THE CONDITION OF HEAT TRANSFER APPARATUS.

The ultimate economy and also the maximum attainable output of any steam installation is very largely determined by the condition and performance of a class of apparatus which may be grouped under the heading of "heat exchangers." These comprise such units as boilers, steam condensers, feed and oil fuel heaters, lubricating oil coolers, evaporators, distillers and refrigerating plant, and they form a very large proportion of the total weight of machinery carried in a vessel.

The existing knowledge of the laws of heat transmission as applicable to the practical design of such apparatus is admittedly sadly deficient, and in view of this it is the universal practice to make ample provision of heating surface in order to cover incalculable factors—in other words, there is a large margin which is frequently doubled by the desirability of duplicating apparatus in case of failure from corrosion. It is obvious that such a procedure tends towards needless weight thus leading to a reduction in the military qualities of the vessel.

The principal unknown factor for which the designer must make an arbitrary allowance is the increased resistance to heat flow due to the formation of deposits upon the heating surfaces under service conditions. Reliable information in this respect, culled from vessels of all types working under widely varying conditions, should enable a more just appreciation of this factor to be made, with ensuing benefit to future designs.

Naval vessels commonly operate at very low proportions of their full power, with the result that the admittedly liberal provision of heat transference surfaces is, under normal cruising conditions, so ample that any gradual deterioration of their efficiency is not generally evident, possibly only becoming so when a demand is made for high power. These circumstances engender the possibility of ultimate failure to produce the power required from the machinery; this danger is usually remote, thanks to the ample margin provided and to the necessity for periodical inspection of these parts if corrosion is to be avoided, but it requires to be borne in mind, and forms one sound argument for maintaining a fairly close watch upon the performance of this class of apparatus.

There is, however, another important factor in this latter regard, namely, that if for any reason the resistance to heat flow becomes abnormally high then the required rate of transmission can only be obtained by increasing the quantity and velocity of one or both of the fluids being handled; this entails the use of more power in the pumping apparatus, so leading to a needlessly high fuel expenditure. The question is, what is the best method in practice for readily determining the condition of the heating and/or cooling surfaces and of expressing this in quantitative terms? It is the purpose of this article to suggest what observations appear to be best suited for this purpose, having in mind the limited appliances available in a ship and the necessity for simplicity and ready interpretation of the results.

Resistances to the Flow of Heat.—It is desirable for the sake of completeness to recall briefly some of the more important considerations and expressions in connection with the flow of heat by conduction and convection.

Conduction.—This form of heat transmission is relatively unimportant in the usual engineering problems, the resistance of the metal walls being so low as to be negligible, while that of the gas or fluid films adjacent to the surfaces is so high that the passage of heat through them is almost entirely effected by convection.

It is sufficient to state that the resistance to flow by conduction is given by the expression :—

$$R = \frac{1}{K} = \frac{L}{C}$$

where K = conductance in (say) B.T.U. per hour per sq. ft. per °F. temperature difference.

L = thickness of metal through which heat is conducted.

C = conductivity in (say) B.T.U. per hour per sq. ft. per ft. per °F. temperature difference.

Average values of " C " for different materials in the above units are :—

Copper ..	230	B.T.Us. per sq. ft. per ft. per °F. per hour.
Brass ..	48–63	,, ,, ,, ,, ,,
Steel ..	26	,, ,, ,, ,, ,,

Convection.—The theory usually propounded to account for the high resistances to heat flow observed at surfaces transmitting heat from one fluid to another, assumes the existence of more or less stagnant films upon the heating surfaces, the conductivity of these films being very low. On this assumption, it is evident that, under given temperature conditions, the best practical means of increasing the flow of heat is by passing the fluids over the surfaces at such a speed that the film is either reduced in thickness or, in the case of eddying flow, is in a state of alternate formation and destruction. In other words, the rates of transmission depend very largely upon the velocity of the fluids, and this has been repeatedly shown to be the case in practice. Theoretically the conductance of such a film should be expressible in the form :—

$$K = A + B\left(\frac{w}{a}\right)$$

where w = weight of fluid passing in unit time.

a = area for flow of fluid.

A is a term which expresses the natural rate of internal diffusion of the fluid, being a constant for a given gas or liquid under particular temperature conditions. B varies with the shape of the channel for flow of the liquid, and also with the temperature of the film and the density and specific heat of the fluid.

In all practical engineering problems it has become evident that the first item is relatively unimportant, while, as the result of empirical investigation, the view is at present generally held that the second term of the above equation is not a linear function of $\left(\frac{w}{a}\right)$. Generally speaking then, it appears probable that the conductance of such films may be expressed as $K = b\left(\frac{w}{a}\right)^n$, where "$n$" has the value of about $0{\cdot}8$.

Resistances in Series.—The overall resistance R of any heating surface on either side of which are fluid films may be considered to be comprised of the sum of a number of partial resistances, that is :—

$$R = r_v + r_t + r_w$$

where r_v = film resistance on one side

$$= \text{say } \frac{1}{K_v}$$

r_t = resistance of the wall itself.

$$= \frac{L}{C}$$

r_w = film resistance on the other side.

$$= \text{say } \frac{1}{K_w}$$

This expresses the condition for clean surfaces, but in practical cases there are, of course, deposits of grease, scale, mud, etc., which impose a very appreciable resistance to the flow of heat; this resistance may be conveniently expressed as r_s, the whole resistance being given by

$$R = r_v + r_t + r_w + r_s$$

In practice the resistance (R) itself is not directly measured but rather its reciprocal, the conductance (K), the units of which have already been referred to. The determination of this factor entails that of two other separate items, namely, the average heat transfer per unit area in unit time (H) and the mean temperature difference (θm) between the fluids on either side of the heating surface, K being given by $H \div \theta_m$. H can, of course, be readily estimated if suitable instruments are available for indicating the quantities of each fluid and their changes in temperature: it should be noted that in practical heaters (or coolers) the absolute value of H necessarily varies considerably in different parts of the heating surface, and hence the values obtained by direct measurement will in nearly every case be a mere average, and thus not representative of the conditions at all positions in the tube nest.

The temperature difference between the two fluids varies throughout the whole surface, and, at the best, only the average value can be determined in practice. The average value of the mean temperature difference is given accurately by $\theta_m = \dfrac{\theta_a - \theta_e}{\log_e \dfrac{\theta_a}{\theta_e}}$

where θ_a and θ_e are the differences between the temperatures of the fluids at entry to and exit from the heating surfaces, respectively. This expression is somewhat cumbersome to use and for ordinary purposes it is sufficient to employ the arithmetic mean between θ_a and θ_e as a working value for the mean temperature difference, which can therefore be readily estimated by means of suitably placed thermometers.

Apart from the case of boilers (with their superheaters) the type of heat exchanger most frequently met with in marine practice is that in which heat is transferred from condensing steam to a metal tube, and thence to water, *e.g.*, condensers or feed heaters. In cases such as these the mechanism of transfer on the vapour side is extremely complex, being dependent not only upon the temperature of the vapour and the thickness of the film of condensed water which collects upon the tubes, but also being greatly influenced by the amount of air entering with the steam and by the distribution of this air in the body of the apparatus. The effect of such factors as these is but little understood, their investigation being a matter of extreme difficulty. The resistance on the vapour side unfortunately may average a somewhat large proportion of the total resistance to heat flow, and this tends to obscure any change in the other partial resistances, especially so when such effects can only be determined by the somewhat approximate measurements likely to be available under service conditions.

The engineer's object is to obtain a rough idea of the extent to which his heating surfaces become foul and to arrange matters so that due warning is given of any circumstances affecting the efficiency of heat transmission. Now, as has already been pointed out, the resistance to the water film varies with the velocity of the liquid itself, and from the form of the equation already given it is evident that the resistance due to this cause should be zero at infinitely great flow; under this latter condition the observed (or predicted) total resistance must, of course, be the sum of the remaining partial resistances.

One practical method of making use of the foregoing argument consists in estimating the overall resistance under standard conditions (as regards the vapour side) at varying velocities of the water. A curve is then drawn of "Resistance" to the reciprocal of the rate of flow of the water, *i.e.*, R to $\dfrac{1}{\text{gallons per min. per sq. ft.}}$;

infinite flow corresponds to zero value of the latter function, and the value of R at this point (obtained by extension of the curve) corresponds to $r_t + r_v + r_s$.

The value of r_t can be calculated, while that of r_v may either be assumed to be about $\frac{1}{4{,}000}$ or be estimated from the results of similar trials with clean tubes. Thence the value of r_s may finally be deduced, giving a quantitative idea of the state of the heat transmitting surface.

In practice it is not usually practicable to measure the quantity of water dealt with, but this can be estimated with reasonable accuracy either from the revolutions of the circulating pump in conjunction with the results of known trials, where such have been run, or else by deduction from the temperature rise of the fluid when handling a known (or estimated) quantity of heat. As an example of the practical use of such methods the following description of the analysis of the performance of the main condensers of certain vessels may be of interest.

Analysis of Condenser Performance.

Object.—To estimate the value of the Overall Cofficient of Heat transfer (U) and to calculate the average partial resistances to heat flow, with a view to determining the state of the tube surfaces.

Data.—In these particular cases the only available data comprised the S.H.P. of the main turbines (torsion-meter), the vacuum at the exhaust connection to the main condensers and the observed temperatures of the circulating water at inlet to and outlet from the condenser.

Estimation of " U."—As a first step it is necessary to find the heat rejected to the condenser in unit time. This entails a knowledge of the steam consumption of the turbine (w) and the total heat per lb. of the exhaust steam (He). The former of these may be found by applying the vacuum corrections (given in Fig. IV a of papers No. 8, page 22) to the steam consumption measured on contractors trials. The total heat at exhaust can be estimated from the Mollier diagram, applying a suitable efficiency; alternatively, if it has already been calculated for the original contractors trials, this value may be amended proportionately to the probable steam consumption.

A check on the steam consumption may be obtained by plotting " Total steam per hour " against " Nozzle box pressures " (for trial conditions) and reading off the probable steam consumption, under any particular condition, from the chart against the observed nozzle box pressure (vide Papers No. X, page 47). The mean temperature difference (θm) is then calculated by the methods already referred to, the temperature of the steam being assumed to be constant and equal to the saturation temperature of steam at the absolute pressure observed at the exhaust orifice (steam tables).

"U" is then directly calculated from $U = \frac{w.He}{S.\theta m}$, where S is the area of the condensing surface.

Quantity and Velocity of Circulating Water.—The quantity of circulating water may be calculated by equating the heat in the exhaust steam to that represented by the temperature rise of the circulating water. The figures given in the table were so obtained.

Alternatively, if the characteristics of the circulating pump are known (say from shop trials) then their output may be obtained with fair accuracy by observing the revolutions of the pump and the total delivery head as measured by suitably placed pressure gauges. This assumes that the characteristics have not been altered by impeller erosion, etc., but in any case the figures so obtained are a valuable check upon those calculated by the first method.

The velocity of the water through the tubes (average) is required for calculating the resistance of the water-film to heat flow.

Partial Resistances to Heat Flow.—The resistances of the water-film and of the tube wall may be found from the following expressions, which give very fair results with $\frac{5}{8}$-in. tubes working under the conditions usual in Naval Condensers :—

Resistance of water film, $r_w = bv^{0 \cdot 8}$

where $b = 185\left(1 + \frac{50}{m}\right)f^{0 \cdot 8}$ for $\frac{5}{8}$-in. tubes.

m = ratio of length to internal diameter of tube.

f = fluidity of circulating water at average temperature of film = $1 \cdot 35$ at 100° F.; $1 \cdot 22$ at 90° F.

The units employed are ft.² lb. °F.-hour, the velocity being in ft./sec.

Resistance of tube wall, $r_t = 7 \cdot 0 \times 10^{-5}$ for $\frac{5}{8}$-in. tubes of Admiralty alloy.

The overall resistance $R = \frac{1}{U}$, and hence the value of Resistance.

Resistance of Scale Air and vapour film, $r = R - (r_w + r_t)$.

Practical Results.

The table shows the value of the foregoing factors as estimated from observations in two vessels both on service and during their original trials.

The following points may be noted, amongst others :—

(1) The value of r on Contractors trials in all vessels was approximately the same. It is to be expected that the tubes were in much the same state of cleanliness, that similar rates of air leakage were experienced, and that the steam film resistances were nearly the same in all cases, and, in view of this, the agreement between the estimated values of r may be reasonably taken as justifying the mode of calculation adopted.

(2) The change in r with time may be seen by comparing columns (*d*) and 6, whence it appears that this factor has increased gradually till it accounts for nearly half of the total resistance, instead of the normal figure of about 15 per cent.

In this particular instance examination showed the tubes to be coated internally with a thin scale which could readily be scraped off with a knife, but which was not removed by brushing. Experiments are in hand to check the estimated heat resistance of this particular scale.

(3) Analyses of this type enable the engineer to see whether his circulating pumps have been driven unduly fast. This is exemplified by the results of the contractors trials of vessel A. The trials were conducted under very similar conditions, yet it will be observed that the pumps in trial (*c*) were run at a very much lower output than in the remaining cases.

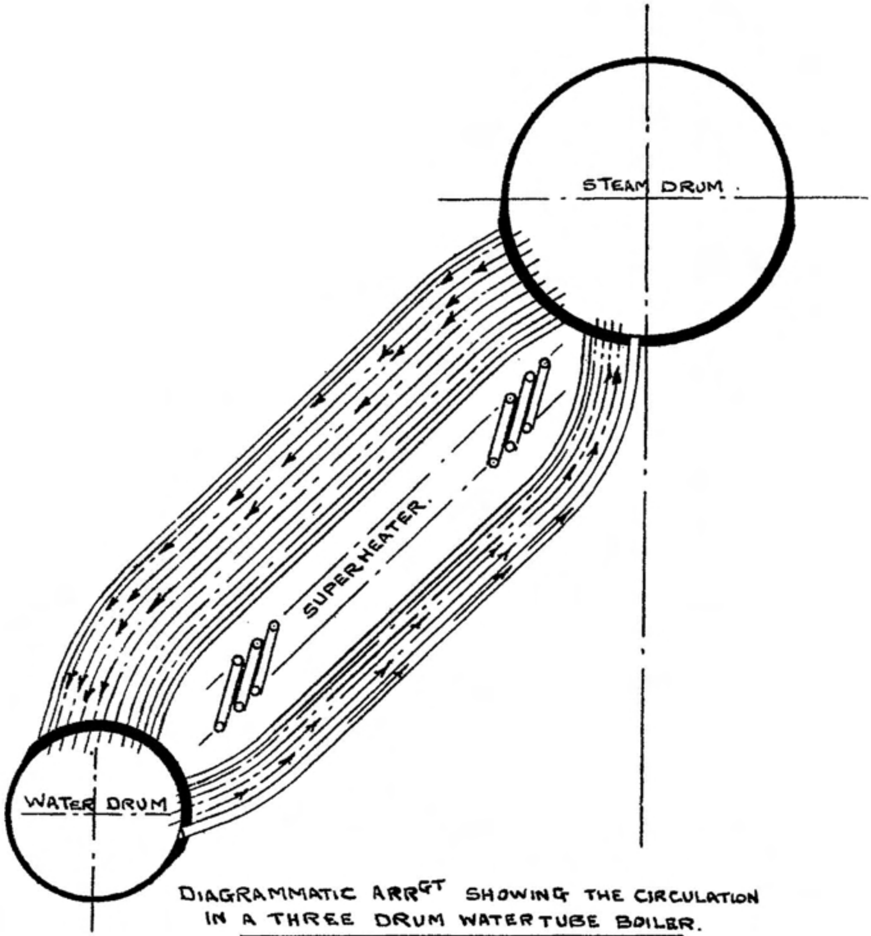
(4) There is a distinct indication that U tends to increase as the temperature of the circulating water becomes higher. This agrees with theory.

It should be observed that r includes three items, which are not separable individually by any practical means. They are, of course, the resistances of (1) the scale; (2) the steam film; and (3) the air in the steam. The values of the two latter will depend upon the temperature in the steam space, upon the quantity of the air and upon the velocity and distribution of the steam—all factors which it is impracticable to evaluate. Under reasonably similar conditions as regards these variables, however, a marked increase in r may be attributed to a change in the scale resistance, and this possibly should thus be explored, bearing in mind that quite a thin scale may exercise a very marked effect upon the heat transference.

In conclusion, it is desired to emphasise the fact that the importance of analyses of this nature lies not so much in the numerical accuracy of the results, as between ship and ship, but rather in the comparative values obtained from time to time in one vessel operating under more or less similar conditions. The observations and ensuing calculations are simple and easily made, and provide a ready meand of checking the performance of the plant and of detecting operational errors, which latter may thus frequently be avoided. Investigations of this nature are to be encouraged as they provide data which may be of value in improving the equipment of new vessels, especially if in reporting results information is given regarding the assumptions made and the methods of measurement employed.

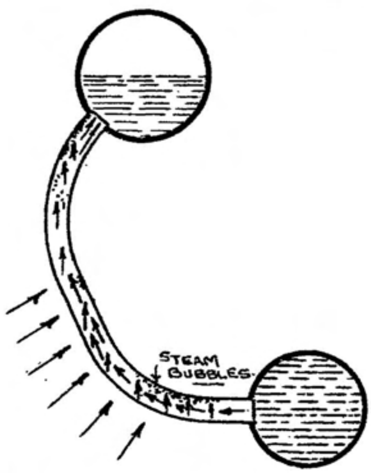
CONDENSER PERFORMANCE.

—	Table I.—Vessel A.							Table II.—Vessel B.		
	Original Trials.			Trials at Sea.				Original Trial.	Trials at Sea. (Condenser tubes badly scaled.)	
	(*a*)	(*b*)	(*c*)	1.	2.	3.	4.	(*d*)	5.	6.
Absolute pressure in Condenser, ins. of Hg.	1·91	1·77	2·34	3·0	3·8	3·5	3·9	2·71	8·2	8·95
Temperature of Exhaust Steam (t_s), °F.	99·4	97·1	106·4	117·4	123·6	120·6	124·6	111·5	153·8	157·1
Temperature of Circulating Water Inlet (t_1), °F.	54·0	62·0	49·0	84·0	84·0	84·0	84·0	50·0	84·0	83·0
Temperature of Circulating Water (t_2) Outlet	77·0	79·0	78·0	102·0	102·0	102·0	102·0	82·0	116·0	116·0
Mean temperature difference, θ_m	32·6	25·9	40·8	24·75	29·8	26·5	30·6	43·8	52·5	56·1
Heat transfer coefficient, U. B.T.Us./hr./ft.²/°F.	633	767	482	858	790	848	743	628	520	524
Circulating Water supplied, gals./min.	13,020	16,900	9,870	17,200	19,000	18,100	18,900	10,070	10,000	10,470
Velocity of circulating water through tubes, ft./sec.	5·3	6·88	4·02	7·02	7·76	7·4	7·72	4·92	4·88	5·11
Resistance of water film (r_w) ..	·001275	·00099	·00175	·00077	·000736	·000736	·000712	·00128	·000943	·00091
Resistance of Tube wall (r_t) ..	·00007	·00007	·00007	·00007	·00007	·00007	·00007	·00007	·00007	·00007
Resistance of steam film, scale and air (r)	·000237	·000243	·00026	·000325	·000487	·000372	·000564	·000239	·000908	·000930
Total resistance $r_w, r_t, r = R = \frac{1}{U}$	·001582	·001303	·00208	·001165	·001267	·001178	·001346	·001589	·001921	·001910



DIAGRAMMATIC ARR^{GT} SHOWING THE CIRCULATION IN A THREE DRUM WATERTUBE BOILER.

FIGURE 1.



FORMATION OF STEAM BUBBLES IN A BENT TUBE.

FIGURE 2.

P.4053, 21193F, 8635, 600, 3/31.