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".

6.2 Superheater Tubes

6.2.1 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 14, 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:

$$Steamratio = 1 + \left[1.7827 \times e^{(-0.0197 \times Superheat)}\right]$$

Where:

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

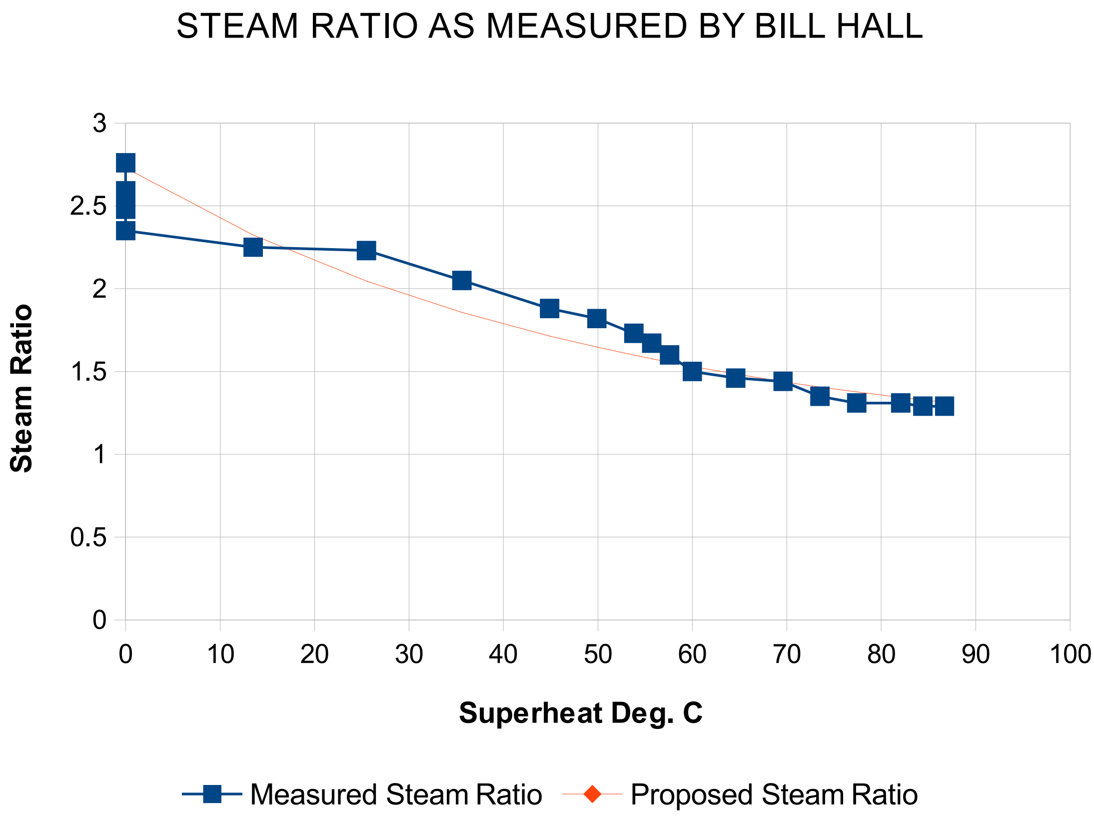

Figure 14 - Steam ratio as measured by Hall on a "Speedy" cylinder

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:

$$Volume = Massofsteam \div density \div Steamratio$$

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]

6.2.2 Different Superheat Options

A widely adopted technique of improving engine performance is to fit radiant superheaters and Figure 15 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.**

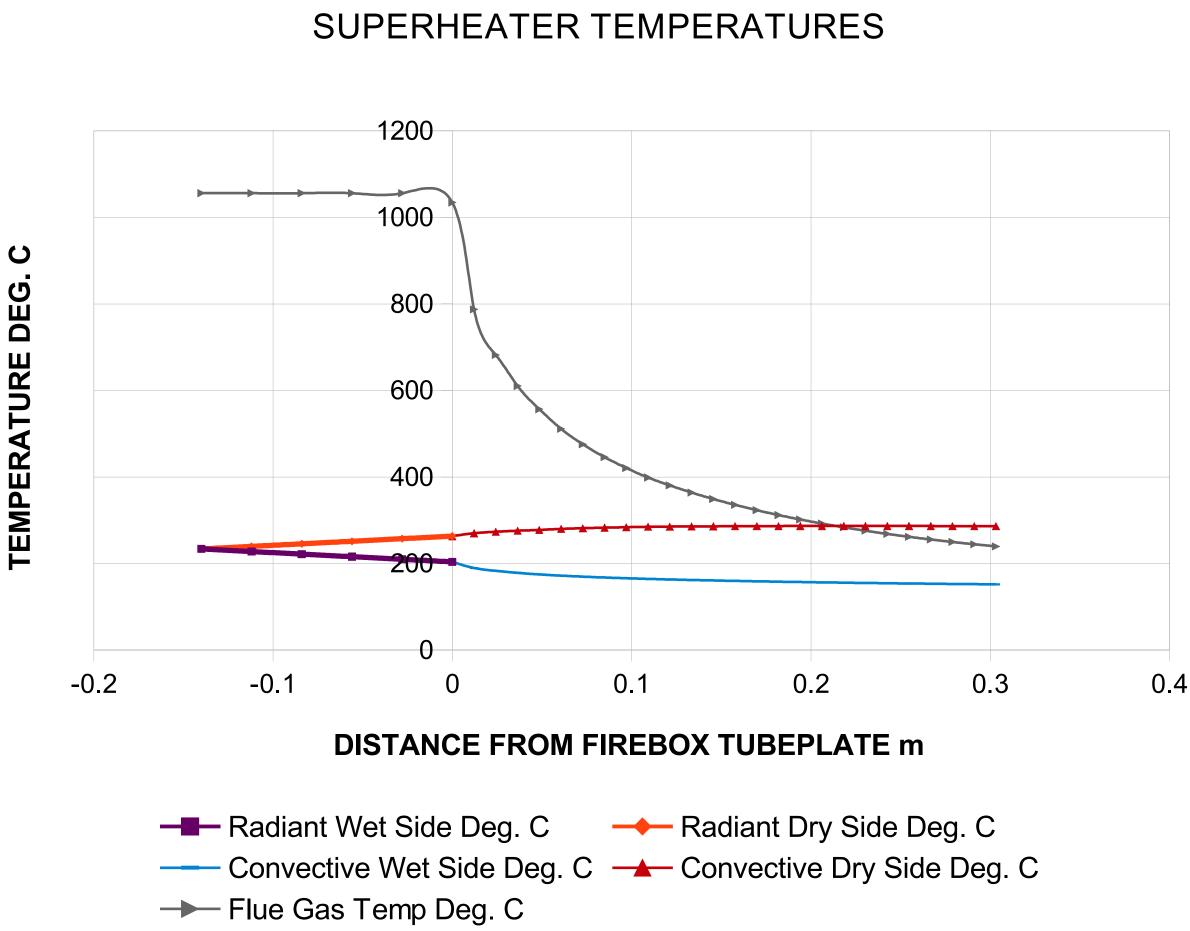

Figure 15 - Performance of Radiant Superheaters on "Speedy"

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.

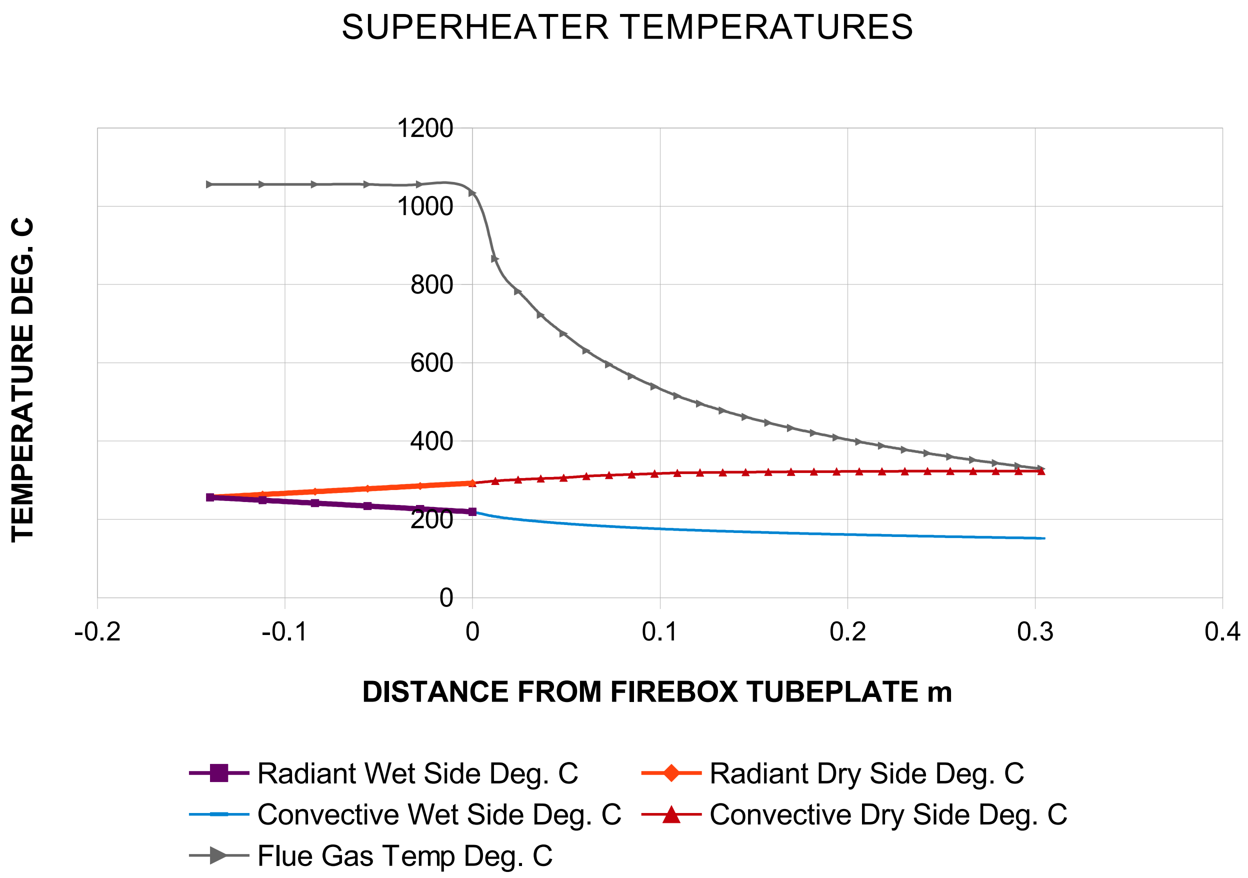

Figure 16: Superheater performance in Roger Froud's "Speedy" boiler design

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 arrangment 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.

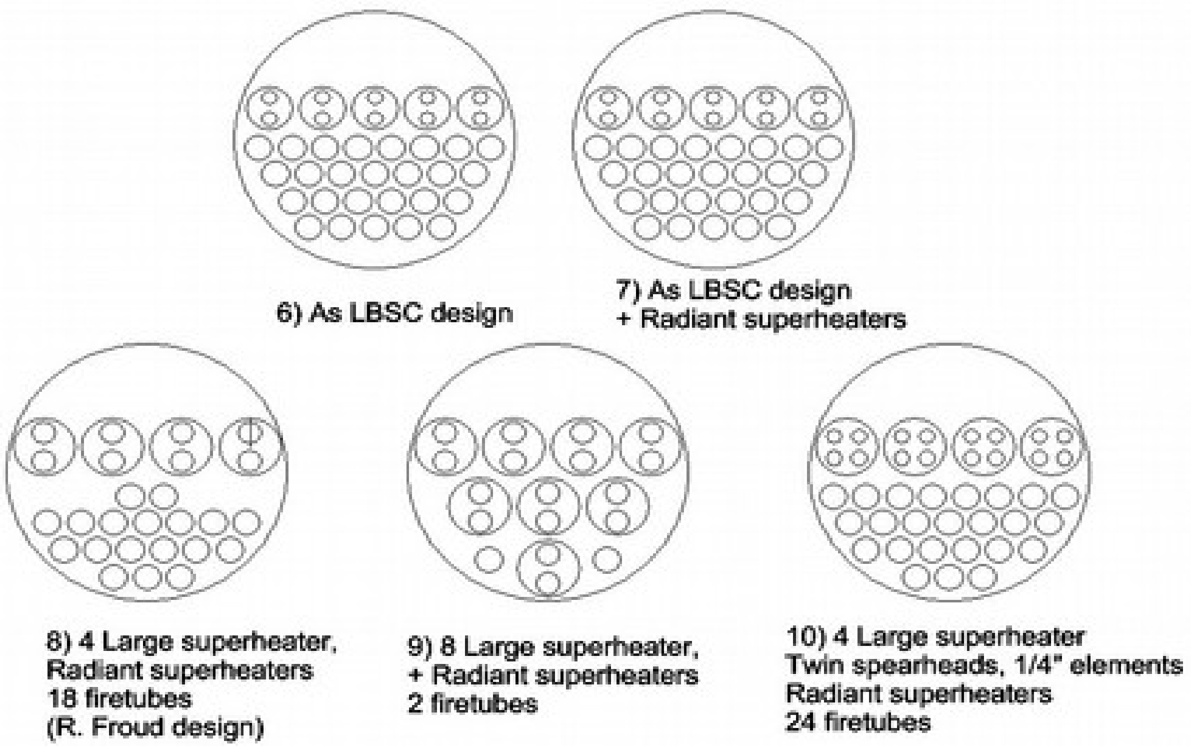

Figure 17: Summary of superheater options tried

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.

7 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; I would also like to get the program up on the net somehow, but I am not keen on starting a web page; if anybody can help I would be pleased to hear from you. 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!

If you would like to contact me, I am Martin Johnson 1 on the Model Engineer forum.

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