THE DEVELOPMENT AND DESIGN OF CONDENSERS AND CONDENSING PLANT.

The general adoption of the steam turbine for propulsion purposes in H.M. Navy was quickly followed by the appreciation of the fact that this type of engine could make economical use of very high vacua, and this greatly stimulated the designers of condensing plant, bringing into being what is practically a new art.

It is, of course, the object of the designer to obtain a high vacuum without an unnecessarily large condensing surface, and thus he must fit the most efficient air extraction plant that considerations of weight, space and steam consumption will allow, while at the same time so disposing the condensing surface that the maximum possible transmission of heat between the vapour and the water is obtained at all parts of the condenser; the ideal to be aimed at is to obtain uniform heat transmission throughout the tube nest.

The extent of the condensing surface to be fitted in any given case depends essentially upon the desired vacuum and upon the relative proportions of the space into which the surface has to be arranged. The problem depends at source upon the factors influencing the transmission of heat from the vapour to the cooling water, namely, the temperature and the velocity of both vapour and water. From the designer's point of view neither the vacuum nor the temperature of the cooling water can be regarded as variables, and he is thus compelled to direct his attention to the two remaining factors, high values of which favour correspondingly high rates of heat transmission.

A high vapour velocity can only be obtained by exposing but a small area of tube surface to the exhaust vapour, a practice which results in the fitting of deep tube nests. Under such conditions the heat transfer to the tubes remote from the exhaust orifice is small, while the resistance to the flow of vapour is very large, as the latter is proportional to the square of the vapour velocity and to the depth of the tube nest. The present tendency of design therefore is towards a very low vapour velocity across the first rows of tubes, the object being to reduce the depth of the tube nest.

For a given velocity of water it will be found that the transmission of heat is favoured by the use of short tubes, since the average temperature of the circulating water passing through the condenser will thereby be lowered. In order, however, to provide the required surface with short tubes their number must be large, entailing a correspondingly increased volume of cooling water to maintain the required velocity through the tubes. Further, since the loss of head through the condenser tubes takes place principally at entrance and exit a reduction in length is not accompanied by a corresponding lessening of the resistance to the flow of water. In this connection it may be of interest to note that stream lined ferrules are being tried in one vessel, and it is hoped by this means to effect a reduction in the entering and leaving losses, while the more regular flow in the tubes may lessen corrosion at their ends. It is difficult to detect leakage through the screwed gland when this type of ferrule is employed, and hence those fitted in new construction merely have their ends rounded to make a better path of entry for the water.

In practice, therefore, the gain obtained by fitting short tubes is more than counterbalanced by the extra weight and steam consumption of the circulating pump, and the length of the tubes in modern underslung designs of the single flow type is as large as space considerations and the size of the exhaust orifice of the turbines allow.

In cases where the space permits of a reasonable length of tube being fitted and where the circulating pipes can be arranged so as to ensure a fairly even distribution of water flow throughout, single flow condensers are in general to be preferred to those in which the water flows twice through the tubes. In many cases, however, condensers with two or more passes must be fitted, and it becomes necessary to augment the head of the circulating pump in order to overcome the increased entering and leaving losses at the tube plates.

The actual water velocities employed in practice are limited on the one hand by the requirement that the flow must be turbulent, a phenomenon which occurs at a speed of about 0·5 feet per second in standard $\frac{5}{8}$-in. condenser tubes, and on the other by the fact that high water velocities are accompanied by heavy demands upon the circulating pump; in practice it is found that very little benefit is to be obtained by the use of water speeds exceeding about 7 ft. per second.

The heat transmission in a condenser is adversely affected by the formation of water films on the outside of the lower tubes due to the condensed vapour from the upper tubes draining on to them, and special means to reduce the loss of efficiency due to this cause are taken in modern condensers.

In the direct drive battleships now in the Service, and in the earlier cruisers and destroyers, condensers of the well known uniflow type are fitted. The object aimed at in this design is to obtain a reasonably constant velocity of vapour past the tubes at all sections of the condenser, so that the heat transmission between the steam and water should be as nearly as possible uniform.

The tube plate of a typical condenser of this type is illustrated at Fig. 1, the most marked features being the wide vapour path at the top, narrowing down as the vapour is condensed to a small passage at the bottom, and the depth of the condenser, the ratio between the breadth at the top and the depth of the tube nests being about ·7 to 1.

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Fig. I

Scale :- 3/8" = 1 Foot.

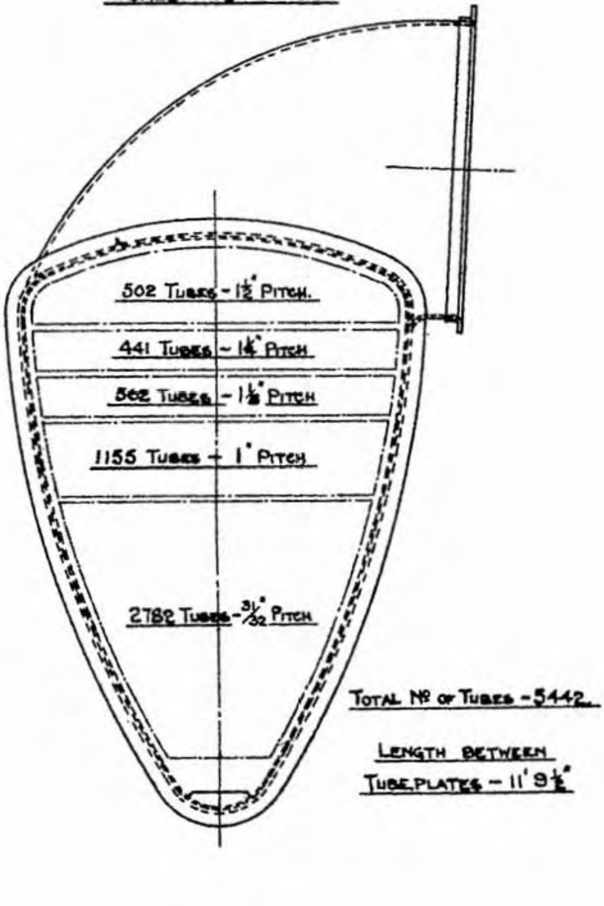
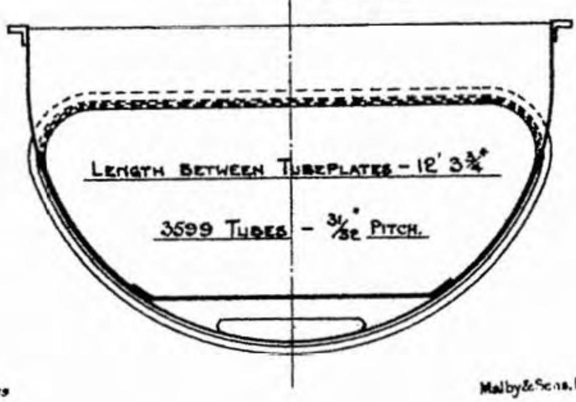


Fig. II

Scale :- 3/8" = 1 Foot



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In the earlier designs the amount of cooling surface provided was about 0·8 sq. ft. per horse-power for battleships, 0·7 sq. ft. for cruisers, and 0·6 sq. ft. for destroyers, the designs providing for lower vacua and less margin for adverse conditions in the smaller vessels than in the capital ships. With the increasing horse-powers of later designs, and the increased necessity for saving weight and space, the surface provided fell to 0·6 sq. ft., 0·5 sq. ft. and 0·4 sq. ft. as average figures for battleships, cruisers and destroyers respectively. The actual increase in heat transmission per sq. ft. of surface was not as great as these figures imply, as the turbines had improved somewhat in efficiency, but a marked decrease nevertheless took place in the cooling surface provided per unit of heat transferred in unit time.

This reduction of cooling surface was arranged in part by reducing the length of the tubes and in part by reducing the cross sectional area of the condenser, retaining its characteristic shape. Reduction in the size of a condenser leads, however, to an increase in the vapour velocity, and in some cases this velocity has reached 450 to 500 feet per second. Difficulty was experienced in obtaining the designed vacuum at full power with many of the condensers with the reduced cooling surface, and there is no doubt that this may be mainly attributed to the increased resistance to the flow of vapour, the pressure drop between the top and bottom of the condenser reaching in some cases as much as 1½ inches of mercury.

Condensers of the underslung type, situated immediately beneath the L.P. turbine, are now universally fitted to the latest designs of geared turbines, the practical advantages of this type of condenser being the elimination of the losses due to the eduction bend and the great saving of weight and space. It will be seen from Fig. 2 that this type of condenser is much wider and shallower than the uniflow design, and thus both the vapour velocity and the length of the vapour path are greatly reduced, any decrease in the efficiency of the heat transmission being outweighed by the decreased resistance to vapour flow. On trials of vessels fitted with these condensers, and with a cooling surface of only 0·35 sq. ft. per shaft horse-power, the vacuum obtained compared favourably with that obtained in vessels fitted with the later types of uniflow condenser with cooling surface 0·5 sq. ft. per shaft horse-power and turbines of similar efficiency.

It should be noted that, although good vacua have been obtained on trials of underslung condensers with cooling surface as low as 0·35 sq. ft. per S.H.P., when ships are new and tubes are clean, the scale formed on condenser tubes during service reduces the efficiency of the heat transmission, and where weight and space allow, it is desirable to provide not less than 0·5 sq. ft. of condensing surface per horse-power.

There is one limitation imposed by placing the condensers underneath the turbines in a warship, namely that owing to the very small distance between the exhaust orifice of the turbine and the upper row of tubes it is not practicable to distribute the steam evenly over the condenser surface. Considerations of this nature have resulted in the development of condensers which are approximately square in plan; even when every endeavour is made to distribute the steam over the tube surface it is probable that portions of the nest are relatively inffective, especially where the L.P. turbines are of the double flow type, receiving steam at the centre.

Typical Modern Condensers.—A typical condenser of late design is illustrated in Figs. 3 and 3A, this unit being provided with rotary water extraction pumps and with steam air ejectors. The general shape of the condenser is not very different from the usual type of underslung condenser, but deep lanes are provided in the tube nests reaching down to about two-thirds of the depth of the nest, and suitable baffles are fitted to guide the vapour. These deep lanes tend to reduce the resistance to the flow of the vapour, and, further, by increasing the extent of the dry surface exposed to the vapour, a reduction in the loss due to the film of water on the tubes is to be anticipated.

The top nests of tubes have a slightly greater pitch than the bottom nests, thus further reducing the vapour velocity and the resistance at this part of the condenser. The actual vapour velocity is about 300 feet per second at the top decreasing gradually to 100 feet per second near the bottom of the condenser.

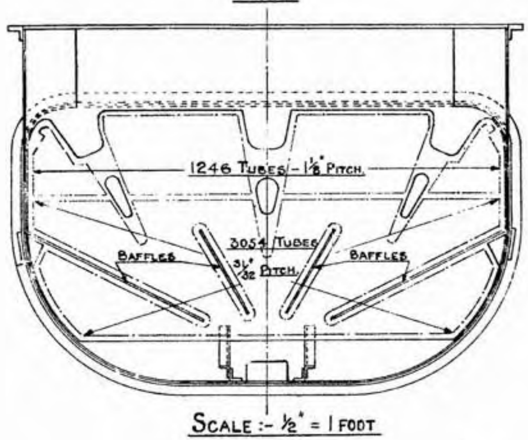
The water extraction pump suction is fitted at the bottom of the condenser, a suitable strainer being fitted over the suction. Two air suctions on each side are fitted, about one-third of the way up the condenser. The wing bottom tubes are separated from the remainder by a baffle so that the air and vapour is drawn by the ejectors from the bottom of the condenser past the wing nests, where the tubes cool the air and vapour, and so further condensing some of the vapour. This reduces the proportion of vapour to air in the mixture which passes over to the ejectors, and slightly increases the density of the air itself, thereby favouring the conditions under which the ejectors are required to work.

The question of the behaviour of condensers under tropical conditions is of increasing interest in view of the number of ships now operating abroad, and the curves given in Fig. 4 show how the vacuum at full power will fall as the temperature of the circulating inlet water increases. These curves have been obtained theoretically but verified by examination of the results of a large number of ship trials in temperate and tropical waters and may be used with confidence.

Whilst steaming at low powers in tropical waters, it is not economical to obtain the highest possible vacuum, as the steam consumption of the main circulators under such conditions will more than offset any decrease in the consumption of the main engines due to the improved vacuum.

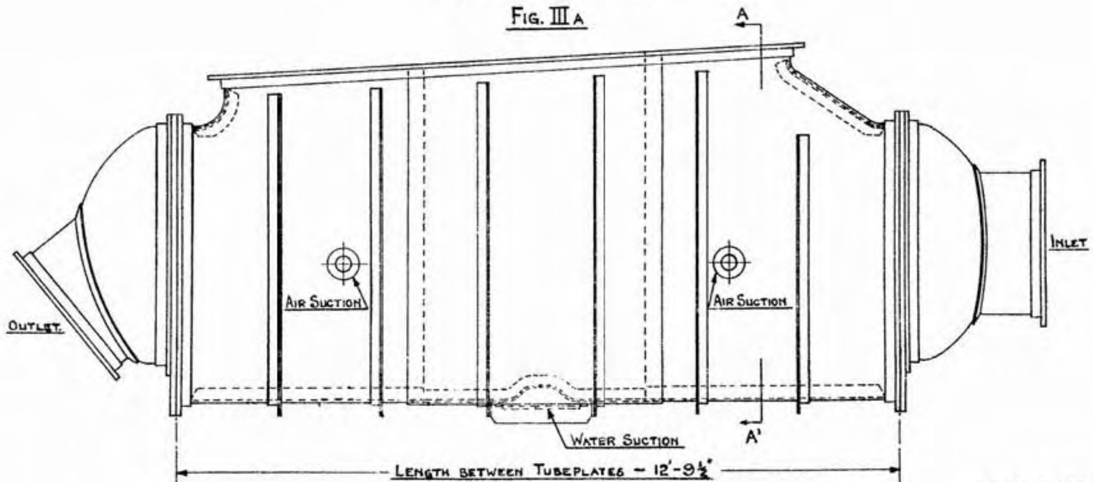
SECTION AT AA'

FIG. III



SCALE :- ½" = 1 FOOT

FIG. III A



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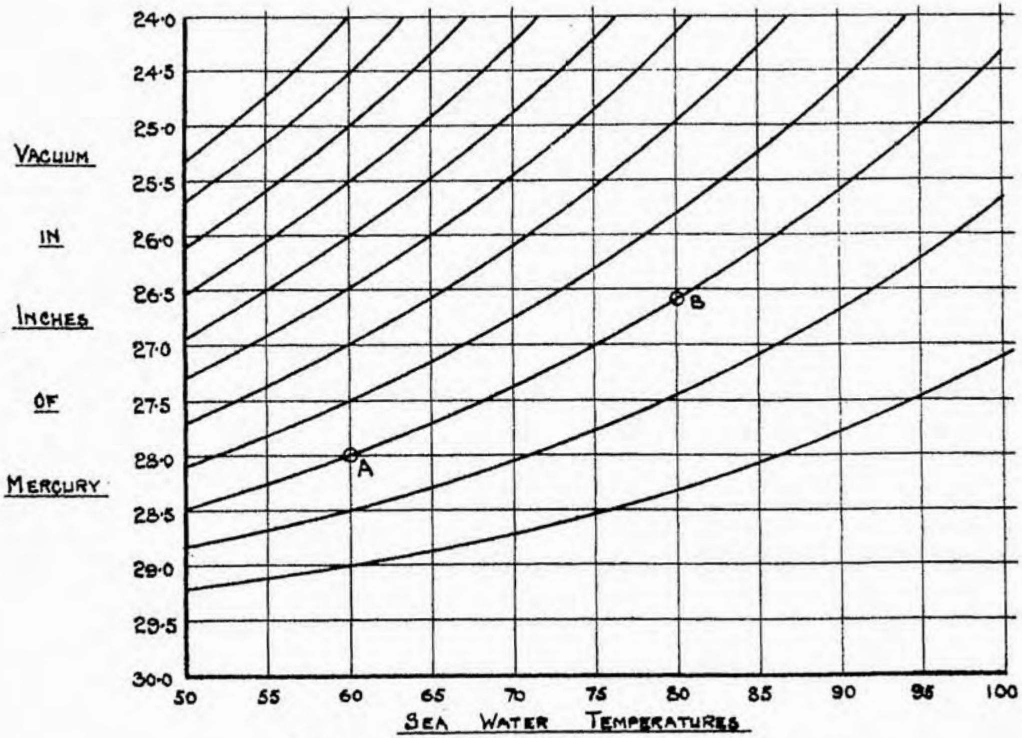

CURVES OF EXPECTATION OF VACUUM.

EXAMPLE

VACUUM OBTAINED WITH SEA WATER TEMPERATURE OF 60°F. IS 28." WHAT VACUUM IS TO BE EXPECTED WITH SEA WATER AT 80°F.?

THE POINT A CORRESPONDS TO 28" VACUUM & 60°F. TEMPERATURE. FOLLOW THE CURVE TILL THE POINT B IS REACHED ON THE 80°F. ORDINATE.

VACUUM EXPECTED IS READ OFF HORIZONTALLY FROM B & IS 26·6"

FIG IV.

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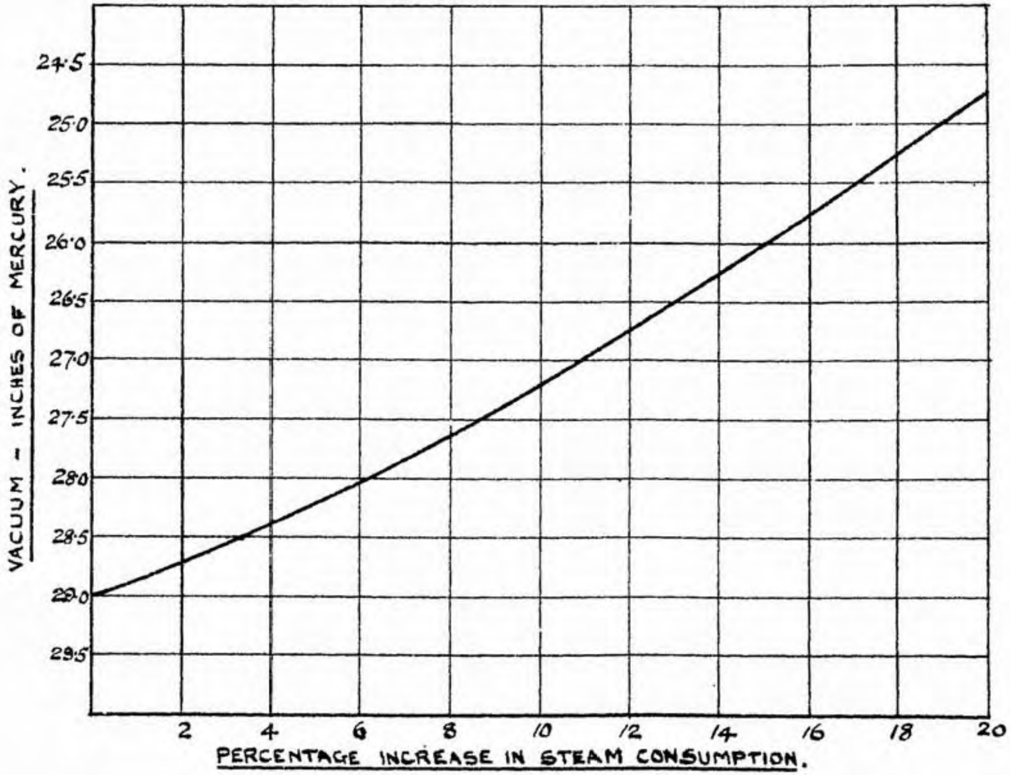

CURVE OF EXPECTATION OF STEAM CONSUMPTION. FIG IVA.

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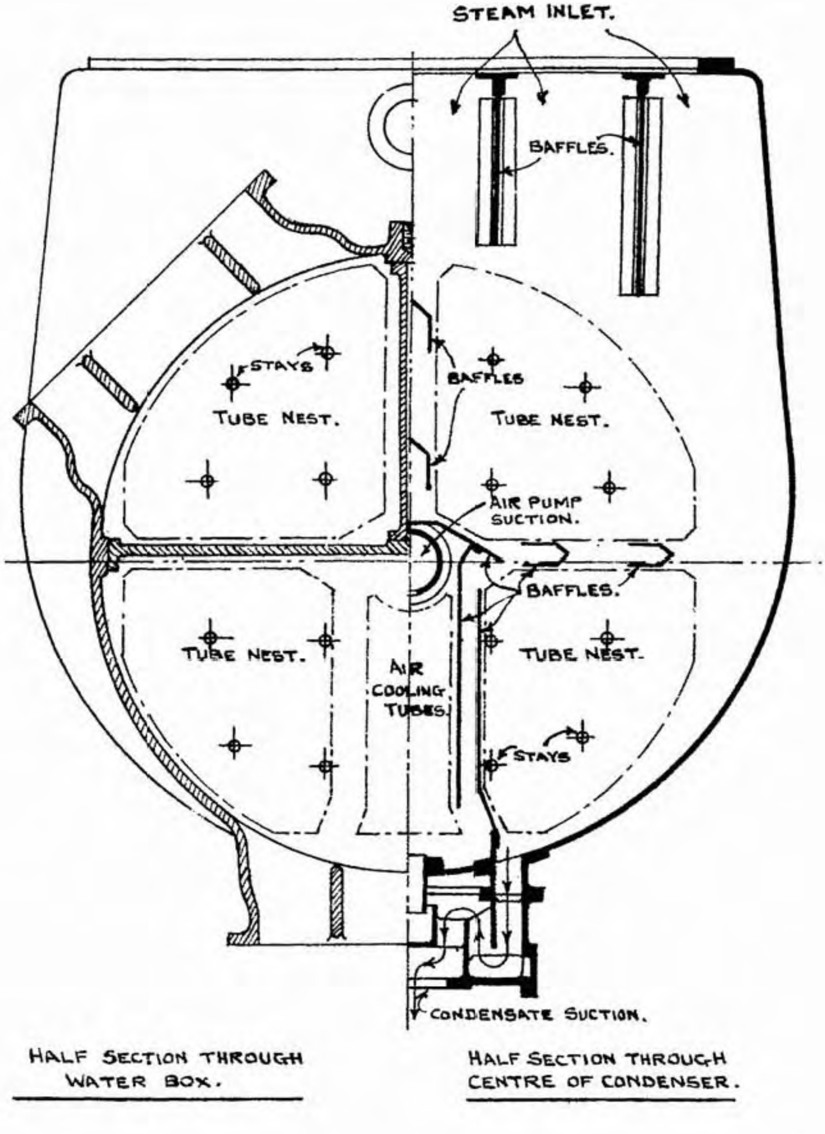

FIGURE V.

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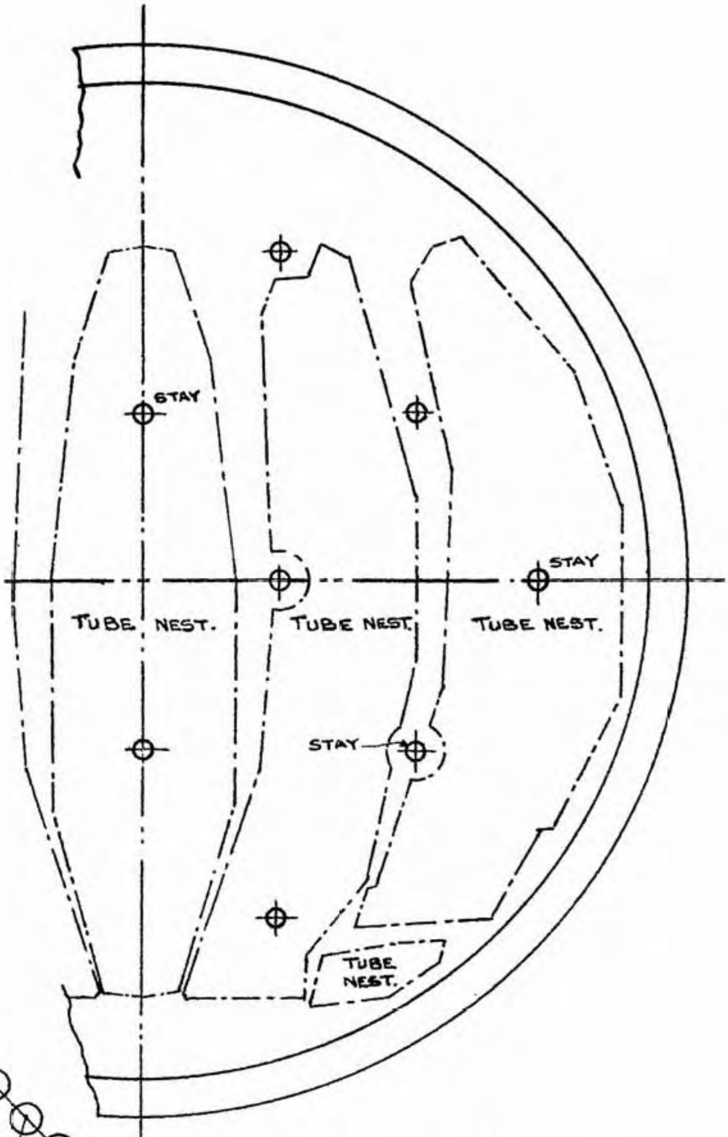
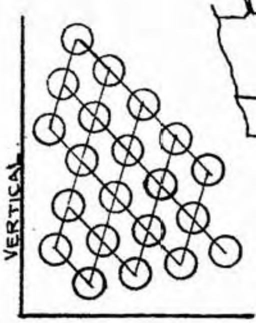


HALF VIEW OF TUBE PLATE.

FIGURE VA.



SKETCH OF LAYOUT OF TUBES.

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Calculations for the case of a particular vessel indicate that maximum economy is obtained under cruising conditions with a vacuum of 27 in. and sea water temperature 80° F., and at $26\frac{1}{2}$ in. with sea water temperature 90° F.

Obviously the most economical vacuum to employ depends on the design of the turbines and of the circulating pumps, but may readily be estimated. The curves given in No. 5 of "Papers on Engineering Subjects" will be found of assistance in estimating the consumption of the circulating engines, whilst that of reciprocating air pumps will be found to vary almost directly as the number of strokes per minute. In Fig. 4A is shown the approximate effect of vacuum upon the steam consumption of the turbines. This curve applies either to Parsons or to Brown-Curtis multi-stage turbines, and gives the percentage increase of consumption with decreasing vacuum. To enable an estimate to be made of the increased weight of steam flow it is necessary to know the actual steam consumption of the turbines under the given conditions. The steam consumption under various power conditions is usually known from trial records or the contractor's estimate.

If the total steam consumption of the turbines at various powers be plotted against horse power, the result will be a straight line or a very flat curve. Hence the overall steam consumption for any given power can be obtained providing it is known at two or more other powers.

Figs. 5 and 5A show the arrangement of the tube plates of two interesting condensers of advanced design used in power stations ashore.

The condenser in Fig. 5 is a design by the Metropolitan Vickers Company, the principal features being that the tube nest is circular in shape and fitted eccentrically in the condenser casing so that the exhaust steam can enter the nest over a very large surface area, the air being extracted at the centre of the condenser. It will be seen that there is a comparatively short path for the flow of vapour through the nest and that, due to the low velocity and short path, the resistance to vapour flow will be very small.

A further advantage claimed for the design is that the temperature of the condensate at the bottom of the condenser is high, approaching that of the temperature of the exhaust, whilst the temperature at the air suction is low.

Fig. 5A shows a condenser fitted in a French power station. The interesting features of this condenser are the number of long vapour lanes, and the fact that no tube is fitted directly under one a row or two higher up. Both features are designed to minimise the effect of the formation of water film, which, as has been previously stated, is a cause of great loss of efficiency. The fact that the tubes are "staggered" from the vertical tends to prevent the drips of water from an upper tube from completely covering the surface of a lower one, although this advantage would hardly be realised in a marine installation. Very good results have, however, been obtained from this condenser on service in shore plants.

PART II.—CONDENSING PLANT.

Main Circulating Engines.—In almost all existing ships the main circulating engines are of the reciprocating type, but in certain cases turbines have been fitted, being coupled directly to the impeller shafts. The usual designed head of the pump is 20 ft. for single flow and about 30 ft. for double flow condensers at full power.

Very high speeds are necessary to keep the steam consumption of auxiliary turbines low without undue demand upon weight and space, but although circulating pumps can be designed for operating at high speeds, the clearances become rather fine and the efficiency of the pump tends to fall off on service due to wear.

In the later vessels geared turbine-driven circulating pumps are being generally fitted, the turbine running at some 4,000 or 5,000 revolutions per minute, and the pumps at about 600 revolutions, thus providing the required compromise and enabling a steam consumption somewhat less than that of the uncompounded reciprocating combination to be obtained. At full power the steam consumption per water horse power per hour of the geared turbine pumps is about 56 lbs. as against 75 lbs. for a single cylinder reciprocating pump, and at three-fifths output the corresponding figures would be about 90 and 100 lbs. At very low outputs, however, it is probable that the reciprocating pump is more economical than the turbine-driven pump.

After a condenser has been in service for some time, a layer of scale is formed on the insides of the condenser tubes while the exterior surfaces become covered with grease. The designed pressure head of the circulating pump has been insufficient in the past to allow of the full output of water being attained against the additional resistance imposed by the scale, and, as the heat transfer under these conditions is also low, very poor vacua have been recorded in certain cases as a result. In new construction, however, the designed head against which the full output of the pump can be obtained has been very greatly increased, the new figures being 60 ft. for double flow and 40 ft. for single flow condensers.

From the above point of view it would be an economic proposition to clean condenser tubes internally, provided that a satisfactory method of so doing can be developed, as in addition to reducing the head on the pumps, cleaning the tubes would result in improved heat transfer. Chemical methods have been proposed for cleaning the tubes and have been actually employed in some cases, but their use is viewed with suspicion since damage

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DIAGRAMMATIC ARRANGEMENT OF COMBINED STEAM JET & RECIPROCATING AIR PUMP.

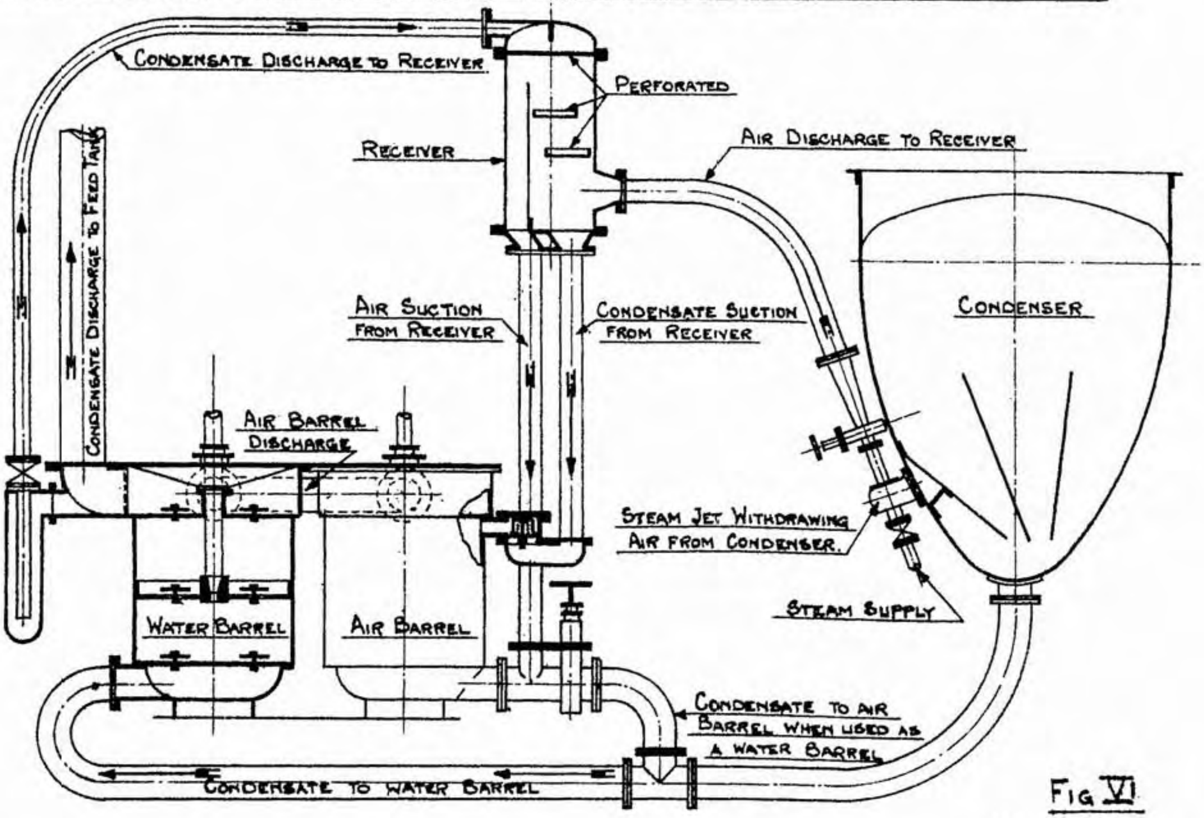

FIG VI

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may be done to the material of the tubes unless the processes are very carefully controlled.

Mechanical methods are in general both ineffective and expensive, while being also liable to damage the tubes. From the point of view of corrosion it is very doubtful whether corrosion films or surface deposits of salt should be removed, since these may act as protective coatings and tend to inhibit corrosion. In view, therefore, of the cost of retubing large condensers, it is doubtful whether internal cleaning of the tubes can be recommended from the financial aspect.

External cleaning of the tubes is, however, wholly beneficial and the method to be employed is laid down in the Engineering Manual.

Arrangements for Extracting Air and Water.—In nearly all the ships now on service a dual reciprocating air pump having wet and dry barrels is fitted for extracting the water and air respectively from the condenser, the arrangement being too well known to need description here.

In certain ships, a steam jet is fitted to draw the aerated vapour from the condenser, the discharge from the jet being passed to a cooler which may be either of the surface or jet type.

The cooler condenses the steam used in the jet and also most of the vapour present, the vapour condensing readily as the pressure in the cooler is higher than the condenser pressure.

The air, the remaining vapour, and the condensed vapour are then passed into the suction of the dry barrel of a turn reciprocating pump.

The wet barrel draws condensate from the bottom of the condenser as usual.

A diagram of the arrangement is given in Figure 6.

The conditions of working of the dry pump are thus much improved, not only on account of the reduced volume of vapour and air with which it has to deal, but also owing to the greater pressure obtaining at the suction of the pump. It is therefore possible either to make use of smaller pumps or to obtain the benefit of an enhanced vacuum.

In both the arrangements described, although the air and water are dealt with by separate pumps, it should be noted that there is no special de-aerating effect, since the two pumps have a common discharge pipe in each case.

With the development of steam ejectors, these appliances in combination with rotary water extraction pumps have entirely replaced dual air pumps in new construction, the advantages of the new system including not only the possibility of attaining higher vacua but also considerable economy in weight and space : the entire absence of working parts in the ejectors should lead to reduction in the cost of maintenance, both in labour and material.

Early experiments with steam ejectors indicated that they would not operate with complete stability under all running conditions with compression ratios of more than about seven to one, and as a consequence two, or three stage ejectors are invariably fitted.

Even with multistage ejectors, if the air leakage with which the ejector has to deal is unusually large, the steam jets will fail to entrain the air, and the vacuum will fall to nil. This limitation of ejectors requires to be borne in mind, and special attention to the avoidance of excessive air leakage is necessary where ejectors are fitted.

The quantity of air leaking into the condenser of a set of turbines remains sensibly constant at all outputs, and thus the steam consumption of the air ejectors becomes increasingly important at the lower powers: this effect is specially marked owing to the small variation in the steam consumption of air ejectors over the whole of their capacity range.

The steam consumption of air ejectors may, however, be very considerably reduced by passing the discharge from each stage through suitable coolers where the condensible vapours are treated and where the air is cooled, with consequent reduction in its volume. Interstage coolers are now being embodied in the latest vessels, the drainage from the coolers being passed back again to the condensers.

In the earlier designs of intercooled ejectors, sea water from the main circulators is used as the cooling medium. On examination of Figure 7, which shows a section through a typical ejector of this type, it will be seen that there is a somewhat complicated system of jointing, which has led to the adoption of fresh water circulation in the later designs, thus precluding any possibility of leakage of sea water into the feed circuit at this point. In these designs the whole of the condensate is pumped through the ejector circulating water system, the condensation of the steam and vapour of the output being carried out entirely within the ejector casing; the final output is led straight to the feed tank, suitable venting arrangements being provided for release of the air.

The steam consumption of the ejectors depends upon the vacuum produced rather than upon the output of the turbines, and thus it may happen that under certain conditions insufficient condensate may be available for cooling the ejectors. In order to prevent a drop in vacuum under such conditions, arrangements are made for the extraction pumps to deliver sufficient water for cooling purposes, the surplus not required for the boilers being returned to the condenser through a hand-controlled byepass valve fitted on the circulating water discharge from the ejectors. Manipulation of this byepass will cause the float-controlled make up feed valve to operate and thus admit into the system an additional amount of water corresponding to the

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THREE STAGE INTERCOOLED AIR EJECTOR.

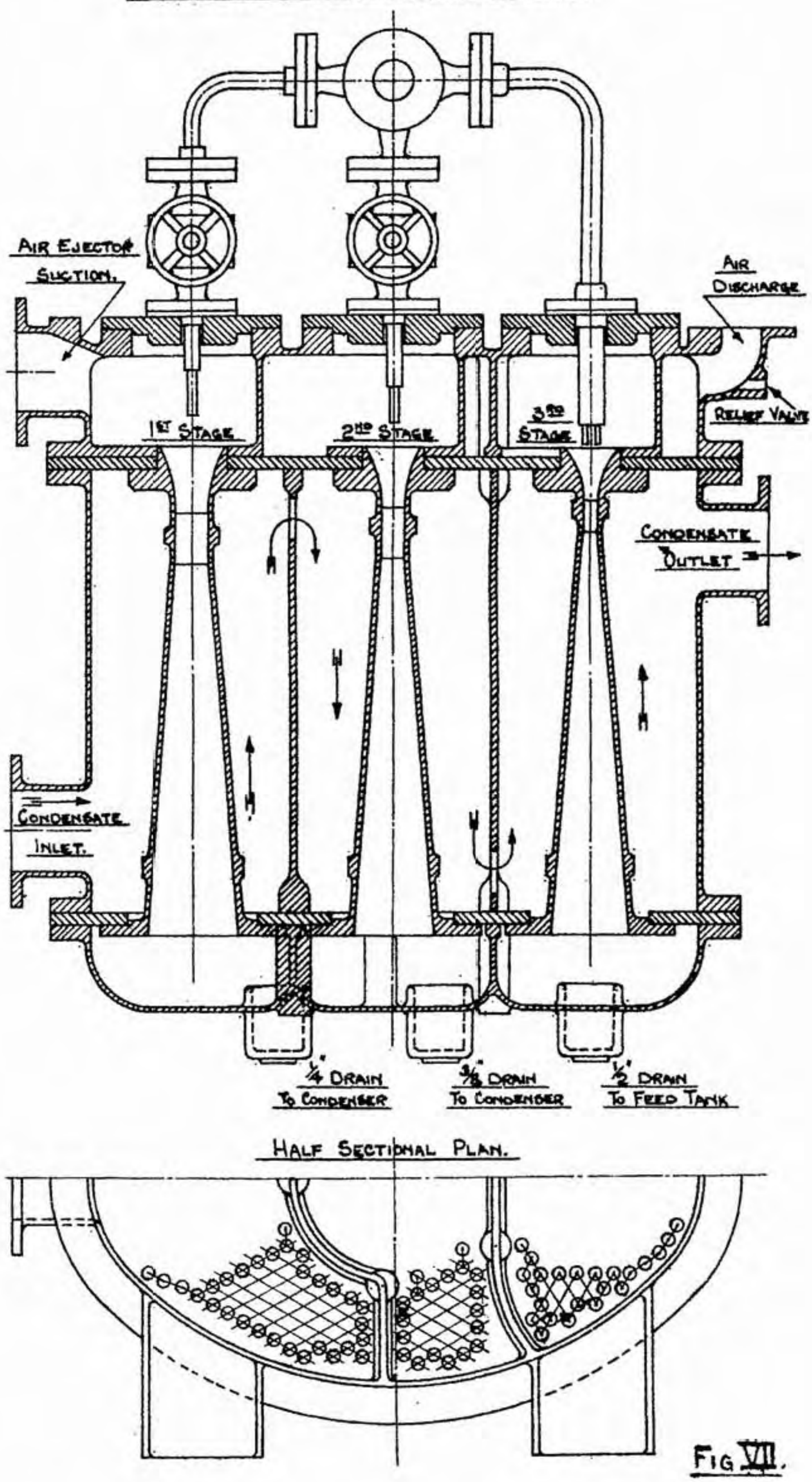

FIG. VII.

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opening of the byepass: the temperature of the air ejectors is thus under direct control. The use of condensate instead of salt water for cooling the air ejectors results in an increase of steam consumption in these units.

The heat of the ejector steam passes into the feed water when condensate cooling is used, but in Naval installations this fact does not result in any appreciable overall gain since there is already available more exhaust steam than is required for feed heating purposes. Under such conditions the addition of further heat to the feed water will only set free exhaust steam which cannot normally be economically disposed of: the maximum gain from this source will only be obtained when the whole of the surplus exhaust is made use of in the evaporators.

The capacity of the air ejectors fitted is about one cubic foot of air per minute per thousand horse power at 28-inch vacuum, and this compares favourably with the capacity of reciprocating air pumps usually fitted.

Extraction pumps are fitted to draw the condensate from the bottom of the condenser. These pumps are of the rotary impeller type, the spindles being arranged vertically in order to obtain the maximum positive head on the suction.

Two impellers are fitted, acting in parallel, with the suction between them, so that the spindle glands are not under a vacuum.

The main pumps are driven by a steam turbine, but where stand-by pumps are fitted, these are driven by electric motors, and are designed to be used for cruising purposes, one motor driven pump taking suction from two condensers, thus reducing the consumption of auxiliary steam.

The steam driven pump is fitted with a governor so that it will not speed up and trip should it pump the condenser dry.

In new construction, a closed feed system is fitted which is similar in principle but differs in several details from that described in Part III of "Papers on Engineering Subjects." With such a system it is essential that the characteristics of the water extraction pumps should be such that they can supply as much water as may be required by the feed pumps at any time. The level of water in the bottom of the condenser is automatically controlled by float operated valves, but experience has shown that there is often a considerable lag in the operation of these valves and it is therefore desirable that the water reservoirs in the bottom of the condensers should be large in order to ensure the water supply on the suction side of the extractor pump.