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Oil Purifying with Continuous Lubrication.

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A good many modern machines, especially Diesel engines and steam turbines of the larger type, are now equipped with so-called continuous lubrication. This is characterized by the oil being conveyed from a large tank, the so-called sump, to the different lubricating points and from these back to the sump in continuous circulation. Usually also a purifying apparatus, centrifuge or filter, is installed to remove impurities from the oil.

Although this arrangement can be regarded as standard, its mode of action and the condition for the attainment of high efficiency are not well known. A close investigation of the problem of continuous lubrication can thus be of a certain importance.

As is always the case, when making an investigation, the first step is to make clear the nature of the problem. For this purpose a comparison will be made with the "ordinary" lubrication, which is still used in most mechanical devices.

With this ordinary lubricating device new oil is led from one or more tanks to the different lubricating points, from which the used oil is discharged as "waste oil". The supply can be intermittent, *e.g.* with ordinary oil holes, or continuous, *e.g.* with sight feed lubricators. The waste oil is either allowed to pass away or it is collected in some kind of receptacle and is often utilized in one way or another.

Not infrequently an oil purifier is used with arrangements of this kind. What is then its function? Simply to restore the waste oil as nearly as possible to its original condition. The waste oil is put into a tank, passes through the purifier and is collected in another tank, from which the pure oil tank of the machine is filled. What is required is thus something of a purely qualitative nature, namely that the oil discharged from the purifier shall be as clean as possible. Quantitatively, *i.e.* with regard to the capacity, there is no demand other than the obvious one that the capacity shall be large enough to treat the discharged waste oil.

Considering a fact, which will be investigated more fully in the following notes, viz. that the purifying efficiency of an apparatus of a given kind generally increases with falling capacity, it is realized at once that the condition for the attainment of the best result is that the purifier should work at the *lowest* capacity at which it still is able to treat the whole quantity of waste oil.

The position is quite different with continuous lubrication. The oil is led to the lubricating points not from the *purifier* but from the *sump*. From a lubricating point of view therefore the condition of the oil discharged from the purifier is unimportant. The chief point is that *the oil in the sump* should be as clean as possible. The rate of oil circulation, measured for example in litres per hour, is so great with installations of the type in question that the whole quantity of oil cannot pass through a purifier of a size acceptable in practice. This is partly due to the fact that the purpose of the circulation system is not only lubrication but also cooling. The oil flow is therefore divided into two parts, one going through the purifier and the other past it. The apparatus works on the by-pass principle. In practice a step further has lately been made and the circulation is divided into two quite independent branches, one for the lubrication and the other for the purification circuit. The former passes from the sump to the lubricating points and back to the sump; the latter from the sump through the purifier and back to the sump. In the following analysis this double circulation system will first be dealt with, as its investigation is simplest.

Let us examine the working of the system.

At the outset the sump and the system as a whole are supposed to be filled with pure oil. The engine and the purifier are started and impurities from the lubricating points are conveyed to the sump oil, which thus gradually becomes contaminated. As long as the sump oil only contains impurities in insignificant quantities the purifier cannot eliminate more than a small quantity of them, hence the percentage of impurities in the sump oil rises. In this way the possibility for the purifier to extract impurities is increased and finally the point is reached at which the *purifier extracts the same amount of impurities in unit of time as is supplied from the lubricating points.* A state of equilibrium is thus reached, and the impurity content remains constant.

The purpose of the lubricating device is to keep the impurity content at this state of equilibrium as low as possible, and our problem is to find the conditions under which this will prevail.

A little consideration will show that in this respect the capacity of the purifier is of dominating importance. Let us take the case of an ideal purifier, *i.e.* one which, at least within certain limits, entirely removes the impurities out of the oil, irrespective of the capacity. This apparatus thus extracts the whole quantity of impurities contained in the oil flowing through it, but the quantities of impurities extracted in a state of equilibrium are equal to the quantity of impurities developed in the engine in unit of time. The amount of oil flowing to the purifier in unit of time will thus contain a certain amount of impurities. It is then evident that the greater the capacity the greater is the quantity of oil in which these impurities are intermixed, *i.e.* the smaller is the percentage of impurities, and vice versa. The condition for obtaining the lowest possible percentage of impurities in the sump oil is thus the maintenance of the highest possible capacity; in fact, the percentage of impurities is inversely proportional to the capacity.

In practice, however, the matter is not so simple, since, as already pointed out, the purifying efficiency of an apparatus is not independent of the capacity. The impurity content of the liquid discharged from a purifier depends on three main factors, viz. the type and size of the apparatus, the nature of the liquid flowing to the apparatus, the magnitude of the throughput.

With regard to the type of apparatus, we may distinguish between filters and centrifuges.

In a filter the liquid is led through narrow passages where impurities above a certain size are retained. It would therefore appear natural to express the course of filtration in the following way:

Impurities above a certain size are extracted, those under a certain size pass, the capacity of the filter exercises no influence. This would, however, give quite a faulty picture of what actually happens. In point of fact the passages in the filter are not all of the same size; moreover, they change during the filtration process by the increasing layer of filtrate. In addition, the still not fully explained adsorption phenomenon may enter. One may thus be justified in saying that all particles above a certain size are retained, but that in an irregular manner some of the smaller particles remain, and some, down to a certain limit, pass through. Under this limit, all particles pass through.

With regard to the capacity, it may be assumed that with increased capacity the higher pressure and the greater velocity of flow will force through certain particles, which would otherwise have been retained, so that the efficiency of purification will diminish with increasing capacity. It must be admitted, however, that so far as the filters used in practice are concerned the conditions are somewhat incompletely known in detail. We can, however, reckon with a very satisfactory purification, which varies only little with the capacity within the applicable limits.

The mode of action of a centrifuge is quite different. The liquid with the particles held in suspension is subjected to an intense centrifugal force. Those particles, the specific gravity of which differs from that of the liquid, will then travel outwards or inwards according as the specific gravity is greater or smaller. The velocity of movement depends on the viscosity of the liquid and on the size and form of the particles. We will call the swifter particles "larger" and the slower ones "smaller", regardless of their actual dimensions.

While the particles are moving in the liquid the latter flows through the centrifuge at a speed dependent on the capacity. The condition to be fulfilled in order that a particle should be separated out and thus not be discharged with the outgoing liquid is evidently that it should reach the "safe zone" before it has been carried to the outlet by the flow of liquid. The safe zone then means that part of the centrifugal bowl, where the particles no longer flow with the liquid *towards the outlet for purified liquid*. The safe zone consists therefore either of the so-called sludge space of the bowl or, in the case of continuous sludge discharge, of that part of the bowl where the flow leads to the outlet for sludge. It is evident that sufficiently large, swift particles reach the safe zone in all circumstances. The smallest particles, in which this applies, are called the limit particles and their velocity the limit velocity. These particles and the larger ones are thus all "saved". But what happens to the smaller particles? Are they "lost"? An example will show how the matter stands.

Let us presume that a river with a steep, inaccessible rock face on one side and a comfortable sandy shore on the other flows to a waterfall. Some way above the waterfall a bridge leads over the river, and on this bridge is a crowd of people, all able to swim but with very different swimming ability. Suddenly the bridge collapses and all who were on it fall into the water. Everybody immediately starts swimming straight towards the sandy shore, while the current carries them on to the fall. Naturally, anyone who can swim the whole width of the river in the same or shorter time than is required for the water to flow from the collapsed bridge to the fall will be saved. Those who require exactly the same time correspond to the "limit particles" and their speed corresponds to the "limit speed".

The above-mentioned swimmers are thus on the safe side but for the slower swimmers it is a question of good or bad luck. If a man who can swim at three-quarters of the limit speed was lucky enough to be standing somewhere within three-quarters of the length of the bridge reckoned from the sandy shore he will be saved, otherwise not. At half the limit speed he must stand within the nearest half of the bridge, at a quarter within the nearest fourth, and so on. The rule is thus that, with chance distribution, the probability of getting saved of a person is equal to the ratio between his speed and the limit speed.

Reverting from this perhaps somewhat dismal example to reality, it is easily realized that with chance distribution the probability of a small particle being separated out, and thus not passing away with the purified oil, is equal to the ratio between the travelling speed of the particle and the limit speed. If, further, we observe that the number of the particles is very great so that the reality practically coincides with the probability, it is obvious that of a number of particles of a certain size such proportion is saved as corresponds to the ratio between the particle speed and the limit speed. The law of centrifuging can thus be summed up in the following propositions :—

(1) All the particles larger than the limit particles or of equal size are saved.

(2) The smaller particles are saved in proportion to the ratio between their own speed and the limit speed.

Although the aim of this exposition is not to explain the relative merits of the different purifying apparatus, centrifuges and filters, it may be appropriate in this connection to call attention to a mistake which is sometimes made. From those people representing the filter interests the following reasoning is not infrequently heard :—"The impurities which were not removed on the first passage through the purifier will not be removed on a second passage for the same reason, *i.e.* they are beyond the scope of the equipment"*

Evidently it is taken for granted that certain particles are centrifugable while others are not; that the former are separated out during the first passage through the centrifuge but the latter are not separated either in the first or any subsequent passage, and remain to cause a constantly progressive contamination of the lubricating oil. It is clear from what has already been stated that this reasoning is wrong and is due to lack of knowledge as to the manner in which a centrifuge works. There are no uncentrifugable particles. A small particle, which during one passage through the centrifuge has had "bad luck" and has not been separated out, can have "luck" at a following passage and be saved. The fact that a particle has passed through the centrifuge once without being retained does not prove that it cannot be retained. It might equally well be contended that in shooting at a rabbit if he has once been missed he could never be shot. On the other hand, it is obvious that as the likelihood of the small particles being separated out is smaller than is the case with the greater, a certain concentration of small particles takes place in the oil, so that when the above-mentioned state of equilibrium is attained, the frequency function for the particles in the oil differs from that of the particles at the moment they are introduced into the oil. This tendency towards concentration is modified, however, by the fact that the small particles have a certain propensity for combining into larger agglomerates, thus increasing the travelling speed.

The aforementioned propositions regarding the centrifuging would render it possible to calculate beforehand the separating result for spherical particles, which follow Stokes' law, if the frequency function for the particles be known. Unfortunately, this is practically never the case. With milk, which from the standpoint of the form of the particles is ideal (small spherical fat globules) the diameter of the limit particles lies below 0·001mm. for modern separators, and it is therefore hardly

* A. Beale: A Paper read before a meeting of the Diesel Engine Users' Association at Caxton Hall, Westminster, 1936.

feasible to find a reliable frequency function for the small particles.

One is thus confined to empiric methods. This, however, does not prevent some conclusions of considerable practical importance being drawn from the above theoretical discussion.

First it is obvious that if, for a certain centrifuge, the capacity be increased and consequently the time in which the particles can reach the safe zone is reduced, the limit will be displaced in a direction towards greater particles; the limit particles are larger than before. Of course, the quantity of impurities retained in oil will then also be greater, at least with continuous frequency function.

Thus: *Increased capacity impairs the purification, and vice versa.*

Further, it is obvious that for a given frequency function, a certain size of the limit particles corresponds to a certain degree of purification.

Thus: *The size of the limit particles can be taken as a measure, on an empiric scale, of the purification.*

Finally, it is obvious that if with a constant frequency function the total quantity of impurities in the liquid circulated through the centrifuge be changed, the quantity of particles within each size-class is changed in the same proportion.

Thus: *The impurity content of the liquid discharged from the centrifuge is proportional to that of the ingoing liquid.*

Now consider the case of a purifying apparatus; it is mainly a centrifuge which is in question, although the reasoning also applies to a filter, working at a capacity of Q litres per hour. The impurity content of the ingoing oil is $=r$ per cent., *i.e.* r kg. impurities per 100kg. of ingoing oil.

Then the impurity content in the outgoing oil may be written:—

$$p=r.f\ (Q) \text{ per cent.} \qquad (1)$$

where $f\ (Q)$ is a function of Q.

From one kg. of oil passing through the purifying apparatus is thus extracted a quantity of

$$\frac{1}{100}(r-p) \text{ or } \frac{1}{100}\,r\ [1-f\ (Q)]\text{kg.} \qquad (2)$$

As the capacity is Q litres or $Q.s$ (s=specific gravity of the oil) kg. per hour, the quantity of impurities extracted per hour will thus be

$$g=\frac{Q.s}{100}\,r\ [1-f\ (Q)]\text{kg.} \qquad (3)$$

The quantity of impurities developed in the engine in unit of time is G kg. If the engine and the purifying apparatus are always in operation simultaneously, then under conditions of equilibrium g and G must be equal. It is, however, common practice for the engine to work continuously, while the purifying apparatus is stopped from time to time for cleaning. It is also possible that the engine may work intermittently whilst the purifying apparatus is in service continuously, or at any rate for long periods without stopping, *e.g.* in land power stations. Assume, quite generally, that during a uniformly recurring period the engine is working T hours and the purifier t hours.

Under conditions of equilibrium $t.g=T.G$ or

$$\frac{Q.s.t}{100}\,r\ [1-f\ (Q)]=T.G \qquad (4)$$

$$\text{hence } r=G.\frac{T}{t}.\frac{100}{s}\ \frac{1}{Q\ [1-f\ (Q)]} \qquad (5)$$

Now, what we desire to know is for what capacity value Q, r will be the minimum. This is ascertained in the ordinary way by derivatives of equation 5, giving the condition:—

$$Q.f^1\ (Q)+f\ (Q)=1 \qquad (6)$$

$$\text{hence } Q_{opt}=\frac{1-f\ (Q)}{f^1\ (Q)} \text{ and} \qquad (7)$$

$$r_{opt}=G.\frac{T}{t}.\frac{100}{s}\ \frac{f^1\ Q}{[1-f\ (Q)]^2} \qquad (8)$$

As will be seen, the most suitable capacity only depends on the function $f\ (Q)$. From equations (7) and (8) it results that, if with constant frequency function the purification were independent of the capacity, thus $f\ (Q)=k$ and $f^1\ (Q)=O$, then $Q_{opt}=\infty$, and $r_{opt}=O$, provided, naturally, that $f\ (Q)<1$, *i.e.* that a purifying effect is present.

This, consequently, leads us a little farther than the previous general reasoning. According to this the sump oil would be kept quite clean, if one had a purifier capable of purifying the through-flowing oil completely and of working at an infinitely high capacity. It is now evident that the first condition is unnecessary. All that is required is that the purifying effect is independent of the capacity; it need not be complete. This shows distinctly the difference between the "ordinary" lubricating arrangement and continuous lubrication, so far as the purification of oil is concerned. A purifying apparatus which at any capacity would retain, for example, half the ingoing impurities, would with the "ordinary" arrangement give a highly contaminated oil, whereas with continuous lubrication it could keep the sump oil quite clean.

In order to find out the most suitable capacity of the purifying apparatus in practical use, $f\ (Q)$ must evidently be determined. As a principle this is very simple. All one has to do is to let the purifying apparatus work at various different capacities and to determine p and r for each capacity. The ratio $p:r$ gives the corresponding $f\ (Q)$ for every value of Q. The function $f\ (Q)$ may then be represented graphically and possibly expressed analytically.

In practice, however, the matter is more difficult.

Most large Diesel engines are installed on board ships and it is not always very easy to obtain permission to carry out tests, which interfere with the ordinary routine in the engine room. Nor is it very easy to find a suitable experimenter who could carry out careful work under the exacting conditions in the engine room of a ship in high seas. In addition to this, the tests must be extended over rather a long period. The particle-frequency in the oil, as pointed out previously, is not exactly the same as with the impurities entering the oil, hence the tests should be carried out after equilibrium has been reached. But for reasons which will be gone into more closely later on, it takes rather a long time until equilibrium is reached again after a change of the capacity of the purifying apparatus has taken place. The time for a complete series of tests is therefore considerable and during them disturbances may easily occur.

The De Laval organisation has, however, succeeded in carrying out some good series of tests, and the results are very interesting.

The tests of which an account will now be given have been made with a marine Diesel engine of 5,600 h.p., the quantity of sump oil amounting to about 21·000kg. The purifying plant consisted of two De Laval centrifugal oil purifiers. The centrifuges worked 3·25 hours per every 4 hours watch; thus $t=3{\cdot}25$; $T=4{\cdot}00$. The specific gravity of the oil $s=0{\cdot}87$. By measuring the quantity of impurities extracted in the centrifuges at that state of equilibrium, G was determined to be $-0{\cdot}285$kg. *per centrifuge*. The total quantity of dry impurities introduced per hour from the engine into the oil is thus 0·570kg.

The function $f\ (Q)$ is represented by Fig. 1.

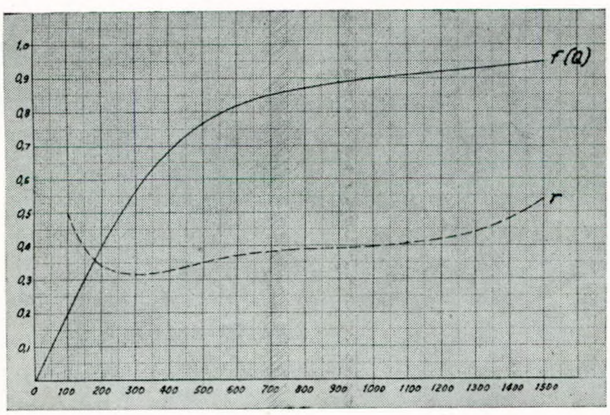


FIG. 1.

In accordance with the equation (5) we get:

$$r=0{\cdot}285\ \frac{4{\cdot}00}{3{\cdot}25}\ \frac{100}{0{\cdot}87}\ \frac{1}{Q\ [1-f\ (Q)]}=\frac{40{\cdot}3}{Q\ [1-f\ (Q)]} \qquad (9)$$

Based on this equation the curve r has been drawn in Fig. 1. As will be seen, this curve reaches its minimum point when $Q=300$, in which case $r=0{\cdot}31$.

For analytical treatment we may try to replace the curve obtained directly by a simple mathematical function.

A very simple and applicable expression is

$$f\ (Q)=1-e^{-BQ} \qquad (10)$$

In accordance with equations (7) and (8) this gives

$$Q_{opt}=\frac{1}{B} \qquad (11)$$

$$\text{and } r_{opt}=40{\cdot}3B.e \qquad (12)$$

$$\text{or generally, } r_{opt}=G\frac{T}{t}\ \frac{100}{s}\ B.e \qquad (13)$$

If $B=10^{-3}\times 2{\cdot}89$

then $Q_{opt}=346$

and $r_{opt}=0{\cdot}317$.

A closer approximation is given by the hyperbola

$$Q=A+\frac{B}{z}-Cz \qquad (14)$$

in which, for the sake of brevity

$$z=1-f\ (Q) \qquad (15)$$

Good values of the parameters are

$A=287$; $B=68$; $C=355$.

As will be seen, Q is expressed here as a function of $f\ (Q)$. Of course, $f\ (Q)$ may be solved as a function of Q in the form of an equation of the second degree, but this is certainly somewhat more laborious. One immediately gets

$$z_{opt}=\frac{A}{2C} \qquad (16)$$

and from this

$$Q_{opt}=\frac{A^2+4BC}{2A} \qquad (17)$$

The introduced values yield

$z_{opt}=0{\cdot}404$; $Q_{opt}=312$ and $r_{opt}=0{\cdot}320$.

Figs. 2 and 3 show the approximated curves and their r-curves. It is thus evident that with centrifuges there is quite a definite capacity at which the best purifying efficiency is obtained.

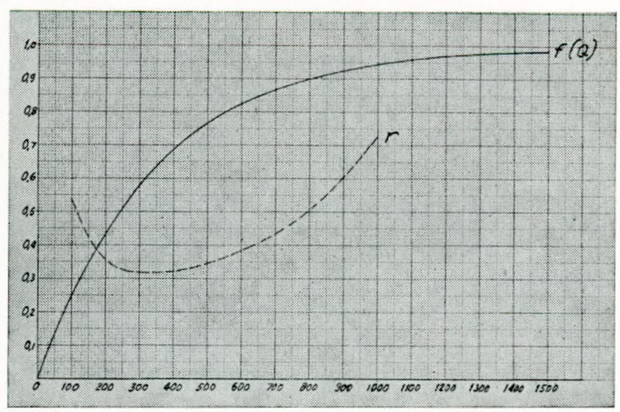


FIG. 2.

A certain control of the validity of the calculations just mentioned is obtained through the following observations.

When the series of tests in question was started the ship had been in regular service for a long time.

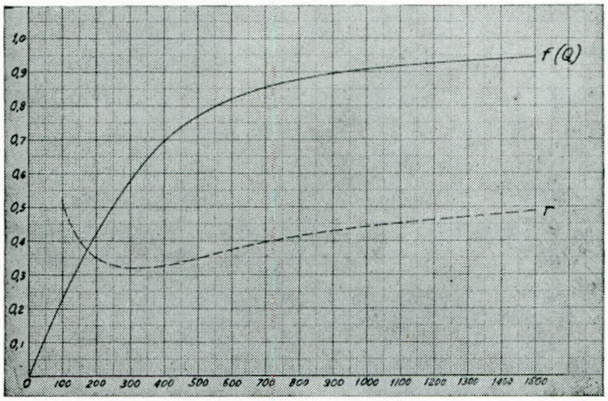


FIG. 3.

The centrifugal separators had been working at a capacity of about 600 litres per hour for so long a period that equilibrium with certainty had been practically attained. The impurity content in the sump oil was about 0·38, which is well in accordance with the curves.

Further, tests were made to measure the quantity of impurities retained in the centrifugal separator at different capacities. It is evident that, if in equation (3) r is considered constant and the maximum value of g is sought, the same conditions are obtained as for the minimum for r with constant g or G, *i.e.* equation (6). Thus, by carrying out short series of tests, during which r has not time to vary to any appreciable extent, and observing how the quantity of impurities extracted depends on the capacity, one may be able to control the value of the most suitable capacity. Such a series of observations with the quantities of impurities indicated in relative numbers is illustrated by Fig. 4. As will be seen, there is a well marked maximum at Q=about 275 which, considering the

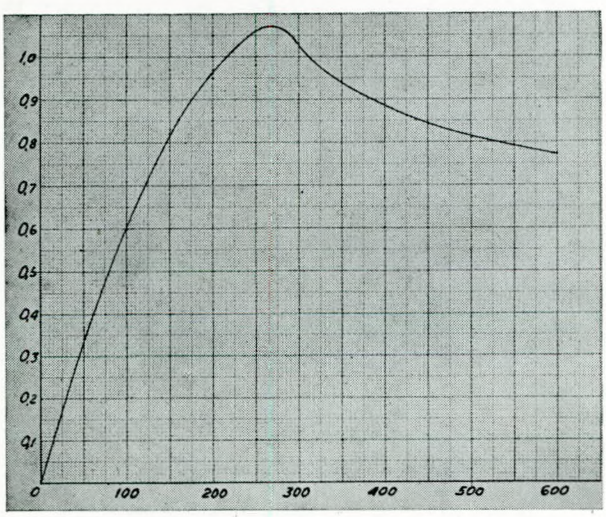


FIG. 4.

circumstances, will probably be regarded as a satisfactory agreement.

The researches mentioned above refer to a certain definite centrifuge of given type and size. One may then ask how the matter stands if a centrifuge of other type and/or size be used. As regards the *type*, there is not much to be added. The fundamental laws for the work of a centrifuge, such as have been laid down in the foregoing, do not apply to any special type of centrifuge but are of universal validity. They are thus applicable to centrifuges of any kind on the obvious condition that the centrifuge is not impaired by any positive defects, such as remixing of the separated products or the like. As regards the size, nothing can be said with certainty before we have exactly defined what we mean by the size of a centrifuge. By the expression that a centrifuge is n (n being an integral number or a fraction) times as large as another, is meant in the following deduction that the former at n times as high a capacity as that of the latter has the same purifying efficiency measured on the liquid being discharged from the centrifuge. This evidently means that n centrifuges of a certain size together are equivalent to one centrifuge of n times as large a size.

We thus start from a certain centrifuge, the purifying efficiency of which and the results subsequent thereupon are known. Assume its capacity to be Q_1. Another centrifuge working at a capacity of $Q_2=nQ_1$ has the same purifying efficiency.

To the former the following equation applies:

$$p=rf\ (Q_1) \quad \ldots\ldots\ldots\ldots \quad (1)$$

To the latter:

$$p=r\phi\ \ (Q_2)=r\phi(nQ_1) \quad \ldots\ldots\ldots\ldots \quad (18)$$

As the purifying efficiency is to be the same, *i.e.* $\frac{p}{r}$ is the same in both cases, evidently:—

$$f(Q_1)=\phi(nQ_1) \quad \ldots\ldots\ldots\ldots \quad (19)$$

Hence:—

$$f'(Q_1)=n\phi'(nQ_1) \text{ or } \phi'(nQ_1)=\frac{1}{n}\ f'(Q_1) \quad (20)$$

To the optimum capacity the following equation applies:

$$Q_1=\frac{1-f(Q_1)}{f'(Q_1)} \quad \ldots\ldots\ldots\ldots \quad (7)$$

and by analogy:—

$$Q_2=\frac{1-\phi(nQ_1)}{\phi'(nQ_1)} \quad \ldots\ldots\ldots\ldots \quad (21)$$

Introduction of the values $\phi(nQ_1)=f(Q_1)$ (19)

and $\phi'(nQ_1)=\frac{1}{n}f'(Q_1)$.............................. (20)

gives:—

$$Q_2=nQ_1 \quad \ldots\ldots\ldots\ldots \quad (22)$$

In an analogical way we obtain

$$r_2=\frac{1}{n}r_1 \quad \ldots\ldots\ldots\ldots \quad (23)$$

This consequently means that irrespective of the nature of the function $f(Q)$:—

The optimum capacity is proportional to the size of the centrifuge, and

The optimum degree of contamination of the sump oil is inversely proportional to the size of the centrifuge.

Consequently, this implies that if in a certain installation the centrifuge (or any other purifying apparatus) be replaced by an apparatus of n times its size then :—

The optimum capacity will be n times as great as previously, and :—

The optimum degree of contamination of the sump oil will be the nth part as great as previously.

These universally valid laws are of great importance both to the manufacturers and the users of centrifugal separators. It will not be necessary to carry out long and laborious series of tests with all the sizes of centrifuges on the market. As soon as the optimum capacity of one size of centrifuge operating under given conditions has been determined in one way or other, the optimum capacity of all the other sizes is also given. All that is required is that we should know the relative size ratio of the different centrifuges, which can be determined once for all in a purely mathematical way.

The investigation just referred to is obviously entirely unbiassed, being founded upon purely theoretical deductions, but precisely because of their theoretical character they will perhaps not carry conviction with practically minded persons, who are not accustomed to give credit to what are called theoretical speculations. It may therefore be of interest to show by a practical example that they are fully in accordance with common sense.

In the example, 0·570 kg. of impurities are introduced into the oil per hour. These were continuously removed by two centrifuges operating at an optimum capacity of 312 litres, the degree of contamination of the sump oil remaining at an equilibrium value of 0·32. The total capacity of the purifying plant thus amounted to 624 litres. Now, suppose that one of the centrifuges is stopped, while the other continues working at its unchanged optimum capacity. This machine alone is not, however, capable of keeping the degree of contamination as low as 0·32, seeing that when operating on such oil it only extracts 0·285 kg. per hour, whereas 0·570 kg. are introduced. The impurity content thus starts rising, the extracted quantity of impurities always being proportional to the impurity content of the sump oil. When this has been doubled, *i.e.* reached the value of 0·64, 0·570 kg. are extracted again, and a new state of optimum equilibrium has ensued. Consequently, *the size* of the purifying plant has been reduced to half (one machine instead of two, $n=\frac{1}{2}$). The optimum capacity Q_2 is now equal to 312 litres instead of previously $Q_1=624$; $Q_2=nQ_1$. The degree of contamination at optimum equilibrium of the sump oil, r_2, is now equal to 0·64 instead of previously

$$r_1=0{\cdot}32 : \; r_2=\frac{1}{n}\,r_1.$$

As will be seen, this is quite in accordance with the theory.

Previously, it has been pointed out that the changes in the impurity content of the sump oil take place quite slowly. As this fact is of a certain importance a more detailed elucidation of the matter is desirable.

If the purifying plant works at a certain capacity Q (optimum or not) and the weight of the whole of the oil in circulation is P kg., the impurity content of the sump oil after θ hours is :

$$x=r-(r-r_0)e^{\frac{-Qs}{P}\,\frac{t}{T}\,[1-f(Q)]\theta} \qquad (24)$$

where r is the impurity content at a state of equilibrim in accordance with equation (5) and r_0 the impurity content at the time$=$O.

It follows from this that the impurity content x approaches the state of equilibrium, r, asymptotically, and thus does not reach this state until after an infinitely long period. After a certain lapse of time the difference will, however, be imperceptible.

Suppose that the engine is started with entirely pure oil ($r_0=$O) and that the centrifuge is working at the optimum capacity. How long will it be until the state of equilibrium has been reached within 10 per cent., *i.e.* $x=0{\cdot}288$? The answer is 271·3 hours, or more than 11 days.

Suppose that the centrifuge has been working for a time at unsuitable capacity so that the impurity content in the sump oil has attained the value of 0·38. The capacity is now changed over to optimum capacity. How long time will it be before the impurity content has fallen to 0·33? Answer : 211·1 hours, or nearly 9 days.

Assume, finally, that both centrifuges have been working for a long time at optimum capacity, so that the state of equilibrium $r=0{\cdot}32$ has practically been reached. Now, one of the centrifuges is stopped for repairs while the other continues operating at unchanged capacity. How much does the impurity content rise during 24 hours? Answer : 0·38 per cent.

As will be seen, the variations take place slowly, which of course renders researches in this field difficult, but, on the other hand, leads to a not inconsiderable increase in the degree of safety.

What has been said above refers to the double circulation. One may ask whether a modification of this would not give a better result. Two possibilities then seem to be at hand.

The first is not to feed the centrifuge with the sump oil, but with the oil returning from the lubricating points. One would then gain the advantage that the centrifuge got to work with a somewhat more contaminated oil, whereby its capacity for removing impurities would increase.

The second is, so to speak, to take the middle course and, while maintaining the continuous lubrication, introduce intermittent purification. Assume that the sump is divided into three preferably equally large tanks, A, B, and C. On a certain occasion the tank A is switched on to the lubricating oil circulation, while, independently of this, oil is passing from C through the centrifuge to B. When C has been emptied B is filled with purified oil. The work is now reversed, so that B is put into circulation while the oil in A, contaminated during the previous period, is passed through the centrifuge and purified and conveyed to C. In this way the work proceeds with continual change of function between the three tanks.

A close investigation now shows that, strangely enough, the optimum capacity in both these cases is just the same as in the original case.

As regards the optimum degree of contamination, its value in the first special case now treated is :

$$r_{opt} = \frac{100\,G}{s}\left[\frac{T}{t}\,\frac{1}{Q[1-f(Q)]} - \frac{1}{S}\right] \ldots\ldots \quad (25)$$

where S is the oil circulation in litres per hour. In the ultimate limit case where S is infinitely great, the degree of contamination will be the same as in the original case. At reduced circulation the purification is ameliorated, although rather inconsiderably, as long as the circulation is maintained at practical values. The best purifying efficiency is reached when the circulation has its lowest value, *i.e.* is equal to the capacity of the centrifuge, in which case, however, the lubricating system has been turned into an arrangement for ordinary, direct purification of the waste oil.

In the case treated previously the degree of contamination would then fall off from 0·320 to 0·215, which seems fairly favourable. Unfortunately, no amelioration even approximately as great can be attained in reality, as the circulation must be kept far higher. Whether one should still use the arrangement in question with the small amelioration which it allows is a matter which in the first place depends on the design of the engine, *i.e.* if without complication or increase in cost it admits of concentrating the whole quantity of the return oil flow into one pipe from which the centrifuge could be fed. This is purely a question of engine technique, and is beyond the scope of this investigation.

As regards the other special case, if the optimum capacity can be maintained, at the end of the working period of a certain tank, when the oil has reached the highest degree of contamination, the impurity content amounts to exactly the same value as the impurity content at a state of equilibrium with the original arrangement. However, as the oil was of course purer at the beginning of the working period the *average* impurity content of the oil was lower than would have been the case with the original arrangement. The contamination of the oil during the working period proceeds linearly with time, and consequently the average impurity content is equal to the arithmetical mean between the degrees of contamination at the beginning and at the end of the working period. In the case treated this means that the average impurity content would be about 80 per cent. of the impurity content at a state of equilibrium. This consequently implies that with this modification a certain purification result may be obtained with a centrifuge of 80 per cent size, or, what is the same thing, that an equally good purifying result may be obtained as would otherwise be possible with a 25 per cent. larger centrifuge. However, the arrangement with three sump tanks including the necessary piping and fittings is likely to involve quite as high an expenditure as the possible saving in the costs for the purchase and operation of the centrifuge and, consequently, this modification is also unlikely to bring about any real advantages over the original system.

From what has been said it is thus evident that there is a certain capacity at which a centrifuge should work in order to give the best possible result with continuous lubrication.

One may then ask whether this optimum capacity is generally used in practice. The reply must regrettably be in the negative. Practically without exception far too high rates of capacity are used.

The cause is easy to realize. According to public opinon, the "size" of a centrifuge is merely represented by the capacity. This is quite justified for several fields of application. If a dairy is to procure a cream separator and different offers are compared, it is considered self-evident that a 5,000 litre separator is larger than a 3,000 litre machine, and the comparison of the prices quoted is made between machines having the same capacity. This procedure is, however, based on the presumption that all the machines have the same skimming efficiency. Should this not be the case the fact must be taken into consideration. If there are two machines of 5,000 litre capacity, of which the one is known to have a better skimming efficiency, this is considered in reality to be a larger and more valuable machine. Although the skimming efficiency of the different separator types is on the whole standardised and fairly well known, buyers often require a guarantee regarding the skimming efficiency, and if the seller can undertake to furnish a particularly good guarantee in this respect this is put to his credit. These comparatively settled conditions are rendered possible, firstly because milk, despite certain variations, is of a fairly consistent nature, and, secondly, by the existence of accurate and generally adopted methods for determining the fat content in the skim milk, *i.e.* the degree of separation.

The position regarding oil centrifuges is quite different. Here, there is neither any standardised liquid to be centrifuged nor any accepted method of analysis. Only in connection with the result obtained can the capacity be considered as a measure of the "size" of the centrifuge and there is no general method of measuring the result, hence the notion of capacity will hang suspended in the air and will not represent the efficiency of the centrifuge, even as an approximation. In reality, the conception of capacity has degenerated to the extent that the capacity, in the case of oil centrifuges, means little more than the greatest possible throughput quantity which the centrifuge can receive without overflowing. Obviously, this state of things is very unfavourable from the point of view of both the supplier and the user. The conscientious supplier wishing to render his clients good service is in an unfavourable position as compared with unscrupulous competitors offering high capacity without considering efficiency.

The client, on the other hand, is often misled, and gives the preference to the less efficient apparatus instead of to the more efficient type. But this is not the worst. The most serious consequence of the present state of disorder is that centrifuges as a rule will be operated at far too high rates of capacity. To the user it seems quite natural, as he has bought and paid for a machine with high capacity, that he should utilize it. Even though the supplier be well conversant with the state of things he must be cautious, in order not to raise suspicions. If one supplier says: "My machine is a 4,000 litre machine and consequently you should operate it at 4,000 litres capacity in order to get fair value for your money", and the other says: "My machine is a 4,000 litre but you should not exceed a capacity of 500 litres an hour with it if you wish to attain the best possible results", there is every likelihood that the business will be obtained by the former.

It is evident that a *really* "larger" machine, employed correctly, renders a better purifying efficiency than a smaller one. This has been thrashed out in detail in the preceding analysis. It is thus always advantageous to use the largest possible centrifuge. The mistake made by a great many people lies partly in confusing a "large" machine with an indicated high capacity, and partly in confusing the optimum capacity of a machine with the maximum capacity indicated by the seller.

The preceding investigation has been confined mainly to centrifuges and their use, although the basic formulæ are applicable to purifying apparatus of any kind. Only the general principles have been referred to, and no comparisons have been made either between filters and centrifuges, or between different types of the latter. However, it may be permissible to hint at some lines of thought for a comparison between the efficiency of a filter and that of a centrifuge.

The formula (5) holds good quite generally for filters as well as for centrifuges, but in order to be able to apply it we must, of course, know the function $f(Q)$. What the nature of this function is for a filter is not known, as already stated. However, as indicated by way of introduction, we may certainly take it that the function rises with an increase of Q_1, but how this rise takes place is unknown.

In order to be quite sure that we are not treating the filter unfairly the assumption most favourable to the filter is made, *viz.* $f(Q)=O$, *i.e.* the filter purifies the liquid complete, irrespective of the capacity.

The formula will then be quite simply:

$$r = G\,\frac{T}{t}\;\frac{100}{s}\cdot\frac{I}{Q} \qquad (26)$$

or, with the same numerical values as previously—

$$r=\frac{40{\cdot}3}{Q}\text{; If } r=0{\cdot}32 \text{ we obtain } Q=126 \text{ litres.}$$

Thus, if the filter can work permanently at a capacity of 126 litres per hour, it is equivalent to the centrifuge in regard to purifying efficiency; at higher capacity it is superior but at lower capacity inferior to the latter.

Therefore, if we wish to compare a filter and a centrifuge of the type and size indicated in regard to the purifying efficiency with a Diesel engine of the kind mentioned, we should *not* look at an oil sample in a test-tube in order to see from which apparatus the purest oil has been discharged, but ascertain whether the filter can work at a higher capacity than 126 litres per hour.

There is no reason to make a pronouncement as to the probable result of such a comparison, so much the less as filters, just as centrifuges, may be built in highly varying sizes; suffice it to say that the comparison should be carried out in the way mentioned above.

The nature of the problem treated here is, in fact, rather simple if we only take care that it is clearly understood. Hitherto the problem has, however, as a rule been misunderstood, with the result that on the one hand mistakes have been made when choosing purifying plants, and on the other that plants in themselves good and suitable have been used in an inefficient way.

If this investigation contributes towards the introduction of an improved state of affairs it will serve its purpose.

ELECTION OF MEMBERS.

List of those elected at Council Meeting held on Tuesday, January 9th 1940.

Members.

Cecil Miln Dalgarno, Rivers S.N. Co. Ltd., 2, Fairlie Place, Calcutta, India.

Edward Ellis, 55, Newcombe Road, Polygon, Southampton.

George Frederick Forsdike, 85, Faraday Avenue, Sidcup, Kent.

Vincent Thomas Gwilliam, Pathend, Rowtan Grange Road, Chapel-en-le-Frith, via Stockport.

George Lionel Lane, 946, Rochester Way, Sidcup, Kent.

Alexander Lucas Taylor, "Carrick", Bassett Dale, Southampton.

Transfer from Associate to Member.

Alexander Fraser Campbell, Greenoak, Wyndham Road, Ardbeg, Rothesay.

Geoffrey Newhouse, 8, Albion Terrace, Kowloon Docks, Hong Kong.

Guy Taite Shoosmith, 2, Hollington Court, Chislehurst, Kent.

ADDITIONS TO THE LIBRARY.

Purchased.

Kempe's Engineer's Year-Book, 1940. Morgan Brothers (Publishers) Ltd., 31s. 6d. net.

Presented by Mr. Thomas Dunlop (Member).

Mechanical World Year Book, 1940. Emmott & Co., Ltd.

Fowler's Mechanical Engineer's Pocket Book, 1940. Scientific Publishing Co., 3s. net.

Presented by the Publishers.

Bulletin de L'Association Technique Maritime et Aéronautique, No. 43, 1939.

The following British Standard Specifications:—

No. 77-1939. Voltages for Transmission and Distribution A.C. Systems.

No. 329-1939. Round Strand Steel Wire Ropes for Lifts and Hoists.

No. 358-1939. Measurement of Voltage with Sphere-Caps.

No. 861-1939. Air-Break Switches (including Isolating Switches, Totally-enclosed and Flameproof Types).

No. 862-1939. Air-Break Circuit-Breakers (including Totally-enclosed and Flameproof Types).

Vol. XLIX of the Journal of the Junior Institution of Engineers, containing the following papers :—

"Canal Locks and other Lifting Devices in Inland Navigation", by McGarey.

"The Warming of Buildings by Electricity", by Mayson.

"Engineering Measurements", by Bowen.

"The Measurement of the Leakage of Compressed Air in Collieries", by Dick.

"Some Applications of Bitumen with Special Reference to Emulsions", by Morris.

"Some Types of Surface Heat Exchangers", by Sutcliffe.

"Roads and Traffic", by Bressey.

"Some Safety Provisions of the Factories Act, 1937," by Taylor.

"Status, Qualifications and Remuneration of the Engineer", by Chatley.

"Engineering in Egypt", by Ablett.

"The Engineer in our Time", by Lawton.

"Applications of the Thermionic Valve for purposes other than Radio", by Riddle.

"Industrial Electric Heat-Treatment", by Ward.

"Woad Mills", by Wailes.

"The Mysteries of Clay Structure", by Chatley.

"The Less Technical Problems of British Airship Development", by Cave-Browne-Cave.

"The Use of Town's Gas in the Engineering and Allied Industries", by Clarke.

"Voith-Schneider System for Marine Propulsion"', by Goldsworthy.

"The Isolation of Vibration and Noise from Engineering Plant and Equipment", by Green.

"Alloy Irons in Engineering and Commercial Usage", by Lowe.

"Modern Concrete Practice", by Cross.

"The Abrasive Wheel and Modern Grinding", by Gutteridge.

"The Machine and the Man", by Ballantyne.

"Engineering on the Stage", by Williams.

"Centrifugal Pump Characteristics", by Lymer.

"The Development of Pottery Firing", by Walter.

"Multi-Tool Lathes", by Lunn.

Principles of Electric Arc Welding. By I. H. Child, B.Sc. The Draughtsman Publishing Co., Ltd., 35 pp., illus., 2s. net.

By a knowledge of the principles underlying the process and a conception of the process actually in operation, the draughtsman can express his designs in terms of the more standard joints and preparations with which the operator will be quite familiar, and can avoid placing before the operator structures which, by their difficulty of handling, poor accessibility, or unbalanced thermal capacities, would in themselves militate against satisfactory welding. The detailed applications of fabrication by welding vary widely between the different branches of engineering, but they are all based on the same fundamental principles, and it rests with the designer successfully to apply these principles to his own particular application. It is with these elementary bases that this pamphlet is chiefly concerned.

Annual Review of Shipping, Shipbuilding and Marine Engineering. "The Journal of Commerce", Liverpool, 2, 238 pp., copiously illus., 2s. net.

The present issue of this annual, the object of which is to review the activities of shipping, shipbuilding and marine engineering during 1939, has again achieved that high standard of contents and production which is expected by those who have been subscribers in recent years. The following summary of the contents will indicate that it contains many articles of special interest to the marine engineer, viz. :—"Fair and Reasonable Rates for Shipping"; "Shipping Now and After the War"; "Chartering Markets in Peace and War"; "Oldest Shipping Line to New Zealand"; "Developments in War Risks Insurance"; "Pioneer Service between Great Britain and India"; "Germany's Tactics in the War at Sea"; "British Port Developments"; "War Effort of the Shipbuilding Industry"; "Public Rooms of Outstanding Ships"; "Electric Arc Welding in Shipbuilding"; "Metallurgical Progress in Shipbuilding"; "Marine Steam Engineering", by J. Hamilton Gibson; "Marine Electricity"; "Significant Motorships of 1939", by A. C. Hardy; "Training of Officers for the Merchant Navy"; and "The War at Sea: A Chronology of Events". Priced at only 2s. its 238 large pages represent striking value.

The Motor Ship Reference Book, 1940. Temple Press, Ltd., 324 pp., 107 illus., 5s. net.

Up-to-date and authoritative information regarding the construction and machinery equipment of motor ships is available in "The Motor Ship Reference Book for 1940", which has just been published. The book has been compiled by the editorial staff of "The Motor Ship" and comprises about 300 pages of valuable facts and figures, together with reproductions from drawings and photographs showing large main Diesel propelling engines and auxiliary generating sets. The details cover all which are in general use in the Mercantile Marine, forming an interesting as well as a fully-illustrated summary of marine Diesel engine practice at the present time.

Every motor ship of 2,000 tons gross or over is listed and the tables have been revised up to the end of 1939. The particulars given include the date of construction, the names of the owners, dimensions and tonnage of each vessel, power of machinery, names of the engine builders and type of engine installed. It has been the object of the compilers to produce a work of reference which shall be of every-day service to the shipowner and his staff as well as to the engineering personnel ashore and afloat. Notwithstanding the quantity of statistics and illustrations in the volume, it remains handy in size. The new edition is the sixteenth that has been published.

American Diesel Engines. By L. H. Morrison. McGraw-Hill Publishing Co., Ltd., 489 pp., 306 illus., 33s. net.

The author has given a well-arranged book, written in a style which is clear and intelligible to the ordinary reader, and which displays a wide experience and profound knowledge of his subject.

The commencement is a brief history of the advance of the Diesel in America, starting from the inventions and experiments of the pioneers, and it is of interest to note that full credit is accorded to Akroyd Stuart for his share in the pioneer work in Europe. Following on the historical chapters is a very complete description of current American practice with respect to air and mechanical injection, heavy-duty Diesels, high-speed truck type Diesels, etc. The differences between American and European practice are clearly brought out, one striking feature being the popularity in America of the common rail injection system. In chapters X to XIII inclusive, which deal with design and materials of mechanical details, the author's practical knowledge is most evident, and a great deal of useful information is given. Other interesting and informative chapters are those dealing with cylinder pressures, fuel systems, etc. The chapter on torsional vibration is all too brief. This is an important subject and some examples and calculations illustrating these problems as they apply to the Diesel engine would have been useful. The chapters on engine installation, lubricating systems, and cooling systems, are rather disappointing. While the author has given a few ideas on installation of land engines, he has not touched on marine installations. It is in marine installations that the real problems of vibration, etc., are encountered, and also lubrication and cooling systems have their special problems at sea.

It is evident that great care has been taken over the compiling of the book and there is no doubt that it will be a valuable addition to any engineer's library.

A.C. Motors and Control Gear. By C. H. Claude Cooke. Crosby Lockwood & Son, Ltd., 88 pp., 20 line diagrams, 3s. 6d. net.

This small volume contains some quite useful practical information dealing with motor drives, the erection of plant, the maintenance of motors and switchgear, etc.

When the author comes to deal with the theoretical aspects of the matter, however, one must take exception to some of his statements which are sometimes misleading and in some cases definitely wrong. For example, he is apparently under the impression that if an inductance is added to a previously non-inductive circuit the current value in amperes will remain unchanged, the only change being the phase position of the current relative to the supply voltage. He also appears to be of the opinion that it will make no difference whether the inductance is added in series or in parallel to the original circuit. In dealing with pole-changing motors the statement is made that "the coil ends of each pole are brought out to a switch having two positions, grouping the windings so that alternate south and north poles are produced in one position of the switch, and all poles have like polarity in the other position. With alternate poles the motor will run normally at the designed speed, but when all poles have like polarity the speed will be reduced to half the former speed". The author probably does not mean precisely what he says when he uses the expression "like polarity" but, in any case, to procure half speed, twice as many poles would be necessary. On page 22 are two statements dealing with the speed of rotation of the rotating field which are obviously mutually contradictory.

It is a pity that what would otherwise be a quite useful little volume is marred by these and other examples of loose and unprecise writing. In view of the fact that there is already an extensive literature of a.c. motors, it would probably have been better if the author had confined himself to practical matters.

JUNIOR SECTION.

Electricity Applied to Marine Engineering (Section 13).

By W. LAWS, M.Sc., A.M.I.E.E.

Alternating Current Theory.

Consider a straight conductor of length l cm. lying in and at right angles to a magnetic field of intensity B gauss (1 gauss is 1 line of magnetic flux per square centimetre), and being rotated at a uniform peripheral speed of v cm. per sec. about an axis o parallel to its own length. The end view of the arrangement is indicated in Fig. 114.

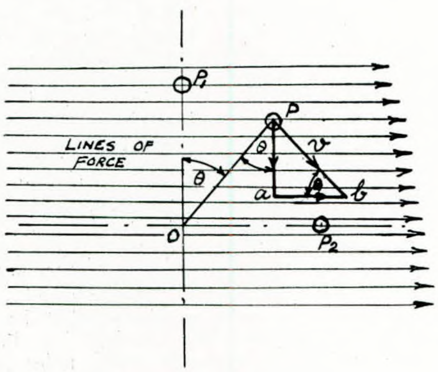


FIG. 114.

When lying in the position P_1, the conductor is momentarily moving in the same direction as the direction of the field, so that there is no cutting of lines of force and consequently no voltage induced in the conductor. When lying in position P_2, the conductor is momentarily cutting the field at right angles, that is, at the maximum possible rate, and will consequently have the maximum possible voltage induced in it. If the conductor is l cm. long, and moving at v cm. per sec., then in one second it will sweep out an area of lv sq. cm., and as each of these square centimetres contains B lines of force, the conductor will have cut through Blv lines.

Now voltage induced per conductor

$$=\frac{\text{lines cut per second}}{10^8}$$

$$=\frac{Blv}{10^8}\text{ volts}=E_{\max}.$$

Consider now the conductor when it is lying in some intermediate position P, such that it has turned through some angle θ from its no-voltage position. Its peripheral velocity v will be momentarily directed at right angles to the radius joining the conductor to the axis O about which it is rotating. Let us represent v in magnitude and direction by a vector and let the vector be resolved into two components, one Pa at right angles to the field, and the other ab in the direction of the field. The latter component is of no use whatever as regards the production of volts. The component $Pa=v\sin\theta$ is, however, useful as regards voltage production, and the momentary or instantaneous voltage at position P is given by $\frac{Blv\sin\theta}{10^8}$ volts

$$=E_{\max}.\sin\theta.$$

The voltage will therefore follow a sine law of variation with respect to time, and if the voltage be plotted against a time base, the well-known sine (or harmonic) curve will be produced. It should be noticed how the mathematics correctly interprets the physical happenings.

When	$\theta=0$	$\sin\theta=0$	E.M.F. $=0$
,,	$\theta=90°$	$\sin\theta=1$	E.M.F. $=E_{\max}$.
,,	$\theta=180°$	$\sin\theta=0$	E.M.F. $=0$
,,	$\theta=270°$	$\sin\theta=-1$	E.M.F. $=-E_{\max}$.

but the E.M.F. at 270° is reversed in direction from what it was at 90°. That is the significance of the negative sign.

Instead of thinking in terms of peripheral velocity let us suppose that the angular velocity of the conductor is ω radians per second.

Then $\theta=\omega t$

where t is the time in seconds which has elapsed since the conductor was in a no-voltage position.

The number of complete cycles per second is the frequency f. One complete cycle is made when the conductor moves through 2π radians (*i.e.* one complete circle).

If the frequency were 1, then $\omega=2\pi$
,, ,, ,, 2, ,, $\omega=2\pi\times 2$
,, ,, ,, 3, ,, $\omega=2\pi\times 3$
and ,, ,, ,, f, ,, $\omega=2\pi f$

so that we can now write the general expression,

instantaneous volts $e=E_{\max}.\sin(2\pi ft)$.

Example.

An alternating voltage has a maximum value of 400, and a frequency of 50 cycles per second. What is its instantaneous value $\frac{1}{600}$ second after having zero value?

$$e=E_{\max}.\sin(2\pi ft)$$
$$=400\sin(2\pi 50\tfrac{1}{600})$$
$$=400\sin(2\times 180°\times 50\times\tfrac{1}{600})$$
$$=400\sin 30°$$
$$=400\times 0{\cdot}5$$
$$=200\text{ volts.}$$

There is a point here which frequently causes beginners some confusion. It will have been noticed that in the above example π was interpreted as 180°. It would have been equally correct to express it as 3·1416 radians; only, if this is done, then when consulting one's trigonometrical tables to find the sine of the angle, one must consult the radian column, thus

$$e=400\sin(2\pi 50\tfrac{1}{600})$$
$$=400\sin(2\times 3{\cdot}1416\times 50\times\tfrac{1}{600})$$
$$=400\sin(0{\cdot}5236\text{ radians})$$
$$=400\times 0{\cdot}5$$
$$=200.$$

Do not forget that just as it is possible to measure a length in either feet or fathoms (or other units) so an angle may be measured in either degrees or radians, only having started with one unit, one must stick to it. It is just the same kind of thing as though the price of a stair carpet were given in a catalogue as so much per foot, and also as so much per yard, and we measured the length we required in yards, and then multiplied the number of yards by the price per foot. Having started with a unit, we must stick to it.

We will now try another type of example over which beginners almost invariably go wrong unless their elementary mathematics is sound.

Example.

A sinusoidal alternating current has a maximum value of 20 amperes and a frequency of 25 cycles per second. After what time after having zero value and increasing in the positive direction will it first have an instantaneous value of 17·32 amperes?

[The word sinusoidal, or sometimes sinoidal, means varying according to a sine law. Some marine engineering friends of the author's always refer to it as "suicidal"—perhaps because this stuff makes them feel that way.]

$$i = \mathrm{I}_{\max}.\sin(2\pi ft)$$
$$\therefore\ 17{\cdot}32 = 20\sin(2\pi ft).$$

You cannot solve for t yet, but this is what many students want to do. $(2\pi ft)$ is an angle, and we must first find out what it is.

$$\text{From above } \sin(2\pi ft) = \frac{17{\cdot}32}{20}$$
$$= 0{\cdot}866$$

therefore, by referring to tables we find that $(2\pi ft)$ is either 60°, or which is the same thing 1·0472 radians.

$$\therefore\ 2\pi ft = 60$$
$$\therefore\ t = \frac{60}{2 \times 180 \times 25}$$
$$= \tfrac{1}{150}\text{second.}$$

Root-Mean-Square Values.

An alternating current or voltage is changing its numerical value continuously as it passes through its cycle of change. What numerical value then can we give to it for purposes of measurement, and comparison? To get the answer to this we ask ourselves the question "Why do we use electricity at all?" and the answer is "To do a job of work", not forgetting that work can be expressed as heat. So we find the direct steady current which would do the same amount of work in the same length of time as the alternating current, and give its numerical value to the alternating current. Then we know that an alternating current of 10 amperes, and a direct current of 10 amperes will have the same energy effect.

Consider Fig. 115, which represents the graph of an electric current which is varying in value from instant to instant with respect to time. Suppose that the whole time under consideration is T seconds and that we consider this as split up into such a great number of parts n, that the mid-ordinates of these parts represent

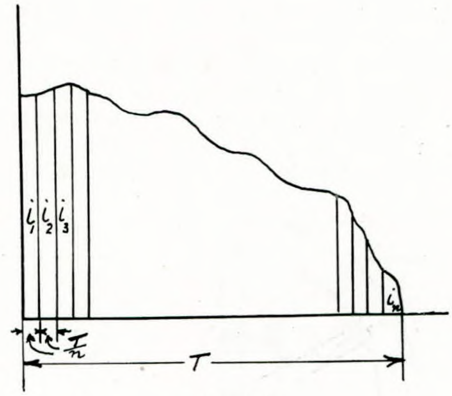


FIG. 115.

the value of the current during the extremely short interval of time $\frac{T}{n}$. Suppose this current to be flowing through a resistance R ohms.

Then the total work done during T seconds is

$$i_1{}^2 R\frac{T}{n} + i_2{}^2 R\frac{T}{n} + i_3{}^2 R\frac{T}{n} + \ \ldots\ i_n{}^2 R\frac{T}{n}$$
$$= RT\left(\frac{i_1{}^2 + i_2{}^2 + i_3{}^2 + \ \ldots\ i_n{}^2}{n}\right) \text{ joules.}$$

Let I be the value of the direct current which would do the same amount of work in the same time. Then the work done by the direct current is

$$I^2RT \text{ joules}$$

but, by our agreed definition, these two works are equal, *i.e.:*

$$I^2RT = RT\left(\frac{i_1{}^2 + i_2{}^2 + i_3{}^2 + \ \ldots\ i_n{}^2}{n}\right)$$
$$\therefore\ I = \sqrt{\frac{i_1{}^2 + i_2{}^2 + i_3{}^2 + \ \ldots\ i_n{}^2}{n}}$$

= the square root of the mean of the squares, or root-mean-square, or R.M.S. value.

The R.M.S. value of a sine wave can be proved to be $\frac{\text{maximum value}}{\sqrt{2}} = 0{\cdot}707$ max.

The average value of an alternating current is the average value of *one half wave.* The average value taken over a complete cycle would be zero, because all of the positive values would be cancelled out by the corresponding negative values.

The average value of a sinusoidal wave can be proved to be $\frac{\text{maximum value}}{\pi/2} = 0{\cdot}637$ max.

The ratio $\frac{\text{R.M.S. value}}{\text{average value}}$ is known as the *Form factor,* and for a sine wave is $\frac{0{\cdot}707}{0{\cdot}637} = 1{\cdot}11$

Whenever an alternating current, or alternating voltage is spoken of as so many amperes, or so many volts, it is always inferred that the R.M.S. value is meant, unless it is specifically stated to the contrary. A.C. instruments record R.M.S. values.

Use of Vector Diagrams in Alternating Current Work —Phase Difference.

Consider Fig. 116, which represents two coils A and B, wound on the same core and rotated at constant speed in a uniform magnetic field, but displaced from each other in space by an angle ϕ. Both coils will have alternating voltages induced in them but that in B will

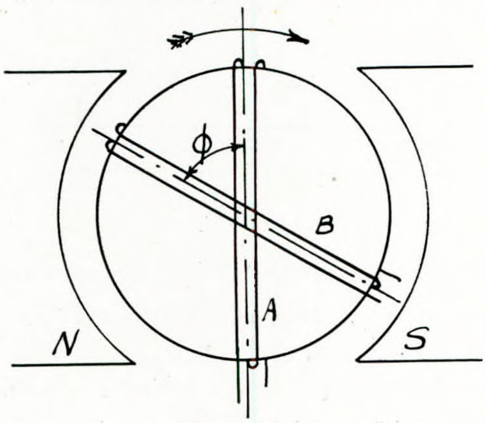


FIG. 116.

lag in time behind that in A. That is A will have moved forward through the angle ϕ from its no-volts position when B arrives at its no-volts position. If their graphs are plotted they will be as shown in Fig. 117, curves A and B.

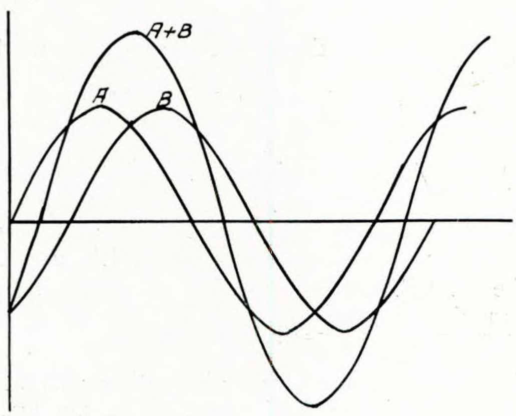


FIG. 117.

Suppose now the end of A is joined to the beginning of B so that they form one complete coil having two parts of itself separated by angle ϕ. What voltage is the new coil going to produce? We can obtain the joint voltage at successive instants in time by adding together the instantaneous voltage in A to the corresponding instantaneous voltage in B, taking into account their signs, positive or negative at the instants considered. We thus obtain a new sine wave $A+B$, and could measure its maximum value from the diagram and then multiply this by 0·707 to obtain the R.M.S. value. This could be extremely laborious, and is, in fact, unnecessary.

Consider Fig. 118. Let OA be a vector drawn to

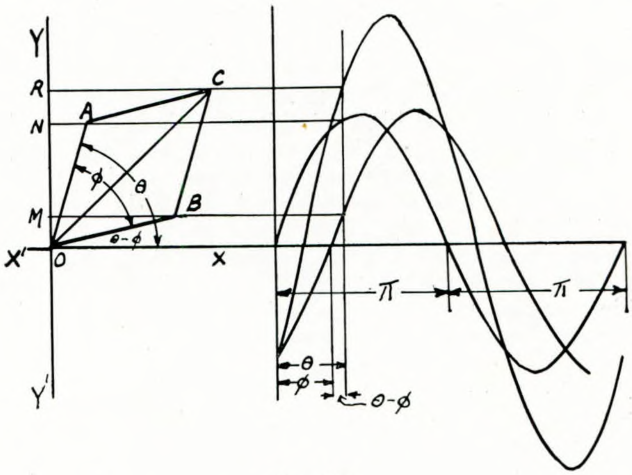


FIG. 118.

scale to represent the maximum value of the voltage induced in coil A, and suppose OA to be rotating about the centre O in a counter-clockwise direction at the same speed in r.p.s. as the armature core carrying coils A and B are rotating. From A drop a perpendicular AN on to the Y axis

$$\text{then } \frac{ON}{OA} = \sin\theta$$

$$\therefore\ ON = OA \sin\theta$$

$$= \text{max. volts of } A \times \sin\theta.$$

ON therefore represents to the scale we are using the instantaneous value of the voltage in coil A for the particular instant we are considering. Suppose then that we project NA through A until it cuts the ordinate through the angle θ, on the angle scale to the right of the diagram we would obtain one point on the sine curve. The complete sine curve for coil A could be derived by carrying out this construction for a series of values of the angle θ. If we draw another vector OB to represent the maximum value of the voltage in coil B, to the scale we have already selected, and draw the vector at an angle ϕ behind OA, we can by the same construction draw the sine wave of induced volts for coil B on the right hand side of the diagram, and as we take the successive values of θ for vector OA we always draw vector OB at an angle ϕ behind OA, the sine wave for B will occupy its correct position relative to the sine wave for A, *i.e.* displaced by an angle ϕ from it. The two waves are said to be *out-of-phase* with each other, ϕ being the angle of *phase displacement.* Now complete the parallelogram $AOCB$ and from C drop a perpendicular CR on to the Y axis. It can be proved geometrically that $OM = NR$, therefore $OR = NR + ON = OM + ON$, that is. OR is the sum of the instantaneous voltages in A and B at that instant. But OR is to scale the instantaneous value of the voltage which would be produced by a sinusoidal wave whose maximum value is represented to scale by OC for that same instant. In short, if we wish to find the maximum value of the wave which would be obtained if two coils such as A and B, displaced by an angle ϕ, were joined together, all we have to do is to draw two vectors to scale at the correct phase displacement and combine them by the ordinary vector laws, such as we use for forces, velocities, accelerations, etc.

Again, the R.M.S value of a sine wave is a definite fraction of its maximum value, to wit, 0·707 max. So if we multiply every vector of the vector diagram by 0·707, we will obtain another diagram of precisely the same shape but smaller which will be true for R.M.S. values, and in which each component vector will have its correct phase position relative to the others.

Example.

Three sinoidal alternating currents are fed into a common conductor. They are of value 10 amperes, 20 amperes and 17·32 amperes respectively. The 17·32 ampere current lags behind the 10 ampere current by 90° and leads the 20 ampere current by 60°. Find the maximum value of the resultant current, and its phase position relative to the 17·32 ampere current.

The vector diagram is shown in Fig. 119. Combining the 10 amperes with the 17·32 amperes a resultant

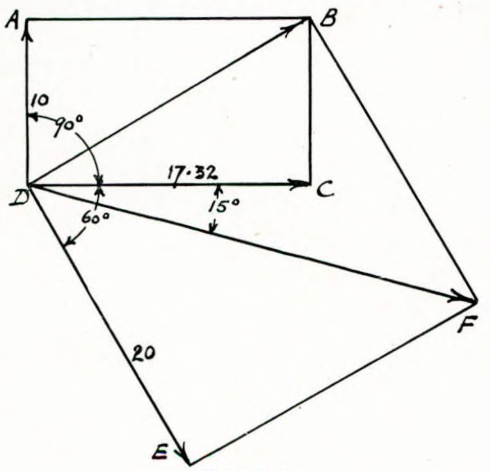


Fig. 119.

DB is obtained value 20 amperes and lagging behind the 10 ampere current by 60°. Combining this resultant *DB* with the other 20 ampere current *DE,* we obtain a resultant *DF* value 28·28 amperes which lags 15° behind the 17·32 ampere current. This 28·28 amperes is the R.M.S. value.

$$\text{Since R.M.S.} = \frac{\text{maximum}}{\sqrt{2}}$$

$$\begin{aligned}\text{maximum} &= \sqrt{2} \times \text{R.M.S.}\\ &= 1{\cdot}414 \times 28{\cdot}28\\ &= 40 \text{ amperes.}\end{aligned}$$

Students are sometimes inclined, when doing a problem of this kind, to combine *AD* and *DC,* obtaining a resultant, then to combine *DC* and *DE,* obtaining a resultant, and then to combine the two resultants thus obtained. A moment's thought will show that this is just the same as if, wishing to add together 3, 4, and 5, we first added together 3 and 4, then added together 4 and 5, and then added together the two sums thus obtained, *viz.* 7 and 9, and took the answer as the summation of 3, 4, and 5.

The diagram could also be drawn as a vector polygon as in Fig. 120.

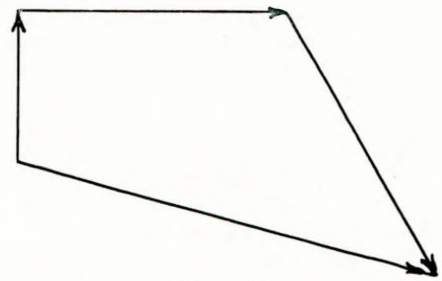


Fig. 120.

Inductance.

This is a property of an electric circuit which must be taken into account whenever the current in the circuit is changing its value. With an alternating current the current is changing its value continuously so that the effect of inductance is of considerable importance. To understand its action we must call up one of our mental pictures. We know that a piece of copper wire having no current passing through it has no magnetic field associated with it. We also know that when a current passes through the wire a magnetic field is associated with it which takes the form of concentric circles of lines of force having their common centre apparently at the centre of the wire, and that the field is denser or stronger in the immediate vicinity of the wire and gradually gets weaker as we move away from the wire. We also know that the stronger the current the further away can this field be detected.

We can therefore form a mental picture of the lines starting from the centre of the wire and radiating outwards into space as the current grows stronger, not unlike the ripples radiating outwards when a pebble is dropped into a still pool. As the current decreases we can think of the lines withdrawing back into the centre of the wire and then vanishing altogether.

Consider Fig. 121. This represents the end of a conductor with a current flowing in it away from the observer. The thin lines represent the magnetic lines of

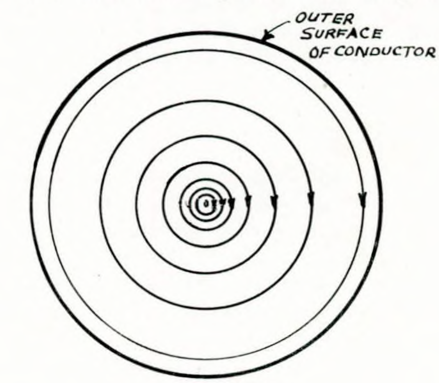


Fig. 121.

force. Suppose the current is increasing. Then these lines of force are moving through the material of the conductor outwards. So if we consider the lines as stationary, we can think of the conductor as relatively shrinking inwards through the lines. Electrically the effect is the same. Now whenever there is relative movement between a conductor and magnetic lines of force, a potential difference or voltage is induced in the conductor, and we obtain the mutual directions of field, motion, and induced voltage from Fleming's Right Hand Rule (thumb—motion of conductor; first finger—field; second finger—p.d. or current). Applying this rule to this case we find that a voltage will be induced which is opposed to the direction in which the current is increased. That is, it is resisting the increase of current. If, on the other hand, the current were decreasing, so that the lines of force were moving back inwards, so that relatively the conductor is moving outwards, the induced voltage is in the same direction as that in which the current is flowing, that is, it is endeavouring to maintain the flow of current. This property of a voltage being induced in a circuit when the current is changing in it is called inductance, and the voltage is said to be self-induced. The self-induced voltage is always in such a direction as to oppose the change producing it. The law governing this property was first enunciated by Lenz. It can be stated in a variety of ways which all actually come to the same thing. One way of stating it is "The direction of the induced voltage is always such as to tend to establish a current which will oppose the change in the flux".

This is closely analogous to the property of mass or inertia in mechanics. If an attempt is made to change the state of motion of a body either from rest to motion, from motion to rest, or from motion at one velocity to motion at some other velocity either greater or less, the inertia of the body will always oppose the change of motion.

Now this property of inductance will vary in different circuits. Some circuits have a much stronger magnetic field associated with them than others. For example, a hundred yards of wire stretched out straight will have a certain magnetic field associated with it when one ampere is passed through it. If this same hundred yards be wound into a close compact coil round a soft iron core, it will have a much greater magnetic flux associated with it when one ampere is passed through it.

We must therefore have a unit of measurement for comparing the inductances of various circuits. The practical unit is the henry.

The Henry.

A circuit has an inductance of one henry when a current change in the circuit at the rate of one ampere per second causes a potential difference of one volt to be self-induced in the circuit. Suppose a coil having 100 turns carries 5 amperes and has 200,000 magnetic lines associated with it. Suppose the coil to be wrapped round a non-magnetic core of wood or brass so that the number of magnetic lines is directly proportional to the current. If then, the current is reduced to 4 amperes the flux will be reduced to $\frac{4}{5} \times 200{,}000$, i.e., to 160,000 lines. The change in flux is therefore 40,000 lines. If the change took place in one second, then the rate of current change would be one ampere per second. Each of these 40,000 lines have cut the 100 turns of the coil, so that the induced voltage is

$$\frac{40{,}000 \times 100}{10^8}$$
$$= 0{\cdot}04 \text{ volts.}$$

But if a voltage of one volt is induced for a change of one ampere per second when the inductance is one henry, a voltage of 0·04 volts is induced for a change of one ampere per second when the inductance is 0·04 henry. The inductance of this circuit is therefore 0·04 henry. The inductance therefore equals

$$40{,}000 \times \frac{100}{10^8}$$
$$= \frac{200{,}000}{5} \times \frac{100}{10^8} \text{ henry}$$
$$= \frac{\text{flux linkages per ampere}}{10^8} \text{ henry.}$$

The number of flux linkages is obtained by multiplying the number of turns by the number of magnetic lines linked with them. The expression for the induced voltage is :—

induced voltage = inductance in henry × rate of change of current in amperes per second

$$\text{or } e = L\frac{I}{t}$$

In mechanics we have

$$f = ma$$
$$= m\frac{v}{t}$$

i.e., the force is equal to the mass multiplied by the rate of change of velocity. It is an interesting fact that if one writes down any equation in mechanics involving force, mass, velocity and time, one can write down a perfectly true equation in electricity by substituting inductance for mass, and current for velocity; for example

$$\text{kinetic energy} = \tfrac{1}{2}mv^2$$
$$\text{energy stored in magnetic field} = \tfrac{1}{2}LI^2$$

This energy appears in the form of a vicious spark when a highly inductive circuit is suddenly opened.

If a body moves at constant velocity, the only force required is that required to overcome frictional resistance. When it is accelerating an additional force must be supplied to accelerate it over and above what is required for frictional resistance. In the same way, when a constant electric current is flowing, the only voltage required is that to overcome the circuit's frictional resistance, or, in short, its ohmic resistance. If the current is to be increased, an extra voltage is required over and above that required for frictional resistance in order to accelerate the current—actually to counterbalance the back voltage of self-induction, which is induced while the current is increasing.

When we press down the knob of an electric light switch, the light appears to come on instantaneously. As a matter of fact it doesn't. The electricity in the wires has to be accelerated and that takes time, such a short time that it is not detectable by the human eye. The final value of the current is what we might term its Ohm's Law Value obtained by dividing the voltage applied to the circuit by the resistance of the circuit.

Let us consider what happens when a man rides a bicycle away from a kerb. Let us assume that the road is perfectly level and of a uniform smoothness (or roughness), and that he exerts a perfectly steady pressure on the pedals. Just before starting he and the bicycle are stationary. Under the action of his propelling force he starts to accelerate. But he does not go on accelerating for ever. Eventually he settles down to a steady speed, say 15 miles per hour. Why and when does he do this? As he accelerates the resistance of the air opposing him gets greater as some power of his speed. Eventually the effective force of the self-produced wind against him plus the frictional resistances of the wheels on the ground and the bearings of the bicycle are exactly equal to the forward propelling force which he is exerting. Then, and only then, does he settle down to a steady speed. If he encountered a patch of road which was rougher so that the frictional resistances were greater, and still continued pushing with the same pressure, the retarding force would be greater than the propelling force and he would slow down a little until once again the retarding force was exactly balanced by the propelling force. Suppose now that we apply an electrical pressure to the electricity resting motionless in an electrical circuit. We start to move the electricity, the rate of movement being the current I. At first this is very small, so that to push this small current against what we might call the frictional resistance of the circuit requires only a small force or voltage IR where R is the resistance in ohms. There will therefore be a surplus of volts available to accelerate the electricity. Just as with the man on the bicycle, we could state a force equation as follows :—

propelling force = force to overcome frictional resistance plus force to accelerate the combined mass of man and bicycle,

so also we can write an electrical equation :—

applied volts = volts to overcome frictional resistance (IR) plus volts to accelerate the electricity (*i.e.* inductance × rate of change of current)

and eventually, for a constant applied voltage we arrive at the stage when all of the applied voltage is entirely used up in overcoming the frictional resistance so that there is none left to accelerate the electricity, that is, increase the current, and the current value remains stationary at

$$I = \frac{E}{R} \quad \text{(Ohm's Law).}$$

Inductance in Alternating Current Circuits.

Suppose that we have a circuit of negligible resistance and inductance L henry, and that a sinoidal alternating current is flowing in it, the instantaneous value of which can be represented by

$$i = I_{\text{max.}} \sin(2\pi ft)$$

as was shown at the beginning of this section; then since the current value is altering all the time, there will always be a back voltage self-induced in it which is given by :—

$$e_s = -L \times \text{rate of change of current}$$
$$= -L \times \text{rate of change of } I_{\text{max.}} \sin(2\pi ft)$$

It can be shown mathematically that the rate of change of $\sin(2\pi ft)$ is given by $2\pi f \cos(2\pi ft)$ so that the self-induced voltage can be expressed as

$$-LI_{\text{max.}} \times 2\pi f \cos 2\pi ft$$

that is, if the current is a sine function, or varies according to a sine law, the self-induced voltage is a cosine function. The significance of the negative sign at the beginning is to show that the self-induced voltage is opposed to the change in current. If the current and self-induced voltage be plotted against a time base, graphs of the form and phase displacement of those

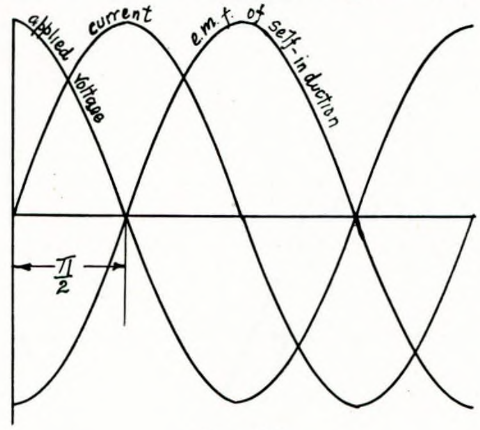


FIG. 122.

shown in Fig. 122 are obtained. Now, despite this back voltage the current is still flowing, which can only mean that, since we are assuming no resistance in the circuit, the volts applied to the circuit must be exactly equal and opposite to the back voltage of self-induction. The graphs of applied volts, current, and self-induced volts will therefore be as shown in Fig. 122. It will be noticed that in a purely inductive circuit the current lags by $\frac{\pi}{2}$ radians or 90° behind the applied voltage. This is very important.

The applied voltage, being equal and opposite to the self-induced volts, will be

$$e = - - 2\pi fLI \cos(2\pi ft)$$
$$= 2\pi fLI \cos(2\pi ft)$$

This is the instantaneous value. The maximum value, when $\cos(2\pi ft) = 1$ (*i.e.* when $t = 0$) will be

$$e = 2\pi fLI$$

and since the R.M.S. value is a definite fraction of the maximum value (·707 max., just as for a sine curve) we can write

$$e_{\text{R.M.S}} = 2\pi fLi_{\text{R.M.S.}}$$

from which we obtain the value of the current

$$i = \frac{e}{2\pi fL} = \frac{e}{x}$$

The quantity $2\pi fL$ is called the *reactance* of the circuit and is measured in ohms.

Example.

A 50-cycle alternating current circuit has an inductance of 0·01272 henry. Calculate its reactance.

$$x=2\pi fL.$$

One must be careful here. It was stated at the beginning of this section that π could be expressed either in degrees or radians when working examples on the equation

$$e=E_{\text{max.}}\sin(2\pi ft).$$

When using the expression $2\pi fL$ for reactance π is always measured in radians, *i.e.* it is 3·1416. The reason for this is involved in the calculus operation employed in finding the rate of change of $\sin(2\pi ft)$.

$$\begin{aligned}x&=2\pi fL\\&=2\times 3{\cdot}1416\times 50\times 0{\cdot}01272\\&=4 \text{ ohms.}\end{aligned}$$

Circuits having both Resistance and Inductance.

In Fig. 123 let i represent the graph of current. If the resistance of the circuit is r ohms, for each successive value of current a voltage of ir will be required

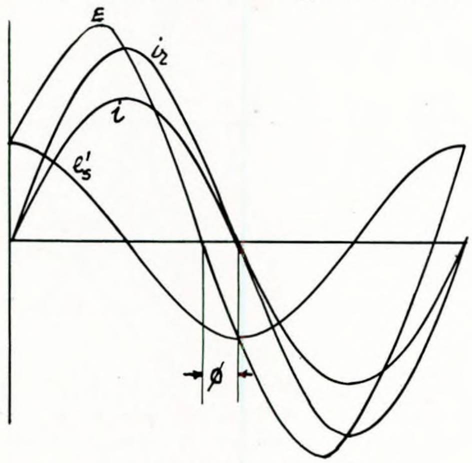


Fig. 123.

to force the current through the resistance. The graph of this voltage will also be a sine wave strictly in phase with i. At the same time the applied voltage must also have a component, $e_s{}^1$, 90° ahead of the current wave to counterbalance the back voltage of self-inductance set up because the current is continuously changing. If we wish to find the total value of the voltage which must be applied to the circuit at any instant, we must add together the voltage required for resistance and the voltage required to overcome the self-induced voltage for that particular instant, having regard to their signs, positive or negative. If we do this we obtain the graph of total applied voltage E, which we see is leading the current but by something less than 90°.

We saw some time back that we could add together sine waves vectorially. Let us draw a vector diagram

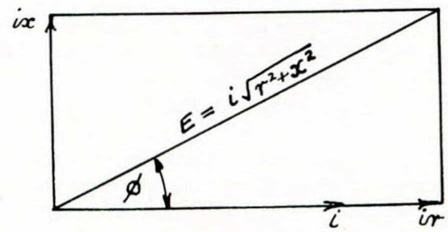


Fig. 124.

for this circuit (Fig. 124).

The vector i is drawn arbitrarily in any direction (but horizontally is probably most convenient) to represent the current i in magnitude. Another vector ir is drawn coincident or in phase with i to represent the voltage used in overcoming resistance, ir, in magnitude. The voltage and current scales can of course be different. A vector ix is drawn 90° ahead (*i.e.* measured forward in an anti-clockwise direction) to represent the component of applied volts which balances the back voltage of self induction. The vectors ir and ix can now be compounded vectorially giving E, the vector of total applied voltage.

$$\begin{aligned}\text{Evidently } E^2&=i^2r^2+i^2x^2\\&=i^2(r^2+x^2)\\ \therefore\ i^2&=\frac{E^2}{r^2+x^2}\\ \text{and } i&=\frac{E}{\sqrt{r^2+x^2}}\end{aligned}$$

This equation is of considerable importance. The quantity $\sqrt{r^2+x^2}$ is called the impedance of the circuit. It is always represented by z.

Summing up, if

$$\begin{aligned}r&=\text{resistance}\\ \text{and } x&=\text{reactance}=2\pi fL\\ z&=\text{impedance}\\&=\sqrt{r^2+x^2}\\&=\sqrt{(\text{resistance})^2+(\text{impedance})^2}\end{aligned}$$

Example.

A coil has a resistance of 3 ohms and an inductance of 0·01272 henry. Calculate the current flowing through it when connected to 100 volt, 50 cycle a.c. mains.

$$\begin{aligned}\text{reactance } x&=2\pi fL\\&=2\times 3{\cdot}14\times 50\times 0{\cdot}01272\\&=4 \text{ ohms.}\\ \text{impedance}&=\sqrt{r^2+x^2}\\&=\sqrt{3^2+4^2}\\&=5 \text{ ohms.}\\ \therefore\ \text{current}&=\frac{100}{5}=20 \text{ amperes.}\end{aligned}$$

Abstracts of the Technical Press

Novel Type of Oil Strainer.

Among the exhibits which were to have been shown at the cancelled Shipbuilding and Marine Exhibition at Olympia, was the "Clinsol" oil strainer illustrated in Fig. 191, which incorporates the principle of edge straining and can be cleaned in service. It consists of a cylindrical pack of straining plates, the spaces between the plates forming the straining orifices. The pack is housed within a body casting with a sump for the dirt below the pack, cleaning being effected by means of knives arranged to deposit the dirt in a place in the pack, exterior to which there is no flow to the slots, thereby reducing the risk of entraining again the finer particles of refuse. As may be seen from the diagram of cleaning operations, the relative movements of the pack in relation to the fixed knives are such as to force the dirt into a space in the pack in the form of a long channel in the face. By means of packing pieces, a zone of no flow is provided in this space. A stop prevents the complete rotation of the pack, but it is placed in such a position that the knives partially emerge from the slots so that the reverse motion, illustrated in the third figure of the cleaning diagram, dislodges the accumulation of dirt from the knives and allows it to drop down the channel to the sump. The pack is strongly constructed, and so designed that any possibility of bursting or rupturing the straining medium is eliminated. Packs have been tested up to a pressure difference of 60lb./in.^{2} between the inlet and discharge, without damage, although a far smaller pressure would destroy a filter of the gauze type. By employing different thicknesses of packing pieces between the straining plates, the strainers can be adapted to different duties and outputs. They are made in sizes from $\frac{3}{4}$-in. pipe bore up to $4\frac{1}{2}$-in. pipe bore for suction and discharge circuits when using oil fuels, oil engine lubricating oils, or turbine lubricating oils.—*"The Engineer", Vol. CLXVIII, No. 4,374, 10th November, 1939, pp. 476-477.*

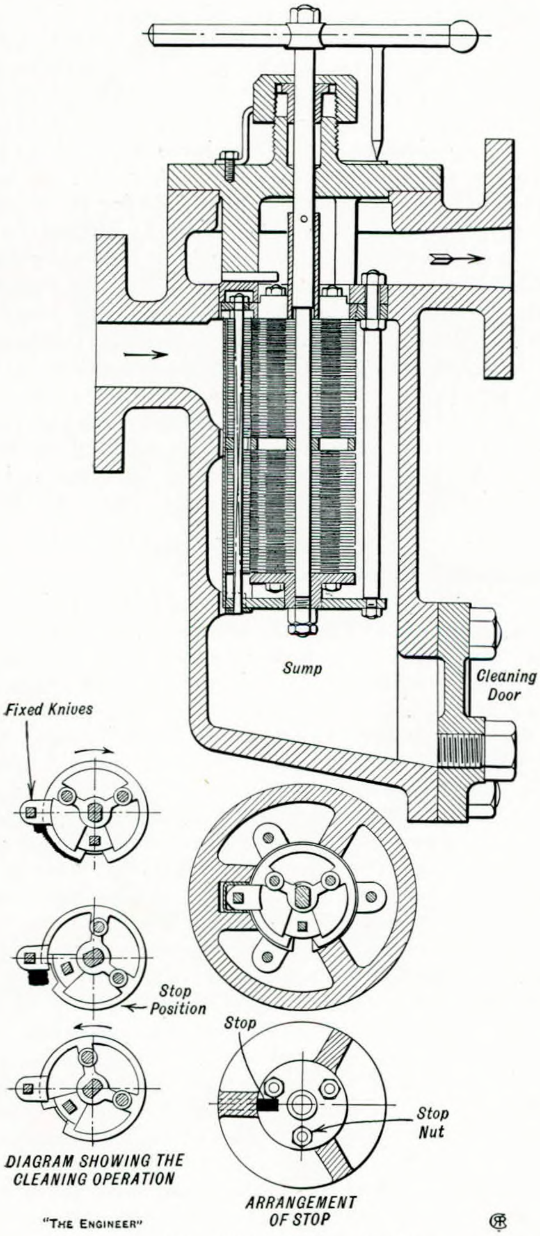


FIG. 191.—Oil strainer—White.

Gas Ferry "Preussen".

The Hamburg-Amerika Line's new m.s. "Preussen" was recently completed for ferry service on the Lower Elbe. The vessel's main dimensions are 146ft. 9in. by 32ft. 9in. by 14ft. 7in., with a maximum draught of 10ft. 6in. on a displacement of 570 tons. The vessel carries 300 passengers and about 20 tons of cargo, in addition to a limited number of motor-cars. The propelling machinery comprises two sets of Humboldt-Deutz gas engines each developing 415 h.p. at 375 r.p.m., the corresponding speed of the ship being 12 knots. These engines operate on producer gas made from anthracite, the producer being fitted with an automatic feed. The ferry averages 8 hours' running per day, the bunker capacity of 30 tons sufficing for a fortnight's operation. The electric steering gear is worked by push-button control, the double rudder giving excellent manœuvring qualities and a small turning circle. The reversing gear of the propelling machinery and the control of engine speed are operated from the wheelhouse by the Pahl patent single-lever system. The deck auxiliaries are electrically driven.—*"Shipbuilding and Shipping Record", Vol. LIV, No. 16, 19th October, 1938, p. 434.*

Stern Tube Lubrication.

A world-famous American oil company has recently developed a system of stern tube lubrication that is designed to permit the use of grease lubrication in the old standard type of stern tube with lignum vitæ bearings, without elaborate equipment. The method is simple, consisting of packing the entire stern tube with a water-resistant grease and providing a means for adding grease to make up for any loss. It can be applied to existing ships without their having to go into dry dock. Most stern tubes are fitted with a water service connection for flushing-through purposes, the water forced out of the tube at the after end preventing the entry of sand and mud. The water-service pipe is disconnected and attached to a grease gun by means of which the tube is pumped full of grease, the gun being left connected up and used to keep a solid pack in the tube. It is desirable to leave the stern gland packing slack until the tube is full, and it should never be set up hard. While the ship is under way any loss of grease from the tube can be made up by means of the grease gun, the amount to be added daily varying, but never amounting to more than a few pounds. Since the grease is not fluid and there is always a pressure of water at the outer end of the tube, there is no natural tendency for the grease to run out, although some will work out owing to the motion of the shaft. An important factor in the success of this type of lubrication is the kind of grease used. It must not be fluid enough to flow, but should be thin enough to pump freely and absolutely impervious to the action of sea water.—*"Motorship and Diesel Boating", Vol. XXIV, No. 10, October, 1939, p. 518.*

High-speed Diesel Ferry of Unusual Design.

A high-speed passenger and cargo ferry for service of Lake Maracaibo, Venezuela, is to be built in the U.S.A. The vessel's dimensions are to be $115\frac{1}{2} \times 21 \times 8\frac{1}{2}$ft., with a draught of only 4ft. 6in. She is to have a service speed of 17 knots and will be driven by two sets of 8-cylinder lightweight Diesel engines with Büchi superchargers, designed to develop a total power of 1,000 b.h.p. at 900 r.p.m. The propellers will run in semi-tunnels. The hull is to be of special high-tensile steel, constructed on the longitudinal system, with four lattice-type girders extending fore and aft from the top of the floor frames to the deck beams. The design provides for all-welded construction.—*"Motorship and Diesel Boating", Vol. XXIV, No. 10, October, 1939, pp. 516-517.*

Ship's Electrical Equipment.

The revised rules of Lloyd's Register relating to the electrical equipment of ships have been coordinated with the new regulations recently issued by the I.E.E. Ship Electrical Equipment Committee, and are published in the Society's *Rules for Steel Ships, 1939-40.* An extract of the electrical rules will be available shortly. They come into force for all new construction for which plans are approved after the 18th October, 1939. Practically every section of the rules has been re-written, and the new requirements prohibit some methods and materials which were permissible formerly, but have proved inadequate owing to the superior materials now available. Section 15, which deals with the extra requirements of tankers, has been expanded to deal more precisely with the type of fittings permissible in various parts of a tanker, and also covers power circuits more completely. The sections on distribution and switchgear have been recast to meet modern requirements. Cables have received special consideration and several types are no longer permitted, *e.g.,* tough rubber sheathing for permanent wiring, armouring without a lead sheath, flat twin- and three-core lead-sheathed cables, and all rubber insulated cables having pure rubber next to the conductor. Important changes in technical requirements appear in the generator section. In future, the series winding is to be on the negative—as in land practice—and not on the positive. The requirements for the fluorescent tube lighting commonly used in passenger vessels have now been added to the rules.—*"Electrical Review", Vol. CXXV, No. 3,234, 17th November, 1939, p. 658.*

"Baronesa" Explosion.

The findings of the Board of Trade inquiry into a boiler explosion on board the 8,663-ton steamship "Baronesa", which occurred last June at Liverpool, has attributed the mishap to a shortage of water in the starboard after boiler, the consequent overheating of the crowns of its wing furnaces and the admission of feed water to the boiler. When these crowns were in that overheated condition the cooler water and condensate coming into contact with the upper surface of the overheated port side furnace caused contraction and a redistribution of stress resulting in the collapse of the port high furnace in the course of which a circumferential fracture developed from the upper surface. The Court found that the second engineer had admitted that since the boiler in question was under steam he had not tested the boiler water-gauge system by manipulating the top and bottom shell cocks, and that if he had done so, the explosion would have been averted. Neither did he avail himself of test-cock methods. Although the second engineer considered that the boiler was overfilled he did not satisfy himself that the boiler water-test valve was open. Two other engineers were both in part to blame, while the chief engineer could not be absolved from all responsibility. It was his duty to see that there was an efficient system of co-operation and that the engineers carried out their duties in a proper manner. The junior engineers had not the necessary qualifications and had not received satisfactory training. The second engineer, who was in a large measure to blame for the explosion, was ordered to pay £20 towards the cost of the inquiry and two other engineers £10 each. The chief engineer and two others, although not altogether free from censure, were not ordered to contribute to the cost.—*"The Shipbuilder and Shipbuilding Record", Vol. LIV, No. 16, 19th October, 1939, p. 443.*

Report on Boiler Explosion in s.s. "Islandmagee".

The report of a Ministry of Shipping surveyor who investigated the cause of a boiler explosion which occurred on the 4th July, 1939, in the 227-ton Glasgow steamship "Islandmagee" (late "Bonahaven"), while the ship was at sea, states that the explosion was due to local thinning of the back plate of the port combustion chamber and its consequent failure to withstand the normal working pressure of 120lb./in.^{2}. The local thinning was caused by corrosion and over-heating of the metal. In his observations the Ministry's Engineer-Surveyor-in-Chief remarks that the 39-year-old boiler had apparently suffered appreciable wastage, and although maintenance and inspection were ordinarily adequate, the local thinning of the plate in this case had extended to the point of failure within 12 months of the previous classification survey. The accident was not of a dangerous nature in itself, but owing to the resulting necessity of drawing the fires in the single boiler, the ship was rendered helpless and she was fortunate in receiving early assistance.—*Lloyd's List and Shipping Gazette", No. 38,992, 25th October, 1939, p. 9.*

Crankpin Turning Machine.

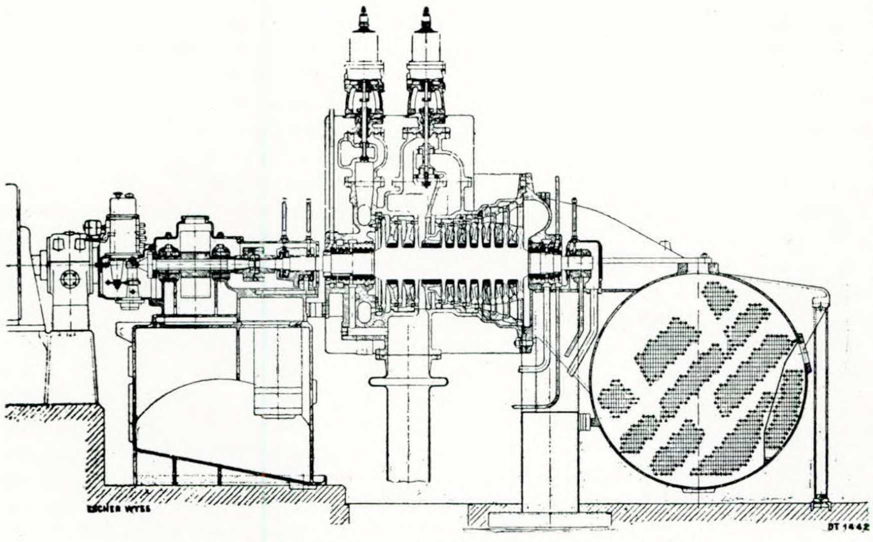
An engineering firm in the U.S.A. has developed a crankpin turning machine by means of which crankpins of marine (and other) engines can be trued up in place. The use of this machine renders it unnecessary to dismantle the engine and refinish the crankshaft in a lathe. Apart from the saving of repair costs thereby effected, the machine enables the heavy lifting and manhandling involved by the work of dismantling and reassembling a large engine to be eliminated. The machine can be attached to the crank and mounted on the crank-case door. Where only one job has to be done the makers will arrange for the supply of a machine on a rental basis, together with an operator.—*"Motorship and Diesel Boating", Vol. XXIV, No. 10, October, 1939, p. 529.*

Cast Iron and Gunmetal.

It is stated that in a Diesel engine of 1,800 b.h.p.—that is, an installation suitable for an 8,000-ton d.w. vessel—the weight of the engine alone would be 240 tons, which would include 8 to 16 cwt. of gun-metal and 1 to 3 cwt. of brass, depending on the design of the engine. This is equivalent to a non-ferrous metal proportion of between 0·2 to 0·4 per cent. of the total weight. Although the quality of cast iron has, during the past few years, undergone a substantial improvement, brass and gunmetal in the form of alloy bronzes have the great advantage of possessing a high degree of resistance to corrosion. At the present time it is probable that the use of various alloy bronzes in merchant ships will have to be reduced to a minimum and replaced, as far as possible, by cast iron. In certain cases—as *e.g.,* in those of propellers—the substitution of cast iron for bronze will result in a certain loss of efficiency in service, but there are hundreds of hull fittings in which the use of good quality cast iron would not affect efficiency. The present-day demand for high-grade cast iron for such purposes as Diesel engine cylinder liners has led to the production of special alloys of the metal, nickel, chromium, molybdenum or tungsten being introduced into the material to provide the requisite heat- and wear-resisting characteristics. What is known as "Diesel quality" cast iron has a tensile strength of 18 tons/in.^{2} as compared with the 9 tons/in.^{2} of ordinary cast iron and it can be said that it is now possible to produce a quality of cast iron suitable for most purposes. Nevertheless, it must also be remembered that such special quality material is often fairly costly, and is only used in special circumstances. It would be possible to replace bronze alloys to a very considerable extent by cast iron alloys, but only at a price.—*"Fairplay", Vol. CLIII, No. 2,946, 26th October, 1939, p. 121.*

Simplified Turbine with Unusual Condenser Arrangement.

A simplified turbine-and-condenser unit which occupies a much smaller space than the usual type of turbine has been developed in Switzerland under the name of Rectaflux. It is particularly suitable for generator sets of up to about 1,500 kW. and similar purposes. The condenser forms part of the turbine itself, a portion of the condenser shell being cast in one piece with a section of the turbine casing. The design of the turbine follows standard practice and consists of a single rotor which may, however, be divided into H.P. and L.P. stages where it is desired to bleed steam from it. The casing at the outlet end is extended to form two exhaust ducts to the condenser, which is located at right-angles below the rear end of the turbine. The Rectaflux design is claimed to eliminate the need for a large basement below the turbine such as is usually necessary with turbine sets of standard design. The H.P. end of the turbine casing is supported by pads resting on the casing of the gear through which the turbine drives the generator or



Sectional diagram of the Rectaflux turbine design.

pump. The gearcase also houses the turbine thrust bearing. The Rectaflux turbine sets are generally built up on small steel frames instead of on the usual form of baseplate, the lateral feet at the rear end of the turbine being arranged to slide on cast-iron slide-plates to allow for expansion. Special provision is made to prevent the undesirable rising of the level of the condensate in case of a stoppage of the air pump, by fitting a float-operated valve which admits oil under pressure from the lubricating system into a servo-motor which actuates a valve at the top of the condenser. This allows the turbine to exhaust freely and at the same time permits the condensate collected at the bottom of the condenser to act by its own weight on a non-return valve, which enables the water to be removed by syphoning. When the evacuation is complete the valves and floats return to their original position in readiness for operation when the level of the condensate again necessitates it. The accompanying diagram shows the general arrangement of the turbine unit.—*"Industrial Power", Vol. XV, No. 170, November, 1939, pp. 307-308.*

Ice Crystals.

Ice used for the preservation and preparation of food is generally supplied in the form of solid blocks which have to be broken up for convenient use. This involves considerable waste and adds to overhead costs. A well-known American firm has therefore developed a compact ice-making unit which supplies ice in the form of snow crystals at a cost claimed to be lower than was previously possible. The machine, which is simple in operation and does not require the constant services of an engineer, comprises a tubular cylinder formed by successive turns of square-sectional aluminium tubing wound to the required diameter. Ammonia or freon is expanded through the tubing and thereby cools the interior of the coils, while water sprayed on the inner surface of the tubular freezing cylinder through spray headers mounted on a revolving rotor, instantly forms a thin layer of ice. The rotor also carries a number of blades which scrape the ice from from the cylinder wall and deposit it in the form of snow-like crystals into a storage bin below. If desired, the ice can be produced in the form of standard cubes by means of an additional attachment. — *"Shipbuilding and Shipping Record", Vol. LIV, No. 17, 26th October, 1939, p. 447.*

Presidential Address at the Annual General Meeting of the North-East Coast Institution of Engineers and Shipbuilders.

The newly-elected President, himself a shipowner, spoke of the increasing difficulties facing the shipping and shipbuilding industries due to war conditions. He suggested that the scheme adopted by the Government for the repair of bridges wrecked by air action could also be brought into operation for ship-repair work. This scheme would merely involve the preparation and distribution at various ship-repairing centres of standardized components for renewing the damaged parts of ships' hulls, etc., with a minimum of delay. The President also referred to the work of the Institution and of the technical societies in general. In speaking of the meetings, papers, discussions and published *Transactions* of the *Institution* he mentioned occasional complaints concerning the insufficiency of so-called "practical" papers and of the efforts made to include an adequately large proportion of such papers in the syllabus. Turning to engineering education and the important part played by the Institution in furthering it, the President stressed the value of a university training in the engineering and shipbuilding industries. In 1915, he said, 97 of the Institution's 1,159 members were university graduates, whereas in 1938, no fewer than 204 members out of a total membership of 1,202 had university degrees, an increase of over 100 per cent. The fact that 83 per cent. of the members had no degrees might, perhaps, indicate that the engineering and shipbuilding industries were not getting the full benefit of the great advances in educational facilities that have taken place during the last 20 years. He said that we were still faced with the problem of the correlation of university training with the needs of industry on the one hand and the accommodation of the maximum number of university-trained men in works on the other. A similar problem had, he knew, not yet been settled in commerce.—*Address by Major T. Russell Cairns, T.D., given at the North-East Coast Institution of Engineers and Shipbuilders, on the 13th October, 1939.*

Simple and Rapid Repair of a Broken Crankshaft.

The accompanying illustrations show a very simple, cheap, rapid and efficient means of temporarily repairing a broken crankshaft forming part of the driving mechanism of a certain machine tool which it was desired to maintain in continuous operation pending the arrival of a new crankshaft. Referring to Fig. 1, it will be seen that the open type crankshaft A had broken along the heavy line across the root of the crankpin B and the web. The broken shaft was set up in a lathe and the web turned down in diameter and thickness as shown by the dotted lines, the broken portion of the crankpin being, of course, faced off during this operation. The shaft was then set up in a milling machine in order to mill the remaining portion of the collar or web into the form of a hexagon (Fig. 2), the set-up being arranged, by means of packing, to produce an inclination on each side of about 7½° to the longitudinal axis of the shaft, the smaller end beginning at the left-hand face of the web as shown at A (Fig. 2). A mild steel cap piece C was machined from a suitable cylindrical billet, the left hand face carrying the crankpin B integral with the body of the cap, and the right-hand face being machined out to fit over the tapered hexagon on the crankshaft. The cap C was then driven on to the hexagon by a few smart blows of a heavy hammer and secured in place by means of a cheese-headed screw D passing into a tapped hole made in the end of the shaft for the purpose. The depth of the hexagonally dished end of the cap C was made ⅛-in. deeper than the thickness of the tapered hexagon on the shaft to ensure an adequate clearance between the faces and to provide a proper grip at the tapered edges of the hexagon collar. The repair proved highly satisfactory, no shift of the cap or pin being apparent after a considerable amount of heavy running. Furthermore, some adjustment in the amount of stroke was obtainable by simply altering the setting of the cap on the tapered hexagon collar so as to alter the clearance provided between the faces of the cap C and the end of the shaft.—*"Industrial Power", Vol. XV, No. 170, November, 1939, p. 317.*

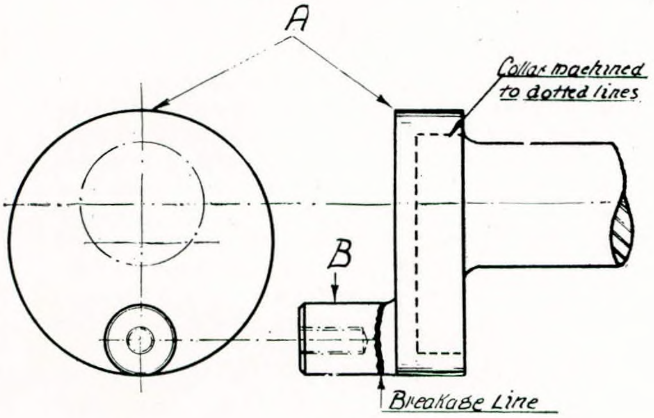


FIG. 2.

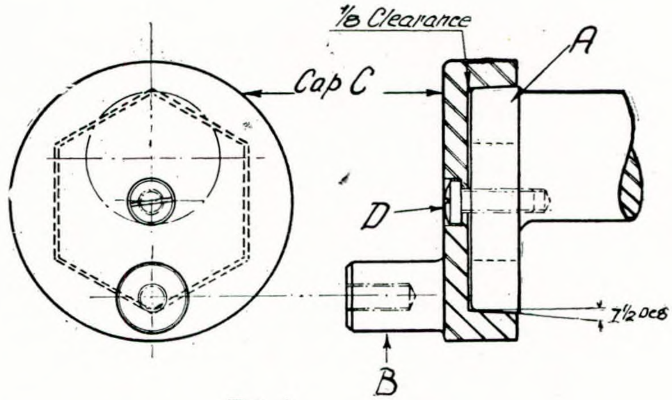


FIG. 1.

An Interesting Crankshaft Repair.

In the course of large repairs recently carried out on the 3,800-h.p. engines of the Furness Withy motorship "Javanese Prince" at the Brooklyn Yard of the Bethlehem Steel Company, the engines were completely dismantled and the crankshafts removed and taken ashore. Such of the old journals, pins and webs as were in good condition were used again, three plain journals, two journal couplings (one for the compressor and one for the timing gear) and four crankpins being renewed on the starboard shaft, while the reconstruction of the port shaft required the renewal of six plain journals and four journal couplings, one for the flywheel, two for the timing gear, and one for the compressor, all eight crankpins (18¾in. dia.) and the after web of No. 5 crank. The crankpins, journals, journal couplings and webs were all forged and rough-machined at the company's works and tested by Lloyd's surveyors. In assembling the shafts, the eyes of the old crank webs were lightly bored and finished to a dead smooth surface. The shrink fits of the new pins and journals were smooth machine-finished to the diameters giving the required shrinkage, and plain dowels were pressed into all the new journals at the positions occupied by the originals. Great care had to be exercised in assembling the shafts. Owing to the short stroke of the engine it was found necessary to shrink one journal and one crankpin into each web simultaneously, to avoid distortion of the assembly. After the entire crankshaft was shrunk and assembled, the journals and crankpins were machined in alignment. One of the features of the repair was the machining of the crankpins in a lathe by means of a specially-designed crankpin turning machine. All the lower halves of the crankshaft main bearings were completely remetalled and bored in place. Finally, the completed crankshafts were taken aboard, bedded in, and deflection readings were taken between the webs.—*"Motorship and Diesel Boating", Vol. XXIV, No. 10, October, 1939, p. 511.*

Some Breakdowns in Marine Diesel Engines.

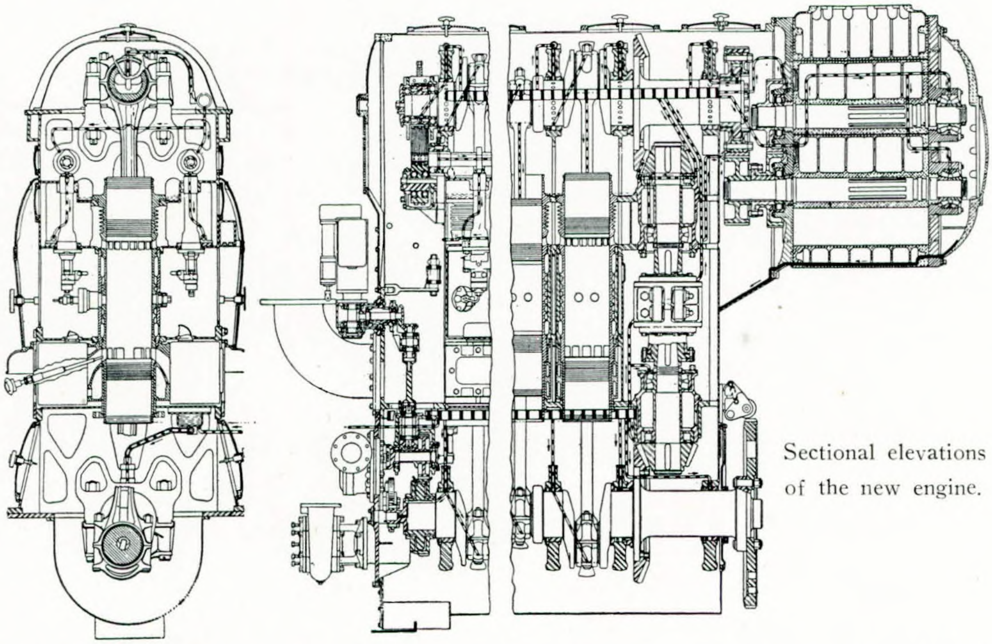
The first of a series of articles under this title concerns a number of breakdowns which have recently occurred in the Diesel engines of various French and foreign motorships. Fractures in forged steel piston trunks have occurred due to original defects in the metal, through internal stresses set up by unequal cooling, and through local corrosion. Several cases of fracture of various parts of cast-iron piston trunks have also been observed. The circular rupture of the bottom of a piston cooled by sea water is described and several instances of diametral fracture of semi-hardened oil-cooled piston bases are also referred to. Star-shaped fractures of piston bases due to various causes, and several cases of piston-base erosion caused by the misuse of welding have been noted. The overheating of a piston trunk and consequent cracking of a cylinder liner occurred in a 2-stroke engine, and several cases of piston seizure in such engines have also been reported. Starting up a 6-cylinder 2-stroke engine of the crosshead type with an exhaust valve closed led to severe distortion of 3 out of 6 crankshafts. Breakages and stretching of studs and bolts for securing cylinders to engine frames and frames to bedplates, have occurred through defective fitting, fatigue of material, and faulty metal. Repeated cases of failure of various bearings in a new Diesel installation were attributed to faulty design, inferior bearing metal and defective lubrication. Breakages of fuel injection valves have been reported in several ships, the mishaps occurring when the engines were started up. Amongst defects observed in Diesel air compressors were several in the form of excessive compressor cylinder wear due to overheating.—*"Bulletin Technique du Bureau Veritas", Vol. 21, No. 8, October, 1939, pp. 164-169.*

Pressure Behind Diesel Piston Rings and Dynamic Stresses in the Running Gear of Diesel Engines.

In the course of a paper recently read by Mr. Robert Sulzer on measurements made on the experimental engine at the Sulzer Works in Switzerland, he established the fact that the greater part of the load on the piston of a Diesel engine is static and that the piston therefore profits from an increase in the fatigue strength of the material with increasing static mean stress. A parallel investigation was also made with the same piston crown in a cold state and removed from the cylinder. It was tested under hydraulic pressure and found to bend about 10 per cent. less than when it was in the engine cylinder, a fact that might be partly explained by the relation between the modulus of elasticity and the temperature. Such measurements of temperature and changes of shape yielded valuable data relating to the probable working conditions of piston rings. It was concluded that the sealing effect aimed at should be realised under all conditions, whatever changes might occur in these. Sliding in grooves which are at a very high temperature, with clearances varying according to the load, and at different inclinations to the running surface, the piston rings exert a varying pressure on the cylinder liner which itself is at different temperatures at different parts. The pressure behind the first ring follows the gas pressure in the cylinder almost exactly, while that behind the second ring reaches a maximum of only 8 atmospheres, showing that the sealing effect is already practically obtained. Mr. Sulzer pointed out that the results were different when the rings had to work in grooves of inadequate width, as in such cases the grooves closed at high loads due to heat distortion and piston ring seizure occurred. Excessive movements of the piston within the liner cause the ring to close in to such an extent that it no longer touches the liner at certain places and allows the gases to blow past. The influence of dynamic stressing on the loading of the running gear had, Mr. Sulzer stated, also been measured. Under the effect of ignition knock the running gear is, because of its own inertia, distorted and vibrated beyond the point corresponding to the position of equilibrium. There is, therefore, an increase in the stresses in the running gear, which rise in proportion to the increase in pressure gradient in the first stage of combustion. With a compression of 35 atmospheres and a maximum cylinder pressure of 60 atmospheres, it was found, said Mr. Sulzer, that the maximum stressing with a sudden increase in pressure was about 80 atmospheres. In reality the conditions are more favourable, since in marine engines working with normal fuel pressure, gradients of about 2 to 4 atmospheres per degree of crank angle are unusual. Consequently, no great excess stressing occurs in such engines.—*"The Shipping World", Vol. CI, No. 2,419, 25th October, 1939, p. 447.*

A New Opposed-piston Diesel Engine.

An entirely new type of opposed-piston oil engine was recently placed on the American market by the Fairbanks Morse Company. In this engine the upper piston is driven by means of a crank and rod from an overhead crankshaft, which is itself driven from the lower crankshaft by a vertical shaft and helical gearing. The scavenging blower is of the rotary type (a design not employed in British opposed-piston oil engines) driven from the upper crankshaft by suitable helical gearing. The engine has 5 cylinders with a diameter of 8in., the stroke of each piston being 10in. The total power developed is 800 b.h.p. at 720 r.p.m. (continuous rating), with an overload capacity of 15 per cent. for one hour. The total weight is about 9 tons, or just over 25lb. per b.h.p., this low weight having been achieved by the adoption of electrically-welded plate construction for the engine housing. The lower piston completely covers the exhaust ports slightly before the scavenging air ports are covered by the upper piston, so that a small supercharge is ensured in the cylinders. The scavenging air intake is provided with Burgess filters and silencers and Burgess silencers are also used for dealing with the exhaust gases. The engine has hollow crankshafts 8in. in. external diameter. Every cylinder has a separate F.M. injection pump on each side driven by separate camshafts, as shown in the sectional drawing. Rotary fuel pumps are used to convey the fuel from the day tanks to the suction side of the injection pumps, filters being fitted in front of the latter. The engine pistons are cooled by lubricating oil, two gear pumps being fitted to maintain the lubricating and cooling oil circuits. Similar engines of 1,300 b.h.p. are built for industrial and marine purposes.—*"The Oil Engine", Vol. VII, No. 79, November, 1939, p. 211.*



Sectional elevations of the new engine.

Propelling Machinery of French Liner "Pasteur".

The "Pasteur" is a ship of 29,253 tons gross, about 698ft. in length overall, with a beam of about 90ft., and a depth moulded to upper deck of about 66ft. The propelling machinery consists of four sets of Parsons turbines—each set comprising one H.P., one M.P. and one L.P. turbines—driving separate propellers through single-reduction gearing. All four propellers normally turn at 200 r.p.m., corresponding to an average service speed of 22·5 knots. Steam at a pressure of 440lb./in.^{2} and superheat temperature of 725° F. is supplied by 10 Penhoët water-tube boilers, each equipped with three oil burners. There are three boiler rooms, the forward and middle ones containing four boilers each, while the after one is divided into two compartments, one of which contains two boilers and the other the refrigerating machinery, air compressors and various pumps. The forward main engine-room contains the port and starboard wing sets of turbines with their reduction gears and associated auxiliaries, together with two turbo-generating sets, while the after main engine room contains the two centre sets of turbines and gears with their associated auxiliaries and two further sets of turbo-generators.—*Ing. Réville, "Bulletin Technique du Bureau Veritas", Vol. 21, No. 8, October, 1939, pp. 158-161.*

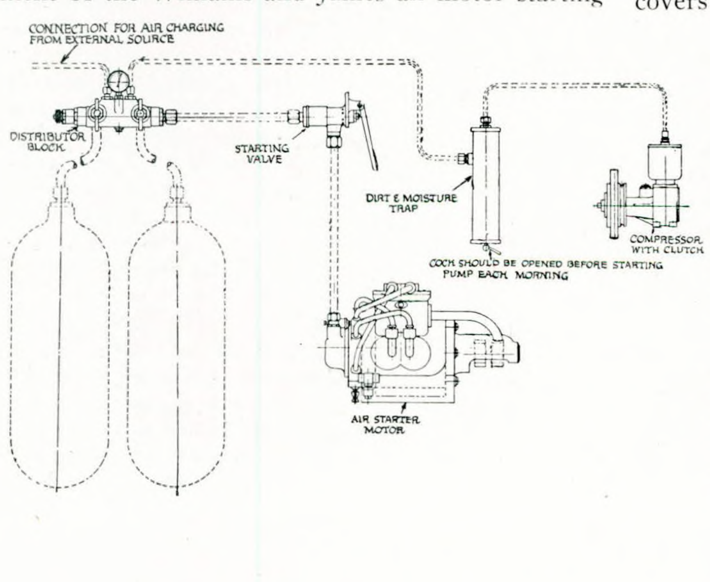
Planning for Cargo Ship Production.

The Presidential Address recently delivered by the newly-elected President of the Institution of Engineers and Shipbuilders in Scotland, was primarily devoted to a consideration of systematic planning and advanced methods of organization for cargo ship production. The speaker dealt with what he described as an imaginary tour through ten stages in the building of a cargo vessel of about 9,000 tons d.w. capacity, in conditions where continuity of production was assured. The ship would be built and delivered in 32 weeks from the time the order was booked, the keel being laid within 6 weeks of that time and the ship launched 21 weeks later. The speaker considered each of the ten stages in detail, beginning with: (I) Securing the contract, in which connection he went on to say that, apart from considerations of financing or the art of salesmanship, a tender for a prospective new vessel should embrace qualities relating to (1) design; (2) specification; (3) performance; (4) price; and (5) delivery time. The next stage would be

(II) The preparation of working drawings and templates, and after this (III) Ordering the material. Within 10 weeks of the signing of the contract, practically all of the steel material—about 2,250 tons—should be delivered to the shipyard. The next phase would be (IV) Constructional and machine shop processes. This embraces the assembly of material, job distribution, continuity of process, special machines, machine shop lag, pre-construction work, overtime work, transport of material, routine work and planning constructional processes. Following on this comes the next stage (V) Erection operations, which include the commencement; the complete erection on the berth of the keel and centre girder with the bottom shell plates riveted in place; the tank margin plates and bars erected and riveted and tank top laid; side frames erected from aft to fore peak bulkhead; bulkheads, beams, deck girders, pillaring and hatchway coamings erected; bow and stern framing complete, all decks plated, midship shell in place, and three skin strakes extended to stem and stern; first ballast tank tested; all wood decks laid; last ballast tank tested; and preparations for launching. Notwithstanding the development of electric welding and its application to shipbuilding, the speaker assumed that the vessel would be of riveted construction, some 450,000 rivets in all being driven into the ship, corresponding approximately to 200 rivets per ton of invoiced steel. In this connection (VI) Riveting as a means of recording progress was advocated by the speaker. The concluding stages (VII) Tank testing, (VIII) Launching the ship, (IX) Fitting out and installing propelling machinery and (X) The trial trip and delivery of the vessel were also briefly reviewed.—*Address given by Mr. Wilfrid Ayre to the Institution of Engineers and Shipbuilders in Scotland, on the 24th October, 1939.*

Air Motor Starting System for Oil Engines.

The accompanying diagram shows the arrangement of the Williams and James air-motor starting equipment for oil engines. The advantages claimed for such starters over the usual system of compressed-air starting for engines of up to 500 h.p. are that the engine starts up from any position without barring round and on full compression; the engine fires at a very low crankshaft speed, as no air is expanded in the combustion space; only a small amount of air is used for each starting operation; the cost of maintenance is negligible and the charge in the air cylinders can be kept for an indefinite period; there is no risk of exceptionally high firing pressures being attained during the starting period; and that there is no need to have a starting-air valve on the cylinder head.—*"The Oil Engine", Vol. VII, No. 79, November, 1939, p. 226.*

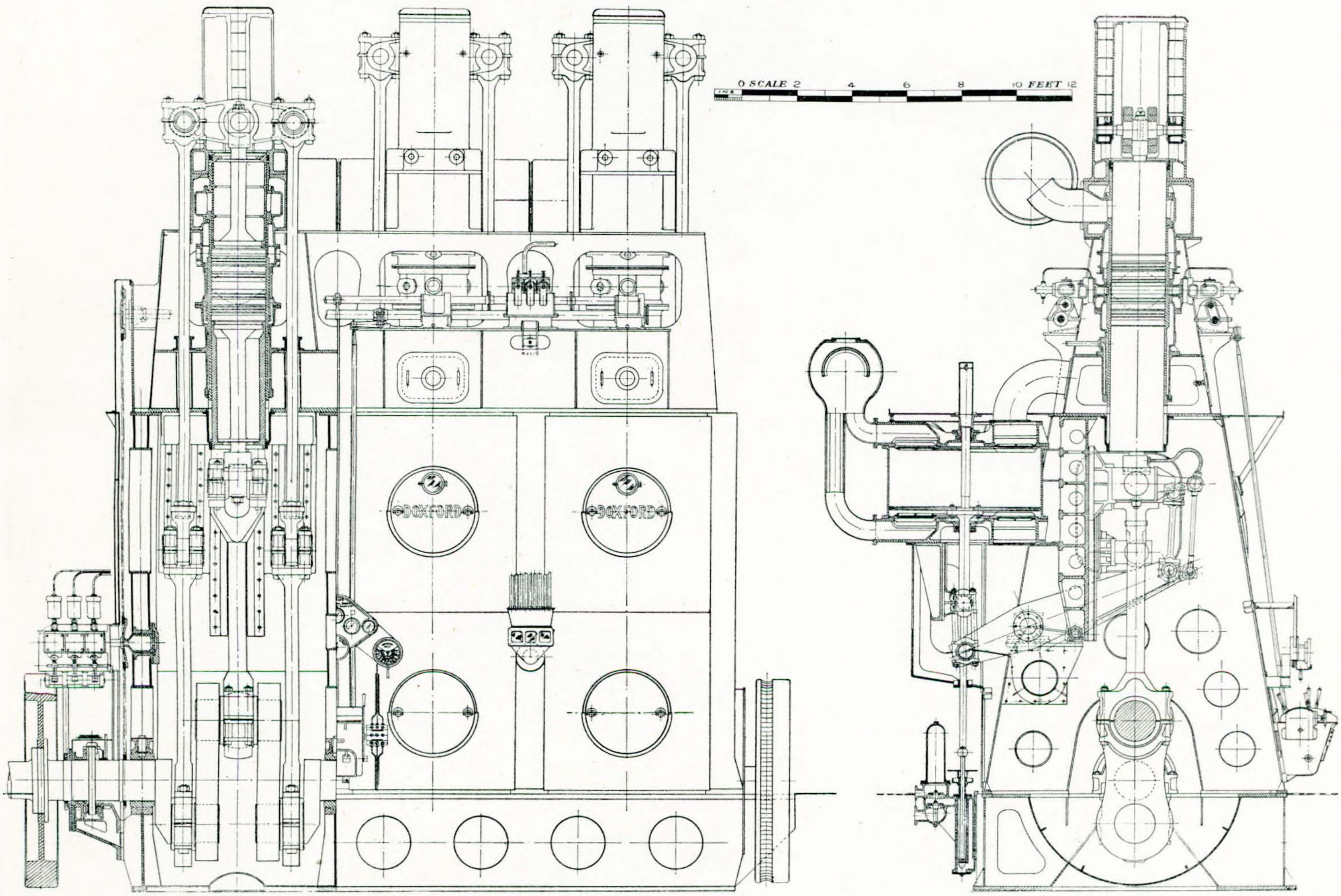


180-b.h.p. "Superscavenge" Marine Engine.

The latest type of Petter "Superscavenge" engine to be put on the market by the makers of these units is a 3-cylinder marine engine with reversing gear and remote control, rated at 60 b.h.p. per cylinder at 500 r.p.m., giving a rated power for the engine of 180 b.h.p. Uniflow scavenging with a gear-driven blower is employed, and twin exhaust valves in the cylinder head, the inlet ports being at the bottom end of the cylinder. The system of scavenging adopted gives excellent combustion with a smokeless exhaust at all normal loads, persisting on considerable overloads, while the fuel consumption is claimed to be very low. Large-size slow-speed cooling pumps are provided and all water jackets are of ample cross-section with large inspection doors or plugs. The transfer passage between the cylinder block and head is a pipe reaching into the cylinder head which directs cooling water straight towards the exhaust passages and atomizer sleeve. The engine is of small size for the power developed and all controls and gauges are grouped in one convenient operation position. The auxiliaries, comprising pumps, compressors and a lubricating-oil filter, are arranged in an accessible position at the forward end of the engine. Large covers for the crankcase and valve gears are provided and the entire working parts of the engine are accessible for examination and refit by the removal of a few covers. A very massive helical gear drive is used for the blower and camshaft, being located immediately next to the flywheel. The governor of the engine has been designed to permit perfectly even running down to about 100 r.p.m., which should make the unit particularly suitable for trawlers and other fishing vessels. The reversing gear has a direct drive through clutches instead of through the usual locked pinion gears, the gear-unit being locked tightly to the crankshaft instead of driving through loose splines or keys, so that it revolves *en bloc* in the "ahead" position. A reduction gear giving a propeller speed of 250 r.p.m. can be incorporated, if desired. The thrust bearings are of large size and of the ball-bearing type. Lubricating oil under pressure is fed to every working part and oil-cooled pistons are used, metered feeds of clean oil being provided for the cylinders and blowers.—*"Engineering", Vol. 148, No. 3,847, 6th October, 1939, p. 397.*

Standard Doxford Engines for Cargo Ships.

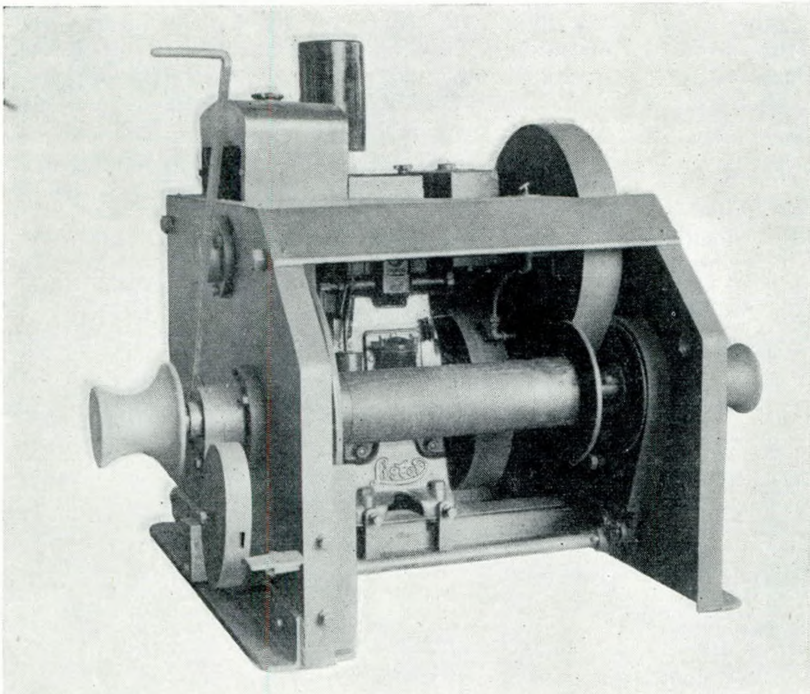
The welded-frame design of Doxford engine originally produced over 6 years ago and developed soon afterwards by the incorporation of an electrically-welded bedplate, has now become standardized, and in its 3-cylinder form, with varying dimensions and revolutions to suit different circumstances, is already found or being installed, in many cargo vessels. The engine illustrated in the accompanying sectional elevations has a cylinder diameter of 600mm., the combined piston stroke being 2,320mm. The two strokes are actually different, that of the lower piston being 1,840mm., while the upper piston has a stroke of 980mm. This arrangement gives a most effective degree of balance, and the engine is exceptionally silent in operation. The output is 2,500 b.h.p. at 108 r.p.m., with an M.E.P. of about 76lb./in.². The two camshafts and three fuel pumps are chain-driven and the engine also drives its own double-acting scavenging-air pump, circulating-water, lubricating-oil and bilge pumps from levers attached to rods operated by one of the crossheads.—*"The Motor Ship", Vol. XX, No. 239, December, 1939, p. 303.*



Sectional elevation of the 2,500 b.h.p. Doxford engine.

Diesel Winches.

The accompanying illustration depicts one of the small sizes of "Sureal" Diesel winches developed by an engineering firm in Kent. The winch is of the friction-drive type, mounted on an all-steel frame with steel side-plates and fitted with a cold-starting Lister radiator-cooled Diesel engine having a self-contained fuel tank. The friction drive consists of machined V-friction wheels of cast iron transmitting a positive drive to a C.I. barrel shaft running in phosphor bronze bushes. The brakes, both foot and hand, are friction-lined and powerful

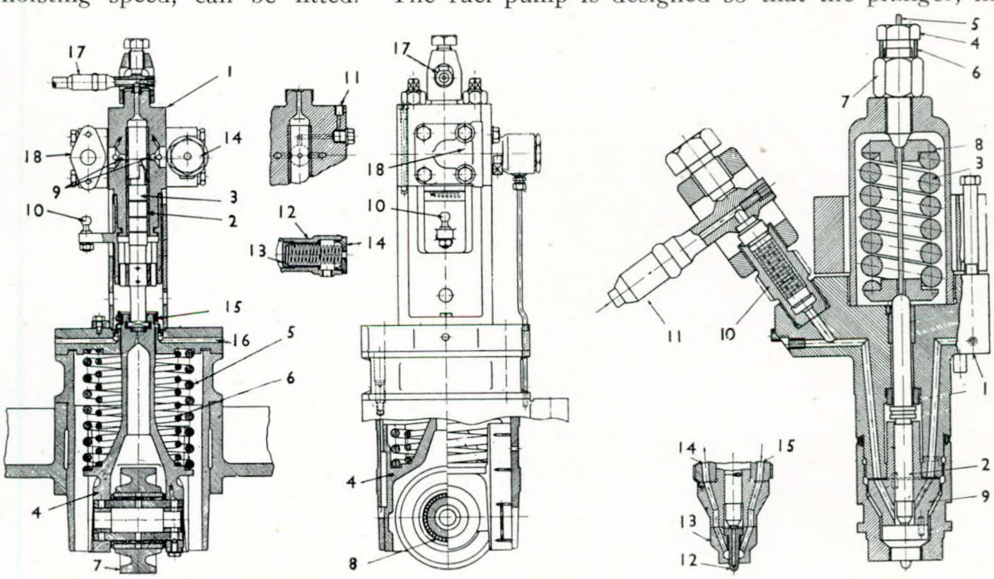


A 15-cwt. "Sureal" Diesel winch.

enough to hold the load in any position. Warping drums of 5-in. diameter are fitted at both ends of the barrel shaft. When fitted in small cargo vessels carrying one lifeboat on the hatches, the construction of this winch enables the lifeboat to be lifted by turning the engine starting handle and placing the control lever in gear. Provision is also made for fitting a messenger chain gypsy wheel on either or both sides of the winch to enable it to be used as an anchor windlass, while for occasional work lifting 2 or 3 cwt. at a time, a dummy barrel giving a considerably increased hoisting speed, can be fitted. The largest size of "Sureal" winch has overall dimensions 6ft. 8in. by 4ft. 11in. by 5ft. high, weighs 33 cwt. gross and can lift 2 tons at 85ft./min. — *"The Shipping World", Vol. CI, No. 2432, 22nd November, 1939.*

Features of the Latest B. & W. Fuel Pumps and Atomizers.

The accompanying sectional views show the arrangement of the main components of the latest Burmeister & Wain type of fuel valve and atomizer, used on single- and double-acting airless-injection engines. The charging or priming pump maintains a constant pressure of from 35·5 to 42·5lb./in.^{2} in the suction chamber of the fuel-injection pump. The latter consists of a housing (1) with a wearing liner (2) in which is loaded by the springs (5) liner and plunger being ground to size and hardened. The fuel, already under pressure, reaches the injection pump through a pipe coupled to the flange (18). The pump plunger is attached by a bayonet joint to the guide (4), which is loaded by the springs, (5) and (6) and fitted over the camshaft. The cam for operating the pump bears on the roller (7), which runs in a needle bearing (8). All the wearing surfaces of the moving parts are lubricated by oil supplied from the forced lubrication system. The fuel priming pump is made of sufficient capacity to supply oil not only to the fuel injection pump and thence to the fuel valve and atomizer, but also to the cooling spaces of the latter. Oil is led from the pump to the fuel valve through a pipe attached to the connection (11) and provided with a non-return valve. This enables a constant flow of oil to pass through the fuel pump and remove any accumulations of air. If heavy grades of fuel are used the continuous cooling of the valve also serves to reduce any formation of carbon on the atomizers. The fuel pump is designed so that the plunger, in



its lowest position, uncovers the ducts (9) which remain open for a relatively long time due to the shape of the pump cam. The amount of fuel delivered by the pump is regulated by means of two recesses milled in the plunger and limited on one side by a steep helical edge, the necessary variation being effected by turning the plunger so as to change the position of the helical edges in relation to the ducts (9). For this purpose the engine control shaft carries a series of levers—one for each pump, the number of pumps depending on the number of cylinders and whether the engine is single-acting or double-acting—which actuates rods, each coupled to a connection (10). When the plunger has moved far enough to uncover the ducts, the suction chamber of the pump is connected to the compression chamber, whereupon the pressure falls instantly and the fuel valve closes, cutting off the delivery of fuel to the atomizer. The shocks in the oil delivery pipe (17) from the pump to the valve, caused by the cessation of injection, are ground-in spring-loaded plunger (13), the outer end of the spring bearing on a cap (14). This absorber takes up the surplus oil from the fuel pump for each injection. At the bottom of the plunger there is a cap (15) to prevent any fuel from reaching the roller guide casing, a drain passage (16) being provided. Maintenance work on the fuel pump is reduced to a minimum, all the tightening and wearing surfaces being ground. The right-hand diagram shows the construction of the fuel valve and atomizer. The housing (1) is machined with a central bore for the valve spindle (2), which is kept on its seat by a spring (3), a screw (4) being provided to adjust the spring compression, while the lift of the valve is indicated by the unrestricted vertical movement of a light spindle (5). A distance piece (6) is fitted between the adjusting screw of the set-bolt (7) which bears on the washer (8) at the top of the spring. The valve spindle works in a bush (9), the spindle and bush being ground with special accuracy owing to the high pressure at this point, while the design of the valve enables both the spindle and bush to be renewed easily and at a comparatively low cost. The spindle is heat-treated and has a flat seating, thereby making it unnecessary to centre the atomizer and spindle with any high degree of accuracy at the time of fitting. A flat seating has, it is stated, proved to be more durable than one of the conical type. All the joints of the valve are ground together to ensure absolute oil-tightness and a special design of filter (10) is fitted to prevent impurities from the inlet (11) reaching the valve passages and choking the atomizer. Referring to the small sectional view of the atomizer, the fuel jets are in the nozzle (12), which is carried in the lower body (13). Cooling oil enters through a passage (14), circulates round the nozzle as near as possible to the tip and is discharged up the orifice (15). The proportions of the valve are such as to obviate any risk of distortion when it is being fitted.—*"The Motor Ship", Vol. XX, No. 239, December, 1939, p. 313.*

New Type of Oil Engine.

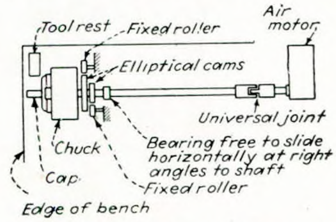
A new and improved type of internal combustion engine is reported to have been developed in Sweden by Mr. Karl Erik Kylen, who has been experimenting with the new design for several years and now considers it sufficiently reliable to be put into commercial production. The engine works on the 2-stroke cycle and tests carried out at the Chalmers Technical College, Gothenburg, have established that the fuel consumption at 33 per cent. of full load is about 30 per cent. lower than that of similar 4-stroke engines running under the same conditions. When running at half-load the fuel saving with this engine amounts to 25 per cent. and when developing full load it is in the region of 20 per cent. In the Kylen engine the crankcase is designed to function as a compressor from which air is led into specially-shaped containers placed on the top of the cylinders. These containers are fitted with automatic inlet valves which are arranged to admit air to the cylinder combustion chambers when the pressure in these is below the pressure in the containers. It is claimed that this enables a 50 per cent. reduction in the time occupied by the scavenging process to be effected, as compared with that of 2-stroke engines of the usual pattern. A remarkable feature claimed for the Kylen engine is that its efficiency is practically constant over the whole range between 30 and 100 per cent. load. The design of the engine includes ball races and roller bearings of the latest type. The inventor states that the working principle of his engine is applicable to all types of internal-combustion engines.—*"The Journal of Commerce" (Shipbuilding and Engineering Edition), No. 34,850, 12th October, 1939, p. 2.*

The Ventilation of Ships.

A reprint of a paper by R. McDonald on the above subject appeared in the August, 1939, issue of the *Journal of the Institution of Heating and Ventilating Engineers.* The author begins by a reference to the Board of Trade requirements for ventilation, which stipulate a minimum of 15ft.^{2} deck area and 830 cu. ft./hr. air supply by mechanical means for each third-class passenger, with an additional 33 per cent. air supply if the accommodation is located in the bowels of the vessel or adjacent to machinery spaces. In actual practice 1,000 cu. ft./hr. per person is the minimum allowed for third-class and crew, which ensures at least 8 changes of air per hour. Ships employed on world tours or in tropical waters are often equipped with ventilating plant sufficient to ensure a maximum of 15 air changes per hour in spaces below deck, which is equivalent to about 2,000 cu. ft./hr. per person. As first- and second-class passengers are allowed more cube or deck area, the same air change in their accommodation may correspond to as much as 3,000 to 4,000 cu. ft./hr. per person. The author emphasises the difference between these conditions and those which apply to the requirements for buildings on land, and the causes from which these conditions arise. In recently-built ships the steam-heating system is generally of the vacuum low-pressure type, the pressure of steam not exceeding 5lb./in.^{2} and the temperature rise being such that the system will be capable of maintaining a temperature of 65° or 70° F. as may be desired in the passenger accommodation. In North Atlantic waters the temperature rise of the air is taken as from 0° to −10° F., and in the Pacific as from 320° F. The author then proceeds to discuss the various insulating materials used in the passenger accommodation and remarks on the fact that little or no insulation is usually provided in the crew's quarters. The ventilation of the passengers' and crew's accommodation, together with that of the sanitary services, public rooms, dining saloons, kitchens and galleys, etc., cargo holds and stores, and machinery spaces, is discussed at some length. With regard to the machinery compartments, the author states that in engine rooms with steam reciprocating machinery 20 to 25 air changes per hour are usually provided, which is increased to 30 to 35 for turbine engine rooms, while in the case of internal combustion engine rooms as many as 40 to 50 air changes per hour may be arranged for. The purpose of an air-conditioning system in a ship is not primarily that of reducing the outside air dry-bulb temperature, but rather the reduction of the relative humidity, as it is this which constitutes the oppressive element in tropical conditions. For example, in the Red Sea the dry-bulb temperature frequently attains 100° F. with 80 per cent. relative humidity, while in the Pacific maximum conditions are about 90° F. and 75 per cent. relative humidity. In order to obtain comfort in these circumstances, it will be the duty of the refrigerating plant not only to reduce the temperature in the ship to, say 95° F. and 65 per cent. relative humidity when the ship is in the Red Sea, but also to, say 85° F. and 60 per cent. relative humidity when in the Pacific. Although this may represent a large refrigerating plant—usually based on the principle of cooling 25 per cent. fresh air and 75 per cent. recirculated air —it is appreciably less than the load which would be involved in cooling 100 per cent. of fresh air. The extraction of the moisture from the outside air, or dehumidifying, as it is called, is the function performed by the cooled air contacting with the warmer air, and producing condensation in the conditioner casing. Moisture is thus extracted and may be used again for cooling purposes in such conditions, but in the operation of humidifying in winter time the conditions are reversed, water being added to the air by evaporation, thus necessitating a certain "make-up" of water from storage, which must be provided for in the ship. The author stresses the importance of a proper system of distribution for the cooled air delivered to any room or saloon, and points out that uniform draughtless distribution is much more essential in a system of this kind than with ordinary mechanical ventilation, due to the creation of lower temperatures and humidities. He declares that unless this is given very careful consideration serious draughts are liable to be caused which would defeat the purpose for which air cooling has come into effect. Strong air movement by the ordinary system when the air is introduced at atmospheric conditions, is not nearly as harmful as a smaller quantity of cooled air not properly distributed, and this is one thing which must be guarded against in the further development and advance of air conditioning in its application to ships.—*"The Engineer", Vol. CLXVIII, No. 4,371, 20th October, 1939, pp. 407-409.*

Cleaning Boiler Handhole Covers.

A very simple device for removing old asbestos gaskets from boiler handhole covers was recently described in the periodical *"Electrical World"*. The device in question enables a man to clean at least 150 covers per day with ease, whereas the hand work was a tiring job which yielded only 40 clean covers in a working day. What the device does is to rotate and oscillate the elliptical cover in such a



way that a hand tool can be held in a fixed position to scrape off what remains of the old gasket. Referring to the accompanying sketch, the cover-plate is held in a chuck having two stationary and one spring-loaded jaws. The chuck is mounted on one end of a 3-ft. shaft, the other end of which connects through a universal joint with the power source. This consists of a portable air drill operated at reduced pressure and controlled by a foot pedal.—*"Mechanical World", Vol. CVI, No. 2,761, 1st December, 1939, p. 509.*

Hard Surfacing Electrodes.

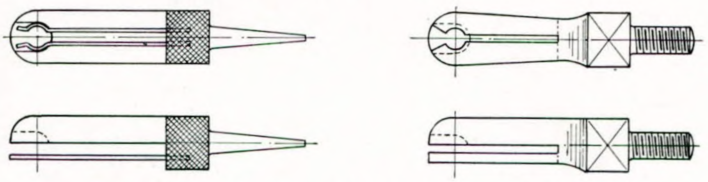
The demand for an electrode for depositing very hard metal on the worn edges of tools and other implements has not been satisfied by the deposition of steel which requires subsequent case-hardening, as it is rarely possible for such tools and implements to be dismantled and heat-treated. A new electrode, known as "Hard Surfacing", has therefore been developed and placed on the market by one of the leading British firms specialising in the manufacture of such equipment. This electrode is stated to be capable of depositing an exceptionally hard surface which is resistant to abrasion and possesses a degree of toughness capable of withstanding a considerable shock or impact. Brinell hardness tests of single runs on mild steel plate made with Nos. 8, 10 and 12 gauge electrodes produced readings of 600, 550 and 575 respectively for narrow runs, and 575, 575 and 600 respectively for wide runs. The same makers produce the "Hardex" electrode with a Brinell hardness of between 630 and 660.—*"The Shipping World", Vol. CI, No. 2,417, 11th October, 1939, p. 398.*

Welding in American Shipbuilding Yards.

A recent report from the President of the National Council of American Shipbuilders states that almost all the U.S. shipyards engaged in the building of ocean-going ships are now making extensive use of welding for that purpose. This has enabled the net weight of a ship to be reduced by anything up to 15 per cent. and it is anticipated that greater experience with welding will also reduce labour costs. A cargo vessel of simple design built during the Great War period may have had as many as a million rivets and any one of these, if damaged in operation, was a potential source of trouble. Welding is already leading to a very great reduction in the number of rivets used in most of the vessels under construction, and in some cases riveting has been almost wholly eliminated. It is anticipated that the limitation of the use of rivets will tend to reduce the size and number of repair bills and to extend the serviceability of a ship. The introduction of welding has eliminated much of the shop equipment and many of the pneumatic tools required for a riveted ship. During the war of 1914-18, the number of men employed in private U.S. shipbuilding yards was multiplied by about eight. The country got good ships, however, which at the peak were being delivered at the rate of 150 a month. Shipyard equipment has since been improved and extensively modified in many respects. The efficiency of present-day vessels is far greater than that of ships of 20 years ago, the weight, space and fuel consumption for the machinery installation of the former being approximately one-half of the values for the ships built during the last war period. The cost of ships has increased enormously in recent years due to higher labour and material costs, higher grade materials, greater safety of life at sea requirements, considerably increased owner and government demands, higher cost of production of plans, increased cost of inspection and higher costs due to legislation.—*"The Iron and Coal Trades Review", Vol. CXXXIX, No. 3,736, 6th October, 1939, p. 472.*

Holding Small Rivets.

Small rivets often have to be inserted in awkward positions inaccessible to the fingers. Fig. 6 shows a tool which can be used to surmount the difficulty and which may be made from carbon steel rod, hardened and tempered, a handy size being about 2in. to 3in. long. A hexagon or square should be cut near one end, the end itself being then turned down and threaded to enable the holder to be screwed into a suitable handle or extension. Alternatively, the end may be shaped to form a tang and have short piece knurled to provide a grip as shown in Fig. 7. Near the opposite end a hole is drilled transversely through the holder to a depth of approximately half its thickness and of a



FIGS. 6 and 7.—A tool for holding small rivets.

diameter to give a sliding fit for the rivet. The bar is then sawn down each way as shown to form a spring clip for lightly gripping the rivet, the slot at the end being bevelled to enable the rivet to enter holes easily. The jaws should next be separated and a seating made in the solid part by means of a twist drill of a diameter corresponding to that of the rivet head, or by using a round-nosed milling cutter of a size to suit the radius of the rivet. The jaws should then be closed together and hardened and tempered to spring temper. The tool can be made with a tang to fit a file handle or it may be made from a hexagon bar and screwed so that it will fit a short wooden handle as well as several extension holders of various lengths. As an alternative to this method of manufacture, half of the rod can be milled away and spring rods substituted for the jaws by fixing them in the hexagon or knurled portion as shown in Fig. 8.—*"Mechanical World", Vol. CVI, No. 2761, 1st December, 1939, p. 507.*

Diesel Engine Seating Design.

A radical change has been made in recent years in the design of Diesel engine seatings in ships by the elimination of the structure above the double-bottom to which the engine bedplate was bolted. Present-day practice is to place the engine directly on the tank top, thereby avoiding the trouble which was formerly experienced with the rivets in the way of the engine bedplate. Nevertheless, trouble sometimes occurs with cast-iron bedplates, and broken or slack holding-down bolts, and even fractures in the engine structure itself are by no means uncommon. The reason for this appears to be that the bedplate of a Diesel engine is subjected to sudden and unequal stresses from the engine, as the working of the hull structure of the ship in a seaway is transmitted to the more or less rigid casting and the combined stresses may cause failure of the material. It is asserted that the substitution of fabricated bedplates, constructed of mild steel components welded together, for the orthodox cast bedplate, will reduce the likelihood of such failures occurring. Mild steel, having a greater elasticity than cast iron, can adjust itself to the working of the hull of a ship to a greater degree, and is able to withstand the external forces to which it may be subjected in a more satisfactory manner. Fabricated engine bedplates and frames of the type under consideration have been used in the construction of a number of recent British motorships, the welded bedplates and frames of the Doxford engines of the "Dominion Monarch" being a case in point. It is improbable that the cost of welded engine bedplates or frames is any lower than that of the cast variety, but they may be anything up to 20 per cent. lighter and—what is of great importance at the present time—they can be produced far more quickly than castings.—*"Fairplay", Vol. CLIII, No. 2,944, 12th October, 1939, pp. 48-49.*

Running Gear for Diesel Engines.

Contributions to a symposium on the above subject were received from 12 British firms engaged in the manufacture of oil engines for marine and general purposes. The points dealt with in these contributions were the following: (1) Arrangement of pistons and connection to connecting rod. (2) Relative advantages of various types of connecting rod big ends. (3) Clearances between pistons and liners, pistons and piston rings. (4) Liner arrangements. (5) Methods of fixing cylinder head. (6) Arrangements for cooling pistons, and (7) Water and gas joints. The marine oil engines dealt with included the Allen 4-stroke trunk-piston type, the Crossley trunk-piston type, the Davey Paxman type with fully floating piston pins, the Fullagar type opposed piston engine, the Harland-B. & W. 2-stroke and 4-stroke with and without crossheads, the Mirrlees Diesel engine, the high-duty 2-stroke Petter engine and the Ruston Hornsby engine.—*Symposium presented for discussion before the Diesel Engine Users' Association on the 12th October, 1939.*

Spiral-flow Heat Exchangers.

A new principle, which has not often been used so far but has much to recommend it in the design of heat exchangers, is the employment of plates arranged in such a manner as to give spiral flow in opposite directions to the two fluids between which an exchange of heat takes place. This type of heat exchanger for the heating and cooling of liquids as well as for the condensation of steam, has been successfully developed by a foreign firm who claim that the spiral-flow principle enables a large area of heat transference surface to be accommodated in a small space, while the use of smooth plates in place of tubes and tube plates permits higher velocities to be employed, with a corresponding increase in the rate of heat transference. It is also claimed that such a design eliminates leakage troubles and provides ready access for the examination of the passages by merely removing the end covers and jointing.—*"Shipbuilding and Shipping Record", Vol. LIV, No. 15, 12th October, 1939, p. 399.*

A Temperature Indicating System for Grain Cargoes.

Grain cargoes in doubtful condition are liable to heat spontaneously, and as periodical "turning over" of the grain is liable to be an expensive matter, it is essential to possess accurate and reliable means of determining the temperature of the grain. A good system is based on the fact that the electrical resistance of a wire of almost any metal varies with its temperature, and in the case of the special metal used for making thermometers, the relationship between resistance and temperature has been determined to within 0·01 per cent. from 0° to 500° F. The apparatus used is based on the Wheatstone's Bridge principle in which a slide wire of uniform cross-section and resistance is connected in parallel with two resistance coils (X and Y) in series, an indicator or galvanometer with a sliding contact being connected across the two parallel leads, while a battery and key are placed in the circuit represented by the slide wire of uniform cross-section and resistance. The resistance coil (X) is of a metal the resistance of which varies greatly with its temperature and if placed in the grain the resistance of this coil is dependent on (a) the temperature of the grain; (b) the area of the cross-section of the wire; (c) the length of the wire; and (d) the material of which it is composed. Since the (a) is the only variable quantity, the alteration in resistance is solely due to the rise or fall of the grain temperature. When applying this principle to the indication of the grain temperature, the arms of the bridge are arranged in such a manner that no current passes through the indicator or galvanometer when the coil X is at the same temperature as the atmosphere in the compartment containing the grain. A rise, therefore, of the temperature of the coil X causes a deflection of the galvanometer, the scale of which is calibrated to read directly in degrees of temperature corresponding to the increase of resistance. The apparatus comprises an indicator and a selecting switchboard placed in a convenient position. Small cables are run from the switchboard and connected to small coils of special wire placed every 5 or 10 feet apart in the grain. These wires are enclosed in heavy steel tubes capable of withstanding the pressure of the grain and also the pull of the grain when the bins or storage spaces are emptied. The tubes are supported at the top by a special fitting, but are free to swing at the bottom. The wires terminate at lever switches on the switchboard, designed to close the thermometer and battery circuits with one movement, the switches being clearly marked to show the exact position in the grain of the thermometers to which they are connected. The whole of the apparatus is worked by a $1\frac{1}{2}$-volt cell, and the current consumption is so small that this cell should only need replacement every 12 months.—*E. Bealing, "The Dock and Harbour Authority", Vol. XIX, No. 228, October, 1939, p. 357.*

Nordberg 6,000-b.h.p. Geared Machinery.

Four of the new C 2 type American motorships are to have propelling machinery consisting of two 9-cylinder single-acting 2-stroke Nordberg Diesel engines geared to a single propeller shaft through hydraulic couplings. The total power output will be 6,000 b.h.p. at 225 r.p.m., corresponding to a propeller speed of 92 r.p.m. The cylinders have a diameter of 21in. and the piston stroke is 29in. The engines are to be capable of developing 7,500 b.h.p. with a two-hour 25 per cent. overload. Under normal running conditions the brake M.E.P. is only 59lb./in.^{2}. The engines are of the crosshead type, but there are no piston rods, the crossheads being attached direct to the bottoms of the piston skirts. Lubricating oil is used for piston cooling, mechanically-operated lubricators being provided for oiling the cylinder liners. The scavenging air ports and exhaust ports are on opposite sides of the cylinders, air entering through automatic valves in the scavenging trunks. The rotary-displacement type scavenging-air pumps are driven by gearing at the after-end of the crankshaft. Reversing gear is provided for each of the pair of geared engines, with interconnecting device to enable the two units to be manœuvred as one. The reversing gear is of the Burmeister and Wain type, the reversing lever moving the starting-air camshaft and placing the ahead and astern starting distributor cams in the correct operating position. Moving the control lever to the starting position opens a valve which supplies air to the starting distributor and, accordingly, to the air-operated valves on the main engine cylinders. There is a brake cylinder which, when the engine is reversed, is supplied with air to prevent the camshaft moving for about 120° after the engine has turned, thus setting the fuel cams in the required position; at the same time the scavenging-air blower reversing valve is moved to correspond with the altered direction of rotation. The control lever is moved from the starting position to the running position, thereby placing all the cylinders on fuel. There is a starting-air valve on each cylinder. Interlocking gear is provided to prevent incorrect operation of the control and reversing levers. Each engine is fitted with a gear-driven centrifugal-type governor designed to maintain any required speed from 50 per cent. full power upwards, with a maximum variation of 5 per cent. A B. & W. independent overspeed governor, which is reset automatically, cuts off the fuel when the engine revolutions exceed 15 per cent. of the service speed, this governor also being capable of adjustment (while the engine is running) to limit the revolutions to 10 per cent. above and 60 per cent. below those of the operating speed at the time. When the overspeed governor comes into action due to the pitching of the ship, it limits the fuel pumps to half load; if this is insufficient a secondary trip acts on the pumps and they are placed in the "idling" position. A separate pump is fitted to each cylinder and any pump can be cut out or replaced if necessary while the engine is running. The same oil is used for the hydraulic couplings as for the forced lubrication and piston cooling systems. Motor-driven pumps draw oil from the sump tanks built, as part of the hull, between the girders below the bedplates. Oil from the hydraulic couplings is also drained into these tanks. A certain amount of oil is delivered through duplex filters and oil coolers to each engine, the remainder being discharged to an overhead tank for gravity feed to the couplings, whence it drains to the sump tanks and is recirculated. The bulk of the oil supplied is for piston-cooling purposes. The lubrication systems of the two engines are separate but can be interconnected in an emergency, while a spare pump and cooler are also installed. The cylinder jackets are cooled by fresh water, 3 motor-driven pumps and coolers being fitted for this purpose; one of the sets is for use as stand-by. The piping and valves are arranged so that any pump can be made to operate in conjunction with any of the coolers, which are supplied with cooling water by 3 salt-water pumps; these pumps also deliver water to the lubricating oil coolers. The construction of the engine embodies tie-rods which extend from the bedplate, below the main bearings, to the tops of the cylinders. These tie-rods are tightened by means of a special type of hydraulic jack, the same stress being produced in each rod. The engine bedplate is cast in two sections and carries a set of A-frames, upon which the cylinder blocks rest; rectangular crosshead guides are bolted on the inboard side of the engine between the frames. The crankshaft, which is drilled through the webs, crankpins and journals for the passage of lubricating oil, is a single forging. The auxiliary machinery includes two motor-driven starting-air compressors, each of 90 cu. ft. of free air per min. capacity, the maximum pressure being 500lb./in.^{2}. Electric current is supplied by two 215-kW. 220-volt Diesel-driven generators. The total length of the engine-room is stated to be about 47ft., the ships themselves having an overall length of 459ft., a breadth of 63ft., and a load draught of 25ft. 9in. The service speed will be 15½ knots.—*"The Motor Ship", Vol. XX, No. 237, October, 1939, pp. 244-246.*

High-efficiency Multi-cylinder Turbines.

Next to reliability in operation the two most important requirements for the working of a steam turbine are: (*a*) that it should attain a high efficiency; and (*b*) that this efficiency should be maintained. A condition for thermo-dynamic efficiency is the maintenance of low steam speeds in the turbine nozzle and blade passages. This involves an increase in the number of turbine stages and in a single cylinder turbine leads either to a large diameter shaft with consequent inter-stage gland leakage, or running through the critical speed. By adopting a two-cylinder design, however, it is possible to use a large number of stages for the utilisation of low steam speeds, the shaft diameter can be kept small and neither rotor runs through any critical speed. Furthermore the two cylinder design has the following points in its favour: (1) high efficiency due to fully staged expansion and small leakage areas; (2) the restriction of the high temperature portion of the steam expansion to one cylinder; (3) the possibility of designing each cylinder for the specific duty it has to perform and the use of simple, short and rigid castings for this purpose, as these would not be liable to distortion; (4) the shortness of the separate rotors enables the running speed to be kept well below the first critical speed; and (5) the elimination of expansion stresses by kinematically correct methods of support and location, the means adopted providing absolute freedom for expansion while still locating the parts with precision. Under operating conditions it is of the utmost importance that the turbine rotor should not run through a critical speed Where this happens the rotary period of the machine agrees with the elastic period of the rotor, with the result that severe vibration occurs if the machine is allowed to run for more than a second or two in this condition. The effect of such vibration is to increase gland clearances throughout the machine with consequent considerable increase in interstage leakage. Where a turbine is so designed as to allow the rotor to run through a critical speed, one bad start or stop is sufficient to increase the clearances to an appreciable extent, and the losses due to this increased leakage will continue until the turbine is refitted. Although the use of two cylinders and a large number of stages involves a more expensive turbine, it results in a far more efficient and durable machine, the low steam consumption of which should enable a substantial saving in fuel costs to be effected.—*"The Allen Engineering Review", No. 4, October, 1939, pp. 9,-12.*

Cable-laying Machinery.

The standard cable-handling equipment fitted in the new Post Office cable ship "Ariel" consists of a steam-driven picking-up and paying-out machine or winch, bow-gears, dynamometers with span lead sheaves, deck leads, bellmouths, spectacles and crinolines. The picking-up and paying-out gear is placed forward between the main and shelter decks with hatches to expose the cable drums when working. The cable drums are independent, with separate engines, but the design permits the coupling of the two drums in the event of either engine breaking down. This change-over can be made immediately on either drum and, in addition, both engines may work on one drum should the maximum lift be required. The engines are placed on the starboard side to leave the fore and after ends of the machine accessible, and facilitate repairs. They are of the 3-cylinder type, each capable of developing 110 b.h.p. when running at 260 r.p.m. The total power of 220 b.h.p. enables the machine to lift a weight of 30 tons at 100ft. per minute on either drum, or half this load on both drums at one and the same time. The cable drums are 6ft. 6in. in diameter and run loose on a fixed shaft. Braking is carried out by rings which fulfil a dual purpose by forming the internal spur rings for driving the drums. The brakes themselves consist of steel straps lined with elm blocks and both brakes may be brought to bear on either drum when the maximum holding power is required. Operation is by means of two handwheels on the shelter deck. Spiral gearing for one handwheel enables the brakes to be applied quickly, while the other, operating through spur gears, gives slow but very powerful control. A continuous lubricant is applied by directing water on to the drums and this also ensures minimum snatch when paying out. All gears are of machine-cut steel, muffled to avoid ringing. The gearing on the first motion shaft slides to give two speeds and a neutral position, and the drum-driving pinions are slid out of mesh when cable is to be paid out. Each drum is provided with knives for fleeting the cable on the drum with inside or outside lead, and they have forward, aft and athwartships adjustment. The hauling-off and holding gears stand on the shelter deck and are driven from the cable drums by chain drives through friction straps and free wheels in such a manner that they are driven in one direction only for picking up, and run free for paying out. The haul-off sheaves can be traversed across the face of the drum to suit an inside or outside lead. Dynamometers with span lead sheaves will be fitted forward to register the strain on the cable. These consist of sheaves carried on a slide which is free to travel up and down a vertical cylinder with a plunger inside, forming a dashpot to steady the movement. Cable leads from the tanks are in the form of single and double rollers carried on pedestals, and thees can be adjusted to suit the line of the cable. In the cable tanks special crinolines are fitted to prevent the cable from kinking during paying out. The whole of the cable machinery is controlled by an operator on the shelter deck.—*"The Shipping World", Vol. CI, No. 2,416, 4th October, 1939, pp. 369-370.*

Dutch Refrigerated Liner.

The recently delivered Holland-America Line's motorship "Sommelsdijk", is the first of two fast cargo and passenger liners designed for the company's Java-New York service. She has a gross tonnage of 9,927, with 3 holds forward of the machinery space and 3 aft. There is a continuous double bottom for fuel oil and liquid cargo. The holds are equipped with deck-controlled mechanical ventilators. Accommodation is provided for 12 passengers, every cabin having a private bathroom attached to it. The refrigerated cargo spaces comprise 6 compartments, 2 of about 1,000 cu. ft. each, 2 of 1,800 cu. ft. each, and 2 of 1,650 cu. ft. each. All of these can be cooled down to a minimum temperature of 0° F. The two small chambers are cooled by galvanized brine grids on the sides and ceiling of 4 air-cooled casings of heavy galvanized plate, containing a number of porcelain rings over which the cold brine is circulated, thus forming a large cooling surface, a special layer of smaller porcelain rings preventing any drops of brine from penetrating into the cold chambers. The air is circulated by double electric centrifugal fans, and the degree of humidity can be regulated by means of by-pass dampers. The wet brine system has the advantage of cleaning the air and retains any odours which may be given off by the cargo. The refrigerating plant consists of 3 vertical single-acting multiple-effect CO_2 compressors, directly driven at 400 r.p.m. by 23-h.p. electric motors; 3 copper coil CO_2 condensers; 3 multiple-effect receivers with counterbalanced float valves; 3 cylindrical evaporators with seamless steel coils located in an insulated evaporator compartment; and 4 electrically-driven brine pumps (one of which is a stand-by), to circulate brine at 4 different temperatures—chilling, fruit preserving, freezing and thawing.—*"The Syren", Vol. CLXXIII, No. 2,249, 4th October, 1939, pp. 20-21.*

Alternative-fuel Engines.

A true alternative-fuel engine is one which can use either of two fuels to as great an advantage as if it had been designed for that fuel alone. Means for quickly switching over from one fuel to the other are envisaged, and any engine requiring the fitment of special pistons or combustion heads for the different fuels cannot be regarded as a true alternative-fuel engine. Up to the present time, interest has centred mainly on the development of an engine that will run on town or producer gas as well as on Diesel oil. One such engine is a 3-cylinder Crossley unit developing 130 b.h.p. at 500 r.p.m., this output being obtainable with all descriptions of Diesel oil or with town gas, and with reasonable quality suction gas, sewer gas, natural gas or blast-furnace gas. The change-over from one fuel to the other is effected by simply moving over a lever while the engine is running, no change of parts being necessary and no delay entailed. If required, the engine can be operated on a combination of oil and gaseous fuels and, in fact, a minute quantity of oil must always be used to produce ignition even when gas is the normal supply. This small quantity of oil can be augmented as desired and the engine can be run on any proportioning of oil and gas up to 100 per cent. of oil. The change can be made without stopping the engine. A 4-cylinder National engine designed to run on either Diesel oil or town gas, has a continuous rating of 220 b.h.p. at 428 r.p.m. In this engine the camshaft operating gear for regulating the proportion of fuel oil to gas is fitted just above the exhaust manifold. Below the left-hand end of the latter is a gauge to indicate what proportion of the total B.Th.U. supply to the engine is contained in the oil being fed thereto. The registration of this gauge is from 10 to 50 per cent., and below the gauge is a conveniently situated control lever. When running on gas the oil required for ignition purposes is normally under 10 per cent. of the total B.Th.U. supplied, which is less than that needed to keep the engine running on "no load". However, the governor is linked up with the oil supply, so that this may be decreased if the engine should tend to race after the gas has been completely shut off.—*"The Power and Works Engineer", Vol. XXXIV, No. 400, October, 1939, pp. 413-414.*

Sterilizing Drinking Water.

Where drinking water is obtained from sources the purity of which is doubtful, sterilization in addition to filtration before use is always desirable. A simple and compact form of sterilizer which appears to be particularly suitable for use on board ship, has been developed in this country. The apparatus utilises ultra-violet rays from a mercury arc lamp as a sterilizing agent, these rays being destructive of all forms of bacteria, protozoa and other elementary organisms. The arc, which has a consumption of 700 watts, is housed in a quartz tube 2-in. in diameter and 13½-in. long, which is arranged along the axis of a brass cylinder of 4-in. internal diameter, chromium plated on the inside, the inlet and outlet connections being so arranged that the water has a whirling motion through the cylinder. The capacity of the unit is 600 gallons per hour, at which rate of discharge the exposure of the water to the rays is 1·75 seconds. The output of this sterilizer can be varied, the exposure of the water being 10 seconds for a flow of 100 gallons per hour, at which rate the rise of temperature between inlet and outlet is only 9° F. The mercury arc tube has a running life of 1,000 hours, which is equivalent to 6 months' use at the rate of 44 hours per week.—*"Shipbuilding and Shipping Record", Vol. LIV, No. 14, 5th October, 1939, p. 375.*

German Submarine Engines.

It is understood that the standard type of Diesel engine installed in modern German submarines is of M.A.N. design, with nine cylinders 460mm. in diameter, the piston stroke being 400mm. The output is stated to be 1,600 b.h.p. Double-acting 2-stroke M.A.N. engines have been designed for battleship propulsion, and a year ago an engine of this type intended to develop 16,000 b.h.p. was completed. It was being subjected to extensive trials but no information is available as to whether it has since been installed in any ship.—*"The Motor Ship", Vol. XX, No. 237, October, 1939, p. 253.*

M.I.P. and Piston Speed Limits.

The superintendent engineer of the Netherlands Steamship Company, Mr. Ysselmuiden, recently read a paper in which he referred to the owners' experience as regards the limits in mean effective pressure and piston speed in Diesel engines. He stated that an M.I.P. of 89·5lb./in.^{2} was not normally exceeded in their Sulzer single-acting engines, although the latter run satisfactorily and with a smokeless exhaust up to 96·5lb./in.^{2}. A piston speed of 1,120ft./min. is considered to be a suitable limit for passenger vessels, but in cargo ships up to 1,240ft./min. may be used as a standard speed with complete success.—*"The Motor Ship", Vol. XX, No. 237, October, 1939, p. 231.*

Increasing Existing Motorships' Speeds.

An increase in the speed of cargo vessels at the present time would be of value for two reasons; first, because a faster ship can more easily avoid attacks by hostile submarines, and secondly, because of the greater amount of cargo which it can transport annually. Several hundreds of British motorships are still propelled by 4-stroke air injection engines, the power output of which could readily be increased by from 7 to 10 per cent. by converting them to operate with airless injection. Such conversions have already been carried out abroad in about 100 cargo motor vessels with complete success. There are at least three thoroughly successful systems of conversion available and the cost (about 12 months ago) in the case of a twin-screw 6-cylinder installation of 4,000 h.p., worked out at about £200 per cylinder, or £2,500 for the whole conversion. Present-day costs would presumably be slightly higher, but if the necessary components for conversion were standardised and built for a considerable number of ships, the necessary work on board could probably be carried out in under 14 days, so that practically no delay would be caused. —*"The Motor Ship", Vol. XX, No. 207, October, 1939, p. 232.*

The Analysis of Ship Wave Resistance into Components Depending on Features of the Form.

The paper describes the calculation of the various components of wave resistance for a series of forms defined by algebraic equations. The dependence of each of these components on particular features of the form is described, and also their variation with speed. Comparison is made with the measured total resistance of five forms of the series, for which models were made and tested at the William Froude Laboratory. Good agreement is found when allowance is made for the effects of viscosity. The profiles of the wave systems along these forms were also measured, and a description of this work is given in the paper, together with the results of some calculations of the wave profiles which would occur with five models of the same waterlines as these, but of great draught with vertical sides. A general agreement in characteristic shape is found between the two sets of profiles. These results have been used to explain the changes in the speeds of the humps and hollows of the model resistance curves for the different models. A section of the paper is devoted to the comparison of these speeds with those deduced from existing theory.—*Paper by W. C. S. Wigley, M.A., "Bulletin of The Liverpool Engineering Society", Vol. XIII, No. 3, October, 1939, pp. 12-35.*

Design and Handling of Oil Engines.

The writer of a letter on this subject describes a case of defective running which occurred in two twin-cylinder, 2-stroke solid injection 55-kW. generator engines which were supposed to run 7 days each in turn, but for which 24 to 48 hours' continuous running proved to be the limit. A thorough examination of the engines revealed the fact that the fuel valves were defective. These valves were of the automatic type, being opened by the fuel pressure from the fuel pumps, and were spring loaded, the springs being set to a definite load suitable for the perfect pulverising of the fuel. The head of each valve was convex, fitting into the spring-carrying washer on to a concave surface and having a small annular clearance round the valve head. In a few months of running, the convex had hammered into the concave recess—only a very small fraction of an inch—and when refitting and testing the valves the spring-carrying washer had, almost invariably, sat on the tiny ridge made by the hammering, the testing being carried out with it in this position. After a certain number of hours' running the washer moved to its correct position, thereby slightly reducing the load of the spring on the valve. This reduction, though very small, affected the correct pulverisation of the fuel and thus caused a material reduction in the power developed in that cylinder, in addition to which it brought about general overheating. To correct this fault, the sides of the concave surface in the spring washer were eased and the surface of contact brought down to the exact diameter of the convex surface on the valve head, and thereafter, when refitting the valves, it was the practice to grind in the head of the valve as well as the mitre. The head was ground with a circular motion, as well as a revolving motion, to preserve as far as possible the convex to concave fitting. The second fault was in the fuel pumps, which were of the variable-stroke-delivery type. In these pumps, even at full load, on such a small engine the stroke is very fractional, and to do full justice to the engine must fill the barrel completely. This necessitates special attention to the lift of the valves, and this lift is critical, particularly in the case of the delivery valve. On these engines the fuel pump valves were not adjustable and, in the course of wear, increased the lift to such an extent that even if the governor opened the pumps "all out" the supply of fuel delivered was still insufficient to support the load. The loss was due to the amount of fuel returned through the delivery valve during its closing, leaving a slack line to the fuel valve which had to be made up on the next delivery stroke before the pressure in the fuel line reached injection point. This trouble was eliminated by removing the solid caps of the delivery valves and fitting caps with adjustable stoppers. In cold weather another fault was experienced. The fuel heating system was only carried to the service tank from which quite a long line conveyed the fuel to the pumps, and during its passage from the tank to the pump the fuel had time to cool off to such an extent that the pumps would not handle it. As soon as the heating was carried right up to the pumps no further trouble was experienced. When the foregoing defects were remedied the engines gave 2 years' steady running, week and week about, without an involuntary stop. A point of purely poor design in these engines was the provision of no more than one strainer in the lubricating system. This meant that until the staff became well acquainted with the engines they were shut down merely to clean a strainer. Moreover, this single strainer was fitted in such a position that the cover was below the level of the oil in the engine sump, which entailed a loss of oil every time the strainer was cleaned.—*D. P. Peel, "Mechanical World", Vol. CVI, No. 2,756, 27th October, 1939, p. 391.*

The Relationship Between the Mechanical Properties of Materials and the Liability to Failure in Service.

The customary industrial tests of materials give valuable information regarding their quality but none of the mechanical properties of materials so determined is an important factor in a normal failure in service. Thus a measurement of ductility, by any of the usual methods, is made after the material has been stretched to the point of failure, whereas service failures take place without appreciable deformation. Moreover, since such failures are not caused by a single application of a static load, a few tons difference in the ultimate strength, in the absence of special circumstances, does not determine which of two materials will fail. A relatively low yield point, though it may be objectionable in so far as it causes a permanent set in a part working with a fine clearance, is not the cause of direct failures; in fact a low yield point may be an advantage. A fatigue test on a cylindrical specimen approximates more closely to working conditions than a tensile test, but being made on a polished specimen does not show the relative behaviour of materials under concentrated stress. The notched-bar test, in the manner it is applied, does not show which of two materials is the less likely to fail, while the bend test, although capable of revealing objectionable defects in the metal, is by its nature insufficiently severe to determine the extent of these defects—moreover, the latter do not represent the mechanical properties of the material. The usual failure is due to faulty design or workmanship, carelessness in operation, or such causes as vibration. The relative suitability of two materials depends on the service conditions. The relative fatigue value should be obtained from notched specimens, as notch sensitivity is of importance. It can also be assessed from the early portion of a load-extension diagram. A somewhat similar measurement can be obtained from a load-extension diagram in the usual static tensile test on an unnotched specimen, as this furnishes the capacity of the material to deform without appreciable increase in the load, the property on which the behaviour of a part subjected to concentrated loading depends.—*Paper by L. W. Schuster, M.A., "Bulletin of the Liverpool Engineering Society", Vol. XIII, No. 4, November, 1939, pp. 11-47.*

Adequate Testing of Water Gauges.

As the result of a preliminary investigation of the causes of an explosion which occurred on the 24th September, in the boilers of the 4,289-ton Cardiff steamer "Nailsea Lass", the Engineer Surveyor-in-Chief of the Ministry of Shipping makes the following observations: "The shortage of water which caused this explosion occurred after the valve on the boiler shell controlling the steam to the water gauge had been shut. The valve controlling the water to the water gauge was left open and the steam condensing in the upper pipe allowed the water to rise and drown the gauge, which was of the hollow column type. Any testing of the gauge thereafter would give a full-bore discharge of water unless the double shut-off method was used, but this method is not commonly used unless there is reason to question the accuracy of the water level indicated by the gauge. Suspicion, however, should most certainly have been aroused by the fact that the boilers had apparently been run for over 16 hours on auxiliary services without extra feed and without bringing the water into sight in either glass, but it is not improbable that the chief engineer may not have been aware of these facts. It is noted that the fitting of valves in lieu of cocks on the boiler for the gauge pipes is unusual and in fact contrary to the present rules of Lloyd's Register. Valves used for this purpose are more liable to derangement through choking, and it is not so readily seen whether a valve is open or shut, but none of these disadvantages need be considered as having contributed to the explosion. It may well be, however, that the junior engineers who "shut down" on the boiler tops on the afternoon previous to the explosion and who had been in the ship only a few days, did not recognise the valves as connected with the water gauges if they had only seen cocks fitted for this purpose in their previous experience, and they may therefore have shut down the gauge valves under the impression that they controlled whistle steam or some other unidentified service. Through allowing the water level to fall below the furnace crowns dangerous conditions were produced, and when the furnace plates ruptured the discharge of water and steam into the stokehold injured two firemen. No exception is taken to the arrangements in the ship and the explosion only emphasises the grave consequences of neglect of adequate testing of water guages".—*"The Journal of Commerce", (Shipbuilding and Engineering Edition), No. 34,892, 30th November, 1939, p. 3.*

New American Tug with Diesel Engines.

The Sheridan Towing Company's new motor tug "D. T. Sheridan" is an all-welded steel vessel for ocean, coastal and harbour towing work. The tug's main dimensions are 116 by 26 by 13ft., and the entire hull space is utilised for the machinery, oil and water tanks, and store-rooms, all living quarters being in a deck house above the main deck. A feature of this deck house is that it enables every compartment in the ship to be reached through internal passages without going on the upper deck. The hull is divided into 7 watertight compartments, of which the foremost (or No. 1) is a forepeak water ballast tank, No. 2 contains double-bottom and wing tanks for 9 tons of fresh water together with bins for deck stores, No. 3 is a galley store room with 25 cu. ft. of refrigerated space, No. 4 contains an 86-ton fuel-oil tank, No. 5 is the engine room, No. 6 contains double bottom and wing tanks for 28 tons of fuel oil, and No. 7 is the after peak containing a reserve fresh water tank with a capacity of 11 tons. Apart from the interior passages throughout the vessel, other special features comprise complete insulation of the engine room, absence of exposed wiring or piping and thorough insulation of the hull against heat, cold and vermin. The main propelling machinery consists of a 6-cylinder Fairbanks-Morse 2-stroke Diesel engine rated at 1,000 h.p. at 275 r.p.m., directly coupled to the propeller shaft. The engine is directly reversible, with oil-cooled pistons, open head combustion, differential fuel injection valves and back-flow scavenging. A 40-kW., 125-volt d.c. generator is driven from the flywheel by a V-belt and works in combination with a 375-amp.-hr. battery and automatic voltage regulators to maintain the current supply while the main engines are in use. There are also two F. M. Diesel generator sets, each consisting of a 6-cylinder 60-h.p. Diesel engine direct coupled to a 40-kW. generator. Each of these engines is provided with its own starting battery. The whole of the engine-room auxiliaries are motor-driven, as is the deck machinery which includes a 15-h.p. windlass.—*"Motorship and Diesel Boating", Vol. XXIV, No. 11, November, 1939, pp. 552-555 and 578.*

Ferro-concrete Floating Dock.

A new floating dock of reinforced concrete has been completed at the Marty Shipyard in Odessa. The dock is 426½ft. long, 100ft. wide and 48ft. high, the lifting capacity being 6,000 tons. The machinery and equipment are stated to be thoroughly up-to-date and include electric welding and air compressor plant. The dock has its own boiler installation and is claimed to have been designed and built by young Soviet engineers. Successful tests are stated to have been carried out with the dock, the new Diesel-electric ship "Trud" weighing 5,400 tons having been raised in it.—*"Fairplay", Vol. CLII, No. 2,951, 30th November, 1939, p. 300.*

Aeronautics and Ship Design.

The paper is intended to give a brief summary of some of the results of the vast amount of aeronautical research work carried out in recent years in so far as they apply to the problems confronting the naval architect. The experimental work dealt with by the author includes three main sections, *viz.*: (*a*) propellers; (*b*) structure; and (*c*) miscellaneous researches. In regard to (*a*), the author discusses the vortex theory, prediction of performance and pressure distribution, optimum efficiency, effect of number of blades, a proposed experimental method, and high speeds. Under section (*b*), the paper deals with the stability of the stiffened plates of aircraft structures under compression, the effect of curvature, combined end and transverse loading, and concentrated loads. The miscellaneous matters dealt with under section (*c*) comprise skin friction and boundary layer control, and the theory of planing craft.—*Paper read by J. Lockwood Taylor, D.Sc., at a General Meeting of the N.E. Coast Institution of Engineers and Shipbuilders, on the 24th November, 1939.*

Opposed-piston Engines for Submarines.

Some of the new U.S. submarines are being equipped with engines of the opposed-piston type. They are of Fairbanks-Morse design, and the standard unit is an 8-cylinder model with cylinders 8in. in diameter, the piston stroke being 10in. and the engine developing 1,000 to 1,300 b.h.p. at 720 r.p.m. The upper pistons are driven from an overhead crankshaft which is itself driven by a vertical shaft and helical gearing. Scavenging air is supplied by a rotary blower driven from the upper crankshaft by means of helical gearing.—*"The Motor Ship", Vol. XX, No. 239, December, 1939, p. 314.*

Extracts.

The Council are indebted to the respective Journals for permission to reprint the following extracts and for the loan of the various blocks.

The Velox Boiler.

By N. I. Poushkin.

Translated from "Soudostroienie", October, 1939.

The Velox boiler is, at the present time, the most formidable competitor of the latest types of natural circulation boilers as well as of the La Mont boiler. As the construction and operation of the Velox boiler have already been dealt with in the technical press, it will be sufficient to note some of the more characteristic features of the principle on which it works.

(1) The burning of fuel under a pressure of from 37 to 50lb./in.^{2} maintained in the combustion chamber by a turbo-blower.

(2) The high velocity of the flue gases (660 to 1,000ft./sec. or more) in the evaporating and superheater elements, and of the water in the feed-heater elements (about 330ft./sec.) due to the relatively high air pressure maintained by the turbo-blower.

(3) The high rate of heat-transfer due to the high velocity and density of the flue gases, the coefficient of heat transfer in the evaporating and superheater elements being of the order of 92 to 129 B.Th.U./h.p.-hr.

In addition to these special features the Velox boiler has a number of others which, although likewise found in several other boilers, are more highly developed in the Velox than in most other types. Among these features the following may be included :—

(*a*) Complete automatic control of the supply of feed water, fuel and air.

(*b*) The adjustable atomizing oil burner which enables perfect combustion to be maintained even when the air supply is relatively small. The Brown-Boveri burners used in Velox boilers are of such a capacity that a single burner suffices for a boiler with an evaporative capacity of 120 tons/hr.

(*c*) Almost complete freedom from carbon formation on the combustion chamber walls due to the screening of the latter by the close-fitting evaporating elements.

(*d*) The use of forced circulation.

(*e*) The efficiency of the steam separation.

(*f*) The rapidity with which the boiler can be brought into operation and the ease with which it can be controlled while the engines are being started, stopped or subjected to variations in load.

Although the Velox boiler has the advantage of being considerably smaller and lighter than drum-type water-tube boilers of equivalent capacity, it must be admitted that it also possesses several apparent disadvantages, viz. :—

(1) The large size (6-in.) tubes of the Velox boiler render it impracticable to make use of the high working pressures employed in the small diameter (1 to 1¼-in.) tubes of certain types of coiled-tube boilers, since these can be made with thicker walls than could be utilised in the case of the large diameter tubes of the Velox boiler.

(2) The fact that in addition to the boiler itself it is essential to use an air compressor driven by a gas turbine and a steam turbine together with a number of pumps and other mechanical appliances for operating the boiler, the overall reliability of the installation throughout a lengthy working period might be considered inferior to that of the ordinary type of boiler.

(3) A lightweight boiler of small dimensions and very high capacity is an uneconomical proposition from the point of view of design, as the maintenance of a high air pressure in a totally enclosed combustion space of limited dimensions coupled with the need for a high velocity of the flue gases and flow of steam through tubes of restricted diameter involves a reduction of some of the most important advantages claimed for the Velox boiler. The highest working pressure yet employed in such a boiler is 1176lb./in.^{2} and the maximum evaporative capacity—80 tons/hr.

In comparing the weight and size of a Velox boiler with those of a standard water-tube boiler, it is necessary to take account of the separator, gas turbine, forced-circulation pump, independent feed-water heater, the piping of the gas turbine and compressor, the combustion chamber under constant pressure, etc., all of which have an appreciable effect on the total weight and size of the Velox installation, more especially at high working pressures. The above components are, nevertheless, essential for the efficient working of the boiler and their various functions are an important feature of the Velox boiler, as it depends on them for its operation.

(4) A possible disadvantage of the Velox boiler for power stations and similar purposes is that it involves the use of expensive oil fuel or gas (or a mixture of both). No pulverised fuel installations for Velox boilers are in operation and, so far as is known, none are contemplated in the near future.

The Velox boiler may be said to possess the following advantages over the ordinary marine water-tube boiler :—

(*a*) It enables a slight saving in weight and space to be effected.

(*b*) Better manœuvrability.

(*c*) Less time required for raising steam.

(*d*) Evaporative efficiency remains practically constant under all conditions of loading between 33 and 100 per cent. of full load.

(*e*) Tubes of boiler and superheater may be easily renewed if necessary.

(*f*) Air required for combustion delivered direct into combustion chamber.

(*g*) Higher boiler efficiency.

(*h*) Higher evaporative capacity and higher rate of heat transfer.

In warships with a relatively low freeboard, the Velox boiler must be disposed horizontally in order to give access to the evaporating and superheater elements for examination, cleaning and repairs, etc. Such an arrangement involves a correspondingly greater floor space, in addition to which there is a departure from the principle of the upward flow of steam, leading to a less satisfactory distribution of water and steam in the collectors of the evaporating and superheater elements, thus nullifying many of the advantages of the Velox boiler. The necessity for disposing the steam generator itself horizontally means that the steam in the horizontally-arranged lower evaporating elements is, in the course of its upward flow, only separated from the high-temperature flue gases by the thickness of the tube walls and as steam is a less efficient conductor of heat than water it absorbs less heat from the tube walls and therefore renders the latter more liable to overheat. In order to eliminate this risk it becomes necessary to increase the rate of evaporation and the velocity of the steam flowing through the superheater outlet. This velocity may, under such circumstances, be of the order of 160-200ft./sec.

It may, therefore, be seen that at the present time there are still serious difficulties to be overcome before high-capacity Velox boilers can be installed in warships.

Furthermore, high-capacity boilers of this type must have very long tubes and the amount of space available in the boiler room must be made large enough to permit of their withdrawal, so that the advantage of having a boiler of relatively small dimensions is thereby lost. A Velox boiler of very high evaporative capacity (120 to 150 tons/hr.) may, therefore, appear unsuitable for installation in a large warship on this score.

In order to overcome these objections it has been proposed that one of the two following arrangements might be adopted:—

(1) To use two horizontal boilers with the stokehold between them and with their tube ends opposite each other so as to save the length of a tube of one boiler in the floor space of the stokehold common to both, or

(2) To locate the boiler and turbines in a single compartment (as is sometimes done in power stations ashore), arranging the boiler opposite the condenser.

It must, however, be remembered that in ships with a limited amount of head room the question of access to the evaporating elements of *vertical* Velox boilers is liable to cause difficulties. The steam and water separator of the boiler must, in any case, be arranged vertically. Existing designs of such separators for reducing the moisture of the saturated steam prior to its entry into the superheater make it necessary to keep the saturated steam outlet from the separator well above the water level inside the latter. Moreover, the small amount of water in the evaporating elements of the Velox boiler makes it essential to provide for a relatively large water space in the steam separator in order to ensure the safety of the boiler when steam is suddenly shut off from the engines or when the load on the boiler is altered. This, in its turn, leads to an increase in the depth of the water space of the separator. Lastly, since the lower part of the separator serves as a reservoir for the feed water, it is necessary to keep the feed-water discharge of the circulating pump well below the outlet from the separator in order to provide for the efficient operation of the boiler circulating pump. All these requirements make it imperative to have a separator of considerable height and its installation in a boiler room of a ship with a limited amount of head room is therefore apt to be a matter of some difficulty.

A vertical arrangement of the separator is, of course, to be preferred from the point of view of the amount of floor space required for the boiler and there are a number of La Mont boilers in operation with horizontal separators which are at a very definite disadvantage in this respect.

The arrangement of the Velox boiler economiser and the gas-turbine exhaust pipe (with a diameter of 24 to 27in. for a boiler of 40 tons/hr. evaporative capacity) as a separate unit involves a considerably greater space than is the case with the economisers of natural circulation boilers or of La Mont boilers having their economisers inside the boiler casing.

The application of forced draught to a stokehold and boiler furnace enables a greater quantity of fuel to be burned within the latter with a corresponding increase in the output of the boiler without necessarily raising the temperature of combustion. Forced draught is used in the Velox boiler in the same way as in most other boilers, but in considering it as a means for increasing the output of the boiler, it must be remembered that:—

(*a*) the high density of the gases or air in confined spaces of limited dimensions (the cylinders of a heat engine, the combustion chambers of boilers, etc.) makes it possible to deal with very large volumes of such gases, their high density enabling the heat transfer between them and the walls of the chamber in which they are confined to be effected under the most favourable conditions, and (*b*) the admission of air under pressure to the furnace ensures more complete combustion of the fuel with a relatively small excess of air ($\alpha = 1{\cdot}2$).

Every form of forced draught involves compression of the air and a corresponding amount of mechanical energy, and this fact raises the question as to whether it is economic to use forced draught at the expense of energy abstracted from the system as a whole. Obviously this must, to a great extent, depend on the manner in which the requisite energy is obtained from the system and in many installations it has proved highly profitable to make use of energy obtained in one form or another from the working process of the system for the purpose of compressing the air blown into the latter with a relatively small amount of mechanical loss.

In the case of the Velox boiler this energy is obtained by the expansion of high-temperature (900° to 1100° F.) flue gases in a gas turbine. Supercharged Diesel engines use gases in the blower which have already performed useful work in the cylinder, the energy of the exhaust gases being used to create a constant flow of air through the latter. In the Velox boiler the gas turbine is driven by flue gases at a temperature best suited to the blades of its rotor (900° to 1100° F.), so that in effect the energy absorbed by the forced draught unit is obtained from the flue gases which have already done useful work in the boiler. These gases are at approximately the same pressure as that required for the air blown into the combustion chamber but at a much higher temperature, so that for any given loading of the boiler all that the gas turbine is called upon to do is to impart the requisite velocity to the air blown into the combustion chamber. In practice it is not possible to make use of flue gases *above* a certain temperature in the gas turbine owing to the temperature limitations imposed by the material (heat-resisting steel) used for turbine blades at the present time. For this reason the power developed by the gas turbine of the Velox boiler would be insufficient to compress the whole of the combustion air required by the boiler when highly loaded and it becomes necessary to supplement it by some other prime mover (steam or electric) which is also used for starting-up the gas turbine and for control purposes. The turbo-blower of the Velox boiler is, therefore, a unit made up of a gas turbine and an electric motor or steam turbine arranged on a single shaft with a turbo-compressor. Once the gas turbine has been started no special regulation of its speed is called for this varying with the flow of the flue gases. Over-speeding of the compressor is guarded against by an automatically-operated valve which by-passes part of the gases from the turbine in the event of excessive pressure. The output of the turbo-blower varies according to its speed and this in its turn is varied automatically with the loading of the boiler. Thus any increase in the amount of fuel consumed by the burner leads to a correspondingly greater generation of flue gases accompanied, in the first instance, by an *initial* rise in their temperature. This causes an increase in the speed of rotation of the gas turbine and a correspondingly greater output of the turbo-compressor or blower, more air being delivered to the boiler combustion chamber at an increased pressure. The electric motor or steam turbine coupled to the gas turbine is automatically cut out of operation when the speed (and power) of the gas turbine attains a predetermined minimum.

The speed of rotation of the turbo-blower unit or the pressure of the air delivered to the combustion chamber enable the actual loading of a Velox boiler to be determined at any time with a high degree of accuracy.

It is worth noting that any overloading of a Velox boiler due to an increased output of the turbo-blower does not involve any risk whatever for the safety of the boiler, which differs in this respect from the standard type of boiler where any overloading caused by an increased rate of consumption of fuel and air supply to the furnace leads to a corresponding rise in the temperature of the flue gases and consequent overheating of the heating surfaces. For this reason the degree of overloading to which an ordinary boiler can be subjected is at best strictly limited, in addition to which the time during which it can be maintained must necessarily be of very short duration in order to avoid any risk of overheating the boiler.

In the Velox boiler the increase in the rate of fuel consumption under forced draught involves a corresponding increase in the amount and pressure of the air delivered into the combustion chamber, thereby ensuring an excess of air within the latter under all conditions of working and preventing any undue rise in the temperature of the flue gases and consequent overheating of the boiler tubes. As the excess of air supplied to the furnace is maintained at a practically constant rate, the combustion of the fuel is at all times almost perfect.

An increase in the rate of heat-transfer in a forced-circulation boiler like the Velox does not involve a corresponding rise in the temperature of the flue gases and walls of the boiler tubes. Local overheating of the kind to which ordinary water-tube boilers are so prone can hardly ever occur in the Velox boiler.

Owing to the high rate of heat transfer maintained in the boiler under all conditions, the overall efficiency of the Velox boiler (and of its auxiliaries) remains practically constant, in fact there is a slight increase in efficiency when the boiler is forced, whereas in ordinary water-tube boilers there is, as is well known, a marked reduction in overall efficiency when the boiler is overloaded.

A Velox boiler designed for a definite maximum output is capable of maintaining it over a very long period. In practice, the only limitation imposed on the Velox boiler's overload capacity is that of its associated turbo-blower unit. There is, of course, a limit beyond which the speed of rotation of the latter cannot be increased (*viz.*, above the critical speed) owing to the limitations imposed by the materials of which the mechanism is constructed, so that in effect the actual output of the boiler is only limited by that of its forced-draught apparatus.

Apart from the provision made to guard against over-speeding the compressor by the fitting of an automatically-operated by-pass valve for the flue gases delivered to the gas turbine, the temperature of these gases can, if required, be maintained below a certain maximum by injecting water prior to their entry into the gas turbine. The limits of this temperature imposed by the heat-resisting capacities of present-day steels is of the order of 1,000° to 1,100° F.

The outstanding characteristics of the operation of the Velox boiler may, therefore, be summarised as follows:—

(1) The burning of the oil fuel under air pressure maintained by a turbo-compressor ensures perfect combustion and an extremely high rate of heat transfer due to the possibility of effecting the necessary mixture of the particles of fuel with the air forced into the combustion chamber throughout a greater range of air pressure drop than is possible in the case of ordinary water-tube boilers. The forced draught blower of a Velox boiler operates efficiently even when the pressure of the air dealt with falls to only 24in. of water. The gas turbine of a high-power Velox boiler normally deals with flue gases at a temperature of round about 1,000° F., and is capable of delivering all the air required for the purpose of combustion at a pressure of 37 to 45lb./in.^{2}. After their passage around the evaporating and superheating elements the flue gases entering the gas turbine are still at 75 per cent. of the discharge pressure of the turbo-compressor.

The power absorbed by the turbo-blower unit in the process of compressing the large volume of air passing through it to as much as 45lb./in.^{2} amounts to something like 25 per cent. of the total power represented by the evaporative capacity of the boiler, so that in the case of a Velox boiler of 75 to 80 tons/hr. capacity, it is equivalent to fully 6,000 h.p. Obviously, associated auxiliary machinery requiring anything like this power could not be operated in conjunction with a water-tube boiler of any other type without bringing about a very appreciable reduction in the overall efficiency of the boiler plant, but a series of exhaustive tests carried out with Velox boilers have proved that the overall efficiency of the boiler and its associated auxiliaries (excluding the feed pump) is, under normal operating conditions, of the order of 90 per cent.

Designers have been quick to appreciate the thermal advantage to be gained by the use of combustion under pressure, since the energy absorbed by the flue gases in the process of compression and their subsequent cycle of operations, is almost wholly converted into useful work. The energy absorbed by mechanical and frictional losses, as well as the amount of heat lost by radiation through the casings of the compressor and turbine, are negligible.

The high degree of compression to which the whole of the air passing through the turbo-blower unit into the combustion chamber is subjected raises its temperature to such an extent as to make it unnecessary to fit an air pre-heater.

(2) Another distinctive feature of the Velox steam generator is the extraordinarily high heat-transfer rate achieved by the process of combustion under pressure owing to the high velocity and density of the flue gases flowing around the evaporating elements as well as around superheater and feed water-heater tubes.

The effect of both these factors on the heat-transfer rate within the boiler has been discussed in several articles which have appeared in the German technical press, in addition to which they formed the object of a number of experiments relating to the behaviour of flue gases of high density at high velocities, carried out by Brown, Boveri & Company in Switzerland, between 1927 and 1930.

The results of these experiments have not been made public, but it may be stated that they, together with those of the researches made into the possibility of fuel combustion under air pressure, formed the basis of the principle on which a new type of boiler—subsequently to be known as the Velox—was designed and developed.

(3) In common with several other types of boiler the Velox makes use of forced circulation and the design includes a steam and water separator of the centrifugal type, the lower part of which acts as a feed-water reservoir. The water from the latter is pumped through the evaporating elements by the boiler circulating pump at a pressure slightly higher than the working pressure of the boiler. No retarders are fitted in the tubes.

(4) In addition to the above distinctive features of the Velox boiler, the complete automatic control system with which it is provided renders the operation of the boiler in service a matter of extreme simplicity.

(5) The Velox boiler responds more rapidly to a sudden change of load than any other form of steam generator. In the case of an installation comprising a Velox boiler and steam turbine with a condenser, the entire plant operates as a single self-contained unit, the control of which for manœuvring purposes is remarkably simple and effective.

Characteristics and Speed Control of the D.C. Motor.

"The Allen Engineering Review", October, 1939.

Many industrial and marine drives demand a motor having a range of operating speeds, and in this article we are reviewing the factors affecting the selection of appropriate plant.

In designing an electric motor, one is faced with the task of providing a machine to develop a given output at a specified speed, which will perform its duty cycle without exceeding a certain "approved" temperature rise. By "approved" we mean a figure selected by the purchaser or complying with some accepted standard.

It will be the designer's aim to employ his material to the best advantage and, consistent with sound engineering practice, the amount of iron and copper used will be kept as low as possible so as to avoid producing an unwieldly and inefficient machine.

Margins in respect of overloads, both electrical and mechanical, are allowed, and a few machines are built which do not comply at least with the figures in B.S.S. 168/1936.

Now, the performance of a motor can be considered under two heads. Firstly, the amount of electrical energy it can efficiently convert to mechanical power and secondly, the speed at which it will run.

With any d.c. motor, the following simple relationship holds :—

$$\text{R.P.M.} = \frac{\text{Armature Volts}}{\text{Field Strength}} \times \text{a constant.}$$

Thus, assuming the supply voltage to remain unaltered, the motor speed may be said to vary in direct proportion to the voltage impressed on the armature and inversely as the strength of the field. The value of the constant will depend upon the design of motor, and will probably be different in every case.

METHODS OF SPEED CONTROL.

It will be seen from the above that the principle of speed control consists in varying either the armature voltage or the field strength, and these operations can be performed in a variety of ways.

The magnitude of the armature voltage can be controlled by :—

(1) Resistance connected in series with the armature.

(2) Combination of series resistance with an armature diverter.

(3) Ward-Leonard control, *i.e.,* connecting the armature to a variable voltage supply.

Similarly, the field strength may be altered at will by the use of :—

(4) A shunt regulator.

(5) Field grouping.

(6) Field diversion.

The practicability of adopting any of these methods in a specific instance will, of course, depend entirely upon the suitability of the motor for variable speed working and upon the type of winding employed.

The methods of field control referred to, result in increased motor speeds, by a reduction in field strength and it might be argued conversely that lower speeds could be produced by "field strengthening". According to our simple formula this appears to be possible but is rendered impracticable by the joint limitations of temperature rise on the field coils and magnetic saturation in the field system.

The general effect of field regulation is not unlike the case of the domestic water tap which can be used to vary the flow up to a maximum determined by the size of the pipe. To increase the quantity drawn from a constant pressure main, a larger pipe must be used. Similarly, the field strength of a variable speed motor can be reduced at will, but any increase to produce a slower speed necessarily involves the use of a larger magnet system, *i.e.,* a larger motor.

When considering electrical equipment for variable speed duty, a knowledge of the nature of the loads is of fundamental importance to the designer.

As is well known, the majority of drives fall into two well-defined categories, these being :—

(*a*) "Constant Torque", or "machine" load, where a constant turning effort is required throughout the speed range.

Since horsepower is a product of speed × torque, it follows that the horsepower in this case will vary in direct proportion to the speed, the torque being constant.

Constant torque drives include reciprocating machinery, compressors, machine tools, printing presses, etc.

(*b*) Centrifugal load, where the torque required by the load varies as the second power of the speed. In this case it follows from the relationship quoted above, that the horsepower developed will vary as the cube of the speed. The majority of fans and centrifugal pumps constitute examples of centrifugal load.

MOTOR CHARACTERISTICS.

With a knowledge of the requirements of a specific application we can proceed to choose a suitable motor. In a previous article (January, 1939) the types of motors were discussed in relation to the different enclosures available and the duties for which they are considered particularly well suited. The operating characteristics of the motor will, of course, depend upon the manner in which it is wound, the usual windings being classified under their well-known names of "shunt", "series", and "compound".

SHUNT MOTORS, have their field coils connected across the mains when the motor is running and the excitation or field strength remains approximately constant. As the armature of the motor is connected across the constant voltage mains also, it will be seen that according to our simple formula quoted above, with both field strength and armature voltage constant, the speed should be constant too, irrespective of load.

In practice, the motor speed will be found to vary slightly as load is applied, and most commercial machines are provided with a series stability winding for the purpose of obtaining a slightly drooping speed-load curve similar to curve in Fig. 1, thus securing stable performance.

The slight drop in speed between no-load and full-load conditions is not enough to be inconvenient for most ordinary uses and a shunt motor is regarded as having an inherently constant speed. We find the term "shunt characteristic" appropriated

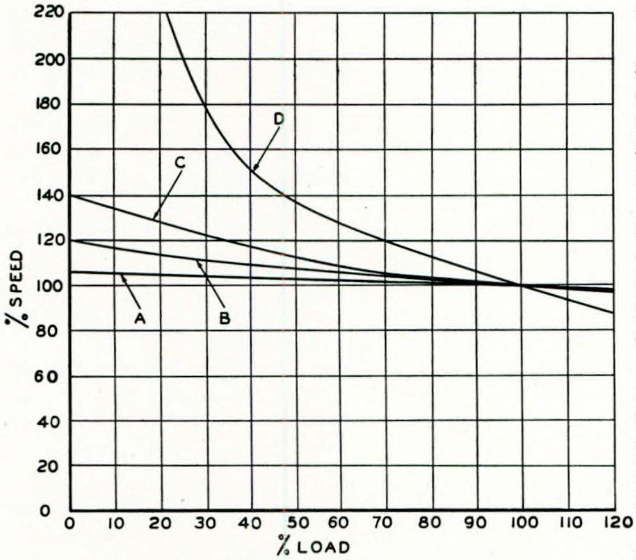


FIG. 1.—Speed load curves of d.c. motors.
Curve A. Plain shunt-wound motor.
Curve B. Compound motor with 20% compounding.
Curve C. Compound motor with 40% compounding.
Curve D. Series wound motor.

and used in connection with certain types of alternating current motors, where a similar speed-load curve is met with.

SERIES MOTORS differ from shunt machines in that the field windings are designed to carry the full-load current of the motor and are connected in series with the armature. The field strength is thus dependent upon the current taken by the armature, since the same current passes through both.

In the series motor, as in the shunt, the armature voltage can be considered to be constant and a glance at the formula previously referred to will show that the speed should vary in inverse proportion to the field strength. In practice it does not quite follow this law owing to the effect of the varying permeability of the magnetic circuit, but for our present purposes we can regard the speed of the series motor as being inversely proportional to the armature current.

For the majority of industrial applications shunt or light compound motors are used, but the pronounced inverse speed-load characteristic of the series motor is, however, particularly well suited to the requirements of traction service and for cranes, hoists and winches, where heavy overloads are common and a motor with a non-overloading characteristic combined with a high starting torque is ideal.

In most of these cases gearing is interposed between motor and load, and this ensures that the motor cannot run absolutely light and attain a dangerous speed which it would otherwise tend to do. With cranes and winches the increase in speed at low loads is most advantageous in that it permits the handling of an empty skip or hook at about twice the normal full-load speed. The speed-load characteristic of a typical series-wound crane motor is that shown in Fig. 1, Curve D.

COMPOUND MOTORS, as their name suggests, are a combination of the two primary types already described. It is convenient to consider a compound machine as a shunt motor with a series winding added, the compounding being heavy or light according to the duty for which it is intended.

The addition of a series winding to a plain shunt-wound motor results in a modified speed-load characteristic, the amount of deviation from the flat speed-load curve of the shunt motor depending upon the degree of compounding (*i.e.,* the ratio of shunt to series excitation) employed.

A light compound motor is generally understood to be a shunt motor with sufficient series turns on the field to result in a drop of speed below the no-load speed, sufficient to give it a non-overloading characteristic when used with a variable load such as a pump.

Many types of compressor require a motor capable of developing a high starting torque, and for this purpose a heavily compounded motor is frequently employed. As will be seen from curve C in Fig. 1, the drop in speed between no-load and full-load is very considerable, but having regard to the fact that on a drive of this sort, the motor load cannot fall below a certain minimum, representing the energy required to rotate the machinery to which the motor is coupled, it follows that the inherent speed regulation is not objectionable.

There are, of course, applications where a high starting torque is required and yet the load fluctuates and the resulting speed variation with a heavily compounded motor would be inadmissible.

We have in mind the ordinary planer drive, where it is customary to fit a heavy series starting winding, this being cut out when the motor is up to speed, leaving it running as a plain shunt machine with the usual flat speed-load characteristic.

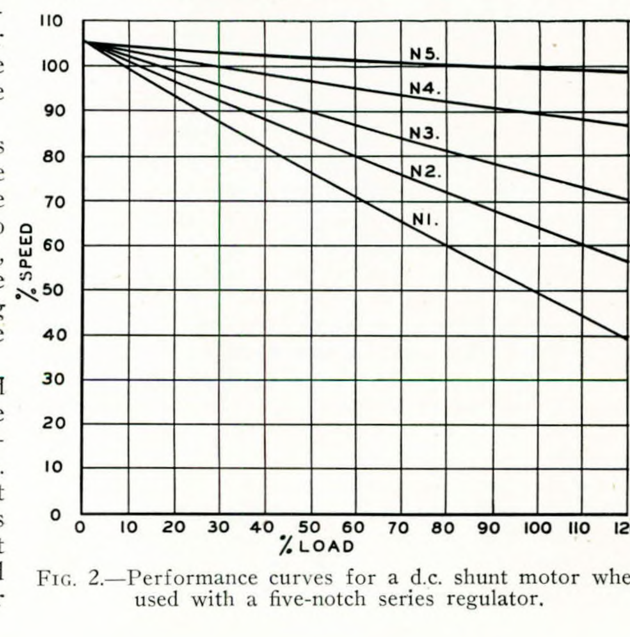


FIG. 2.—Performance curves for a d.c. shunt motor when used with a five-notch series regulator.

SPEED CONTROL OF SHUNT MOTORS.

When discussing methods of speed control we referred to six ways in which this can be done, and of these Nos. 1 and 4 are the most common where shunt motors are concerned.

METHOD 1.

Insertion of resistance in the armature circuit to reduce armature voltage.

This method is often used to obtain speeds below the "normal" speed at which the motor will run under full-load conditions. It can be used to give reduced speeds on a variety of drives such as fans, pumps, compressors, printing presses, etc., but it must be borne in mind that a considerable amount of power may be lost in resistances on constant torque loads.

The effect of inserting resistance into the armature circuit is to modify the speed load characteristic of the motor. The curves in Fig. 2 are drawn to

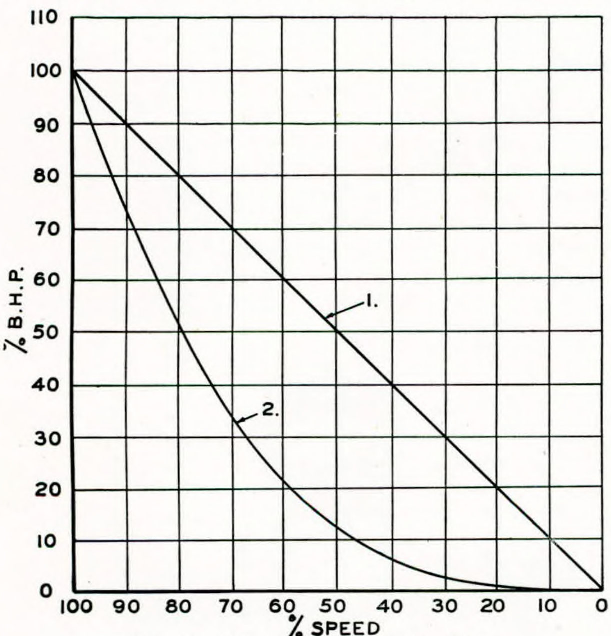


FIG. 3.—B.h.p.—speed relationship.
Curve 1. Constant torque drives.
Curve 2. Centrifugal pumps and fans.

represent the altered characteristic obtained with a five-step series regulator designed to give approximately equal speed increments with a constant torque load. It will be noticed that with the resistance adjusted to give the correct speeds at full-load the motor will run faster on loads below full-load and correspondingly slower on overload. With this scheme of control it is essential that the resistances be designed for the correct working load in order to obtain satisfactory operation.

The curves in Fig. 3 have been prepared to show the relationship between b.h.p. and speed for both constant torque and centrifugal loads, and from these, diagrams indicating the approximate amount of power lost in resistances have been calculated, and are illustrated in Fig. 4. With constant torque drives it will be seen that the power lost is the same

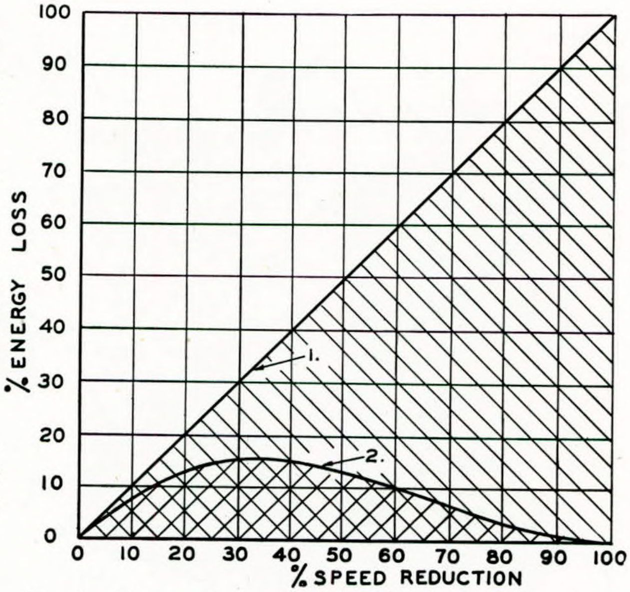


FIG. 4.—Rheostat losses with series regulation. The shaded areas below the curves represent the power lost in resistances for a given reduction in speed.
Curve 1. D.c. shunt motor on constant torque drive.
Curve 2. D.c. shunt motor on centrifugal load.

percentage of the total input as is the speed reduction of the full-load speed; *i.e.,* 25 per cent. speed reduction represents 25 per cent. energy lost in resistances.

With centrifugal loads the maximum energy dissipated in resistances in 15·4 per cent., and occurs at a speed reduction of 33⅓ per cent.

An interesting example of series regulations is that shown in Fig. 5. The equipment shown is a heavy-duty drum type combined starter and series regulator, with resistances housed in a ventilated compartment attached to the back of a totally-enclosed controller. It is used with motors up to 20 h.p., and the resistances are usually designed to give a speed reduction of 50 per cent. against full-load torque.

METHOD 4.

Insertion of resistance in the field circuit, by means of a shunt regulator to reduce the field strength.

The use of shunt regulation is undoubtedly the more general form of speed regulation employed, but it must be noted that with commercial machines it can only be used to increase the speed above the normal full-load speed of the motor because any attempt to obtain reduced speeds by means of "field strengthening" would be rendered impracticable by the effect of magnetic saturation.

The effect of shunt regulation is to create a series of more or less parallel speed-load curves above the datum representing full-field performance, each curve corresponding to a regulator position. It is of interest to contrast the curves shown in Fig. 6 with those in Fig. 2, and it will be seen that with shunt regulation the speed is approximately constant from no-load to full-load.

It is possible to obtain an increase in speed

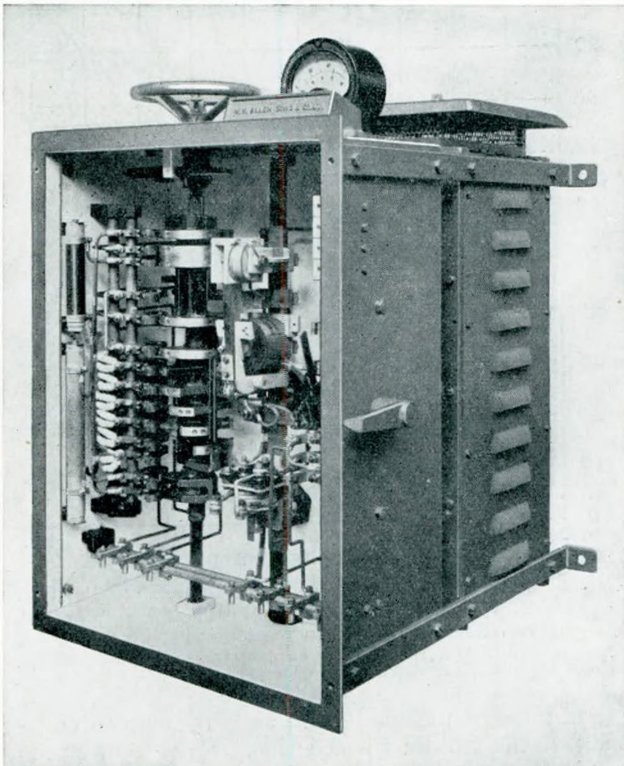


FIG. 5.—20 h.p. Allen drum type combined starter and series regulator with contactor circuit breaker.

amounting to 300 or 400 per cent. of the normal full-load speed with a suitably designed motor and speed increases of 50 to 100 per cent. are quite usual.

Occasionally, where a very wide range of speed is required, a combination of series and shunt re-

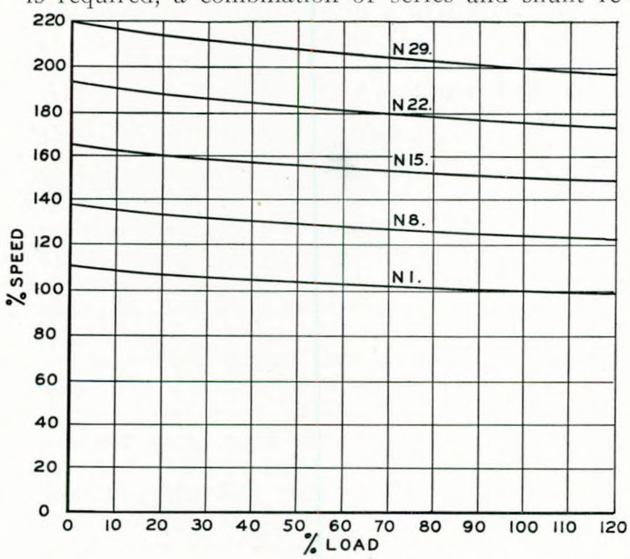


FIG. 6.—Performance curves for a d.c. shunt motor used with a shunt regulator. (The curves are drawn for several positions on the regulator as indicated by the contact numbers on the curves).

gulation is employed, and in Fig. 7 we illustrate a motor mounting controller manufactured for use with a special pump of the positive displacement type. The motor developed 5 h.p. at top speed and the equipment was designed for a speed range of 250/600 r.p.m. by series control and 600/1,800 r.p.m. by shunt. The construction of the panel is such that the shunt regulator cannot be used until the whole of the series resistance is cut out of the armature circuit, and similarly the series regulator is so interlocked as to prevent its use when the motor is running on the shunt controlled portion of its speed range.

SPEED CONTROL—SHUNT OR SERIES REGULATION?

The method adopted for obtaining the desired speed regulation will be largely determined by economic considerations influenced to some degree by local factors applying in a particular instance.

The following generalisations will, of course, apply :—

(1) The motor having the higher "normal" or full-field speed will be the cheaper. Thus a 25/50 h.p. machine having a speed range of 500/1,000 r.p.m. by series regulation, will have a frame size based on an output of 50 h.p. at 1,000 r.p.m.

If, however, shunt regulation is to be employed, the motor must be rated to carry the current corresponding to 50 h.p. and also, must have a stronger field to permit the lower speed of 500 r.p.m. to be

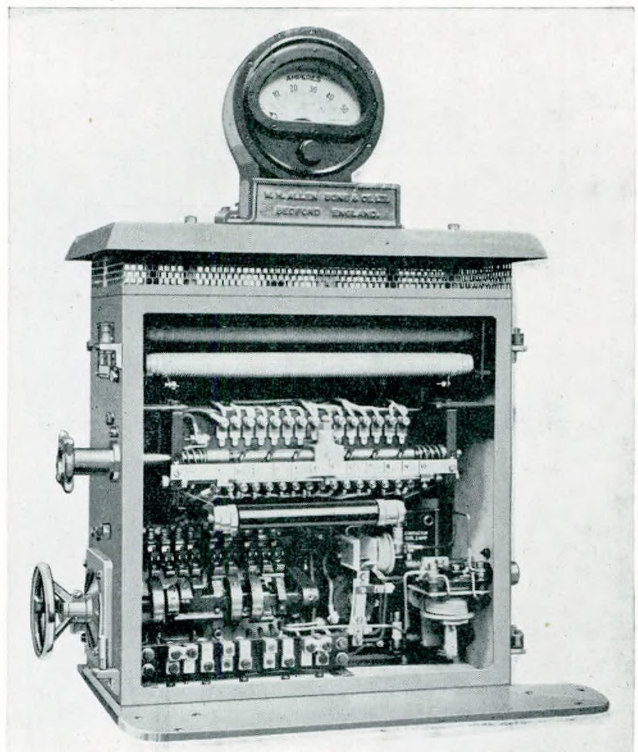


FIG. 7.—5 h.p. Allen combined starter, series and shunt regulator, arranged for motor mounting.

obtained. For the reasons already discussed, this will mean a larger and hence more expensive motor.

For a continuously-rated machine of the enclosed-ventilated type, such as that illustrated in Fig. 8, the slower speed motor will cost approximately 40 per cent. more, initially.

Apart from other considerations, the higher speed motor will therefore be preferred, on the score of lower first cost and hence smaller depreciation and, lesser dimensions.

(2) Motor control gear incorporating series regulation is considerably more expensive than the corresponding equipment embodying shunt regulation, except perhaps on very small sizes below 1 h.p.

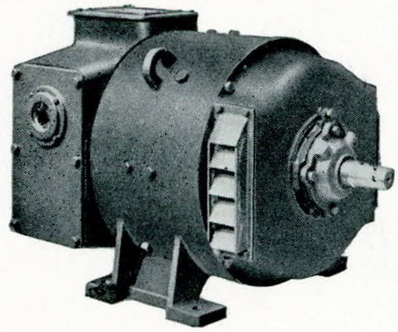


FIG. 8.—Typical Allen enclosed-ventilated drip-proof d.c. motor.

(3) The power costs when running at reduced speeds will be higher when using series regulation, on account of the rheostat losses referred to above.

This applies particularly to constant torque drives, but diminishes in importance as the period of slow running decreases. Thus a large machine tool such as a wheel-lathe, may require slow speeds for setting up and perhaps for passing "hard" spots. The use of series regulation in this case is limited, because the motor is usually running at full speed and therefore does not materially affect the running costs, but in this instance does result in the use of a smaller and less expensive motor.

(4) Series regulation is not particularly suitable for use with fluctuating loads, as with each variation a corresponding speed fluctuation must occur, as will be seen from the curves shown in Fig. 2. This is very undesirable on centrifugal loads for certain technical reasons connected with the regulator design, which we need not consider here in detail, and even with constant torque drives may occasion inconvenience if the desired speed range cannot be obtained owing to the load falling short of that estimated.

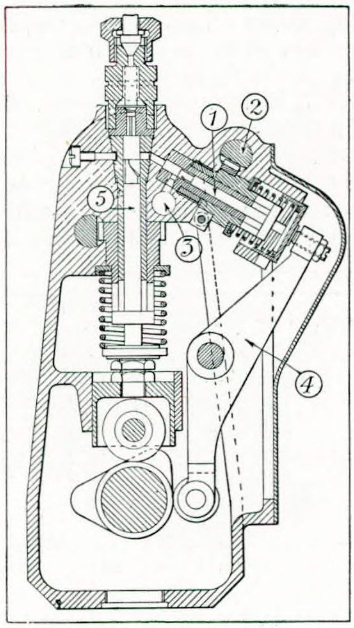
Recent Inventions of Oil Engine Interest.

"The Oil Engine", October, 1939.

A Pilot Injection Scheme.

An injection pump claimed to reduce "Diesel knock" is shown in patent No. 507,940 by The Austin Motor Co., Ltd., and R. Colell, both of Birmingham. The pump injects a small amount of fuel just prior to the main charge, and it is this feature which is said to reduce the knock.

The main plunger (5) and its allied mechanism are of conventional construction, the pilot injection being performed by an additional plunger (1)

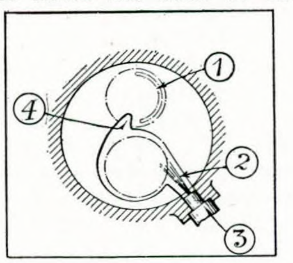


Austin fuel pump.

operated from the camshaft *via* a rocker (4). Fuel drawn from the supply port (3) fills the working space, and as the plunger (1) moves upwards, the early injection commences as soon as the supply port is covered. It terminates when the helical edge again opens the supply port, the plunger being adjustable for output by the usual rack mechanism (2). All the foregoing action occurs before the main plunger comes into operation, the exact phase angle being governed by the cam setting.

A New Petter Combustion Chamber.

Combustion chamber design is the subject of patent No. 508,192 by Petters, Ltd., and others. The proposed scheme employs a cylindrical recess in the piston crown, with a tangential groove (2) leading to the injector (3). The piston reaches practically to the cylinder head at all other points, so that the recess holds the bulk of the air charge.



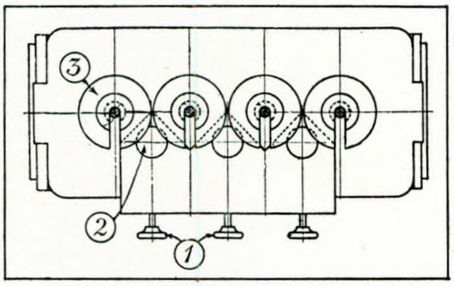
Petter combustion chamber.

Rotary motion is imparted to the air by its being forced into the pocket *via* a channel (4); this slopes gently from the piston crown to the bottom of the pocket. The position of the valves is shown by dotted lines; it will be noticed that the inlet valve is fitted with a deflector (1) to initiate the swirl.

Pump Seizure Causes Engine Stoppage.

An injection pump in which a seized plunger immediately cuts off the whole fuel delivery is advocated by R. Onions and Ruston & Hornsby, Ltd., Lincoln, in patent No. 509,501.

Irrespective of where it seizes, a plunger always comes to rest at the top of its stroke, being forced there by its cam, the spring being insufficient to return it. Advantage is taken of this fact in the design of the stopping device. Between each pair of tappets is a spring-raised plunger (2) which bears against the spring-retaining washers (3). Normally, these plungers never rise, owing to the "one up, one down" action of the tappets. Should, however, a pump plunger stick at top-stroke, the next upward motion of its neighbour allows the plunger (2) to rise to its full height where it is locked by a spring catch (1). This renders the second pump plunger inactive, and the same action then takes place with the third, and so on, until all the pump plungers are locked at top-stroke.



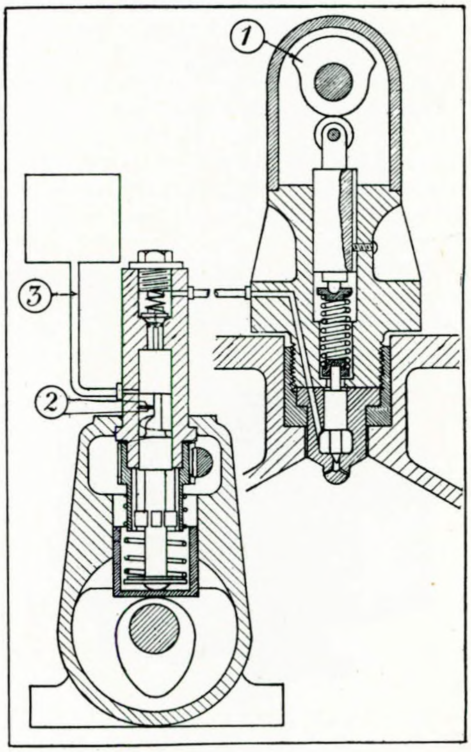
Ruston-Hornsby fuel pump safety device.

A Swiss Pilot-injection Scheme.

Patent No. 508,945, by B. Bischof, Winterthur, Switzerland, shows an injection scheme in which a small fuel charge is injected ahead of the main quantity, the object being to avoid too sudden a rise of pressure in the cylinder.

This arrangement employs a special form of injector, and a pump of slightly modified design. Dealing first with the pump, the plunger is provided with an additional groove (2) which, as it passes the supply port (3), has the effect of causing a definite break in the delivery pressure, resulting in injection in two distinct spurts.

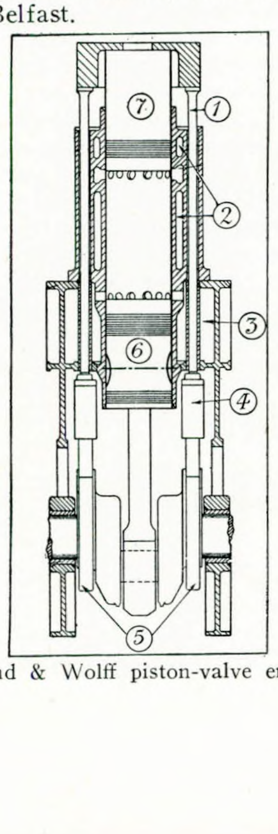
The injector has the usual spring-closed needle-valve, but the force of the spring is varied in timed relationship with the engine by means of a two-diameter overhead cam (1). The cam and the injection pump are so timed that the first injection spurt has to lift only the weaker spring setting whilst the following main charge is subject to the intensified spring pressure.



Bischof two-stage injection equipment.

More Compact Piston-valve Engines.

Reduction of the overall length of piston-valve engines is the object of patent No. 508,464, by V. Mickelsen and F. Rebbeck, both of Harland & Wolff, Ltd., Belfast.



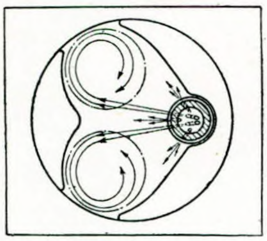
Harland & Wolff piston-valve engine.

The drawing shows a cylinder in which the working piston (6) is connected to the usual crank-pin, whilst the valve piston (7) acts through a pair of eccentrics (5). The eccentric straps are attached by connecting rods to crosshead guides (4) from which extend sliding rods (1) to the upper piston. The saving in engine length arises from the fact that the rods (1) are kept close to the cylinder, actually passing through the scavenge air-chest (3), whilst higher up they are very close to the exhaust ports. Water spaces (2) prevent the spread of high temperatures to the valve rods.

New Combustion System by Büchi.

An improved shape for a combustion chamber formed in the crown of the piston is shown in patent No. 509,086 by A. Büchi, Winterthur, Switzerland. The invention applies only to engines employing a thin-walled pre-cumbustion chamber into which the fuel is primarily injected.

Several forms are described, one of which is



Büchi combustion chamber.

illustrated herewith. Pockets are formed in the piston head, one under each valve, and sloping upwards to a common small pocket under the pre-combustion chamber. A feature of the patent is that the pockets are deep enough to clear the valves when fully open; this is to enable a straight-through scavenge to be given.

Water Softening for Steamships.

"The Engineer and Iron Trades Advertiser", 9th November, 1939.

The water-softening plant of the t.s.s. "Mauretania" operates on the principle of cream lime treatment, with filtration through closed pressure sand filters, in conjunction with base-exchange treatment, to reduce the resulting 5° total hardness to zero without the addition of any material quantity of soluble salts. The normal capacity is 100 tons of softened water per 24 hours, *i.e.*, 9,300lb./hr., while the settling time in the reaction tank is 3 hours. A rate-of-flow indicator of the mercurial type in the raw water inlet main shows the volume of water passing to a preliminary mixing tank and two main reaction and precipitation tanks, each 9ft. in diameter and 9ft. high, equipped with a central soft-water collecting tube, a sludge agitator, and sludge valve. The preliminary mixing tank is also equipped with a mechanical agitator which serves to mix the hard water effectively with the desired amount of lime cream. The latter is mixed in a lime container of 24-hour supply capacity by means of agitator gear, the cream being discharged to the preliminary mixing tank by a reciprocating pump fitted with a graduated crank disc and movable crankpin, so that the stroke of the pump and therefore the amount of lime per stroke, can be adjusted as required. The drive is by means of a $\frac{3}{4}$-h.p. electric motor coupled by reduction gearing to a horizontal shaft from which the three agitators are driven. Temporary hardness (due to bicarbonates) is removed by the lime, and the sludge collecting at the bottom of the reaction tanks is drawn off as necessary through the sludge valves. The almost clear water is pumped under pressure through two cylindrical sand filters, each 2ft. 6in. in diameter, containing quartz sand resting on pebbles. The plant includes a patent manifold strainer system, having a number of collecting orifices distributed in such a manner as to ensure a uniform draw-off of the filtered water as well as equal distribution of the compressed air (supplied by a steam injector) and the washing water used at intervals for cleaning the sand and discharging the separated material. The clear and partially softened water undergoes final treatment in a base-exchange plant comprising two containers or softening units, each 1ft. 9in. in diameter, charged with "Basex" granular material, resting on a bed of pebbles, fitted with soft-water collecting and brine-distributing systems with the necessary inlet, outlet, waste, washing water, drain and automatic air relief valves, as well as a water meter to indicate the amount of water treated with brine between regenerations. For this purpose there is also a brine-preparing tank fitted with a salt basket and the necessary piping and valves, connected to each softening container.

An Interesting Amphibian Vehicle.

"The Engineer", 24th November, 1939.

An amphibian vehicle for exploratory and rescue work in swamps and jungles, which incorporates some interesting features, has been built in Florida. The hull, 20ft. long and 8ft. wide, comprises an open box-shaped body, the side portions of which end in closed cabins fore and aft. The forward or control cabin is nearly 8ft. high, while the after cabin, which houses the 110-h.p. engine, is 6ft. 8in. high. Up to 40 passengers can be accommodated in the amidships space, which is 9ft. 6in. long, 5ft. 4in. wide, and 3ft. 2in. deep. A total load of just over three tons can be carried at speeds of between 15 and 25 m.p.h. on level ground, and at from $8\frac{1}{2}$ to 10 m.p.h. in the water. Propulsion on both land and water is by means of endless chains on the sides of the vehicle, T-shaped curved cleats, 7in. high and 4in. wide at the top being bolted to the chains. On land the cleats grip the ground, whilst in the water they act as paddles. Tension on the driving chains is adjusted by moving forward the front idle sprocket against the pressure of two shock-absorbing springs. The vehicle is steered by disengaging the clutch and stopping the chain on the side towards which the turn is to be made. Sufficient fuel is carried for a cruising radius of about 400 miles.

Belgian Shipbuilding Costs more than Doubled in Two Months.

"Journal de la Marine Marchande", 16th November, 1939.

The unprecedented rise in the cost of ship construction in Belgium between the 1st September and 1st November, 1939, is causing serious concern in that country. Before the outbreak of the present war, a motorship of 8,000 tons d.w.—such as the "Alex-Van-Opstal"—cost from £125,000 to £140,000, whereas at the present time Belgian shipbuilding yards are quoting from £330,000 to £350,000. Various suggestions for meeting this difficulty have been put forward, one of them being a proposal to purchase German ships, but it is considered that the only practical remedy will be the taking over of all future shipbuilding orders by the State.

MINISTRY OF SHIPPING EXAMINATIONS.

List of Candidates who are reported as having passed examinations for certificates of competency as Sea-Going Engineers under the provisions of the Merchant Shipping Acts.

Name.	Grade.	Port of Examination.
For week ended 4th January, 1940 :—		
Donnan, Arthur E. E. C. ...	2.C.M.	Belfast
Kerr, Andrew J. L.	2.C.M.	"
Marr, Ronald E.	2.C.	Cardiff
Thomas, Thomas Alfred ...	2.C.	"
Vaggers, Owen Harry Riordean	2.C.M.	"
Morgan, John Richard G. ...	2.C.	"
Briggs, William R.	2.C.M.	"
Fitzpatrick, John	2.C.	Glasgow
Russell, Alex.	2.C.	"
Nicol, John Alex.	2.C.M.	"
Ormiston, David	2.C.	Leith
Stormont, John A.	2.C.	"
Taylor, Alexander	2.C.	"
Wanless, Richard	2.C.	"
West, Thomas B.	2.C.	"
Davidson, Francis A. ...	2.C.M.	"
Higham, Philip R.	2.C.	Liverpool
Morrison, John A.	2.C.	"
Paterson, David B.	2.C.	"
Quirk, Harry Lewis... ...	2.C.	"
McAthey, Edward	2.C.M.	"
Harris, David W.	2.C.	London
Lindley, Jack Clifford ...	2.C.	"
Lunberg, Geo. H. A. ...	2.C.	"
Swain, Stanley W.	2.C.	"
Hastie, John V.	2.C.M.	"
Craig, Arthur	2.C.	Newcastle
Freight, Norman	2.C.	"
Rowe, Harry	2.C.	"
Short, James Carr	2.C.	"
Robertson, Alan	2.C.M.	"
For week ended 11th January, 1940 :—		
Phillips, Barry A.	1.C.S.M.	Cardiff
Moon, John I.	1.C.M.E.	"
Simpson, James	1.C.	Glasgow
Anderson, John	1.C.M.	"
Gray, Frederick B.	1.C.M.	"
Wilson, Alasdair C. ...	1.C.M.E.	"
Hines, John A.	1.C.M.	Hull
Anderton, Charles E. ...	1.C.	Liverpool
Culshaw, George	1.C.	"
Dewar, William	1.C.	"
Smith, Stanley	1.C.	"
Stevenson, George B. ...	1.C.	"
Vincent, John Vernon ...	1.C.	"
Lawe, Robert	1.C.S.M.	"
Redfearn, Godfrey F. ...	1.C.	"
Allen, Philip B.	1.C.M.E.	"
Hughes, Harry P.	1.C.M.E.	"
Wetherilt, James	1.C.M.	London
Elder, William B.	1.C.	Newcastle
Buckingham, Allan D. ...	1.C.M.	"
Howey, Reed S.	1.C.M.	"
Mitchell, John W.	1.C.M.	"
Walton, Thomas	1.C.M.	"
Smith, James F.	1.C.S.E.	"
Berryman, George W. ...	1.C.M.E.	"
Bulley, William	1.C.M.E.	"
Johnson, William B. ...	1.C.M.E.	"
Murphy, William	1.C.M.	Belfast
Donnelly, Alexander L. ...	1.C.	"
Anderson, James McE. ...	1.C.S.M.	Glasgow
Ferguson, Thomas	1.C.	Leith
Hay, Alexander	1.C.M.E.	"
McCracken, Robert J. ...	2.C.	Liverpool
For week ended 18th January, 1940 :—		
Hogg, Thomas Edward ...	1.C.	Newcastle
Innes, Andrew	1.C.M.	"