INFLUENCE OF MARINE BOILER CONDITIONS ON FURNACE RATINGS AND DESIGN

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ABSTRACT

The extent of the influence of steam pressure and degree of superheat, feedwater temperature and preheated air temperature on recent marine boiler ratings and furnace exit gas temperature-for specified designs of marine boilers are the territory of investigation in this paper. The locus of the marine standard steam conditions w.r.t. the plotted charts is aimed to be displayed and discussed.

NOMENCLATURE

AF	Actual air fuel ratio (--)
cp	Specific heat at constant pressure of superheated steam (Btu/1b$_m$. °F) or (KJ/kg.k).
C.F.M	Required Cubic feet per minute combustion air (ft^3/min) or (m^3/min).
EV	Required boiler evaporation (1b/hr) or (kg/hr).
FR	Fuel rate (firing rate) (1b/hr. ft^2 RHA) or (kg/hr.m^2).
F.W.T.	Feed water temperature (°F) or (K).
H_i	Energy (heat) input to the boiler (Btu/hr) or (KJ/hr).
H_o	Energy (heat) output of the boiler (Btu/hr) or (KJ/hr).
H_n	Net heat input to the boiler (Btu/hr) or (KJ/hr).
HAR	Heat absorption rating (Btu/hr. ft^2 RHA) or (KJ/hr.m^2).
H.C.V.	Higher calorific value of fuel (Btu/1b$_m$) or (KJ/kg).
HRR	Radiant heat surface heat release rating (Btu/hr.ft^2.RHA) or (KJ/hr.m^2).
HLR	Furnace volume heat release or liberation rating (Btu/ft^3.hr) or (KJ/hr.m^3).
$I_{F.W}$	Feedwater enthalpy (Btu/1b$_m$) or (KJ/kg).
I_{sat}	Enthalpy of dry and saturated steam (btu/1b$_m$) or (KJ/kg).
I_{sup}	Enthalpy of superheated steam (Btu/1b$_m$) or (KJ/kg).
L.C.V.	Lower calorific value of fuel (Btu/1b$_m$) or (KJ/kg).
P	Absolute Pressure of steam (Psia) or (KPa).
RHA	Radiant heat absorbing surface (Ft^2) or (m^2).
T_a	Preheated combustion air temperature (°F) or (K).
T_F	Furnace exit gas temperature (°F) or (K).
V	Furnace Volume (Ft^3) or (m^3).
ΔT	Degree superheat of steam above saturation temperature at boiler pressure (°F) or (K).
WF	Weight of burnt fuel per hour. (1b/hr) or (KN/hr).
η_b	Boiler thermal efficiency (-----).

INTRODUCTION

In the early stages of the design of marine oil fired water tube boilers a trial and error procedure was performed in order to roughly determine the radiant heat absorbing surface and consequently the furnace gas temperature; a matter which needed several trials before a satisfactory answer was reached. It was customary to rate the marine boilers in horsepower defined as the evaporation of 30 pounds of water per hour at 10 psig from feedwater at 100°F. The definition was altered to the evaporation of 34.5 1b/hr at 212°F representing 33,975 btu/hr. Nevertheless, the term boiler horsepower is now obsolete [1].

Other boiler ratings criteria came into appearance namely: the evaporative rating expressed in Btu per square foot of steam generating surface and the equivalent evaporative rate based on the absorption of 970 Btu. Both ratings became obsolete.

Besides, the generating, surface heat release rating expressed in Btu per square foot of steam generating or even per square foot of the total heating surface were adopted, however both became of little importance in boiler design [1].

Recently used boiler ratings [1] could be summarized as follows:

a. The radiant heat surface heat release (HRR). expressed in Btu per square foot of radiant heat absorbing surface. It is a criterion of the furnace heat absorption rate HAR but it does not indicate the tube temperature or even the increase in HAR due to preheating the combustion air.
b. The furnace volume heat release or liberation rate (HLR). expressed in Btu per cubic foot of furnace volume which is a measure of the time taken by the combustion gases to cross the furnace to the tube bank [2] and whose limitations given in [3] for merchant ships range from 80,000 to 100,000 Btu/ft^3.hr.
c. Heat absorption rating HAR expressed in Btu per square foot of radiant heat absorbing surface RHA which represents the true loading on the furnace tubes, the most highly loaded boiler component.

In contrast to HRR, it takes into account the relationship between the absorbing and the radiating surfaces as well as the effect of the preheated combustion air. Limitations imposed on HAR as given in [3] range from 100,000 to 150,000 Btu/ft^2.hr for merchant ships.

Likewise mechanical stresses HRR, HAR and HLR represent thermal loading on the furnace which should withstand and resist.

Similarly, another two resulting furnace loading criteria are the fuel rate or fire rate FR expressed in 1b/hr.ft^2 (RHA) and the furnace gas exit temperature TF whose theoretical and empirical evaluation is best illustrated in [4]. Limitations of furnace gas exit temperature in merchant ships occupy the range from 2400°F to 2850°F [3] when firing Bunker "C" fuel with 15% excess air.

It is needless to emphasize that the furnace design and exit gas temperature affect too the superheater gas temperature and heating surface. Further details are widely discussed in [5,6] where marine standard steam conditions [7] are applied.

Additionally, an acceptable conventional range of Fr is bounded by 8 and 12 1b/ft^2.hr.

The relationship combining HRR, HAR and T_F with 15% excess air and a mean line of emissivities is demonstrated in [5].

Relationships between HAR and T_F-for different emissivities ranging from 0.1 till perfect radiator Figure (1) and also between HAR and T_F for different fuel ratings Figure (2) are presented in [4].

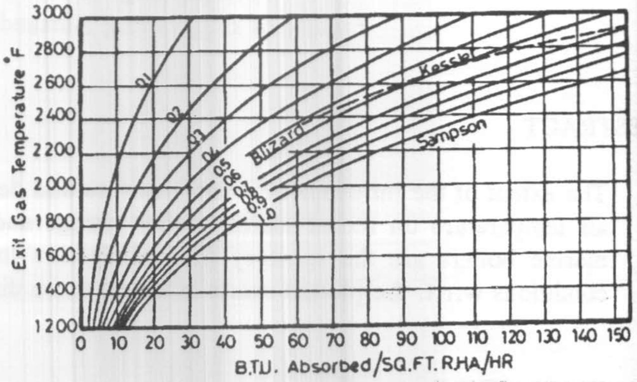

Figure 1. Exit Gas temperature versus enissivity and HAR.

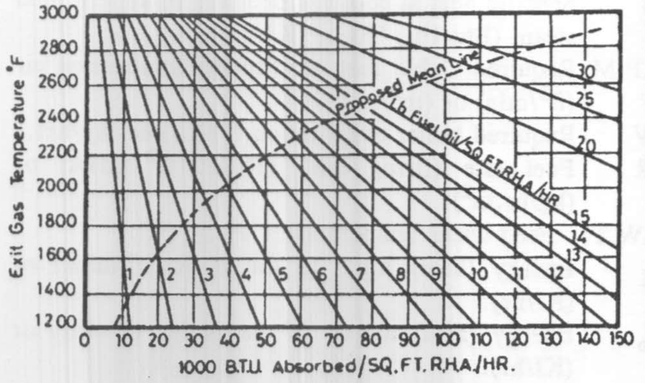

Figure 2. Exit Gas temperature versus fuel rate and HAR.

Moreover, a powerful graph-Figure (3)-used when deviating from the normal 15% excess air ratio and combining HRR, T_F and excess air ratio is included in [4] too. On the other hand, another appreciable graph Figure (4) relating the relationships between HRR, T_F and HAR is demonstrated in [5] for 15% excess air ratio and based on the proposed mean line [4]. It is worthmentioning that this graph was processed to the digital computer to carry out the third degree parabolic interpolation around the required value by Gauss-Jordan successive elimination numerical method in the solutions of this study.

In order to determine the RHA, the effectiveness factor-Figure (5)-reproduced from [4]-should be multiplied by the projected area of the furnace sides where waterwalls are located.

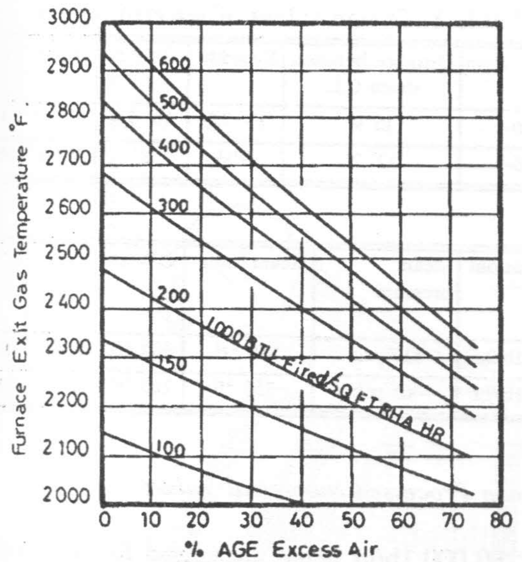

Figure 3. Furnace exit Gas temperature versus % exces air and HRR.

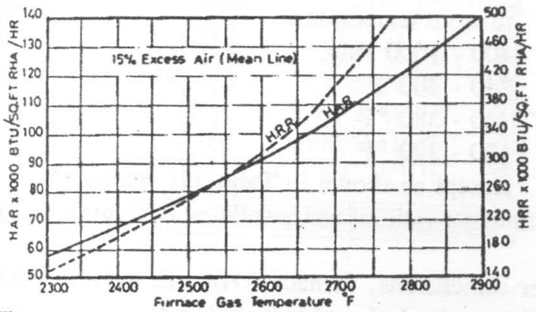

Figure 4. Relationship between HRR, HAR and T_F.

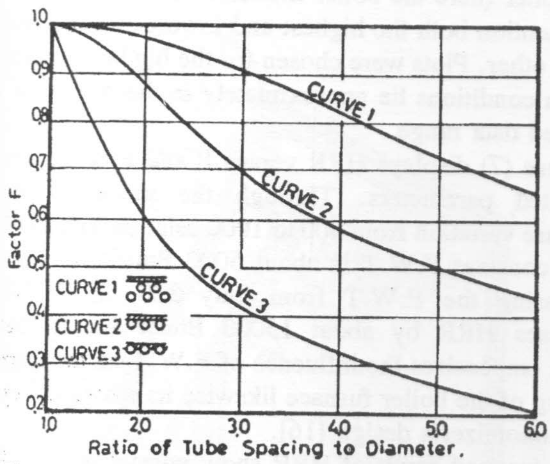

Figure 5. Effectiveness factor.

Furnace design Principles

In [5,8] are cited the factors affecting the furnace dimensions which together with the steam conditions, evaporation required, feedwater temperature, preheated combustion air temperature and desired boiler efficiency govern all furnace ratings and exit gas temperature.

To encounter concisely the major fundamentals influencing the furnace dimensions which are briefly reproduced for convenience:

1. Type of boiler under consideration and its general design which are discussed in details in [3,9].
2. Fire room space limitations.
3. Service requirements.
4. Burners number, size, location, arrangement and clearances.
5. Prerequisites for proper thermal and circulation performance.
6. Amount of furnace waterwalls.

Scientific disagreement on boiler ratings and furnace waterwalls

The aforementioned HLR criterion according to which boilers were early designed is not the best criterion, since it does not take into account the scope of furnace water walls [2,3].

Considering two-cube-shaped-furnaces, the first has 10 ft while the second has 20 ft in each dimension and the boiler conditions being the same, the second would be suitable on a liberation rate for eight times the capacity of the first while the RHA would be only four times that of the first and the firing rate would be twice as high [10]. Since as mentioned before, some considerations are not involved in HRR it would not represent the best criterion too. Regarding HAR, according to [2,3], increasing RHA lowers both HAR and T_F with the associated consequences of less critical and sensitive circulation, reducing the superheater slagging and corrosion and finally elongating the life of firebrick too. Knowingly, HAR depends on the Stefan-Boltzman law but radiation involves some factors that are far beyond mathematics where each designer has developed his own constants and empirical formulae.

Scientific discussions or even "disputations" cited in [11] conclude that it is very difficult or even impossible to agree on values of HAR which represent the best criterion. It is contested to be a comparison measure in furnace design. The final recommendation was to select the firing rate criterion even though it does not take into consideration the combustion air temperature, excess air ratio and burner arrangements.

In reference to the amount of furnace waterwalls, the appreciated merits of reducing T_F are outweighed which imply in turn, increasing the waterwalls surface.

Nevertheless, the necessity of some amount of refractory exists in order to radiate heat to flame and assure stable fires particularly at low loads. Whatever the merit of this argument, fully water cooled furnaces have been in service and this trend is dominating.

For further informations related to furnace and boiler design, reference is made to [12, 13, 14].

Boiler furnace in Warships

Combatant ships [15] are characterized not only by space and weight restrictions but also by particular prerequisites such as power (evaporation, steam conditions and vacuum), speed, reliability and manouvrability whose reflex action is represented in much higher furnace ratings and shorter time taken by the combustion gases to cross the tube bank [1, 2].

Objectives of This Study

One of the studies presented in [4]-Figure (10)-is the relationship between boiler evaporation (ranging from 10,000 to 60,000 1b steam/hr and the fuel consumed in 1b/hr. Selected F.W.T were 240°F and 320°F whereas steam conditions were chosen as (465 psia, 750°F) and (800 psia, 800°F or 1000°F).

Normally, each evaporation (or small range of evaporations) corresponds to a specified boiler and consequently furnace design with its own dimensions. In contrast and as an extended contribution to this study, the question arises if an already existing boiler properly designed for certain requirements of load, conditions of steam. F.W.T. and combustion air temperature what will occur if it were to operate at other conditions apart from or excessively beyond its designed point.

In other words, supposing that, capacity, mechanical and thermal stresses were taken into consideration, it may be doubtful if the furnace will thermally withstand the new resulting ratings and exit gas temperature.

For the first instant, the answer may be no. while exactly the contrary will be revealed in this paper accompanied by engineering logical analysis.

Digital Treatment and Discussion

Repetitive computations were executed whose brief fundamentals are summarized in the appendix. Two designed vertical superheater marine boilers for cargo ships were selected to the study Figure (6-a,b) and whose design data are indicated in Table (1). Calculations were carried out in F.P.S. system for the sake of quick comparison with the published standards.

Table 1. Design values of selected boilers

Item / type	Distance between drum C.L.	Breadth	Width	RHA	Volum
Fig. 6-a	15'¼	10' 7/8	10' 3/4	523 ft^2	1470 ft^3
Fig. 6-b	12' 7"	9'½	8'	357 ft^2	860 ft^3

Evaporationt	Steam pressure	Steam temp.	F.W.T.	Ta	b
80000 1b/hr	615 psia	850 °F	280 °F	120 °F	0.88
50000 1b/hr	465 psia	750 °F	240 °F	120 °F	0.875

Common Processed Numerical Values

EV 80,000 1b/hr being unchanged for both boilers. The aim being to investigate how much excessive load (on the small boiler) affects the ratings and furnace temperature.

P 400 - 1000 Psia

ΔT 240 - 400°F

F.W.T 240 - 380 °F

T_a 120 - 320 °F

η_b is kept as shown in Table (1) unaltered.

Pitch/diameter ratio of waterwalls = 1.7 (curve 2, Figure (5)).

Boiler dimensions, furnace RHA and volume are held invariable as the designed values. It may be sufficient and satisfactory to avoid monotony-to plot only the results of any boiler (here the boiler illustrated in Figure (6-a) and just mention both the highest and lowest obtained results of the other. Plots were chosen for the big boiler since its design conditions lie approximately in the middle of the scanned data range.

Figure (7) displays HRR versus P and F.W.T. for the indicated parameters. Through the scope of steam pressure variation from 400 to 1000 psia, the rise of HRR for a constant F.W.T is about 5000 Btu/ft^2.hr whereas decreasing the F.W.T from only 320 °F to 240°F increases HRR by about 15000 Btu/ft^2.hr. A matter which emphasizes the influence of F.W.T on the thermal loading of the boiler furnace likewise its strong effect on the economizer's design [16].

The scanned result of HRR show normal or even low values where the most lowest and greatest values of HRR ar 187,500 and 217,500 Btu/ft^2.hr.

The corresponding values for the smaller boiler additionally overloaded with 60% of its load are 275,000 and 319,000 Btu/ft^2.hr respectively.

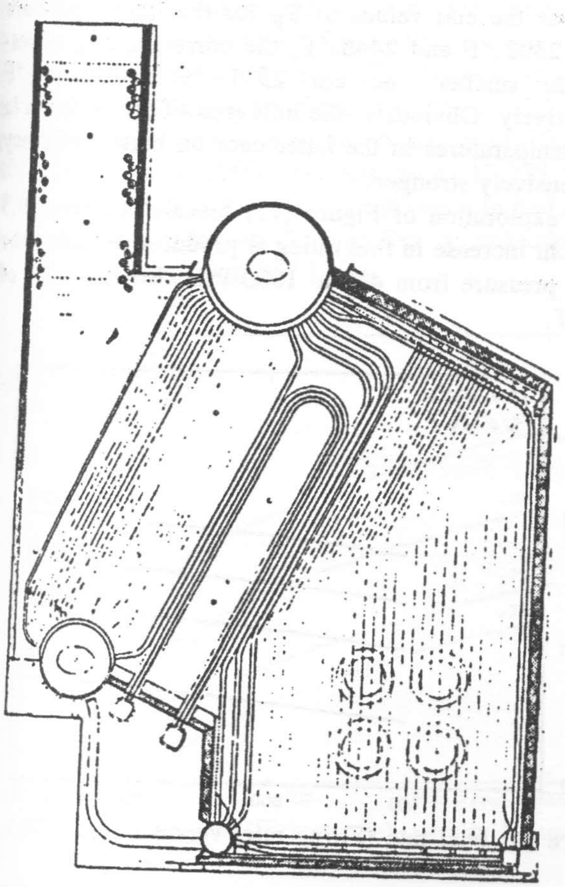

Figure 6-a.

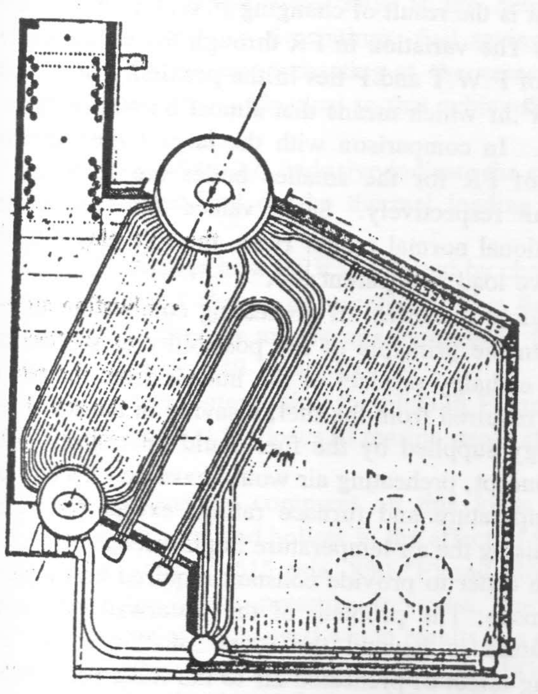

Figure 6-b. Sectional views of the selected boilers.

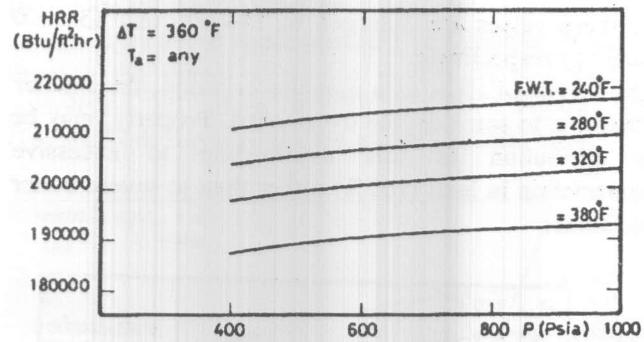

Figure 7. HRR versus P and F.W.T.

In what concerns Figure (8), the maximum effect of increasing the steam pressure from 400 to 1000 Psia on HAR does not exceed 1000 Btu/ft^2.hr provided that the F.W.T is kept invariable. In contrast to P about 3000 btu/ft^2.hr increase in HAR are assumed when lowering the F.W.T. from 320 °F to 240 °F. The furnace proves to be at any conditions thermally under loaded.

Scanning extreme values of HAR yields 67, 750 and 73,000 Btu/ft^2.hr for the large furnace whereas 99,250 and 107,000 Btu/ft^2.hr are picked out for the smaller one; such values lie in the normal range.

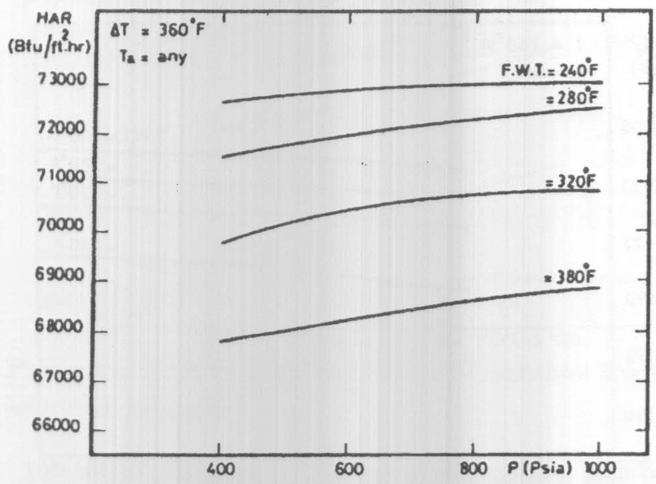

Figure 8. HAR versus P and F.W.T.

Considering Figure (9), the whole range of steam pressures-at a specified F.W.T adds about 2000 btu/ft^3.hr in HLR whereas the prementioned decrease in F.W.T provides additional thermal loading reaching about 5000 Btu/ft^3.hr on the furnace. Here, w.r.t. HLR, the furnace is almost normally loaded at low F.W.T (240 °F) and thermally under loaded at higher values of F.W.T.

Limit values of HLR reach about 68,000 and 78,500 Btu/ft^3.hr for the larger furnace, however, for the smaller one these values are enlarged to 116,000 and 134,000 $Btu/ft^3.hr$ respectively.

On a basis of HLR as a measure criterion, the smaller furnace, is to some degree overloaded. Properly, may be the attribution of this overloading to excessive augmentation in load (160%) rather than to severe boiler conditions.

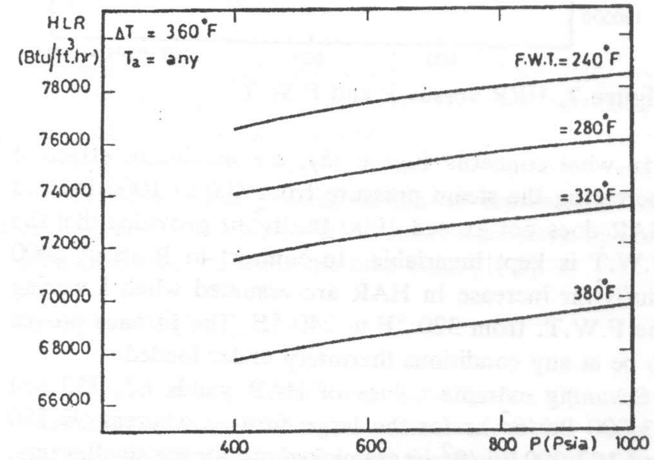

Figure 9. HLR versus P and F.W.T.

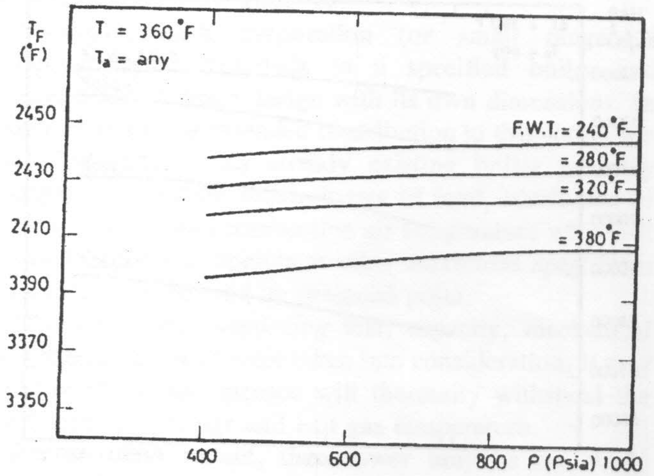

Figure 10. T_F versus P and F.W.T.

When examining the plot of the furnace exit gas temperature-Figure (10-a) maximum deviation reaching about 55 °F is registered through the whole graph. Such deviation, despite seeming minute, if prevails till the stack gas temperature and-assuming no change in radiation losses causes a deviation by about 0.8% in boiler thermal efficiency, which corresponds to a variation of about 55 1b/hr in fuel consumed or, in other words, 153 tons fuel per navigating year of 260 days. Whereas the end values of T_F for the large boiler are about 2392 °F and 2448 °F, the corresponding values for the smaller one are 2535 °F and 2625 °F respectively. Obviously, the influence of the deviation in limit temperatures in the latter case on boiler efficiency is extensively stronger.

The exploration of Figure (11) reveals that about 0.3 $1b/ft^2.hr$ increase in fuel rating is produced at raising the steam pressure from 400 to 1000 Psia, for any value of F.W.T.

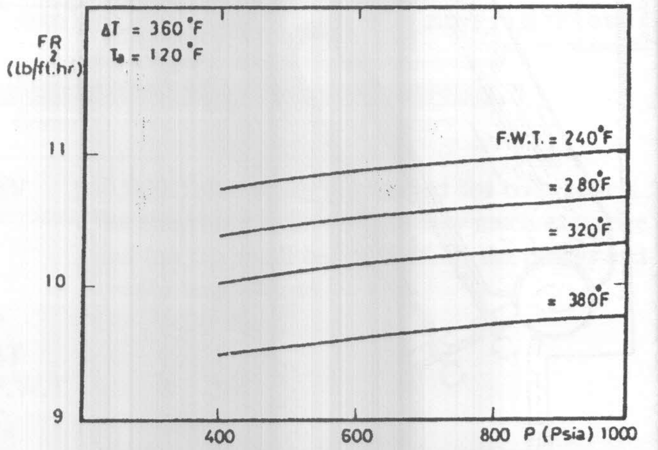

Figure 11. Fuel rate (Firing rate) versus P and F.W.T.

On the contrary, a change in fuel rating reaching 0.75 $1b/ft^2.hr$ is the result of changing F.W.T from 240 °F to 320 °F. The variation in FR through the whole scanned ranges of F.W.T and P lies in the proximity from 9.5 to 11 $1b/ft^2.hr$ which means that almost normal ratings are yielded. In comparison with the large boiler, extreme values of FR for the smaller boiler are 13.9 and 16 $1b/ft^2.hr$ respectively. Such values are beyond the conventional normal range. Here, the role played by the excessive load is apparent too.

Regrading the effect of preheating combustion air and based on the principle of the possibility to subtract the rate of enthalpy product of the hot air from the rate of energy required from the fuel, a saving in the actual rate of energy supplied by the fuel could be realized. With such concept, preheating air would have no effect on exit gas temperature and furnace ratings except fuel rating since raising the air temperature implies reducing the fuel burnt in order to provide constant required heat input to the furnace. The problem of the waterwall tube metal temperature is beyond the scope of this study. The radiating effect of preheated air to the walls needs rather experimental instrumentation and will be discarded as well. As an avoidance of repetition only the relationship of fire rating versus steam pressure for various air temperatures is displayed in Figure (12).

The discussion of Figure (12) demonstrates that preheating combustion air from 120 °F to 320 °F if possible reduces the fuel rating from about 10.62 to 10.28 $lb/ft^2.hr$ at 1000 Psia and from 10.375 to 10 $lb/ft^2.hr$ at 400 Psia. The average alteration in fuel rating is 0.3575 $1/bft^2.hr$ which signifies a reduction in consumed fuel attaining 187 lb/hr.

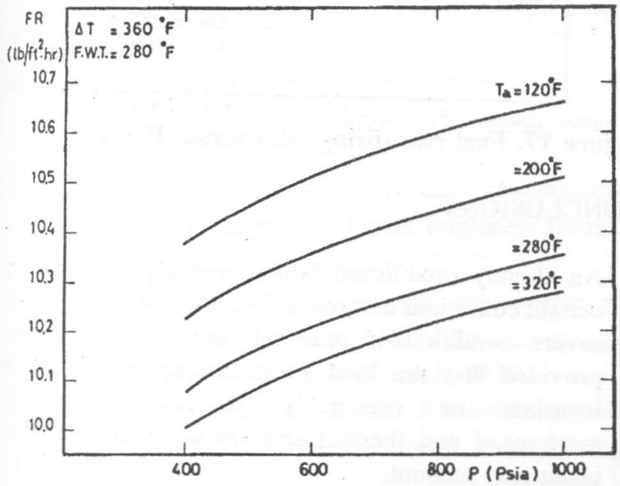

Figure 12. Fuel rate (Firing rate) versus P and air temperature.

Since the radiant heat areas ratio is the magnifying constant relating the two fire ratings, fuel saving of the small boiler due to air preheating-at the same above mentioned conditions-is identical to that achieved by the larger one.

Figures from (13) to (17) inclusive indicate the effect of the degree of superheat on the thermal loading on the furnace.

Figure (13) displays how HRR is affected by changes in steam pressure and degree of superheat. Five thousand $Btu/ft^2.hr$ are noticed as an average increase in HRR-at a specified ΔT from 240 °F to 400 °F.

The dotted curves intersecting the plots in Figures (13) through (17) represent the locus of marine steam standard conditions.

A simple explanatory comment on this locus is the attribution of the crest and hollow to the effect of the two standard conditions of (615 psia, 850 °F) and (865 psia, 850 °F). As the steam pressure increases, the steam temperature remains unchanged which results in a drop in the degree of superheat. Considering Figure (14), HAR is augmented by about one thousand $Btu/ft^2.hr$ through the scope of the scanned pressures and is considerably increased by about two thousand five hundreds $Btu/ft^2.hr$ through the range of studied degree of superheat. In completion, all values of HRR and HAR Figure. (13), (14) for both furnaces do not exceed normal ones.

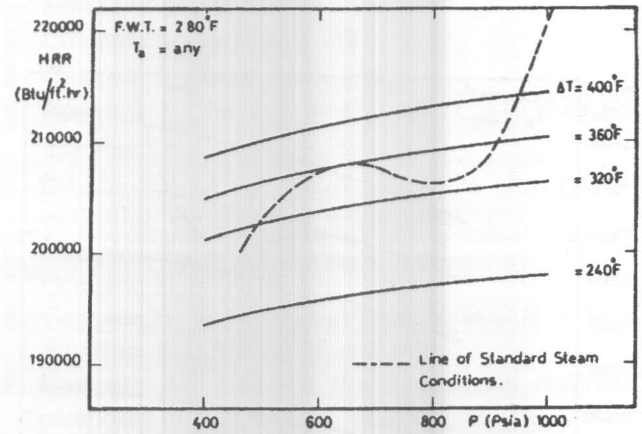

Figure 13. HRR versus P and degree of superheat above saturation temp. (ΔT).

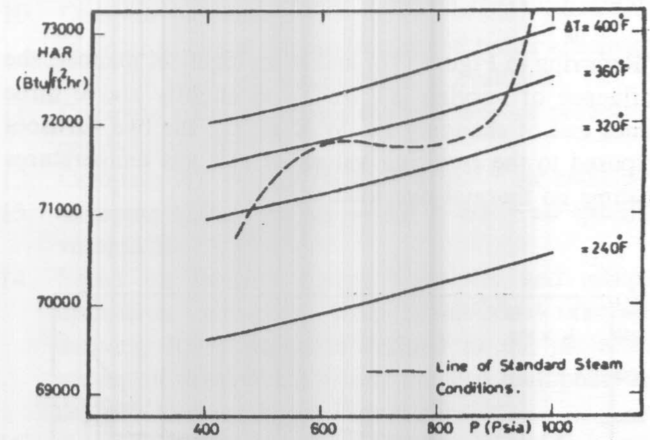

Figure 14. HAR versus P and degree of superheat above saturation temp.

Inspection of Figure (15) shows that increasing each or both P and ΔT thermally overloads the furnace based on HLR criterion. Additional two thousand $Btu/ft^3.hr$ approximate increase in HLR is the result of raising the pressure through its extreme limits. The influence of shifting ΔT from 240 °F to 400 °F on HLR is almost three times that of shifting the pressure through the range under discussion. HLR values for the large boiler are just below the normal range; adversely, values corresponding to the smaller one are rather excessive. The designed volume ratio of the two furnaces approaches 1.71 while the designed RHA ratio reaches about 1.465; a matter which explains why-in contrast to HRR and HAR,HLR becomes excessive for the second boiler particularly its load was raised by 60%. This is the fact known as the ratio of augmentation in volume is exceedingly larger than that of increase in area.

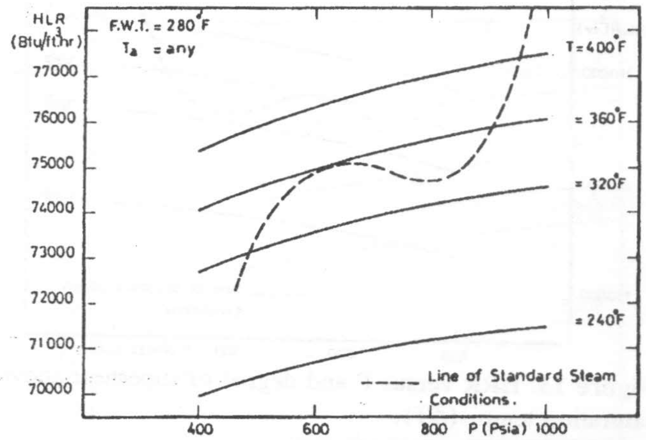

Figure 15. HLR versus P and ΔT.

Referring to Figure (16) and in an identical manner, the influence of varying ΔT on T_F is slightly above three times that of changing P. W.R.T. T_F, the two furnaces exposed to the resulting values of exit gas temperatures assume no thermal overloading.

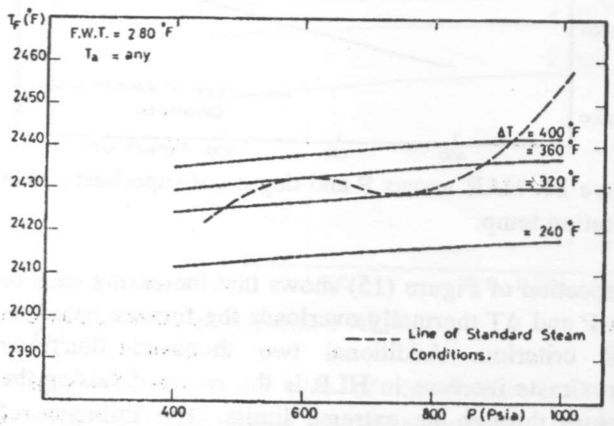

Figure 16. T_F Versus P and ΔT.

Concluding with Figure (17), the change of ΔT produces an approximate alteration in firing rate attaining 0.8 1b/ft^2.hr which is roughly three times the influence of the scope of pressure variation. The aforementioned ratio holds good for the small boiler too.

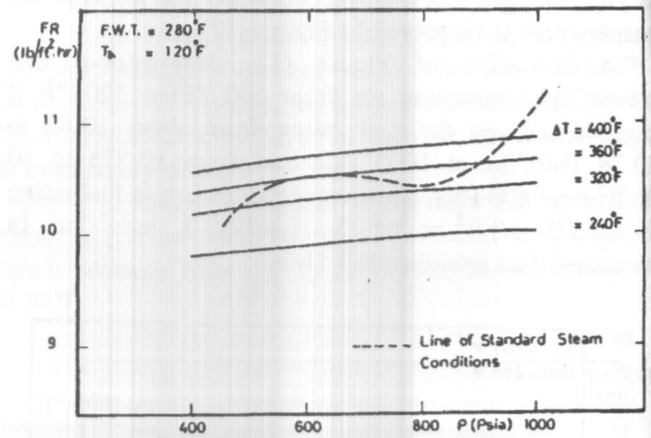

Figure 17. Fuel rate (firing rate) versus P and ΔT.

CONCLUSIONS

1. An already established boiler properly designed for certain conditions assures withstandability of any other severe conditions-if practical and not hypothetical provided that the load (evaporation) is either held invariant or reasonably increased. Capacity, mechanical and thermal stresses are assumed to be taken into account.
2. The effect of F.W.T on furnace ratings and exit gas temperature is sufficiently considerable.
3. Less significant is the influence of degree of superheat since the growth of steam enthalpy w.r.t. ΔT is almost with a little constant rate.
4. The steam pressure assumes the most feeble influence due to the rapid decay of the growth of steam enthalpy at high pressures. The effect of the whole scope of pressure variation is not so sensitive to changes in F.W.T.
5. Tangible fuel or energy saving could be achieved by air preheating; a matter which deserves an optimizing analysis of the steam generating equipment. Despite the relatively low value of specific heat at constant pressure of preheated air, its amount, in contradiction is extremely considerable.
6. HLR and HAR could not be considered similar criteria in judging the thermal loading on a boiler furnace.
7. The locus of the marine standard steam conditions is displayed and discussed. With the large boiler, this locus does not represent any thermal overloading at all, while with the smaller one, only HLR and FR could exceed the conventional upper limits due to excessively raised evaporation.

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APPENDIX:

Concise Procedure of Computations (FPS System)

1. Compute F.W.T from final heat balance of the marine steam power plant.
2. Compute $I_{F.W}$=F.W.T -32
3. Compute I_{sat} from steam tables .
4. Compute $I_{sup} = I_{Sat} + C_p \cdot \Delta T$, $C_p = 0.48 - 0.6$ Btu/1bm.°F
5. Decide desired η_b which should be checked when evaluating the stack gas temperature.
6. Compute $H_O = EV * (I_{sup} - I_{F.W})$
7. Compute $H_i = H_o/\eta_b$.
8. Compute $H_n = H_i$ * L.C.V/H.C.V, H.C.V = 18,500 Btu/1bm, L.C.V/H.C.V = 0.945.
9. Compute AF [16] for the fuel whose analysis is composed of : 87.64% C, 11.0%H, 0.53% S, 0.26% N, 0.57% O and traces of H_2O. With 15% excess air, AF equals 16, while AF equals 14.57 if the excess air ratio is only 5%.
10. Compute percentage CO_2 in combustion gases [16] in order to check the predetermined boiler efficiency once the stack gas temperature is evaluated.
11. Compute from air tables the enthalpy of preheated combustion air.
12. Compute WF = Hn/ (H.C.V + AF * air enthalpy)
13. Compute C.F.M for air = AF * WF * air specific volume/60.
14. Based on furnace design principles and related discussion compute from established furnace drawing RHA which is defined as the projected waterwall area which (sees) the fire multiplied by the effectiveness factor Figure (5).
15. Compute V in a manner similar to point 14.
16. Compute HRR = Hn/RHA
17. Compute HLR = Hn/V
18. Compute FR = WF/RHA
19. With the knowledge of HRR enter Figure (4) to determine both T_F and HAR. The successive elimination algorithm of Gauss-Jordan was used in third degree parabolic numerical interpolations on these graphs. HAR denotes the rate of supply of energy minus the sum of the rate of heat enthalpy of the products of combustion added to the rate of radiation per unit RHA.
20. All ratings and T_F should satisfy the design limitations agreed on.