ON THERMAL ANALYSIS OF MARINE ECONOMIZERS

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ABSTRACT

Fuel Saving or, in other words raising thermal efficiencies together with surface area reduction remain of the most momentous objectives of researchists in energy sciences. The marine economizer, being a rather complicated case of multi-variable heat recovery equipments is thermodynamically analysed for each of its influencing parameters. Surface ratio distribution between the main bank of tubes and the economizer is investigated too. Executed versions of digital programs adopt the Foster-Wheeler's incorporation design procedures.

INTRODUCTION

The marine economizer consists of a series of horizontal tubular elements by means of which heat is recovered from combustion gases leaving the boiler and added to the in-coming feedwater. The amount of heat absorbed in an economizer is dependent on the temperature difference between gas and water, the heat transfer rate and the amount of heating surface per meter of tube length. Economizers can be considered to fall into two categories: bare tube and extended surface types, the tube shape has the form of hair pin or multi-loop elements. Due to the fact that the coefficient of heat transfer is relatively low in the gas side of an economizer tube-as compared to that in the water side-it is desirable to have some form of extended heating surface on the outside of the tube so as to increase the overall rate of heat transfer. Each type of extended surface economizer has a heat transfer rate which depends on the shape, finishing of the surface, the material employed and the method of fabrication. Most marine economizers are designed for counter-flow of gas and water, that is water comes down through elements and gas comes up outside of elements in order to take the advantage of greater temperature difference between gas and water. Usually, the average temperature difference lies in the range of 80°C, i.e. from 200°F~250°F. Under these conditions, the heat transfer rate can not be increased without incurring a substantial resistance to gas flow and the use of extended surface to increase the heating surface per lineal meter becomes desirable. The heating surface must be kept free of deposits. The coexistence of moisture and deposits will cause corrosion and accordingly the spacing of the elements, the quantity, effectiveness of soot blowers should all be carefully considered. Moisture can come from a number of sources namely: down the stack, leaks in economizers or boiler pressure parts and condensation of moisture in gases. Attention should be made to the fact that tremendous amount of corrosion, erosion damage can result in a very short time. While the economizer is designed principally for steady steaming conditions, proper consideration should be given to the design of elements so that steam will not be generated within the economizer during maneuvrings, i.e. there must be suitable water velocities and pressure drop which will assure adequate circulation in each element. Under most conditions there is sufficient differential between saturation and feedwater temperature to prevent the generation of the steam. However, in order to obtain the required economizer's heat absorption, the conomizer length, width, number of tubes high (vertically spaced) and number of tubes wide (horizontally located) may have to be varied to obtain the best arrangement with respect to the space constraints and the thermal performance conditions. The objectives of this paper include such interesting points of research. Emphasis should be made to the permitted gas pressure loss and water pressure drop. Economizers are designed so that they can be by-passed if a leak occurs to allow the boiler to remain in service until the necessary repairs can be made. This means that economizer should withstand entering gas temperatures without any feed water flow through the tubes. With the economizer by-passed, the loss in efficiency results in an increased fuel firing rate and increasing draft loss causing a higher fan load. Recently designed ship boilers are characterized by high steam temperatures due to higher gas flow and increased furnace temperature, Ref. [1-11].

Economizers may be considered advantageous from the point of view of saving the up stack temperature, thus improving the boiler efficiency. In addition, their relative compactness due to the high specific heat of water is regarded as a distinguishing merit. Adversely, their high draft losses with the accompanied high fan power, the necessity of soot blowers, to guarantee an Oxygen-impurities free feed water and their difficulty of repair are characterized as drawbacks. Besides, the less effective performance of economizers at high F.W.T is not considered as a privilege. Some considerations helping in designing economizers and known as a rule of thumb are that the inlet gas temperature to the economizers exceeds that of the saturation temperature of steam by about 150°F. Moreover, the rise in F.W.T approaches one third of the gas temperature drop through the economizer. The exit water temperature from the economizer should be at least 50°F below the saturation temperature at the corresponding boiler pressure. Shortly, gas temperature drop through the economizer ranges from 200-300°F which corresponds to water temperature rise in the extent of 70-100°F. Concerning the rate of combustion gasses crossing the economizer, it could be approximately anticipated as about 1.25 the boiler's evaporation based on air-fuel ratio equals 16 and excess air factor equals 15%.

Recent developments in marine economizers, particularly in materials used are decisively inevitable due to the tendency of adopting high steam conditions, lowering the stack temperature, employing coal-fired boilers, rapid innovations in steam or binary nuclear-steam war ships, the introduction of waste heat boilers with supercharging marine Diesel engines and finally the up to date orientation to marine combined cycles-whether COGAS or CODAS Ref. [12-18]. Since most well reputed american boilers manufacturers still use F.P.S system of units, and their designing charts are [19,20] still dealt with in british system, therefore, the same system of units will be applied here. Nevertheless, fewer authors apply either the M.K.S or S.I. units Ref. [21,22].

NOMENCLATURE

AF Actual air fuel ratio (-)

a/d Spacing of main tube bank perpendicular to the passage of gas per tube diameter. (-)

C_p Specific heat at constant pressure of combusti gases BTU/1b.°F (kJ/kg.K)

c_p Specific heat at constant pressure of wa BTU/1b.°F (kJ/kg.k).

d Tube diameter of the main bank ft(m).

d..... Infinitiesmal change of .... (any units).

d_{eq} Equivalent diameter of the finned tubes of economizer ft (m).

ds Infinitesmal change of economizer's heat surface ft^2 (m^2).

Eb Economizer's breadth.

EV Boiler's evaporation 1b/hr (kg/hr).

FA Free area for the passage of gases through economizer ft^2 (m^2).

F_a Arrangement factor for the main tube bank, (

F_n Factor corresponding to the number of rows,(

F.W.T Feed water temperature coming from the heater to the economizer F° (k).

H.C.V Higher calorific value of fuel BTU/$1b_m$ (kj/k

H.S Total heating surface of both main tube ba and economizer ft^2 (m^2).

$I_{f.w}$ Enthalpy of the feedwater. BTU/1b (kj/kg).

I_{sup} Enthalpy of the superheated steam BTU (kj/kg).

L.C.V. Lower calorific value of fuel BTU/$1b_m$ (kj/k

L.O.R Length over ring (effective length of one tube economizer) ft(m).

MF Mass flow rate of gasses 1000 1b/hr ft^2 free a (1000 kg/hr m^2 free area).

MF' Gas mass flow through the main tube bank 1(1b/ft^2.hr (1000 kg/m^2.hr).

N.H Number of economizer's tubes spaced hi (vertically)

N.W Number of economizer's tubes spaced wi (horizontally)

R Overall heat transfer coefficient assumed const BTU/ft^2.hr.°F (kj/m^2.hr.K).

R' Overall heat transfer coefficient for the main tu bank. BTU/ft^2.hr.°F (kj/m^2.hr.k)

S Total heating surface of economizer ft^2 (m^2)

S' Total heating surface of the main tube bank (m^2).

S.R The ratio of heating surface of main bank tubes to the total heating surface of both the ba of tubes and the economizer.

T Temperature of gasses at any point through the economizer °F (K).
T_1 Inlet gas temperature to economizer °F (K).
T_2 Outlet gas temperature from economizer (stack gas temperature) °F (K).
T_N Inlet gas temperature to economizer °F(K)$=T_1$.
T'_N Inlet gas temperature to the main bank of tubes F° (K)
T_{N+1} Stack gas temperature °F (K) $= T_2$.
T'_{N+1} Exit gas temperature from the main bank of tubes = Inlet gas temperature to the economizer $=T_N$ °F (K).
T_S Saturation temperature of steam at boiler's pressure. °F (K).
t Temperature of water at the corresponding point to T through the economizer °F (K)
t_1 Temperature outlet of the economizer °F (K)
t_2 Inlet feed water temperature to the economizer °F (K)
W Rate of flow of gases lb/hr (kg/hr)
WF Fuel rate lb/hr (kg/hr).
w Rate of flow of water lb/hr (kg/hr).
α Heat transfer coefficient by convection in the main tube bank BTU/ft² hr °F (kJ/m².hr.k).
γ An average temperature dependent factor (-).
$\Delta\eta_b$ Incremental change in boiler's efficiency (-).
η_b boiler's efficiency (-).
η_{ec} Economizer's efficiency (-).

COMPUTATATIONAL ASSESSMENT AND DISCUSSION

An introductory part of this section is to define figures from (1) to (4) inclusive. Figure (1) is a schematic illustration of the temperature's distribution in counter-flow heat exchangers, whereas Figure (2) represents a pictorial performance chart of the economizer's efficiency in terms of its principal parameters (equation (17)-Appendix I). Concerning Figure (3), it assists in the determination of the boiler's efficiency for a specified fuel oil analysis, provided that both the stack temperature and the percentage CO_2 in flue gases are established. Considering Figure (4), it graphically demonstrates the variation of the overall heat transfer coefficient with respect to the mass flow and the dimensions of economizer's tubes and fins. The pre-mentioned charts belong to the Foster Wheeler incorporation design charts collection.

In this research, a thorough thermal analysis of the economizer in a specified two-drum, water bent tubes, vertical walk-in superheater is carried out.

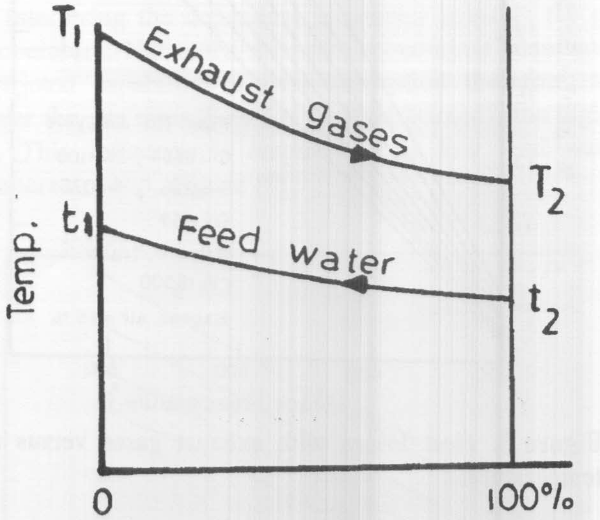

Figure 1. Percentage economizer's heating surface versus temperatures.

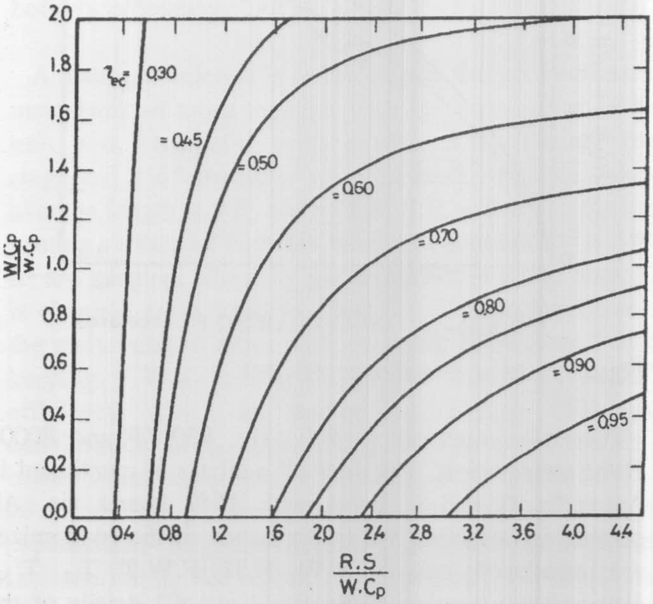

Figure 2. Performance curves of economizers.

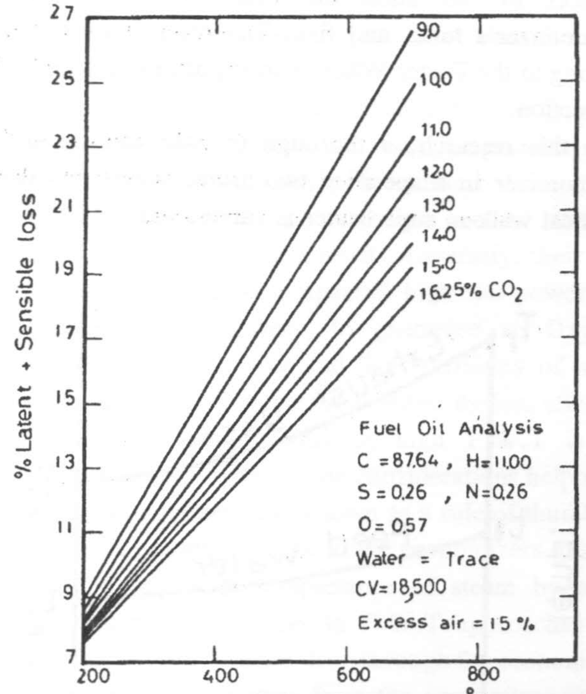

Figure 3. Heat losses with exhaust gases versus stack temperature.

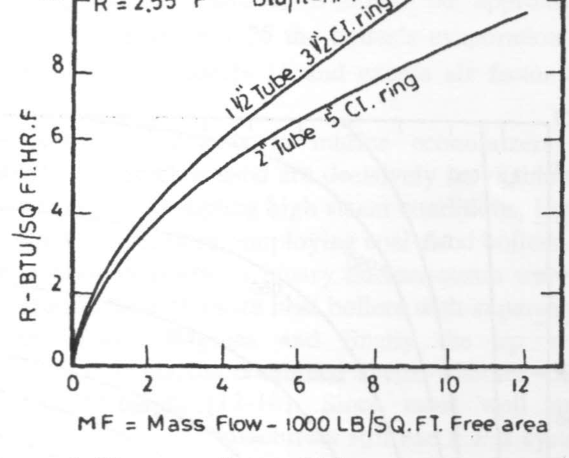

Figure 4. Heat transfer coefficient.

The boiler operates at 600 psig, 850 °F and 80000 1b/hr evaporation. The fuel oil analysis is mentioned in Appendix II and is burnt with 15% excess air. All parameters affecting the performance of the economizer are introduced; namely N.W, N.H, F.W.T, T_N, T'_N, economizer's heating surface and η_b. All details of the digital computations whether fundamental proofs or design procedure for a typical objective are demonstrat in Appendices I and II. To proceed, Figure (5) illustrat the variation of η_b with respect to T_N and F.W.T at t indicated parameters. Both T_N and F.W.T are inversel proportional to η_b at the fixation of the other paramete Increasing feedwater temperature from 240 °F to 380 ° assumes a drop of about 3% in η_b while increasing T from 500 °F to 700 °F assumes a drop of less than 1 in η_b.

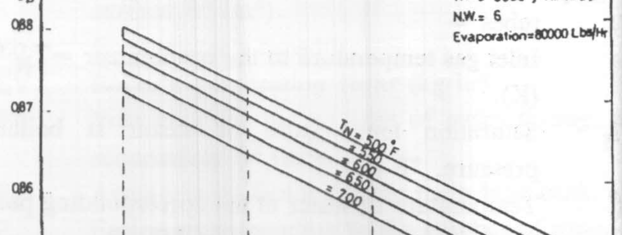

Figure 5. Effect of feed water & gas inlet temperatur on boiler's efficiency.

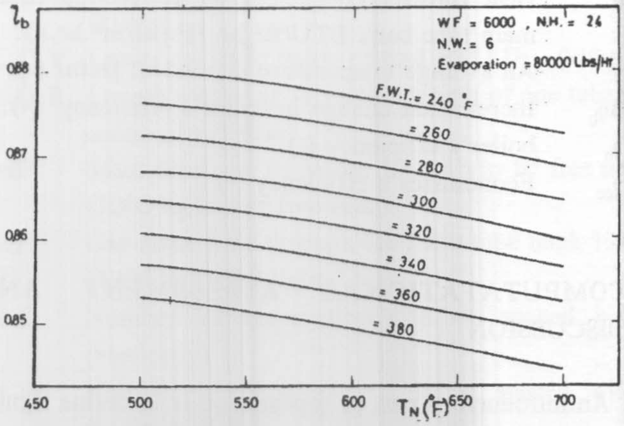

Figure 6. Effect of gas inlet & feed water temperature on boiler's efficiency.

This certifies that the influence of F.W.T is mor predominant than that of T_N. The dotted lines shown o all figures at F.W.T = 240, 280 °F and 320 ° illustrate the cases when only two feedwater heaters wit atmospheric deaerator, two feed water heaters wit double atmospheric deaerator and three feed water heater usually with an air preheated are incorporated into th steam cycle respectively. Figure (6) is a modifie alteration of Figure (5) but emphasizes the significant role of F.W.T on η_b. In what concerns Figure (7), it represents the effect of both N.W and F.W.T on η_b at fixed N.H. Evidently, N.W affects the free area, hence affects the gas mass flow and consequently varies the overall heat transfer coefficient. Almost the influence of the variation of F.W.T at a certain value of N.W is identical to that mentioned in the explanation of Figure (5). On the contrary, the smaller F.W.T, the higher the effect of N.W on η_b. Obviously, due to the fixation of N.H, the higher N.W, the higher η_b but excessive increase of N.W diminishes the corresponding increase of η_b.

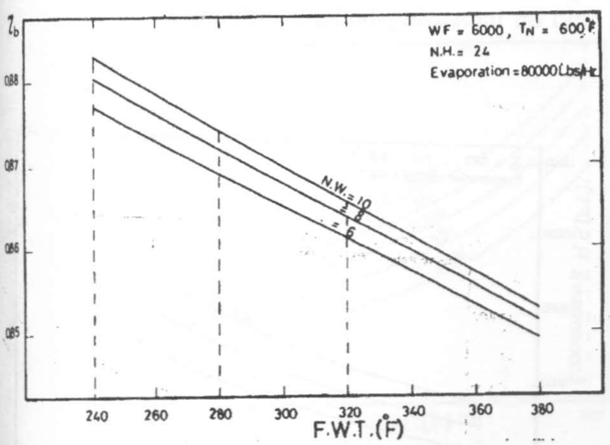

Figure 7. Effect of feed water temperature & no' of tubes wide on boiler's efficiency.

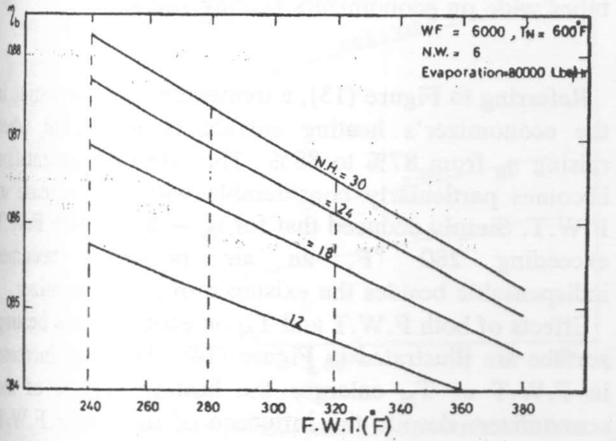

Figure 8. Effect of feed water temperature & no. of tubes high on boiler's efficiency.

Figure (8) explains how much η_b is affected by the deviation in any or both N.H and F.W.T while N.W and T_N being held constant. An elementary consequence is that the higher N.H, the higher η_b; but at a certain change in N.H, a considerable increase in η_b is assumed specially at low F.W.T. Surprising enough, observing that changing average values of N.H has a stronger influence on η_b rather than varying higher values.

In order to visualize the effect of gas mass flow through a specified economizer on η_b, the amount of burnt fuel should be varied. A hypothetical assumption is made that, inlet gas temperature to the economizer could be-some how -kept constant. The aim, being to interdict the effect of interfering the dependent parameter namely, the gas temperature. Normally, a certain economizer possesses a restricted capability of heat absorption therefore, the lower the gas mass flow (or the fuel burned), the higher η_b. This influence is emphasized at low feed water temperature too, Figure (9).

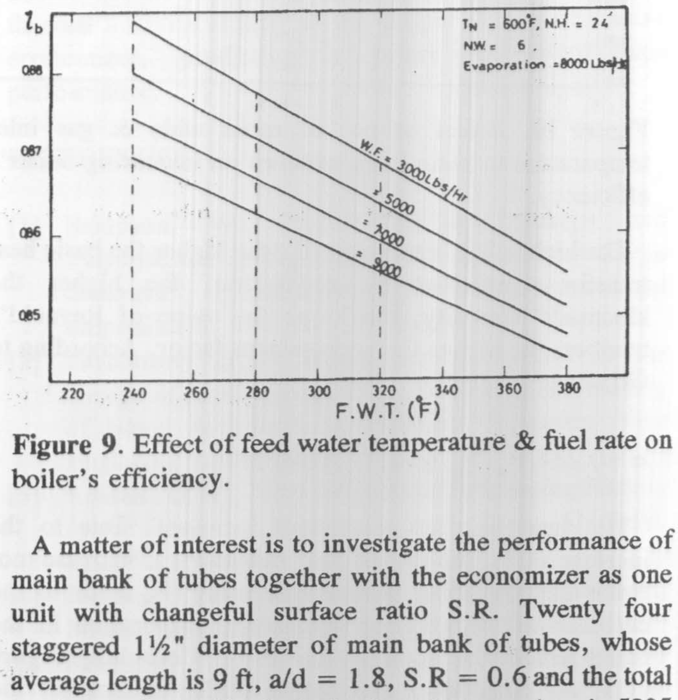

Figure 9. Effect of feed water temperature & fuel rate on boiler's efficiency.

A matter of interest is to investigate the performance of main bank of tubes together with the economizer as one unit with changeful surface ratio S.R. Twenty four staggered 1½" diameter of main bank of tubes, whose average length is 9 ft, a/d = 1.8, S.R = 0.6 and the total heating surface of both the bank and economizer is 5805 ft² are adopted. The inlet gas temperature to the bank T_N is changed from 1200 to 1400 °F. The performance of the main bank of tubes with changing N.W from 5 to 8- keeping F.W.T at 280 °F-on the change of boiler's efficiency ($\Delta\eta_b$) is studied in Figure (10). An amelioration in η_b approaching 0.15% is realizable; a matter which signifies sparing about 10.25 1b/hr fuel, i.e 28.6 tons fuel per navigating year of 260 days. A brief explanation, why the higher T_N results in a higher increase in η_b for specified main bank of tubes and economizer could be easily illustrated from equation (18)-

Appendix I, which may be rewritten for the main tube banks in the form

$$T'_{N+1} = T_N = \frac{T'_N - T_S}{e^{\frac{\acute{R}.\acute{S}}{W.C_P}}} + T_S$$

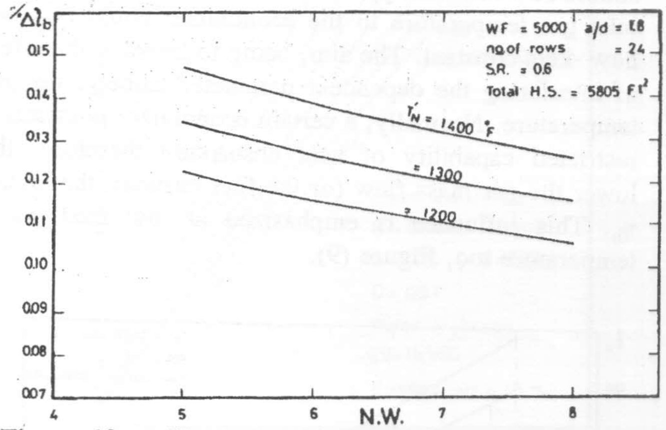

Figure 10. Effect of no. of tubes wide & gas inlet temperature to main bank of tubes on increasing boiler's efficiency.

The higher T'_N, the higher C_p, the higher the basic heat transfer coefficient by convection, the higher the kinematic viscosity, the lower the value of Reynold's number, the higher the arrangement factor. According to Grimison's equation namely:

$$\alpha = 0.284\ F_a . F_n . \gamma . d^{-0.39} . (\acute{M}F)^{0.61}$$

The heat transfer coefficient increases. Due to the increase of T'_N the rate of heat radiated will increase too. From the preceding points, increasing T'_N leads to the decrease of T_N (raising percentage evaporation in the bank) hence, the improvement in η_b. Consulting Figure (11), we conclude that likewise N.W, the smaller the value of S.R, the higher the increase in η_b. But this rate of increase in η_b decays with excessive decrease in S.R.

The relationship among the economizer's heating surface, N.W and F.W.T is displayed in Figure (12) for pre-required η_b and T_N. For fixed economizer's inlet and outlet gas temperatures, higher F.W.T has a significant effect on increasing the economizer's heating surface. Whereas, a less important effect is influenced by the increase of N.W on the increase of heating surface. For a pre-required heating surface either a small value of F.W.T together with a relatively large value of N.W are adopted or vice versa.

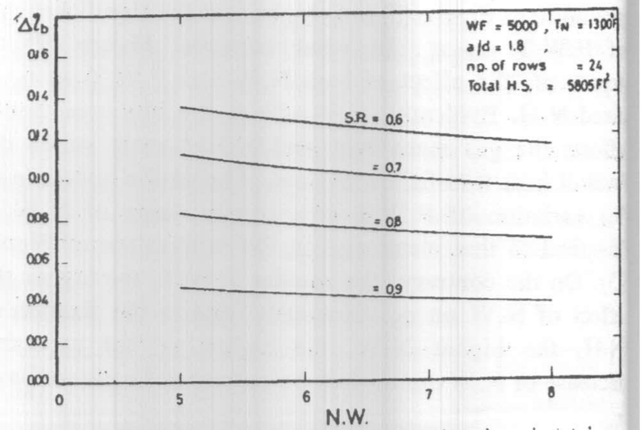

Figure 11. Effect of no. of tubes wide & heating surface ratio on increasing boiler's efficiency.

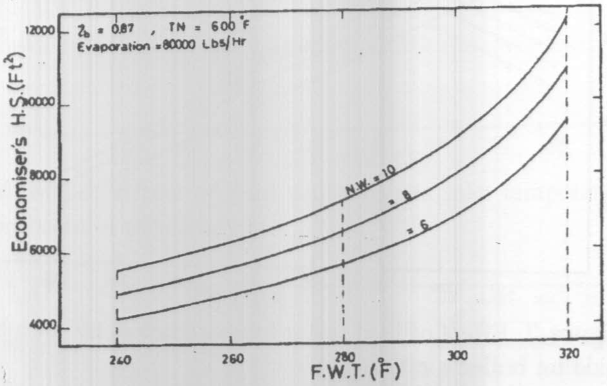

Figure 12. Effect of feed water temperature & no. of tubes wide on economizers heating surface.

Referring to Figure (13), a tremendous augmentation in the economizer's heating surface is noticeable when raising η_b from 87% to 88%. The rate of augmentation becomes particularly considerable with the increase of F.W.T. Simply deduced that for η_b = 88%, with F.W.T exceeding 280 °F, an air preheated becomes indispensable besides the existence of the economizer.

Effects of both F.W.T and T_N on economizer's heating surface are illustrated in Figure (14). An either increase in F.W.T or T_N enlarges the heating surface of the economizer; despite the influence of increasing F.W.T corresponds to an exponential increase in the graph. In order to maintain a certain value of economizer's heating surface any parameter of T_N or F.W.T could be sacrificed, i.e. selecting a high T_N with a low F.W.T or vice versa.

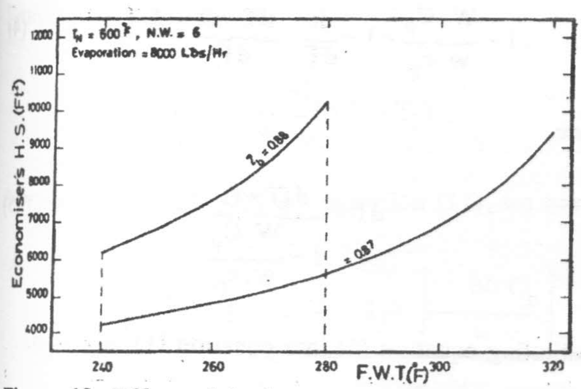

Figure 13. Effect of feed water temperature & required boiler's efficiency on economizer's heating surface.

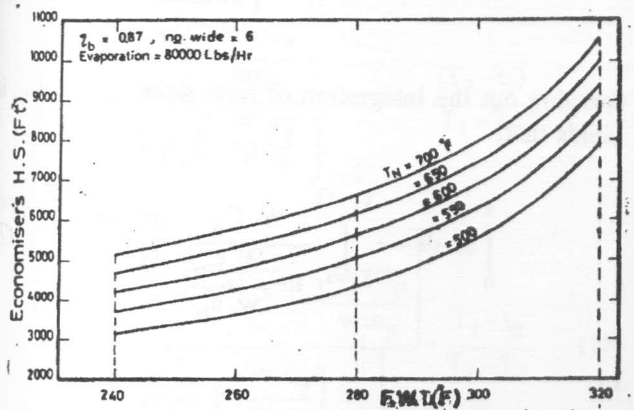

Figure 14. Effect of feed water & gas inlet temperatures on economizer's heating surface.

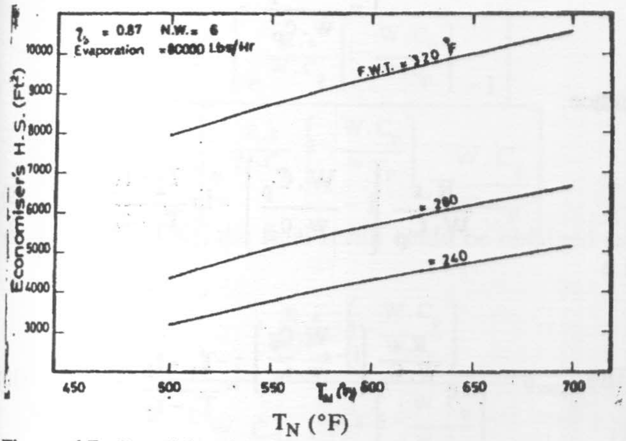

Figure 15. Considerable increase of economizer's heating surface with feed water temperature increase.

A graphical transformation from Figure (14) is visualized in Figure (15) but at standard marine F.W.T. Chosen values of F.W.T are 240, 280 and 320 °F. These values correspond to two feed water heaters with low pressure atmospheric deaerator, two heaters with high pressure double atmospheric deaerator and three feed water heaters with usually an air preheater incorporated in the marine regenerative steam cycle respectively. Evident the considerably large economizer's heating surface at F.W.T of 320 °F.

CONCLUSION

Knowingly, all studied parameters namely, F.W.T, N.W, N.H, T_N, T_N' and S.R assume of course an influence on both η_b and economizer's heating surface. But distinguished is the role played by F.W.T; less characterized are those effects of N.W, N.H and S.R; while seemingly, normal expected effects could be attributed to both T_N and T_N'. Adoption of air preheaters if essential is discussed in the light of the scanned results. Moreovre, the paper represents in detailed discussions a thermal analysis aiming to the proper design of marine economizer, predicting a priori its approximate performance.

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Appendix I. Economizer's Efficiency

Considering a Counter flow economizer and referring to Figure (1), we can write

$$R.ds.(T-t) = -W.C_p.dT \qquad (1)$$

Moreover, it could be written that

$$W.C_p.dT = w.c_p.dt \qquad (2)$$

From equation (2) it follows:

$$\frac{W.C_p}{w.c_p} = \frac{dt}{dT} \qquad (3)$$

From Equation (3) it follows:

$$1-\frac{W.C_p}{w.c_p} = 1-\frac{dt}{dT} = \frac{dT-dt}{dT} = \frac{d(T-t)}{dT} \qquad (4)$$

Hence,

$$dT = \frac{d(T-t)}{1-\frac{W.C_p}{w.c_p}} \qquad (5)$$

Substituting equation (5) into equation (1), it results:

$$R.ds.(T-t) = -\frac{W.C_p.d(T-t)}{1-\frac{W.C_p}{w.c_p}} \qquad (6)$$

carrying out the integration of both sides of equ. (6), it yields that:

$$\int_0^s R.ds = -\int_{T_1-t_1}^{T_2-t2} \frac{W.C_p}{1-\frac{W.C_p}{w.c_p}}.\frac{d(T-t)}{(T-t)} \qquad (7)$$

or:

$$R.s = -\frac{W.C_p}{1-\frac{W.C_p}{w.c_p}}.\ln\frac{T_2-t_2}{T_1-t_1} \qquad (8)$$

hence:

$$-\frac{R.s}{W.C_p}\left\{1-\frac{W.C_p}{w.c_p}\right\} = \ln\frac{T_2-t_2}{T_1-t_1} \qquad (9)$$

i.e

$$e^{\frac{R.s}{W.C_p}\left\{1-\frac{W.C_p}{w.c_p}\right\}} = \frac{T_1-t_1}{T_2-t_2} \qquad (10)$$

and

$$e^{\frac{R.s}{W.C_p}\left\{1-\frac{W.C_p}{w.c_p}\right\}} - 1 = \frac{(T_1-T_2)-(t_1-t_2)}{T_2-t_2} \qquad (11)$$

From equation (2), we get

$$t_1 - t_2 = \frac{W.C_p}{w.c_p}(T_1 - T_2) \tag{12}$$

Substituting equation (12) into equation (11), we have

$$e^{\frac{R.s}{W.C_p}\left[1-\frac{W.C_p}{w.c_p}\right]} - 1 = \frac{(T_1 - T_2)\left[1-\frac{W.C_p}{w.c_p}\right]}{T_2 - t_2} \tag{13}$$

Rearranging the terms of equation (13) and adding one to both sides we get:

$$1 + \frac{1 - \frac{W.C_p}{wc_p}}{e^{\frac{R.s}{W.C_p}\left[1-\frac{W.C_p}{w.c_p}\right]} - 1} = 1 + \frac{(T_2 - t_2)}{T_1 - T_2} \tag{14}$$

or:

$$\frac{\left[e^{\frac{R.s}{W.C_p}\left[1-\frac{W.C_p}{w.c_p}\right]} - \frac{W.C_p}{w.c_p}\right]}{\left[e^{\frac{R.s}{W.C_p}\left[1-\frac{W.C_p}{w.c_p}\right]} - 1\right]} = \frac{T_1 - t_2}{T_1 - T_2} \tag{15}$$

The reciprocal of both side of equation (15) gives

$$\frac{T_1 - T_2}{T_1 - t_2} = \frac{\left[e^{\frac{R.s}{W.C_p}\left[1-\frac{W.C_p}{w.c_p}\right]} - 1\right]}{\left[e^{\frac{R.s}{W.C_p}\left[1-\frac{W.C_p}{w.c_p}\right]} - \frac{W.C_p}{w.c_p}\right]} \tag{16}$$

From equation (16), the final result could be obtained as

$$\frac{T_1 - T_2}{T_1 - t_2} = \frac{1 - e^{-\frac{R.s}{W.C_p}\left[1-\frac{W.C_p}{w.c_p}\right]}}{1 - \frac{W.C_p}{w.c_p}.e^{-R.s\left[1-\frac{W.C_p}{w.c_p}\right]}} = \eta_{ec} \tag{17}$$

It is to be noted that equation (17) is defined as the economizer's efficiency η_{ec}.

In case of boiling water economizer (or screen tubes or main bank of tubes) $t_1 = t_2 = t = T_S$ = constant.

By rearranging the terms of equation (1) and carrying out the integration we get

$$\int_0^S \frac{R.ds}{W.C_p} = -\int_{T_1}^{T_2} \frac{dT}{T-t}$$

$$\frac{R.s}{W.C_p} = -\ln\frac{T_2 - t}{T_1 - t} = \ln\frac{T_1 - t}{T_2 - t}$$

or

$$e^{\frac{R.s}{W.C_p}} = \frac{T_1 - t}{T_2 - t} \tag{18}$$

Equation (18) is applied for screen tubes and main bank of tubes too.

Appendix II:

Brief Design Procedure:

The economizer as a multi-variable heat recovery equipment may be designed in a thermal equilibrium manner for any of the variables entering, leaving or for its heating surface itself (N.W, N.H and L.O.R). The latter case is shortly displayed

1. On the basis that the oil fuel is composed of 87.64% C, 11% H_2, 0.53% S, 0.57% O_2 and 0.26% N_2 and traces of H_2O with H.C.V = 18500 BTU/$1b_m$ and is burnt with 15% excess air and from elementary combustion theory, it follows:

$$\left[\frac{87.64}{12}C + \frac{11}{2}H_2 + \frac{0.53}{32}S + \frac{0.57}{32}O_2 + \frac{0.26}{28}N_2\right]$$

$$+10.1917 * 1.15(O_2 + 3.76N_2)$$

$$\longrightarrow 7.3CO_2 + 5.5H_2O + 0.15625\ SO_2$$

$$+\ 44.076\ N_2 + 1.5288\ O_2 \tag{19}$$

which gives:

$$AF \simeq 16\,(1b_{air}/1b_{fuel})$$

and $\%\ CO_2 = 13.1\%$

Assuming 2% radiation losses, η_b could be determined from Figure (3) where the stack temperature is determined on the basis of the dew point of sulphuric acid and the economizer material.

2. Compute

$$WF = \frac{EV(I_{sup} - I_{f.w})}{\eta(H.C.V. + AF * \text{Enthalpy of entering air})} * \frac{L.C.V}{H.C.V} \text{ 1b/hr} \quad (20)$$

then:

$$W = (AF + 1) * WF \quad \text{1b/hr} \quad (21)$$

3. T_N is the outlet gas temperature from the main bank of tubes and F.W.T is determined from the final heat balance of the steam cycle (°F).

From equation (17 Appendix I); we have:

$$\eta_{ec} = \frac{T_N - T_{N+1}}{T_N - F.W.T} \quad (22)$$

4. it is known that:

$$C_p = 1.025 \ (BTU/1b_m \, ° F)$$

While, C_p could be calculated from the variation of the specific heat of combustion gases at the mean average temperature (BTU/$1b_m$ °F) (Table 1)

Table 1. Specific Heat at Constant pressure of Combustion gases (15% Excess Air).

T (°F)	200	400	600	800	1000	1200	1400
C_p (BTU/1bm.F	0.25	0.26	0.27	0.277	0.289	0.290	0.297

Compute $W.C_p./w.c_p$.

5. From Figure (2) which is a pictorial graph of η_{ec} with the knowledge of $W.C_p/w.c_p$ and η_{ec} determine $R.S/W.C_p$ and compute

$$R.S = \left[\frac{R.S}{W.C_p}\right] * W.C_p \quad (23$$

6. Select N.W based on the breadth of the economize the dimensions of the tubes and fins and the allowabl clearances.

7. Compute the equivalent diameter of the economizer' tubes and fins according to the definition [21].F 2"x5" finned tubes, d_{eq} = 3½" [19]:

$$d_{eq} = \frac{4 \text{ x area of the passage occupied by gases}}{\text{wetted perimeter}}$$

8. Compute $FA = (Eb - N.W * d_{eq}) * L.O.R \text{ ft}^2$ (2

9. Compute: $MF = W/(1000 * FA) \text{ 1b/ft}^2\text{hr}$ (2

For 2"x5" cast iron Gilled ring economizer's tubes

10. Compute: $R = 2.55 \text{ x } MF^{0.535}$ BTU/ft^2.hr.°F Fig.(4) [19] (2

Hence,

$$S = (RS)/R \quad (2$$

11. Compute the heating surface of one row

(2

$$= 4.1 * (L.O.R) * (N.W) \text{ ft}^2\text{/one row}$$

where, for this specified type of finned tubes the heatir surface/lineal feet = 4.1 ft^2 [19]

12. Finally,

$$\text{Compute: } N.H = \frac{S}{\text{Heating Surface of one row}} \quad (2$$

Either the integer portion of N.H is kept or N.H rounded to the nearest integer and the computatior should be finely readjusted.