10.

NOTES.

(a) DIESEL-ELECTRIC AUXILIARIES.

The use of Diesel generators for providing power for driving the auxiliaries of a steam plant is a practice which can hardly be adopted in Naval installations owing to the unacceptable demands upon weight and space thereby entailed. The system has, however, been used with considerable success in more than one mercantile vessel, and it may therefore be of interest to consider briefly the gain that may be expected.

The efficiency of the electrical transmission gear between the Diesel engines and the auxiliaries themselves, that is the generators and motors, may be expected to aggregate about 85% as a maximum. The probable fuel consumption of the Diesel engines would be of the order of $0 \cdot 4$ lb. per B.H.P. per hour, and thus, allowing for the efficiency quoted, the auxiliary power could be provided for an expenditure of about $\frac{0 \cdot 4}{0 \cdot 85} = 0 \cdot 47$ lb. per H.P. per hour.

On the other hand, if the auxiliaries are driven directly by steam engines and if the whole of the exhaust heat is used for feed heating purposes then the cost of each H.P. at the auxiliaries would amount to $\frac{2545}{19000} \times 0 \cdot 75 \left(\textit{i.e.,} \; \frac{\text{Heat equivalent of 1 H.P./hour}}{\text{Calorific value of Fuel} \times \text{Boiler efficiency}} \right)$ or $0 \cdot 179$ lb. of oil per hour. Assuming an efficiency of exhaust disposal of 90 per cent. and a mechanical efficiency of 90 per cent., this figure is increased to $0 \cdot 22$ lb. per hour, which is readily realised in practice.

This comparison shows that from the point of view of efficiency in driving the auxiliary engines there is nothing to be said for the Diesel-electric system, while in respect to reliability, weight, space, cost of upkeep, and original capital expenditure, there is much that might be said against it. There is, however, another important aspect to be considered, namely, the efficiency of the main turbines and the influence of bleeder feed heating thereon.

The gain to be achieved by bleeding the main turbines varies with the initial and final steam conditions and also with the point at which the bled steam is extracted—in fact, as has already been pointed out (p. 73, Papers No. X), there is an optimum condition for each case, corresponding to a definite feed temperature and to a given number of stages of feed heating. As an example, it is estimated that in the case of a turbine typical of modern naval designs, the use of single stage bleeder feed heating may result in a reduction in the heat consumption of the main turbines of about $5\frac{1}{2}$ per cent., as a maximum.

The figure just quoted assumes that only the bled steam is employed to heat up the condensate from the main condensers. If, however, the auxiliary exhaust steam is also used for such a purpose, the feed being then passed to " bleeder " heaters, it will be found that the maximum gain in economy to be obtained from the turbines is reduced to the order of 2 per cent. The effect of using the auxiliary exhaust for feed heating is thus to cause a loss amounting to $5\frac{1}{2}—2$ per cent. $= 3\frac{1}{2}$ per cent. of the heat consumption of the main turbines, as compared, that is, with the condition when all the feed heating is done by bled steam.

The use of Diesel-electric auxiliaries solves the problem of exhaust disposal and thus permits full advantage to be taken of the possibilities inherent in the bleeding process. In other words, with steam-driven auxiliaries a low heat consumption for the auxiliaries is associated with but a small possible gain due to bleeding the main turbines, while with Diesel-electric auxiliaries the reverse is the case. The balance of advantage is evidently determined by the ratio of the horse power of the main turbines to that of the auxiliary engines.

This ratio $\frac{\text{Main}}{\text{Auxiliary}}$ Horse Power is about 45 to 1·0 in a naval installation, and the comparison between the two systems of working is made in the following table on that basis.

	Steam driven Auxiliaries.	Diesel-electric Auxiliaries.
Fuel per H.P./hour Main engines only	0·778 lbs.	0·752 lbs.
Ditto Auxiliaries	0·220 ,,	0·47 ,,
Fuel per hour Main	77,800 lbs.	75,200 lbs.
,, ,, Auxiliary ..	480 ,,	1,050 ,,
Total Fuel per hour	78,280 ,,	76,250 ,,

The foregoing assumes :—

(1) that the exhaust of the steam driven auxiliaries is used to heat up the feed, subsequently further heated by bled steam at maximum possible efficiency ;

(2) That the main turbines are bled at their most efficient point when Diesel-electric auxiliaries are employed ;

(3) The S.H.P. of the installation is 100,000.

In conclusion it may be noted that as the percentage output of the main turbines is reduced, two factors gradually become of importance in making the comparison less favourable to the Diesel-electric system. These are : (*a*) the ratio $\frac{\text{Main}}{\text{Auxiliary}}$ Horse Power becomes gradually less, and (*b*) the maximum possible gain from bleeder feed heating becomes smaller. This is an argument against using such a system (Diesel-electric) in installations which operate frequently at very low proportions of full output.

(b) Failure of a Superheater Tube.

The rapid developments that have taken place in recent years regarding the employment of higher steam temperatures and pressures in both marine and shore plants attach a special interest to the following account of an explosion which occurred from a superheater tube on board the Clyde steamer S.S. " King George V." A brief description of the machinery of this vessel was given in Papers No. 7, from which it will be seen that steam is generated at 500 lbs. pressure, leaving the superheaters at a temperature of about 750° F.

The type of boiler will be seen from the sketches accompanying the article referred to. The steam makes four passes through the superheating elements, the initial pass being arranged at the back, and the final one at the front of the boiler. It will be recalled from a report on an explosion of a generator tube in one of these boilers (Papers No. X) that the preheated air supply enters the ashpit at the back of the boiler, at which part of the grate the fires are thus usually the thinnest and the flame fiercest, actually impinging upon the generator tubes.

During the early life of the vessel a considerable quantity of smoke was given off at the funnels, and during the winter season 1927–28 alterations were made in the air distribution to the fires with the object of improving the combustion. These changes made it possible for air to be admitted near the upper surface of the fire at the back should the fires become thin. There appears to be little doubt that at a certain stage of this process the quantity of top air would ensure complete combustion, with an extremely hot flame.

No trouble was experienced with the superheater tubes prior to the alteration referred to above, but it is significant that the accident here described occurred after only a few days' service with the modified air supply arrangement, especially as mechanical tests of sample tubes drawn from each pass of the superheater during the winter season immediately preceding the explosion showed the metal to have been unaffected by use. Contemporary microscopic examination of the tube material showed, however, that the structure of the metal of the legs nearest the fire had changed completely as compared with that of those remote from the fire. This change would have no practical effect upon the life of the tubes (nor upon their mechanical properties), but indicated that they had been subjected to prolonged heating at about 650° C.

The actual explosion occurred when the vessel was underway, and was indicated by a dull report, accompanied by a slight hiss of steam which enabled the defective boiler to be located. The water level in this boiler remained steady, showing that the damage was probably in a superheater tube. The defective tube proved to be in the row nearest the fire, being the third tube in the second pass from the back end of the boiler. Subsequent examination showed that an aperture $5\frac{1}{2}$ in. long by $2\frac{7}{16}$ in. maximum width had opened in the tube, which had increased in diameter from $1\frac{1}{8}$ in. to $1 \cdot 4$ in. and $1 \cdot 28$ in. respectively above and below the fracture, over a length of 18 in. At the position of maximum opening, the tube wall had been reduced in thickness from $0 \cdot 104$ in. to $0 \cdot 048$ in. The tube was covered externally with a heavy heat scale, which was seamed with longitudinal cracks varying from 2 in. to 5 in. in length. All the tubes of the 2nd pass nearest the fire showed signs of overheating, and had increased in diameter by different degrees. The fractured tube was coated internally with black iron oxide, the product of the reaction between the steam and the heated metal, showing that a temperature of about 750° C. had been reached.

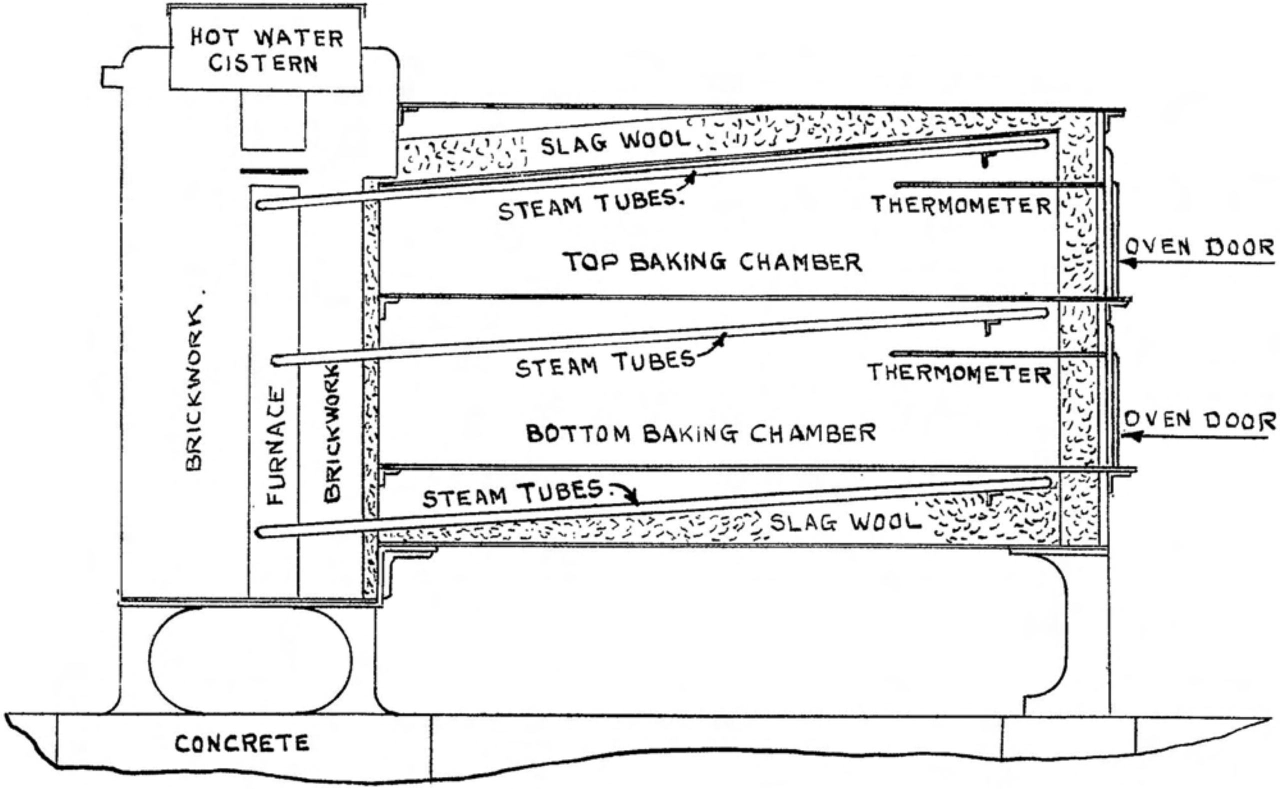
Observation windows were fitted after the explosion in order to ascertain whether the flame penetrated as far as the superheater elements. This was seen to be the case, thus providing an explanation of the damage. It is not known whether the flames impinged upon the superheater tubes prior to the alteration to the air supply, but there is little doubt that this was so. The heat from this flame appears, however, to have been insufficient to cause failure of the tubes until the modified air arrangements rendered possible the production of very intense local temperatures.

The tubes in the first pass might well be able to withstand conditions such as these by virtue of the rapid rate of heat transfer rendered possible by the presence of moisture in the steam. In the second pass, on the other hand, the steam would be drier, and the rate of heat transfer appreciably reduced, leading in turn to a higher temperature of the tube wall. This led to overheating and rapid oxidation of the tubes with final rupture of the tube under consideration.

The thickness of the heat scale upon the external surfaces of the tubes, together with the presence of the longitudinal cracks, indicates that the process proceeded in stages. The scale would first form when the tube temperature was high enough for the action to take place, and would gradually thicken, weakening the metals. The tube wall would swell under the action of the internal pressure, this cracking the scale and permitting further attack of the underlying metal by the flame and the oxidising atmosphere of the flue gases. More scale would form, grooving the metal and weakening it still further, till finally fracture occurred along one of the cracks, as might be expected.

In conclusion, it may be observed that the conditions in the superheaters of this boiler are more severe than in any boilers at present operating in H.M. Service, and would appear to indicate the working limit for tubes of the material used in their particular case.

P4053.


HOT WATER CISTERN
SLAG WOOL
STEAM TUBES.
THERMOMETER
OVEN DOOR
TOP BAKING CHAMBER
STEAM TUBES
THERMOMETER
OVEN DOOR.
BOTTOM BAKING CHAMBER
BRICKWORK.
FURNACE
BRICKWORK
STEAM TUBES.
SLAG WOOL
CONCRETE

(c) EXPLOSIONS IN BAKERY OVENS.

An examination of the reports of inquiries into boiler explosions held by the Board of Trade under the Boiler Explosions Acts reveals that the greatest single cause of accidents is the explosion of tubes in baking and other similar ovens of the steam-heated type.

These ovens are made by several makers, but are all of practically the same construction; they are generally worked by people having no mechanical knowledge, and are often run for years without competent inspection or proper repairs. This type of oven is also fitted in H.M. Service for bakery purposes, and though the standard of inspection is naturally higher, tube explosions have occurred from time to time.

The construction of these ovens is simple in the extreme, with a view to being operated by unskilled labour, and, provided they are kept in a condition approximating to the makers' design, they perform their duties very satisfactorily. As an example, a sketch is attached of a typical oven of this type, as supplied to H.M. Service for bread baking. It consists of an oven having two decks and lined with non-conducting material. This oven abuts a furnace formed by two brick walls about 9 in. apart. Three rows of tubes, inclined at an angle of about 5°, are fitted, running right across the oven, through the brickwork and into the furnace. The top and bottom rows of tubes project right across the furnace, the bottom row acting as the firebars, while the centre row being in the hottest part of the furnace, only project a part of the way into the furnace. The tubes are self contained, each tube being of solid-drawn steel and hermetically sealed at both ends, tested to 6,000 lbs./in.^{2}, and containing a quantity of water. The width of the furnace and the length of projection of the middle row is arranged on the basis of the firm's experience to give the requisite oven temperature, and an equal distribution of heat between the three rows of tubes without undue local heating of the tubes at the furnace end.

Some interesting experiments were recently carried out with the object of ascertaining whether the quantity of liquid in the usual form of bakery tubes exercised any appreciable effect upon the pressure exerted in the tube and upon the temperature of the tube wall.

In these experiments a length of about 9 in. of the tube was inserted in a gas-fired muffle, provided with a pyrometer, while the other end of the tube was closed with a plug in which was inserted a suitable pressure gauge. The portion of the tube outside the muffle was partially lagged, and a second pyrometer was arranged to give readings of the temperature of the tube wall at one selected position. Simultaneous observations were then made of the two pyrometers and of the pressure gauge under various conditions of temperature in the muffle, and with varying quantities of water in the tube: it may be observed here that the so called "special liquid" in the commercial form of bakery tubes is almost invariably water. At first sight the results of these tests, which were very carefully conducted by a thoroughly competent observer, appear to be entirely inconsistent as regards the relation between the temperature of the tube and the pressure observed therein.

The pressures indicated by the gauge may, however, confidently be assumed as correct and calculations made on this basis show that in all cases the tube contained saturated steam and water, or in other words the amount of water originally placed in the tube has little or no effect upon the pressure therein, provided that the quantity is sufficient to ensure that at the maximum pressure experienced the steam remains saturated. In the system under test it is obvious that the heat supplied to the tube is equal to that radiated away by the portion outside the muffle, and this latter factor thus controls the temperature of the tube wall under given conditions in the muffle.

The tube must necessarily be subjected to a temperature gradient varying from a maximum within the muffle to a minimum outside the latter; the highest temperature will be somewhat above that corresponding to the temperature of the saturated steam inside the tube, thus permitting heat to flow from the muffle to the contents of the tube; the minimum temperature similarly will be somewhat below that of the steam in the tube, establishing by these means a flow of heat from the steam to the atmosphere surrounding the tube. It appears probable that when conditions are steady the major portion of the heat flow will be by conduction in an axial direction through the wall of the tube.

The temperature readings taken at the wall of the tube in these particular experiments tell us little except that at the point in question there was a flow of heat from the fluid and steam to the tube wall and thence to the atmosphere. The maximum temperature difference between the steam and the tube was about 64° F., and it is probable that a similar difference, but in the opposite sense, existed at the muffle end of the tube.

Assuming that the tube contained saturated steam and water at the maximum pressure experienced, the temperature of the tube wall within the muffle would be about 690° plus 64° = 754°F., under which conditions the limiting creep stress would be approximately 14 tons/in.^{2}, giving a factor of safety of about 7·0, an ample margin.

It will, therefore, be seen that provided the length of tube exposed to the furnace remains sensibly the same as designed by the makers, there should be little risk of failure.

If the rate of heat transfer from the furnace to the tube is increased either by increasing the furnace temperature or exposing a greater area of surface to the furnace, the temperature may be raised above the critical temperature of water, viz., 706°F., 3,200 lbs./in.^{2} pressure above which water cannot exist as such. With further heating the steam will behave as a gas and the pressure would continue to rise. Owing to the absence of water the circulation within the tube would be impaired, and in consequence of the increased resistance to heat transfer the temperature of the tube wall rises still further, till ultimately failure occurs under internal pressure.

In practically all cases of failure, examination shows the furnace brickwork to be badly eroded, allowing a considerably increased surface of the tubes in that area to be exposed to the furnace with consequent overheating and failure.

Generally only a few tubes in way of the eroded brickwork are affected and the overheating of these tubes is not reflected on the oven thermometers.

Owing to the small capacity of each tube, the explosions are not generally of a violent nature, and sometimes pass unnoticed, though in several cases injuries to the stoker (due to the scattering of the fire) have been recorded. Until recently it was the practice of the makers to use lap or butt welded tubes so that in the event of overheating the weld would open out and give a less violent explosion than if the tube end blows off.

(*d*) A METHOD OF PREVENTING AIR ESCAPE FROM BOILER ROOMS THROUGH FAN ENGINE SAVEALL DRAINS.

The drains to boiler-room bilges from fan engine savealls, etc., are frequently ineffective owing to the escape of air from the boiler room through the open ended pipe usually fitted. The following method of preventing this which was successfully employed in one of H.M. Ships may be of interest.

The end of the drain pipe is led into a receptacle full of water. The air pressure in the boiler-room forces the water up the drain pipe to a height corresponding to the air pressure in use and any drainage water descending the pipe merely accumulates in the receptacle until it overflows. The size of receptacle required obviously depends upon the size of the drain pipe and the maximum air pressure likely to be employed.

The same result can be obtained by bending the end of the drain pipe to a U-shape; in this case the height of the leg open to the boiler-room above the base of the U must be equal to at least half the maximum air pressure likely to be met with in use.