BALANCING OF LARGE TURBINE ROTORS

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As the size and flexibility of turbine rotors increase, the problems associated with their smooth running become more complex. There is a growing awareness that the function of low speed balancing machines is quite inadequate for many types of flexible rotor. Both the rigid and flexible modal behaviour of such rotors is therefore discussed at some length and balancing machines are considered in this context. Modal balancing techniques are described, comparisons are drawn between this and low speed methods and the limitations and advantages of each are demonstrated.

It is shown that modal balancing can be carried out by the application of a comparatively simple set of rules and standard graphical constructions. Although available elsewhere, these are included here for convenience of reference. Their use is amplified by discussion of the practical application of the rules to particular examples of balancing problems.

INTRODUCTION

The theoretical approach to the balancing of large flexible rotors has been the subject of a number of papers by Professor Bishop *et al* and the practical application of modal balancing has been covered in a paper by Moore and Dodd[1].

A large turbine rotor falls into this general pattern of flexible rotors, but greater difficulty is encountered in applying the techniques because of the problem of driving a fully bladed spindle at high speeds. The manufacturer is therefore inevitably faced with the decision of whether low speed rigid rotor methods can be extended to fit a particular rotor, or whether a full modal balance must be undertaken in order to ensure smooth running on site. The purpose of this paper is to describe the fundamental differences between high speed and low speed balancing methods.

BEHAVIOUR OF A FLEXIBLE ROTOR

General

In order to understand the basic principles of balancing a flexible rotor, it is necessary to consider in general terms the pattern of the modal behaviour of such a rotor. This is its response to unbalanced centrifugal forces, when running over a wide range of speeds, supported by its own bearing pedestals.

As speed of rotation increases, an unbalanced flexible rotor will deflect into each of its modal shapes in turn. The amplitude of deflexion in any one mode will depend on the degree of defect in that mode and it will build up to a maximum at the corresponding critical speed and then die away again. The critical speeds and the modal shapes will to some extent be influenced by the flexibility of the bearing pedestals, but it must be appreciated that extremely flexible or soft bearings do not permit a rotor's high speed behaviour to be examined at a low running speed. The reason for this is explained as follows.

If a large flexible rotor is mounted in very flexible bearings, as in a low speed balancing machine, it will exhibit first and second rigid modal shapes at the critical speeds of the suspension, (low speed balancing machines normally work at a speed well above either of these criticals). Fig. 1 shows to a greatly exaggerated vertical scale, the manner in which the rotor will vibrate at its first and second rigid mode criticals, also the shape it may be expected to distort into at each of its first three criticals when running in its own bearings. Figs 1(a) and 1(b), the rigid mode shapes, represent the one extreme of vibration in a single plane (usually horizontal), about the mean position which is the line joining points A and B. On the other hand, the modal shapes associated with normal running conditions [Figs 1(c), 1(d) and 1(e)], represent a distortion which takes place in one plane in the rotor, but then generates an imaginary solid of revolution about the same mean position; for the purpose of this paper these will be called flexural modes. In the diagram the bearings are represented by the points A and B. Under normal running conditions they will be semi-rigid supports, but the associated modal shapes will not pass through, nor will they necessarily be symmetrical

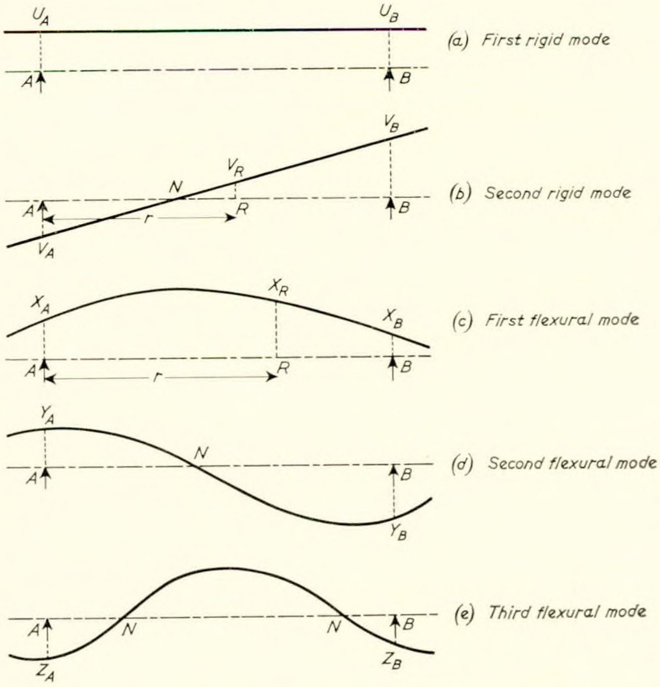

FIG. 1—*Modal shapes of a flexible rotor on both soft and semi-rigid bearing supports*

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about A and B. The behaviour of a flexible rotor running under the influence of each of these modal shapes in turn, is now discussed in some detail.

Under the Influence of Rigid Mode Shapes

Consider a perfectly balanced flexible rotor running in soft bearings in a low speed balancing machine and having associated first and second rigid modal shapes as shown in Figs 1(a) and 1(b).

To clarify the discussion, assume that the rotor can be run at a speed where it is only under the influence of the first critical of the suspension. Then, with a deliberate unbalance added anywhere along its length, the vibrations measured at the bearings will be equal to and in phase with each other. For a given unbalance and a given speed the bearing vibration will be constant whatever the axial position of the added weight and if the weight is traversed in a constant radial plane, the balancing machine will indicate a constant angle of unbalance. Conversely, a rotor which was unbalanced could be corrected in this mode by the addition of a weight anywhere along its length and the amount of correction required would be independent of the axial position at which it was added. This is usually referred to as the "static" component of unbalance.

If the rotor could be run at a speed where it was only under the influence of the second critical of the suspension (this is again an academic assumption), the traverse weight test would give the following results. The vibrations measured at the bearings would be in anti-phase and the vibration at bearing A would be $\frac{AV_A}{BV_B}$ times the vibration of bearing B. Starting from point A, as the position of the added weight approached N, the mass centre of the rotor, the measured effect would decrease to zero and then beyond that point would build up again, but in the reverse direction. That is, for a given unbalance and a given speed, the bearing vibration would be proportional to the axial distance of the added weight from the centre of mass of the rotor, and if the weight was traversed in a constant radial plane, the indicated angle of unbalance would reverse as point N was passed. A rotor which was unbalanced could be corrected in this mode with a single weight if necessary.

The foregoing is a highly theoretical assumption, as a rotor on the flexible supports of a balancing machine would be influenced by unbalance in both modes simultaneously and so would usually be corrected in the second mode by an out-of-phase pair of weights, added towards the two ends. Equal weights would be chosen to cancel each other in their effects on the first rigid mode and the amount required would be determined by the distance between the planes in which they were added. This is usually referred to as the "dynamic" component of unbalance.

Under the Influence of Flexural Mode Shapes

Consider a perfectly balanced flexible rotor running in its own bearing pedestals and known to have associated modal shapes like those shown in Figs 1(c), 1(d) and 1(e).

If the rotor is run at a speed near to its first flexural critical with a deliberate unbalance added anywhere along its length, the vibrations measured at the bearing pedestals will be in phase with each other and the vibration at bearing A will be $\frac{AX_A}{BX_B}$ times that measured at B. For a given speed and a given unbalance, the amplitude of vibration in this case will be proportional to the "modal sensitivity" at the axial position chosen (if the weight is added at a distance r from bearing A in Fig. 1(c), the modal sensitivity will be RX_R). The first flexural modal shape can therefore be obtained by traversing a constant unbalance along the length of the rotor, at a speed near to first critical and plotting the vibration of, say, bearing A on a base of the distance of the plane of the added weight from bearing A (the modal shape obtained from the vibration readings at bearing B would be of a similar form but the vertical scale would be reduced in the ratio of $BX_B : AX_A$). If the deliberate unbalance is traversed in a constant radial plane, then for all practical purposes the angle of the measured vibrations will remain constant. A rotor which was unbalanced in this mode could be corrected by the addition of a single weight anywhere along its length on the opposite side to the initial error, but for a given error, the amount of correction necessary would be inversely proportional to the modal sensitivity at the point of addition. If it is required to add weight to the rotor for other purposes, without upsetting its state of balance in this mode, then the additional weight must be split into a minimum of two parts, and the individual parts must be so placed axially and radially, that Σ mass $\times$ radius $\times$ modal sensitivity $= 0$.

If the rotor is run at a speed near its second flexural critical, its behaviour can be described in a similar manner to the above, but with the following exceptions:

i) the vibrations at the bearing pedestals will be in anti-phase to each other;
ii) the amplitude of the vibration of bearing A will be $\frac{AY_A}{BY_B}$ times that at bearing B and there is no direct relationship between $\frac{AY_A}{BY_B}$ and $\frac{AX_A}{BX_B}$;
iii) on Fig. 1(d) the point marked N is a node and the rotor can be neither unbalanced nor corrected by the addition of weight at this axial position, as far as this mode is concerned;
iv) in plotting the modal shape from a traverse weight, the curve obtained from readings of vibration of bearing B would be opposite in phase to that obtained from readings of vibration of bearing A;
v) a rotor which was unbalanced in this mode could be corrected by the addition of weight anywhere along its length, except at N. Correction could, if necessary, be effected by addition of a single weight in a radial plane opposite that of the initial error, if error and correction were on the same side of N, or in the same radial plane as the initial error, if error and correction were on opposite sides of N. For a given error, the amount of correction necessary would be inversely proportional to the modal sensitivity in this mode at the point of addition.

When the rotor is running near its third flexural critical, the bearing vibrations will be once more in phase with each other and the relation between the vibrations at the two ends will be as $AZ_A : BZ_B$. In addition there will be two nodal points or axial positions at which the rotor will not respond to added weight. Correction of a rotor unbalanced in this mode could again be achieved by the addition of a single weight if necessary. For a given error the correction would be required in one radial plane if it were placed between the nodes and in the opposite radial plane if it were placed outside the nodes.

As Speed is Varied Near a Critical

If the rotor under discussion is unbalanced in any one of the rigid or flexural modes just detailed and is then run through the associated critical speed, the measured vibrations will follow normal resonance response curves in both amplitudes and angles. A typical example is shown in Fig. 2. In each case the peak amplitude and the spread of noticeable effect will depend on both the magnitude of the original unbalance and the degree of damping of the system concerned.

There are, however, two fundamental differences between conditions near the rigid mode criticals and those near the flexural criticals. Firstly, although the amplitude of vibration increases to a maximum at a rigid mode critical, the shape of the rotor is unaltered. On the other hand, as speed is increased near a flexural critical, the distortion in the corresponding mode builds up to a maximum and then dies away again. Secondly, the rigid mode criticals as encountered in a low speed balancing machine are associated with a system having very little damping. A large flexible rotor running in normal journal bearings has appreciable damping.

As Encountered in Practice

The foregoing explanations are again idealized in so far as they assume that a rotor will be under the influence of errors in only one mode at any particular speed. This is rarely completely true and more realistic conditions are now considered.

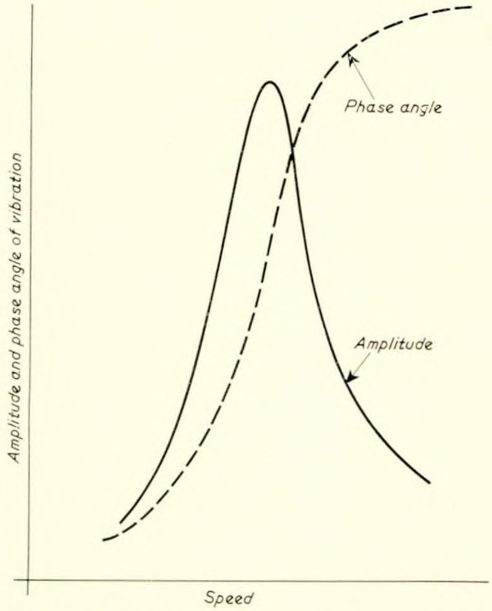


FIG. 2—*Typical resonance response curve as rotor runs through a critical speed*

All low speed balancing machines work at a speed well above the first and second criticals of their suspension. The system can, however, still respond measurably to unbalance in either of the rigid modes. A rotor running in this environment and unbalanced both "statically" and "dynamically" will therefore vibrate in its first and second rigid modes simultaneously and the measured vibrations will be related to a mixed modal condition.

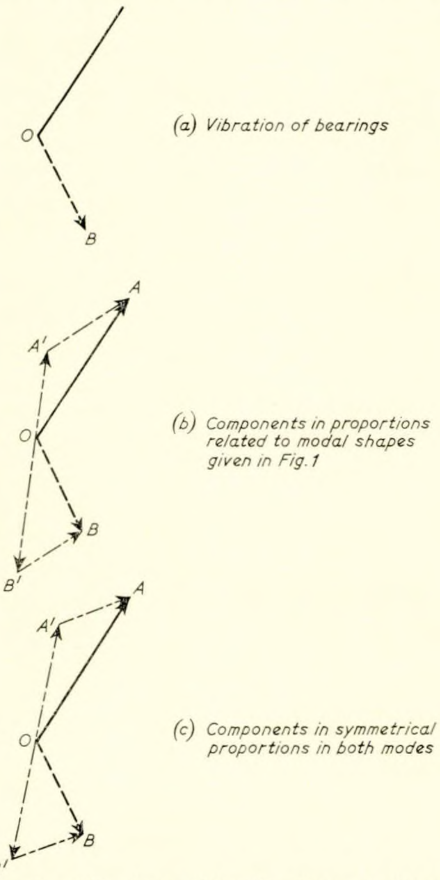


FIG. 3—*Mixed modal vibration condition*

Under normal running conditions on semi-rigid supports, because of the high degree of damping, a particular modal defect will influence the vibration of the bearing pedestals quite strongly over a considerable speed range. A rotor which is unbalanced in several flexural modes can therefore suffer significant distortion in two or even more modes simultaneously over a wide range of speeds. Consequently, the measured vibrations are frequently related to mixed modal conditions. As this produces a complex situation, it is appropriate here to consider the fundamentals of mixed flexural modal response in some detail.

A rotor with flexural modal shapes as shown in Figs 1(c), 1(d) and 1(e), which was running at a speed where it was under the influence of unbalance in second and third modes, might well give vibrations OA and OB at its bearing pedestals as shown in Fig.3(a). These would be composed of a second modal component OA', OB', where $\frac{OA'}{OB'} = \frac{AY_A}{BY_B}$ and a third modal component $A'A$, $B'B$, where $\frac{A'A}{B'B} = \frac{AZ_A}{BZ_B}$, see Fig. 3(b). It should be noted that unless the proportional effect of each mode on the two bearings is known, the individual components cannot be positively identified, as the same vectors OA, OB, could be resolved into an infinite number of combinations; a particular example is shown in Fig. 3(c) where $OA' = OB'$ and $A'A = B'B$. The angular difference between the two components in Fig. 3(b). could be the result of the individual errors being in different planes in the rotor. Alternatively, the two distortions could be caused by a single error; the chosen speed being at different points on the resonance characteristics for the two modes could then account for their angular displacement.

Experimental Determination of Mixed Modal Shapes

A single weight traversed along an otherwise perfectly balanced rotor will not infrequently give vibrational effects equivalent to the condition shown in Fig. 3(b). Interpretation of traverse weight measurements into modal shapes is then a complex process.

Consider a large perfectly balanced flexible rotor running at such a speed that addition of weight will cause it to deflect into second and third modes and that the plane of measured effect will be at different angles for the two modes.

Let the second modal shape be such that:

$$\frac{\text{effect on end A}}{\text{effect on end B}} = m.$$

Let the third modal shape be such that:

$$\frac{\text{effect on end A}}{\text{effect on end B}} = n.$$

In Figs 4(a) and 4(b), let OA_1, OB_1 be the effect of fitting the traverse weight at axial position 1 and let OA_2, OB_2 be the effect of fitting the traverse weight at axial position 2.

Let the modal components of these effects be:

$$\left.\begin{array}{l} OA_1', OB_1', \text{ for second mode} \\ A_1'A_1, B_1'B_1 \text{ for third mode} \end{array}\right\} \text{for position 1.}$$

$$\left.\begin{array}{l} OA_2', OB_2' \text{ for second mode} \\ A_2'A_2, B_2'B_2 \text{ for third mode} \end{array}\right\} \text{for position 2.}$$

Then, by definition $\frac{OA_1'}{OB_1'} = \frac{OA_2'}{OB_2'} = m$

and $\frac{A_1'A_1}{B_1'B_1} = \frac{A_2'A_2}{B_2'B_2} = n$

and for all practical purposes

OA_1' will be parallel to OA_2'

and $A_1'A_1$ will be parallel to $A_2'A_2$

If these statements are expressed in terms of the co-ordinates given in Figs 4(a) and 4(b) then by equating real and imaginary parts the following sets of equations are obtained:

$$\left.\begin{array}{l} P_1 = ma_2 + na_3 \\ Q_1 = mb_2 + nb_3 \\ P_2 = mc_2 + nc_3 \\ Q_2 = md_2 + nd_3 \end{array}\right\} \quad (1)$$

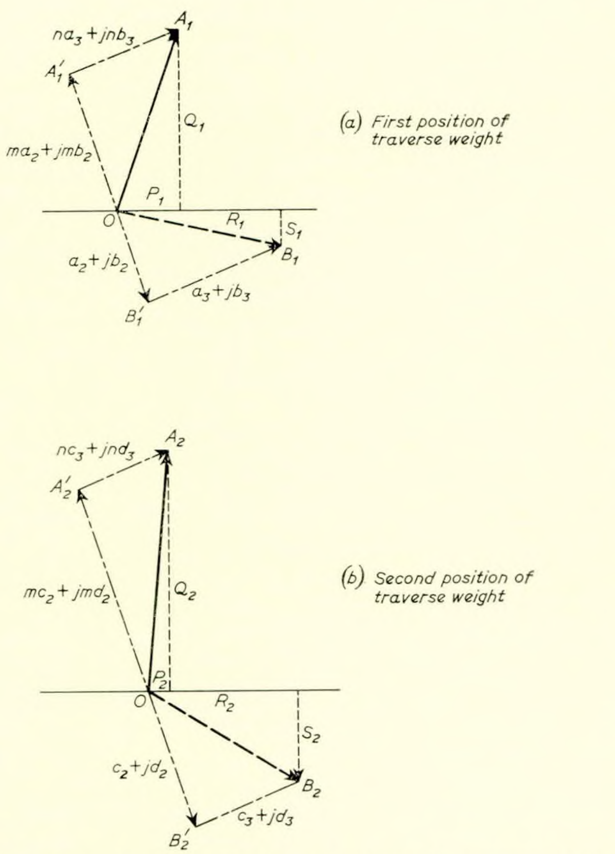


FIG. 4—*Separation of modes in a mixed mode traverse weight test*

$$\left.\begin{aligned} R_1 &= a_2 + a_3 \\ S_1 &= b_2 + b_3 \\ R_2 &= c_2 + c_3 \\ S_2 &= d_2 + d_3 \end{aligned}\right\} \qquad (2)$$

$$\left.\begin{aligned} \frac{a_2}{b_2} &= \frac{c_2}{d_2} \\ \frac{a_3}{b_3} &= \frac{c_3}{d_3} \end{aligned}\right\}. \qquad (3)$$

Substituting from equations (2) in equations (1):

$$\begin{aligned} P_1 &= ma_2 + n(R_1 - a_2) \\ Q_1 &= mb_2 + n(S_1 - b_2) \\ P_2 &= mc_2 + n(R_2 - c_2) \\ Q_2 &= md_2 + n(S_2 - d_2) \end{aligned}$$

Hence:

$$\left.\begin{aligned} a_2 &= \frac{P_1 - nR_1}{(m-n)} \\ b_2 &= \frac{Q_1 - nS_1}{(m-n)} \\ c_2 &= \frac{P_2 - nR_2}{(m-n)} \\ d_2 &= \frac{Q_2 - nS_2}{(m-n)} \end{aligned}\right\} \qquad (4)$$

Substituting these values in equation (3):

$$\frac{P_1 - nR_1}{Q_1 - nS_1} = \frac{P_2 - nR_2}{Q_2 - nS_2}$$

which can be solved for n.

It can be shown that m is the other root of the same equation and in the example illustrated m would be negative while n would be positive. The values of m and n can therefore be deduced by analysis of the effects of the traverse weight in two positions along the rotor.

Given these values of m and n, the effects of the traverse weight in the remaining positions along the rotor can be resolved into their modal components either graphically by the construction given later in Fig. 10, or mathematically by use of equations (4) and (2). In either case the second and third modal shapes can be obtained by plotting values of OA' and $A'A$ respectively against the distance of the traverse weight from one bearing.

Given the effect of the traverse weight in a minimum of six positions, the above analysis can theoretically be extended to the separation of a mixture of three modes, but this is beyond the scope of the present paper.

METHODS OF BALANCING

Low Speed Balancing Machine

The degree to which a low speed balancing machine can be used to balance large flexible rotors must be considered in relation to their general behaviour under the extreme conditions of the very flexible supports of the machine and the relatively stiff conditions of the site assembly.

An unbalanced rotor in a low speed machine responds to unbalance in its first and second rigid modes only. The machine will resolve these into corrections in two pre-selected balancing planes. The more sophisticated machines which claim to determine the individual modal components then take the vector sum and difference of the two plane corrections. The difference is referred to as the "dynamic" or "couple" correction and has to be divided equally between the pre-selected planes which are usually near the ends of the rotor; the sum is referred to as the "static" correction and has to be added half way between the end planes. It is interesting to note that if the second rigid modal shape is unsymmetrical, the point midway between the end planes will not coincide with the nodal point. This produces a situation in which addition of the "static" correction will balance the rotor statically, but addition of the "dynamic" correction on its own will not balance the rotor dynamically.

It has already been shown that a weight anywhere along the body of the rotor can produce unbalances both in the rigid and flexural modes. It is likely therefore that an unbalanced rotor will have defects in several modes, and the relationship between these defects will depend not only on the modal shapes but on the axial disposition of the unbalances. As the balancing machine considers the rigid modes only, some residual unbalance will remain in the flexural modes which will result in vibration when the rotor is run to speed in its own bearings.

The errors which might arise in these modes are best illustrated by the following example.

Figs 5(a) and 5(b) show a rotor unbalanced by two different

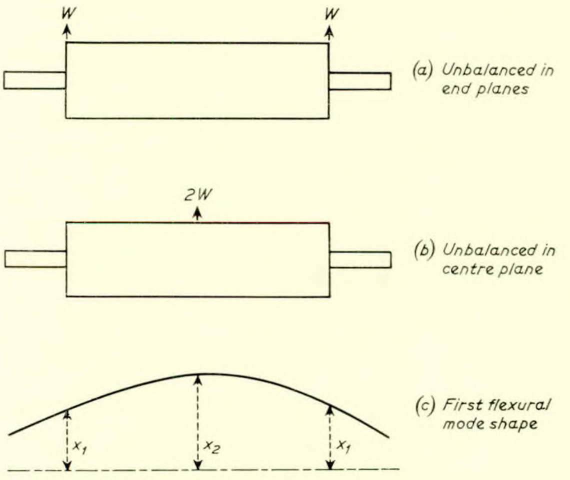


FIG. 5—*Flexible rotor with the same unbalance in first rigid mode but different unbalances in first flexural mode*

weight conditions. If the flexural mode shapes are generally as indicated in Fig. 5(c), the unbalance in the first flexural mode will be proportional to $2(Wx_1)$ in Fig. 5(a) and $2Wx_2$ in Fig. 5(b). The difference therefore between the two unbalances would be proportional to the difference between x_1 and x_2. On the other hand the rigid mode unbalance indicated by a balancing machine would be the same numerical value for each.

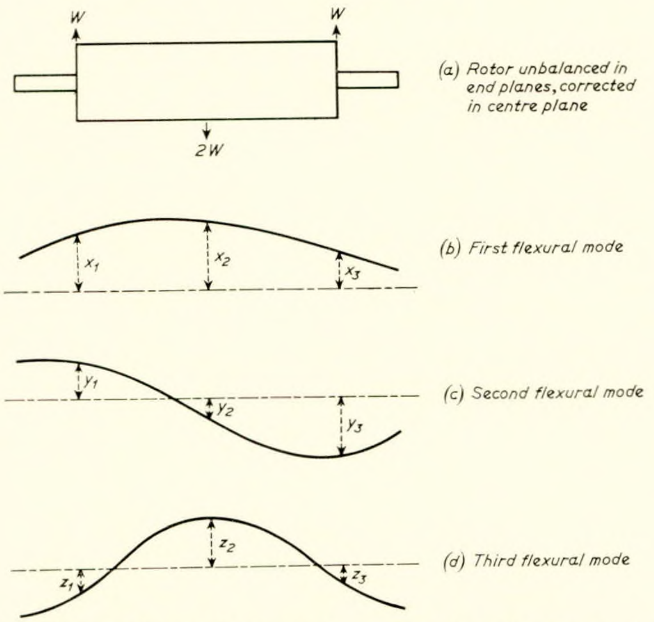


FIG. 6—*Resultant effect on flexural modes when a flexible rotor is corrected in a low speed balancing machine.*

If a rotor were unbalanced in its end planes but corrected in a low speed machine using a central balancing plane, there would effectively be a configuration of three weights as indicated in Fig. 6. The resultant unbalance in the first three flexural modes shown would now be as follows:

1st mode unbalance: proportional to $x_1 - 2x_2 + x_3$
2nd mode unbalance: proportional to $y_1 + 2y_2 - y_3$
3rd mode unbalance: proportional to $z_1 + 2z_2 + z_3$

Therefore, although the balancing machine would indicate a balanced rotor, it would still not be completely corrected in its first flexural mode and the problem of correcting the higher modes would have been increased.

The same effective weight configuration would arise if a rotor unbalanced in its centre plane had fortuitously been corrected in the end planes.

It can now be seen that a low speed balancing machine can have only limited use in balancing large flexible rotors. Nevertheless, there are situations in which a fair degree of successful full speed running can be expected. For example, a low speed machine could be limited to flexible rotors whose running speed is either below the first flexural critical or is well above first and well below second critical. Within this range successful balancing might be achieved providing the rotor has a reasonable degree of axial symmetry and is likely to have unbalances fairly equally distributed along its length. If the total unbalance is small, little error would be left if correction was made only in the end planes. On the other hand, if the total unbalance is large, some proportion could be attached to a central plane and the remainder to the end planes. How large an initial unbalance could be corrected in this way and what fraction should be added to the central plane can only be judged on the success or failure of a number of similar rotors treated in a similar manner.

High Speed Balancing Machine

A high speed machine is electronically similar to the low speed type, but mechanically it has means for varying the flexibility of the supports. A low speed machine is designed to balance rigid rotors and, once set for this purpose, can be operated by non-technical personnel. It should be appreciated that a high speed machine supposedly designed to balance flexible rotors does no more computation and therefore virtually modal techniques have to be employed in the final stages, demanding some technical ability on the part of the operator. Its chief merit is the ability to drive a rotor at high speed on fairly rigid bearing supports and this feature should permit a full modal technique to be employed. A high speed balancing machine therefore is best considered in this context.

Modal Balancing

General

The rotor to be balanced is mounted on pedestals which as far as possible simulate site conditions. Vibration transducers are suitably attached to measure either pedestal or journal vibrations and means are provided to determine the phase of the vibrations with respect to some fixed, but arbitrary angular reference on the rotor. As already stated, a high speed balancing machine can be used for modal balancing, provided that the bearing stiffness and instrumentation is adequate to satisfy the above conditions.

The method of determining the effect of an unbalance in any particular axial plane of the rotor is common to most methods of balancing.

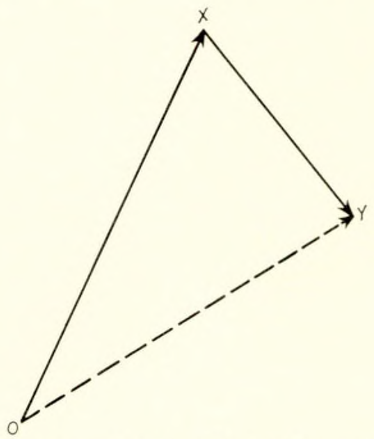


FIG. 7—*Basic principle of balancing operation*

In Fig. 7 let the original unbalance at one bearing be represented in amplitude and phase by the vector OX.

Let the vector OY represent, on the same scale, the vibration of the same bearing when a calibrating weight is attached in a particular axial plane.

Then vector XY represents, to the same scale, the effect of the calibrating weight on the bearing under observation.

Therefore, in order to nullify the original unbalance, the calibrating weight must be increased in the ratio of $\frac{OX}{XY}$ and moved circumferentially through an angle OXY.

Following this principle, but observing the vibrations of both bearings, the defects in the flexural modes are corrected in turn in the following manner.

From now on all mode shapes and critical speeds referred to are flexural modes and associated critical speeds.

The rotor is run at a convenient speed approaching the first critical and the vibrations are reduced to a minimum by the addition of a single weight near the middle of the rotor. It is not essential at this stage to consider the effect of the weight on other modes. Having completed the first modal correction, the rotor is run near to second critical speed with a pair of weights added near the ends of the rotor to correct the second modal defect. They are added on opposite sides so that their effects are additive for second mode, and are proportioned so that they cancel each other in their effects on the first mode. Similarly a configuration of three weights is applied for the third mode.

They are so proportioned that their combined effects are zero on both first and second modes.

In attempting to apply the basic theory of modal balancing to an actual rotor, it invariably becomes apparent that the rotor is distorting in two modes simultaneously in the manner already described. Various means are adopted for the separation of the individual modal components. The necessary graphical constructions are discussed in the context of a typical example of an application of each. Details of the two final conditions can be found in a previous paper(1), but are included here for convenience of reference.

Balancing When There Is a Mixed Mode Condition Near to a Critical Speed

Under this condition it is likely that the unbalance vibration and that produced by the addition of calibrating weights, will be predominantly in one mode. Providing the existence of a small contribution from another mode is accepted and understood, little difficulty is encountered.

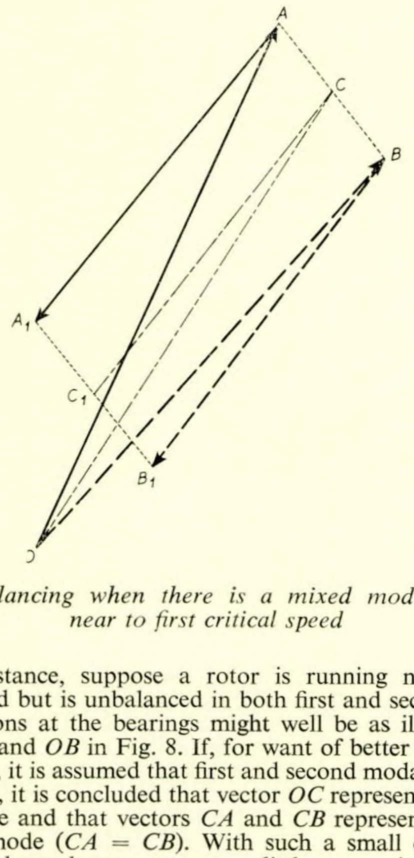


FIG. 8—*Balancing when there is a mixed mode condition near to first critical speed*

For instance, suppose a rotor is running near to first critical speed but is unbalanced in both first and second modes. The vibrations at the bearings might well be as illustrated by vectors OA and OB in Fig. 8. If, for want of better information at this stage, it is assumed that first and second modal shapes are symmetrical, it is concluded that vector OC represents the defect in first mode and that vectors CA and CB represent the defect in second mode ($CA = CB$). With such a small contribution from second mode present, very little error is introduced if the correction for first mode is arranged so that the "in-phase" component of its effect cancels the "in-phase" error OC. That is:

In Fig. 8, let OA_1, OB_1 represent the vibrations at the bearings with a first mode calibrating weight attached to the rotor.

Then, vectors AA_2, BB_1 represent the effects of the calibrating weight and if A_1B_1 is bisected at C_1 vector CC_1 represents the "in-phase" effect of the calibrating weight.

Therefore, the first mode calibrating weight should be increased in the ratio $\frac{OC}{OC_1}$ and moved circumferentially through an angle OCC_1.

A variant of the foregoing may be encountered when a rotor is running near to second critical speed and is unbalanced in second and third modes. The measured vibrations would then be of the form shown in Fig. 9, vectors OA and OB again representing the original unbalance. A reasonable state of balance can often be attained by assuming that the calibrating weights for second mode give a measure of the asymmetry in that mode, and that third mode is symmetrical as detailed:

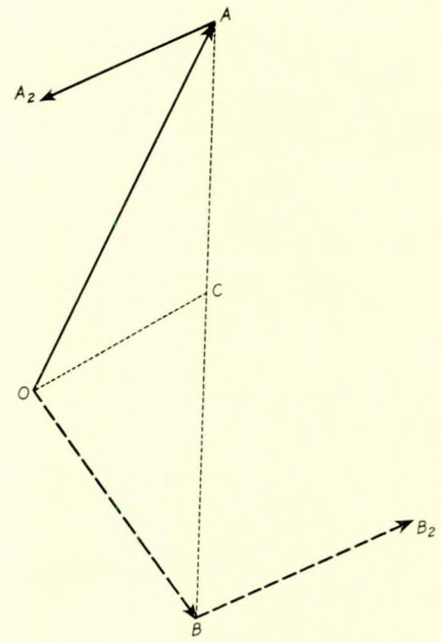


FIG. 9—*Balancing when there is a mixed mode condition near to second critical speed*

In Fig. 9, let OA_2, OB_2 represent the vibrations of the bearings when a calibrating pair of weights is added for second mode.

Then, vectors AA_1, BB_2 represent the effect of the calibrating weights on the bearings.

Let AB be divided in C so that $\frac{CA}{CB} = \frac{AA_2}{BB_2}$.

Then, adjust the calibrating pair of weights so that their effect cancels the vectors CA, CB.

That is, increase the calibrating weights in the ratio of $\frac{CA}{AA_2}$ (which is the same as $\frac{CB}{BB_2}$) and move both circumferentially through an angle CAA_2 (which is the same as angle CBB_2).

Balancing for a Mixed Mode Condition Remote from Either Critical Speed

Consider the case of a rotor which has been balanced in first mode which is still unbalanced in second and third modes, and which at full speed is running well below either second or third critical speeds. Under this condition there might well be a significant contribution from both modes. The measured vectors of vibration have first to be split into modal components where the asymmetry of the modes will not be known, and then each component has to be corrected.

In Fig. 10, let vectors OA, OB represent the original unbalance.

Let OA_2, OB_2 represent the vibration measured when a calibration pair of weights is added to the rotor mostly to affect second mode.

Then vectors AA_2, BB_2 represent the effect of the calibrating weights for second mode.

Let OA_3, OB_3 represent the vibration when the calibrating weights for second are removed and are replaced by a three weight calibrating configuration which will largely affect third mode.

Then vectors AA_3, BB_3 represent the effect of the calibrating weights for third mode.

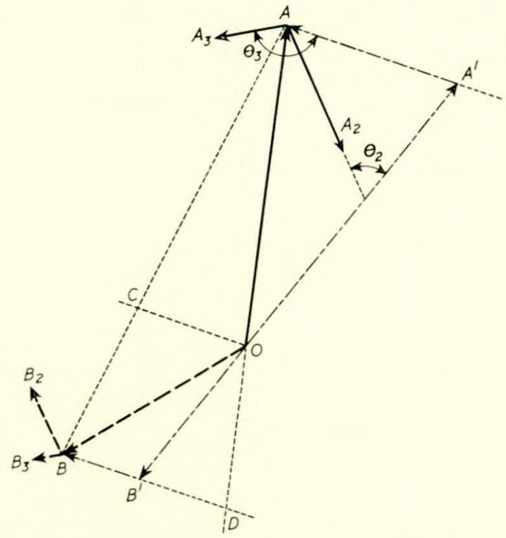


FIG. 10—*Balancing for a mixed mode condition remote from either critical speed*

For the purpose of the balancing operation it can be assumed that $\dfrac{AA_2}{BB_2}$ represents the asymmetry of second mode and $\dfrac{AA_3}{BB_3}$ represents that of third. Thus it is necessary to split the original unbalances into second and third modal (or "out-of-phase" and "in-phase") components of the same proportions as deduced from the calibrating weights. To do this:

In Fig. 10, divide AB in C so that $\dfrac{CA}{CB} = \dfrac{AA_2}{BB_2}$ and call this m.

Let $\dfrac{AA_3}{BB_3} = n$.

Construction: Join CO and draw lines parallel to it through A and B. Produce AO to meet the parallel line through B in D. Divide BD in B' so that $\dfrac{B'D}{BB'} = \dfrac{n}{m}$. Join $B'O$ and produce to meet parallel line through A in A'.

Then original vectors OA, OB are equivalent to an "out-of-phase" component OA', OB' plus an "in-phase" component $A'A$, $B'B$, and it can be shown that $\dfrac{OA'}{OB'} = m$ and $\dfrac{A'A}{B'B} = n$.

Therefore, the "out-of-phase" component can be corrected by adjusting the calibrating weights for second mode and the "in-phase" component can be corrected by adjusting the calibrating weights for third mode.

Therefore, increase the calibrating weights for second mode in the ratio of $\dfrac{OA'}{AA_2}$ and move them circumferentially through the angle labelled θ_2 in Fig. 10. Increase the calibrating weights for third in the ratio of $\dfrac{A'A}{AA_3}$ and move them circumferentially through the angle labelled θ_3 in Fig. 10.

In both cases the proportions between the individual weights in each configuration must be maintained and as it happens, in the example illustrated, the correction for second would have to be placed clockwise of the calibrating position and the correction for third would have to be placed anti-clockwise from the calibrating position, always regarding the rotor from one end throughout.

This construction is equally applicable to balancing for a mixed mode condition when running closer to the critical speed associated with one of the modes, providing ideal balancing planes are available so that weights added for each mode have no effect on the other.

Balancing for a Mixed Mode Condition with a Limited Choice of Balancing Planes

It can occur that, with a limited choice of balancing planes, the calibrating weights added mainly for one mode will have a small but not insignificant effect on the other. It follows that as the rotor will be running at a different point on the resonance response curve for each mode, the two effects will be at different phase angles. This results in the situation illustrated in Fig. **11.**

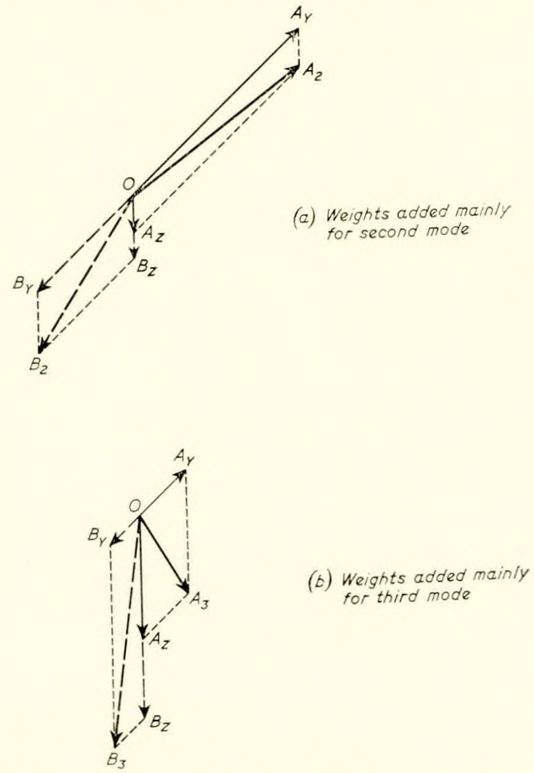


FIG. 11—*Effect of calibrating weights on a mixed mode condition with a limited choice of balancing planes*

Fig. 11(a) is related to weights added mainly for second mode. The vectors OA_Y and OB_Y represent the wanted effects on second mode, and there will also be present the effects represented by vectors OA_Z and OB_Z; these are the unwanted but unavoidable effects on third mode. The resultant measured effects will therefore be vectors OA_2 and OB_2. Fig. 11(b) is related to weights added mainly for third mode. In this case vectors OA_Z and OB_Z represent the wanted effects and vectors OA_Y and OB_Y represent the unwanted but unavoidable effects on second mode. The resultant measured effects will then be vectors OA_3 and OB_3.

The configuration of weights added to the rotor mainly for second mode will therefore have effects on the bearings which are not exactly in antiphase to each other. Similarly, the configuration of weights added to the rotor mainly for third mode will have effects on the bearings which are not exactly in phase with each other. A modified form of the construction given in Fig. 10 is then required.

In Fig. 12, let vectors OA, OB represent the original unbalance.

Let OA_2, OB_2 represent the vibrations of the bearings when calibrating weights are added mainly for second mode.

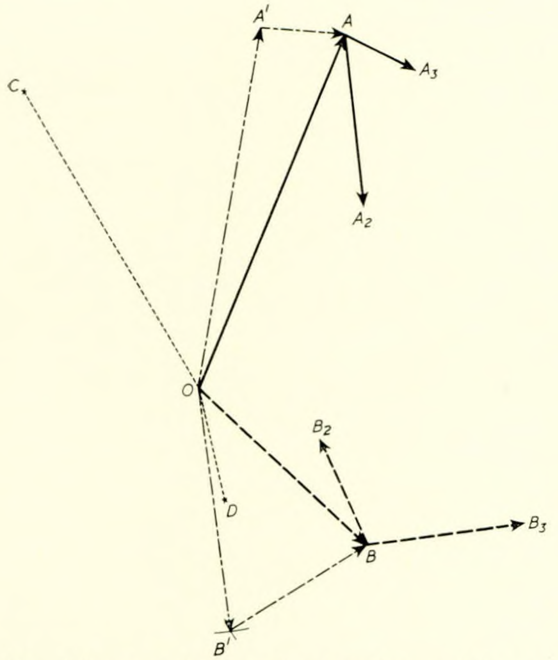


FIG. 12—*Balancing for a mixed mode condition with a limited choice of balancing planes*

Then vectors AA_2, BB_2 represent the effect of the calibrating weights added mainly for second mode.

Let $\dfrac{\text{vector } AA_2}{\text{vector } BB_2} = \gamma$, a complex operator.

Let OA_3, OB_3 represent the vibrations of the bearings when calibrating weights are added mainly for third mode.

Then vectors AA_3, BB_3 represent the effect of the calibrating weights added mainly for third mode.

Let $\dfrac{\text{vector } AA_3}{\text{vector } BB_3} = \delta$, a complex operator.

It is now necessary to find components of the original unbalance which are not only numerically of the same proportions as found with the calibrating weights, but are also biased from exact antiphase and exact in-phase by the angles represented in the complex operators γ and δ.

Construction: Draw OC equal to γ times vector OB, and OD equal to δ times vector OB. Measure CA, AD, DC. Determine the numerical value of $\dfrac{AD}{DC}$ times OB and call this x. Determine the numerical value of $\dfrac{CA}{DC}$ times OB and call this y. Find B' so that OB' equals x and BB' equals y. Draw vector OA' equal to γ times vector OB'. Draw vector $A'A$ equal to δ times vector $B'B$ (remembering that γ and δ are complex operators).

Then the original vectors OA, OB are equivalent to an approximately "out-of-phase" component OA', OB' plus an approximately "in-phase" component $A'A$, $B'B$ and it can be shown that OA', OB' can be corrected by adjusting the calibrating weights added mainly for second mode and that $A'A$, $B'B$ can be corrected by adjusting the calibrating weights added mainly for third mode.

A very common situation in which this construction is of value arises on a semi-rigid rotor, particularly when balancing planes are available only at the ends of the rotor. By definition, such a rotor runs below its first critical speed. Nevertheless, it can respond to unbalance in both first and second modes and the planes available for correcting the unbalance will almost certainly result in complex effects of calibrating weights.

CONCLUSIONS

There is little or no relationship between the rigid and flexural mode shapes of a rotor. A low speed balancing machine considers only the rigid mode conditions and a rotor corrected by this means can still be unbalanced in its flexural modes. This may result in high vibration when it is run to speed in its correct environment.

The modal balancing method considers all the relevant modes of the rotor when mounted in simulated site foundation conditions and this ensures satisfactory running throughout the entire speed range. Even the condition of site foundation flexibility which causes a rotor to pass through both rigid and flexural criticals in running to speed is automatically covered by this method.

It may be however, that other than technical considerations dictate that only low speed balancing is used. In this event an understanding of modal behaviour and some knowledge of modal balancing should ensure the best use of the low speed facility, and prove invaluable if subsequent site correction is necessary.

ACKNOWLEDGEMENTS

The author wishes to acknowledge the assistance given by Miss E. G. Dodd in the preparation of the text and also to thank C. A. Parsons and Co. Ltd. for permission to publish this paper.

REFERENCE

1) MOORE, L. S. and DODD, E. G. 1964. "Mass Balancing of Large Flexible Rotors". *G.E.C. Jnl Science and Technology*, Vol. 31, No. 2.

Discussion

MR. T. W. BUNYAN, B.Sc. (Member) opening the discussion, said that he did not know about others present, but until he read the paper, the term modal balancing had had a somewhat sinister ring. Most were familiar with modes and antimodes in vibration, and now modal balancing opened up a new field. When serving their apprenticeships, it must have been a common experience for them to watch the man at the balancing machine, perform miracles of balancing with small strategically placed pieces of plasticine.

The paper sharply focused on the serious limitations of slow speed balancing machines, with soft bearing supports, for balancing turbine rotors of any size; having unavoidable but significant flexibility. Mr. Bunyan had no doubt that many would have had unsatisfactory experiences resulting from balancing done on such machines. The author himself had mentioned an instance of that.

Mr. Bunyan referred to an instance involving a three-bearing turbo alternator set of 1200 kVA.

The slow speed wheel with bearings on either side was coupled to the alternator rotor which had an outrigger bearing at the after end. All would be familiar with the type of construction. Two unsatisfactory factors evolved in the balancing. The first was that a satisfactory balance could not be obtained, i.e., the vibration velocity on any bearing was not less than 0·2 in/s r.m.s. The unbalance was in the low speed branch, the high speed branch was entirely satisfactory.

Secondly, it was found that the balance of the alternator, which had been previously balanced with a dummy shaft at full speed, and was excellent, when it was put on the slow speed balancing machine coupled to the alternator, they had to upset the balance of the alternator rotor to achieve a result which was unacceptable.

One could not accept an unbalanced alternator rotor because any rotor might in time have to be fitted to any one of three alternators on the ship.

It was suggested by the manufacturer that simple field balance in that case would put the job right. One could imagine a chief engineer trying to cope with a simple field balance in the middle of the Pacific—most impracticable. Under the circumstances, the best that could be done was to have a gearwheel unit carefully balanced at twice the speed at which they were previously balancing it, but working to much closer limits, and have the alternator rotor balanced using a carefully trued stub shaft as previously. When the two were coupled together, it was found that the 0·2 in/s limit was easily reached.

It was probably a little out of line with the terms of reference of the paper, but Mr. Bunyan asked for Mr. Moore's comments on that.

It was interesting and understandable that the flexural modes, associated as they were with the whirling criticals of the rotor itself, had to be balanced at speeds near to the whirling speeds and supported in bearings of similar stiffness to that of the turbine itself.

Mr. Moore had already mentioned the matter of bearing flexibility which in ships was quite a problem. Mr. Bunyan wanted to limit his comments to marine turbines and raised the following points: as an example, he took an L.P. main propulsion turbine rotor, developing 15 000 shp, and it would have an overhung L.P. astern turbine, solid with the L.P. rotor as that was now the order of the day. Of what order of relationship to the running speed would be the first, second and third modes?

The author had given an example on the feed pump motor, where one had values of 1500, 3800 for the second mode, and 6000 for the third mode.

The overhung L.P. turbine must considerably complicate and influence the balancing problem, particularly where the L.P. branch tuned with the first or second critical of the rotor. What was Mr. Moore's experience in that matter? Mr. Bunyan's own experience had been quite spectacular in one instance and provided the explanation for the persistent disintegration of the bearing between the main and astern turbines.

Mr. Bunyan wondered whether the author had any experience with carrying out an exploratory check of the three modes by exciting the stationary rotor flexibly supported in its bearings, and obtaining the critical frequencies. As all balancing deductions were made by readings of vibration amplitude and phase at the bearings, it would be very difficult to differentiate between the first rigid mode and the first flexural mode. He supposed the answer could be that static balancing on knife edges might deal sufficiently with the first rigid mode.

Another question, which was slightly "off-beat" was the problem of oil film whirl occasionally experienced with high speed auxiliary turbo-generator sets. It was probable that Mr. Moore had also been involved with that matter. What concerned the speaker was whether such a whirl condition could also be generated by the second or third mode of the rotor.

The last paragraph of the conclusions in the paper, he had found to be somewhat inconclusive. He thought that that was due, no doubt, to his own lack of experience with modal balancing. The body of the paper stressed the ineffectiveness of the low speed balancing machine with soft bearings. The last paragraph of the paper said: "It may be, however, that other than technical considerations dictate that only low speed balancing is used. In this event an understanding of modal behaviour and some knowledge of modal balancing should ensure the best use of the low speed facility, and prove invaluable if subsequent site correction is necessary."

Site balancing of main turbine rotors in a ship was extremely difficult, in many cases it was quite an impracticable job. To give point to the argument, it would have been better if the last paragraph, as indeed the paper did, clearly specified the limitation of the use of the low speed balancing machine with flexible bearings.

MR. W. S. ROSS, B.Sc. (Member) suggested that one of the reasons why low speed balancing of turbine rotors had persisted for so long was probably that a very large number of turbine rotors, especially those for the larger turbine, appeared to be in a category mentioned by Mr. Moore, which was that they ran well above their first critical speed and well below their second critical speed. Fortunately, they did not appear to develop the second or third mode shape to any significant extent when running at their full speed.

The result had been that even in the case of very large power station turbines, where a number of rotors were coupled together in tandem, it was still possible to obtain satisfactory running, even though the rotors had only been balanced in a low speed balancing machine with soft bearings.

However, it was becoming clear in the largest sizes of power station plant at present in operation, i.e. 500 MW, where one had five steam rotors and one electrical rotor in tandem, that the largest rotors of those tandem lines would benefit markedly by the application of modal balancing principles and that this trend would continue for the still larger machines now being designed.

Furthermore, there had always been rotors, sometimes marine propulsion rotors, sometimes rotors like the boiler feed pump which Mr. Moore had mentioned in his introduction to the paper, which had not responded to treatment in the low speed balancing machine. There were cases of rotors which had not run properly, and been sent back to the low speed balancing machine for correction where no one had been able to find anything wrong with the balance, i.e. the low speed balancing machine had not been able to reveal any fault in the balance.

In such instances, there could be no doubt that a great deal of time and often a great deal of money had been expended in applying inappropriate remedies for the vibration problem simply because up until now the theory and practice of modal balancing had not been widely understood.

Modal balancing required the running of the rotor at high speeds. If it was a turbine rotor, that had to be done under vacuum. That meant expensive test facilities had to be employed. It was not enough to spend money to provide the test facility. It was also necessary to make sure that the operators of the test facility had the adequate technical training mentioned by the author.

The personnel who were responsible for carrying out the balancing operations had to be able to do the vector diagrams, given in the paper and understand what they were doing. Usually, one could find people capable of doing that in the engineer's design office in the plant. However, such facilities, because of their expense and heavy capital investment, had to be operated round the clock and engineers had to sleep some time. Mr. Ross asked the author if he had experienced any difficulty in training balancing personnel to apply the methods properly without continuous supervision.

Many people thought that if one ran a rotor at speeds up to its maximum operating speed and put it into a state of balance, that one had carried out a modal balance. It would be quite clear from what had been said that this was not necessarily so.

MR. J. F. F. JOHNSON in a contribution read by Mr. A. E. Franklin (Member) said that no doubt the author, in the work described in this paper, had in mind some criteria, presumably in terms of the type and number of natural modes of vibration of a rotor falling within its running speed range, which determined whether a given rotor was to be classified as "flexible" or "rigid"?

Bearing in mind the limited choice of balancing planes usually available in a particular case and the admitted complexity of the full modal balancing techniques, one was very much inclined to believe that the immediate practical consequence of this paper would be a better appreciation of the factors involved in making the best use of such limited alternatives as were available in applying the two-plane force and moment method.

Finally, one wondered what was the author's attitude to damping? Was it friend or foe? It would appear that so far as the application of modal balancing was concerned, mixed modes were an unwelcome complication, and a system with low damping and well defined resonances was much to be preferred. This contrasted sharply with the attitude of some engineers who aimed for a highly damped system and deliberately included devices, e.g. squeeze film bearings, to achieve their aim. Mr. Moore's comments on this point would be particularly welcome.

Correspondence

MR. N. BLOUNT wrote that, with reference to the subsection headed *As Speed is Varied Near a Critical*, he could not agree that the differences between conditions near "rigid mode" criticals and those near flexural criticals were fundamental. Although, for practical purposes it was accepted, partially due to:

i) damping effects;
ii) the low ratio of rotor deflexion to bearing deflexion;
iii) the fact that rotors are not operating at high or critical speeds;

that rotors in soft bearing balancing machines did not normally deflect, the difference between (a) and (c) in Fig. 1 appeared to be only a matter of degree.

The deflexion of a given rotor at the first critical speed increased as the bearing stiffness was increased, there being a transition from the extreme "rigid" mode into "flexural" modes, the difference being arbitrary rather than fundamental.

The section "Methods of Balancing" did not make it sufficiently clear that the soft bearings of the balancing machine were only one of an infinite range of bearing stiffnesses which could "result in vibration when the rotor is run to speed in its own bearings". The differences between x_1 and x_2 at the first mode would vary (increasing from zero for "rigid" modes as the mode became more "flexural") thus it would appear that the only sure way of correcting the first flexural mode at one arbitrary axial position was to rotate the rotor in bearings identical in stiffness and weight with those to be used in service. It was stated that pedestals should simulate site conditions but the character of and reason for the similarities was perhaps not made sufficiently clear.

When not running exactly at a flexural critical speed the true modal shape was distorted by the amount and position of unbalance. Were such distortions significant and would they be discovered by obtaining a shape at a speed slightly below and again at a speed slightly above the critical?

Author's Reply

Mr. L. S. MOORE, replying to the discussion, said that Mr. Bunyan's question, on the balancing of a three bearing turbo-alternator, raised a fundamental point in all balancing work. Where the mechanical design dictated that a dummy shaft extension had to be fitted to a rotor in order to run in a balancing machine, there was a risk that the axis of rotation under this condition would be different from that at site. In the example quoted there was little reason to doubt that the problem would not have arisen if greater care had been taken to ensuer that the axis of rotation was the same for both site and balancing house conditions.

With regard to Mr. Bunyan's remarks on overhung L.P. turbine rotors, the modal balancing techniques automatically allowed for unbalances in any axial position along the rotor. The effect of a large overhung mass, however, might produce very unsymmetrical modal shapes and this point should be appreciated when proportioning the weights for the higher modes. It was likely that the rotor would be sensitive in second and third modes to unbalance in the overhang, but this could be corrected using normal balancing planes. If the overhang on its own had a resonance within the running speed range, it was highly desirable to support it with a steady bearing both during balancing and in the final site assembly.

The critical speeds and modal shapes obtained by exciting a stationary rotor could be useful as a rough guide but it should be remembered that, because of the dynamic characteristics of the oil film, the running criticals and associated modal shapes would be somewhat different. If this point was appreciated some useful but limited empirical knowledge could be accumulated by such tests.

It was quite true that static balancing on knife edges, as Mr. Bunyan suggested, would for all practical purposes deal with the first rigid mode critical, if it occurred within the running range. None the less it would have introduced a superfluous operation, as modal techniques would embrace this critical automatically. If modal balancing was to be used on a particular rotor, no advantage would be gained by prebalancing either on knife edges or in a low speed machine. Indeed, as the paper demonstrated, the problem could be aggravated by this means.

Mr. Bunyan raised an interesting point on oil whirl and wondered whether it might be associated with the second or third criticals of the rotor. As the phenomenon occurred at a speed corresponding to at least twice the critical speed involved, the question became rather academic. The author said that he had never encountered a turbine rotor of such flexibility that its running speed would be as high as twice second critical and had, therefore, never experienced oil whirl associated with other than first critical frequency.

It was interesting to note that Mr. Ross confirmed that although some flexible rotors responded satisfactorily to low speed balancing techniques, there was always a degree of risk and it was not possible to predict beforehand whether the process would be satisfactory.

Mr. Ross also mentioned the vexed question of training of balancing personnel. In the author's experience it was rather a question of indoctrination. It was difficult to train "Old Joe, the man on the balancing machine", who had preconceived ideas on balancing and a deep conviction of the intransigent nature of large rotors. Generally speaking, personnel with only a limited experience of balancing found it easier to accept the idea of a logical series of highly controlled and accurate operations. Once it was appreciated by the man concerned that with proper and accurate control, rotors would be made to behave in a predictable and logical manner, the rest of the training would be absorbed quite quickly, with interest and indeed with enthusiasm.

The limited choice of balancing planes mentioned by Mr. Johnson was a situation that arose far too often in balancing flexible rotors, and demonstrated the need for a better understanding of the subject by the design staff. The choice of suitable balancing planes should be settled at the design stage rather than waiting until the rotor had a physical existance.

The implication behind Mr. Johnson's final remarks that mixed modes produced a very difficult situation was not borne out in practice. Once it was appreciated that a rotor could distort into at least two modes simultaneously and the methods outlined in the paper were mastered, little difficulty should be encountered.

On the specific question of damping, there could be little doubt that it would be unwise to have a low damped system in an attempt to make the balancing operation easier. Indeed, if critical speeds were passed through during run up, or in normal operation, the problem of producing and maintaining a good state of balance would be increased.

The relevance of Mr. Blount's initial comment was not immediately apparent. The whole paper dealt only with the two extremes of the very flexible low speed balancing machine and the relatively stiff site bearing supports. Under these two specific environments the difference in the behaviour of the rotor in relation to balancing was quite fundamental.

It was quite true that the only way to balance the flexural modes exactly was to do so in bearings of identical weight and stiffness to those used in service. In practice, however, quite a fair degree of difference could be tolerated before the change in balance between the two conditions became significant.

Mr. Blount's final question was somewhat obscure. If a particular mass asymmetry produced components of unbalance in a number of modes, at any particular speed the rotor would distort to some extent into all of the modes simultaneously. What proportion of each would be present in the resultant shape would depend on the axial position of the asymmetry and the proximity of the various criticals to the running speed. This pattern of behaviour applied equally when running at an exact critical speed. For example, in the region of first critical there was often a small but quite measurable second modal component of unbalance. This would be equally detectable just below, just above and exactly at first critical.

Marine Engineering and Shipbuilding Abstracts

No. 4, April 1969

Intermediate Cargo Liner

Tyr, the first of a series of two intermediate cargo liners ordered from Sasebo by Wilhelm Wilhelmsen's for their Europe–India service, was delivered, just seven months after her keel-laying.

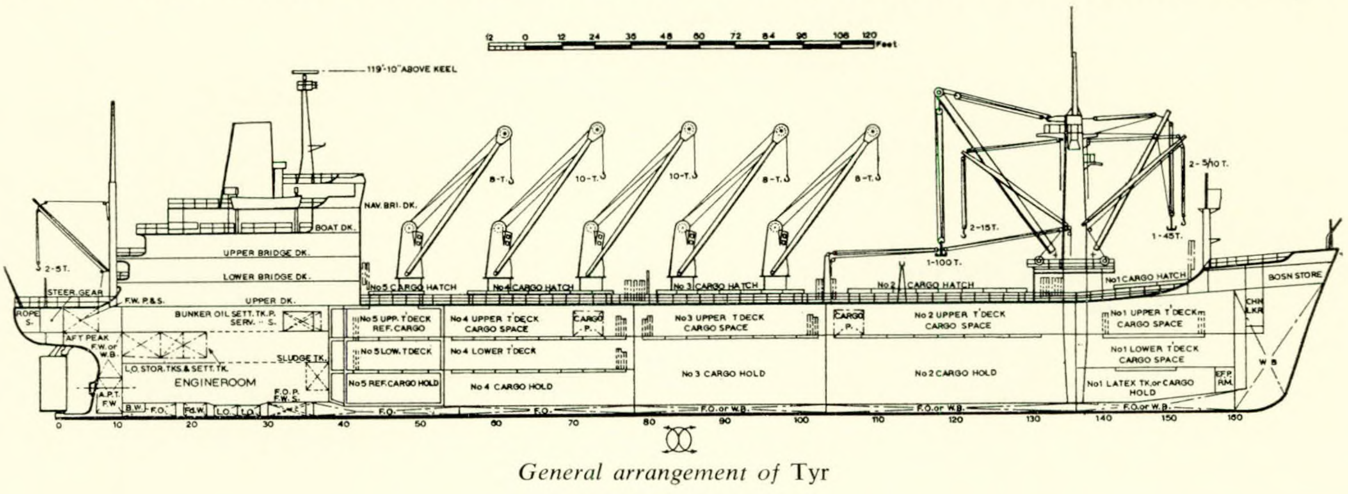
Built to the highest requirements of Lloyd's Register of Shipping, the vessel is of somewhat unusual configuration; a five-hold, machinery and accommodation right aft arrangement with heavy lift facilities well forward between holds Nos 1 and 2. The convenience of this loading layout, however, is offset to a degree by an increase in the longitudinal bending movement of the vessel in both laden and light condition—a 7395 ft^3 fore peak tank is necessary for sea water trim ballast—with an inevitable compensatory increase in the hull steel weight.

The choice of twin rather than the more usual triple hatch arrangement of today's open ship means that the deck beams either side of the centreline are cantilevered and that the only underdeck pillaring necessary is along the centreline where it also provides the necessary support for the travelling deck crane tracks. All hatches are closed by steel folding covers operated by a deck crane runner; 'tween deck covers are flush fitting and capable of supporting a laden four-wheel fork lift truck of eight tons all-up weight.

The service which the vessel is to operate called for the installation of comprehensive ship-borne cargo handling facilities.

Three eight ton and two ten ton electro-hydraulic deck cranes with long (62 ft 4 in) jibs are fitted on a fore and aft track between the hatches to give maximum spot loading facilities. The operating speed of the cranes at full load is 143/295 ft/sec.

Tyr is powered by a reversible twin NKK-Pielstick installation comprising two ten-cylinder PC2V units, each with an m.c.v. of 4000 bhp and a service rating of 3630 bhp at 428 and 415 rev/min respectively, turning through a two into one reduction gear-box, a four-bladed 16 ft 5 in diameter, 13 ft



General arrangement of Tyr

3 in pitch and 111·5 ft² developed area propeller at 135–131 rev/min at maximum/service speed. The main engines, which operate at a b.m.e.p. of 14·5 kg/cm², are equipped with IHI-BBC VTR400 exhaust turbochargers.—*Shipbuilding and Shipping Record, 8th November 1968, Vol. 112, pp. 598–601.*

Economics of Medium-speed Power Packs for Ship Propulsion

Increasing numbers of ships are being built in which the machinery installed performs not only its primary propelling duty but participates also in the care and handling of the cargo. Vessels engaged on world-wide tramping are frequently involved in long periods of estuarial navigation or riding at anchor in a tideway and are consequently faced with propulsion power demands of a very low order. Multi-unit systems are capable of producing power of this form economically by the simple expedient of selective shut-down of one or more prime movers.

One of the many advantages inherent in medium-speed applications is that of standardization. Operators of widely diversified fleets can select freely in powers ranging from 2000 to 30 000 hp from single and multiple engine systems embodying prime movers having a common cylinder capacity.

Typifying these identical and varying power combinations are the arrangements shown in Figs 1 and 2.

Fig. 1 illustrates two engines of identical power, co-planar reduction gear coupled to a single propeller. Combined flexible clutch couplings provide the means of separation between engines and gearing.

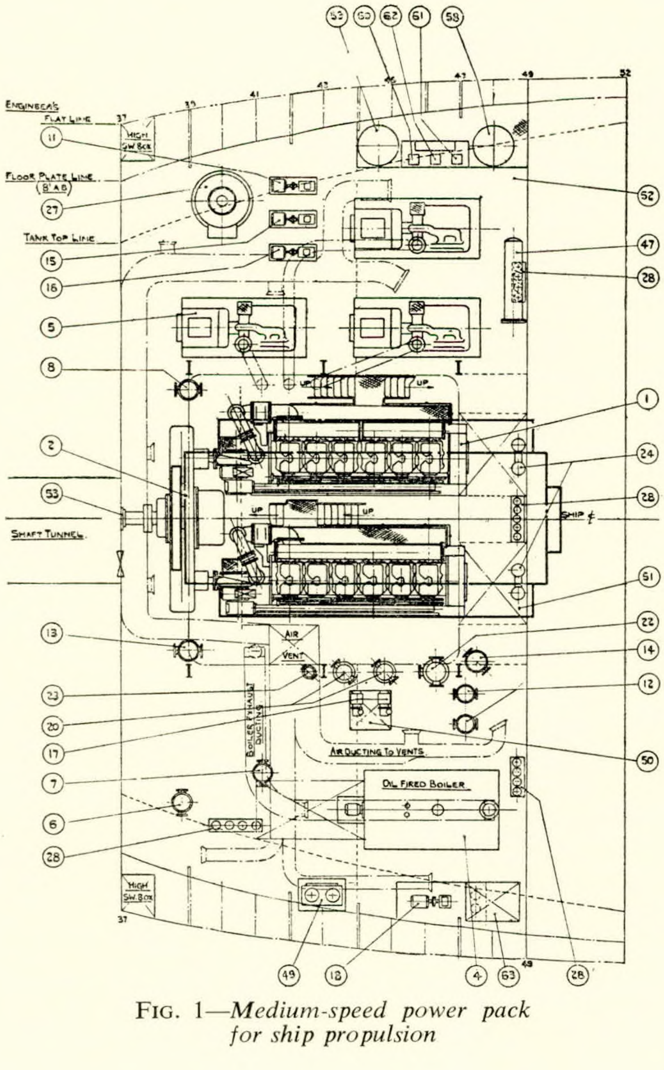


FIG. 1—*Medium-speed power pack for ship propulsion*

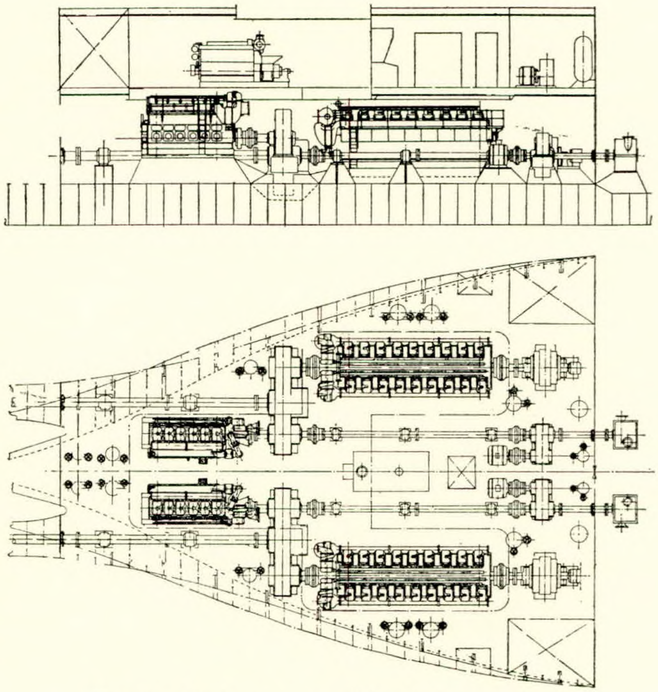


FIG. 2—*Twin-screw multi-unit installation incorporating ancillary drives*

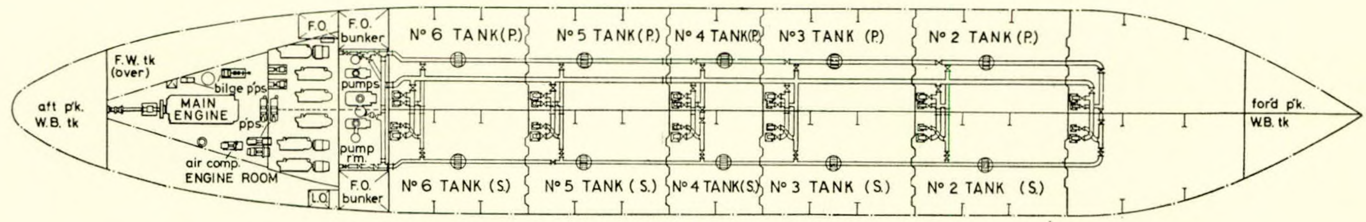
Where higher installed power is required that can offer a similar diversity of usage the arrangement shown in Fig. 2 ensures economic power production from various prime mover combines. Matching the characteristics of the equipment shown in Fig. 1 and combining this with a higher manoeuvrability factor, the twin screw proposal outlined in Fig. 2 increases the element of safety without detracting from the operational flexibility.

A contributory factor to the slow acceptance of multi-engine installation is without doutb the numerous sub-contract suppliers that are necessarily involved in the supply of what is, to the operator, essentially a composite power package. Rapidly increasing demand is, however, resulting in closer co-operation between associated suppliers which can only prove beneficial.—*Adley, A. A., Shipbuilding and Shipping Record, 1st November 1968, Vol. 112, pp. 570–574.*

Largest Japanese-owned Ore Carrier

The largest Japanese-owned ore carrier, of 102 000 dwt, was recently delivered to the Nippon Yusen Kaisha Line from the Tamano yard of the Mitsui Shipbuilding and Engineering Co. Ltd. Named *Furyu Maru*, the new vessel has a length, b.p., of 249 m, a breadth, moulded, of 39·6 m, a depth, moulded, of 19·7 m and a draught, loaded, of 14·48 m. To enable different types of ore to be carried simultaneously, the main cargo hold is subdivided into four compartments by non-watertight bulkheads: all hatches are fitted with electro-hydraulic hatch covers of the Ermans type.

The main engine is a Mitsui-B. and W. 984-VT2BF-180 unit capable of a maximum continuous rating of 20 700 bhp at 114 rev/min, with a normal output of 18 900 bhp at 110 rev/min, giving a service speed of 14·85 knots. Extensive automatic and remote control systems are fitted to the main and auxiliary machinery which are supervised from a central control room in the engine room. *Furyu Maru* has a total complement of 34 and is to operate on the owner's service between South America and Japan.—*Motor Ship, September 1968, Vol. 49, p. 287.*



Petroleum products carrier cargo tank and piping arrangements

Petroleum Products Carrier

Built by Cochrane and Sons Ltd., a Ross Group subsidiary, at their Selby shipyard, *Rudderman* is a 2957 dwt tanker for C. Rowbotham and Sons (Management) Ltd., London. This vessel will mainly be used for coastal work carrying petroleum products.

Rudderman, which was side-launched, has been built to the latest rules of Lloyd's Register for classification ✠ 100 A1 Oil Tanker and to satisfy Board of Trade requirements, Class VII, long international voyages for vessels under 1600 grt. This is a single-screw ship with an all-welded hull of round bilge form having a radiussed sheerstrake and raked round nose plate stem with an elliptical stern. There is a single streamlined semi-balanced rudder.

Principal particulars are:

Length, o.a.	274 ft 0 in
Length, b.p.	260 ft 0 in
Breadth, moulded	41 ft 0 in
Depth, moulded	20 ft 0 in
Gross tonnage	1592
Deadweight	2957 tons
Oil fuel capacity	130 tons
Machinery output ...	1530 bhp at 350 rev/min
Speed	12 knots

Rudderman has a continuous main deck and raised forecastle with a poop and deckhouse aft. The hull is subdivided into fore peak tank, six cargo tanks, pump room, engine room and after peak tank.

The cargo tanks are divided by a longitudinal oiltight centreline bulkhead.

Propelling machinery consists of a Drypool-Brons type 16.GV pressure-charged, direct reversing two-stroke engine capable of developing 1530 bhp at 350 rev/min. Engine control is effected through electro-pneumatic push-button remote control equipment. The control panel has start, ahead, stop and start astern buttons, speed lever and alarm lights. In the event of lubricating oil pressure or cooling water temperature of the engine exceeding predetermined figures, audible and visual alarm warnings are given in the engine room and wheelhouse. In the event of failure in either of these systems the engine is automatically shut down to idling speed. This engine drives a four-bladed manganese bronze propeller of Lips manufacture having a diameter of 2100 mm and a pitch of 1400 mm.—*Shipping World and Shipbuilder, December 1968, Vol. 161, pp. 1907–1910.*

Tests on Diesel Engine Cylinder Oil

It is well known that the various designs of two-stroke marine Diesel engines differ in their cylinder wear rates and deposit-forming characteristics, and practical experience has shown that some engines have particular cylinder lubricant requirements. Thus, before the introduction of the cylinder oil Mobilgard 504, an extensive programme of development tests, both in the laboratory and on board various ships, was carried out by the manufacturer.

The first phase of the development programme dealt with a study of oil formulation factors involved in the control of deposit formation and wear carried out in laboratory, bench and engine tests was designed to explore the following influences:

1) oil viscocity;
2) alkalinity level;
3) additive type.

A large number of experimental formulations were discarded since they were found to be deficient in one or another of the factors in the laboratory, bench and engine tests; finally 14 formulations were selected for shipboard trials.

Each of these oils was then tested at sea over a period of 2000 to 3000 operating hours in several cylinders of a 750 mm-bore, two-stroke turbocharged engine operating on 1500 sec Redwood No. 1 and 3·5 per cent sulphur fuel. The remainder of the engine cylinders were operated on a reference oil with a high alkaline total base number and SAE 50 viscosity. Measurements of cylinder liner wear rate and port clogging for all the oils concerned were checked at regular intervals and compared as percentage increases or decreases against the results from the reference oil. It was noted that neither viscosity change, alkalinity level, additive metals nor ashless additive materials produced consistent or large decreases in liner wear rates or port clogging. Large and significant decreases, however, resulted from changes in the organic structure of the additives.

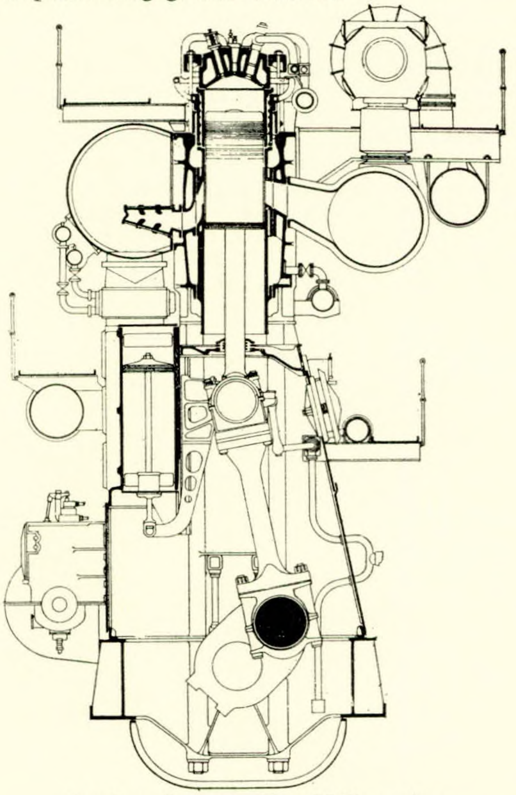
From these trials and their subsequent findings, one oil was chosen which gave the best results and this was then further tested in 12 different ships on varying services throughout the world and with different types of two-stroke, crosshead engines. The results of these tests, from 44 cylinder units totalling 137 743 operating hours, were again compared against the reference oil and in some cases a high-alkaline competitive cylinder lubricant. For the most part, these results paralleled those obtained in the initial ship tests.

However, the degree of response differed and it was found that a few of the engines showed no change in port clogging rates but a decrease in wear rates while several others showed the opposite. In all the cases no limitations in application were found during the six to eight months' period of the tests.—*Motor Ship, September 1968, Vol 49, p. 294.*

Fiat's New 780 mm Engine

The Lauro Shipping Group has ordered three fast 12 730 dwt cargo liners from the Genoa-Sestri yard of Italcantieri.

The propelling machinery of each vessel is to consist of a seven-cylinder Fiat 780S engine of 14 000 bhp m.c.r. at 126 rev/min. Two engines will be built by Fiat Grandi Motori at Turin, and the third by the San Andrea engine works of CRDA at Trieste. These are the first orders for the new 780 mm engine, a two-cylinder prototype of which has been running at Fiat's Stura works for some time. Distinctive features of this new size which is intended to supplement the 750 mm range are a much longer stroke/bore ratio (2·05) than Fiat have hitherto used. Bedplate and columns are fabricated, the former having a flat bottom, again new to Fiat practice. The cylinder liners have strongbacks to relieve tensile stress in the iron under the higher working pressures and permit the inner walls to be used. The cast CrNiMoVa steel cylinder cover is secured by a separate cast steel ring and 16 studs while the main bearing caps are held down by two jack studs. The pistons, too, are of strongback type with chromium-plated ring groove surfaces.



Section through Fiat 780S engine

This new engine has high specific output, for an eight-cylinder engine producing 16 000 bhp is just over 14·2 m long compared with the 17 m of the ten-cylinder 750 engine which is necessary to produce the same power. The cylinder liners of large Fiat single-acting engines are always open to the atmosphere and the new engine conforms to this practice. The turbochargers operate on the constant-pressure system.—*Marine Engineer and Naval Architect, December 1968, Vol. 91, p. 521.*

General Cargo Ship

Built by Henry Robb Ltd. for Ellerman and Papayanni Lines Ltd., *Mediterranian* is an improved open shelter deck version of *Athenian*, also built by Henry Robb Ltd. The intention to use this ship for the carriage of general cargoes is reflected by the cargo handling arrangements and ample lashing facilities provided in the cargo spaces and on deck.

This ship has been built to Lloyd's Register requirements under special survey, for Classification ✠ 100 A1 ✠ LMC, and also complies with the 1965 Merchant Shipping Life-saving and Fire Appliance Regulations. *Mediterranian* has a raised forecastle and raked stem with a soft-nosed bow and a curved transom stern having a cruiser profile.

Principal particulars are:

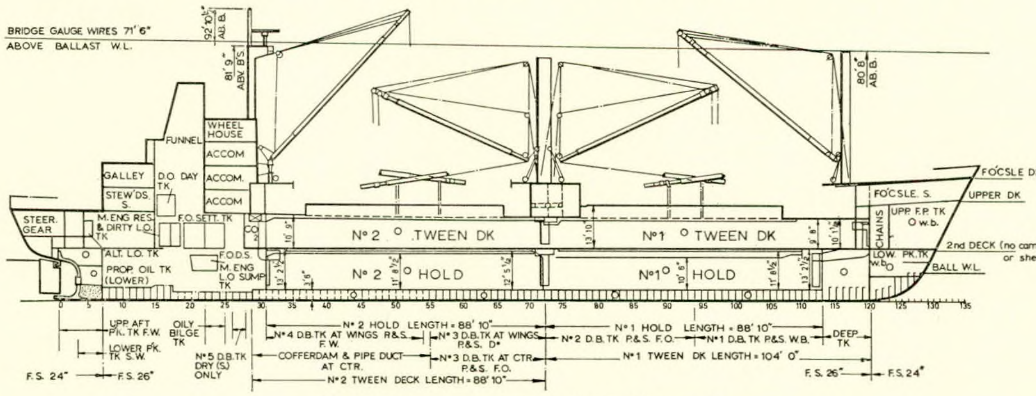
Length, o.a.	309 ft 1$\frac{1}{4}$ in
Length, b.p.	280 ft 0 in
Breadth, moulded	47 ft 6 in
Depth, moulded to upper deck ...	26 ft 0 in
Depth, moulded to main deck ...	16 ft 9 in
Mean summer draught	16 ft 8 in
Displacement	4151 tons
Deadweight	2476 tons
Gross tonnage	1459
Net registered tonnage	729
Bale capacity	179 200 ft^3
Service speed	13 knots
Trial speed	14·8 knots
Oil fuel capacity	323 tons
Fresh water capacity	156 tons

The large cargo holds and 'tween decks are served by four crane-derricks, two of 10-tons and two of five-tons safe working load. These are each operated by one man from a centralized control position. Norwinch hydraulic winches are used for the slewing, topping and hoisting movements. The 10-ton derricks are positioned each side of a single mast between Nos 1 and 2 holds with the five-ton capacity derricks positioned at the fore and aft ends.

The propelling machinery comprises a six-cylinder Mirrlees National turbocharged Diesel engine developing 2580 bhp at 280 rev/min driving a Novoston four-bladed controllable pitch propeller of Stone KaMeWa design having a diameter of 8 ft 2 in.—*Shipping World and Shipbuilder, December 1968, Vol. 161, pp. 1893–1896.*

Three-blade Rudders on Seagoing Ships

The Jenckel three-rudder system, and its satisfactory performance led to its being considered as a means of improving the manoeuvrability of East German stern trawlers, in particular that of a series of freezer trawlers (length, b.p., 145 ft, displacement 29 800 ft^3, and a service speed of 12 knots) building at the VEB Elbe-Werft, Boizenburg. The first few trawlers of this type had a nozzle rudder; this did not give satisfactory manoeuvrability, particularly during trawling in heavy seas and side winds and the fifth and later ships of the series were fitted with Jenckel three-rudder systems behind fixed Kort nozzles. A controllable-pitch propeller has been fitted from the outset.



General arrangement of Mediterranian

This rudder and nozzle arrangement, adopted after tank tests had been quickly carried out, is giving satisfaction. The manoeuvrability is now as good as that of a ship with an active rudder. The advantages due to the fixed Kort nozzle and those due to the Jenckel rudders, are listed.—*Jenckel, F. W., Schiffbautechnik, December 1967, Vol. 17, pp. 694–696; Jnl Abstracts B.R.S.A., July 1968, Vol. 23, Abstract No. 26 526.*

Large Coastal Barge

A British coastal shipowner who has brought into service several purpose-designed specialized vessels is James Fisher and Sons Ltd. of Barrow-in-Furness, and the latest addition to this fleet was recently delivered from the Krager yard of A/S Tangen Verft.

The new ship, a coastal bulk carrying barge powered by two Rolls-Royce engines driving well-type Schottel propeller rudder units, is owned by Anchorage Ferrying Services Ltd. and was ordered from the Norwegian builders, by A/S Bulkhandling of Oslo. Named *Odin*, the vessel is built to the requirements of Lloyd's Register of Shipping ✠ 100 A1 classification and in accordance with the British Board of Trade Class 8 rules as well as Den norske Skipskontroll. Besides having its own propulsion machinery the vessel is arranged for towing, unmanned, as a seagoing barge, but is essentially designed for service within two miles of the coast. Principal particulars are:

Length, o.a.	240 ft 0 in
Length, b.p.	234 ft 9 in
Breadth, moulded	40 ft 0 in
Depth to deck	23 ft 0 in
Draught, for scantlings	22 ft 0 in
Draught, designed	18 ft 9 in
Corresponding deadweight ...	3380 tons
Hold capacity (grain)	153 160 ft^3

Of all-welded construction, *Odin* has a single continuous deck, no forecastle, transom stern and a small deckhouse aft with accommodation for a crew of five. To achieve good visibility from the wheelhouse over the length of the vessel, it has been arranged some eight feet above the top of the accommodation deckhouse, access being by an enclosed stairway leading from the accommodation to the upper deck and thence to the wheelhouse.

Main propulsion of *Odin* is by two Rolls-Royce eight-cylinder DV8TM engines each developing 495 bhp at 1800 rev/min and each driving a Schottel SRP 300 rudder propeller well-type unit. The drive from each engine is taken to the upper gear-box of the Schottel units, which are fitted with hardened steel, helicoidal bevel gears. From this box, power is transmitted to the propeller units, some 14 ft 6 in below, through specially constructed wells in the engine room double-bottom. The propellers are made from Ni Resist alloy and are of the push type. For lubrication the complete rudder propeller units are filled with oil which is constantly circulated through the underwater gear-box for cooling. Steering and speed control are hydraulically-powered from the wheelhouse, the steering being assisted by servomotor units.

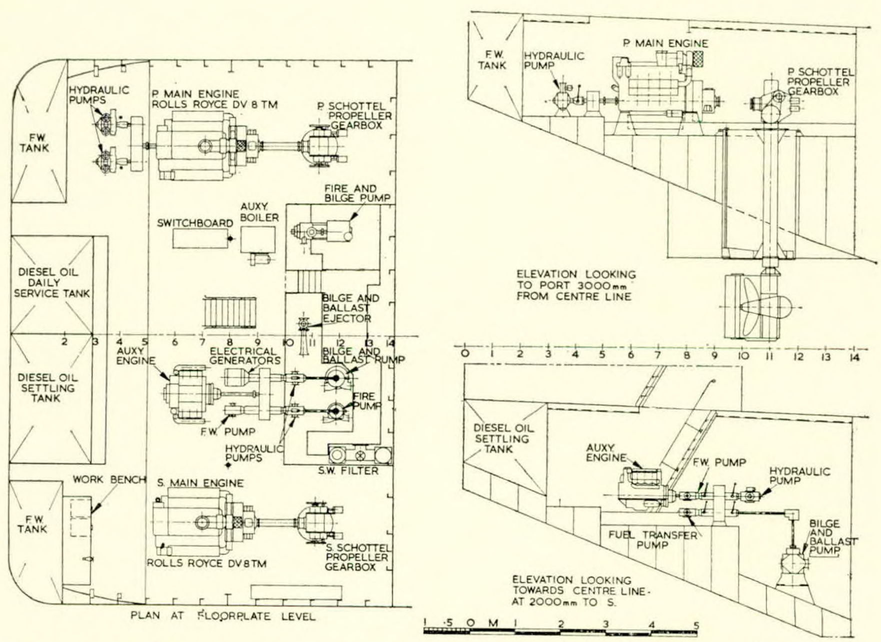
To supply hydraulic power for steering and the deck equipment, there are two 38 hp hydraulic pumps driven by the port main engine through a suitable gear-box and clutches. *Motor Ship, December 1968, Vol. 49, pp. 443–444.*

Hapag Lloyd Container Ships

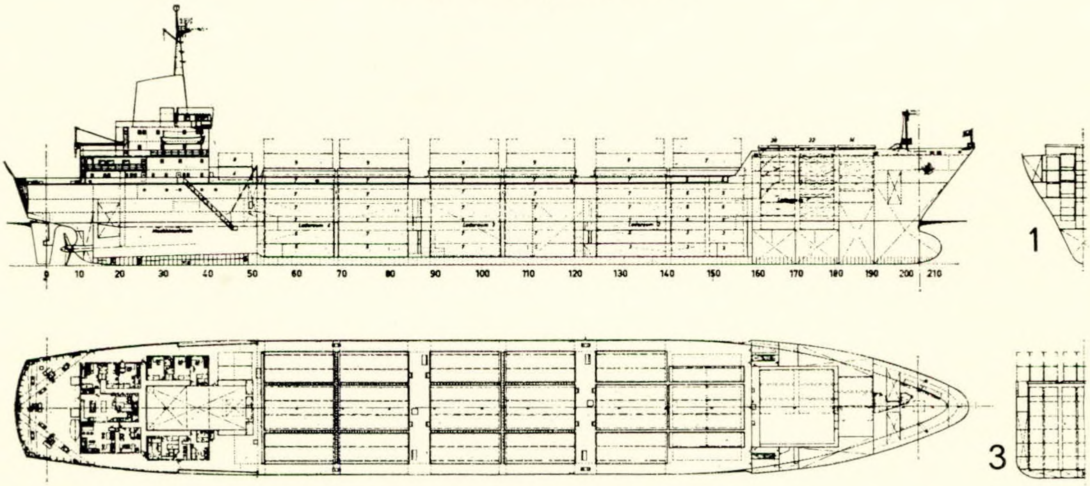
The container ship *Weser Express*, the North German Lloyd will operate jointly with two others on order from Blohm and Voss for the Hamburg-Amerika Line. This vessel, which is the first German-flag ocean-going full container ship, has the following principal particulars:

Length, o.a.	560 ft 0 in
Length, b.p.	508 ft 1 in
Breadth, moulded	80 ft 5 in
Depth	47 ft 11 in
Draught	25 ft $8\frac{1}{2}$ in
Gross measurement	14 500 tons
Deadweight	10 800 tons
Containers (20 ft)	728
Service speed	20·6 knots
Passengers	12

These ships are of very advanced design and will be fitted with fin-type stabilizers, automatic heel correction and stability indicators. The machinery consists of a Bremer Vulkan-M.A.N. nine-cylinder K9Z78/155E engine of 15 750 bhp at 122 rev/min. Auxiliary power will be provided by shaft-driven generators at sea and Diesel generators in harbour. The machinery has been arranged for unattended opera-



Engine room arrangement of twin-screw Odin *with service speed of 9·5 knots*



General arrangement of Weser Express

tion for periods of up to 16 hours. Accommodation will be provided for 12 passengers and a ship's company of 35/36 in single-berth cabins, each with private shower and toilet facilities.

As the profile drawing indicates, there are four main cargo compartments. The after two are closed by tripled hatches giving stowage for seven containers abreast (two in the wings and three on the centreline) and nine stowed abreast in two tiers on deck. Due to the narrowing forebody a modified arrangement is necessary for No. 2 cell and the long forecastle is raised to a height equivalent to that of two containers. This enables all the stowage in this area to be under cover of three slab-type hatches. The drawing also shows that there is deck stowage for containers immediately forward of the bridge and above the engine room.

The profile is characterized by a heavily raked stem with bulbous bow, transom stern and suspended balanced rudder. The engine room hatch is immediately forward of the bridge. —*Marine Engineer and Naval Architect, August 1968, Vol. 91, p. 311.*

Coastal Inspection Vessel

Aalborg Vaerft has recently completed m.v. *Aegir*, a coast inspection and salvage vessel designed for North Atlantic service and built to the order of the Icelandic State. Complying with the highest class of Lloyd's Register of Shipping and strengthened for navigation in ice, the vessel has an overall length of 229 ft., a moulded breadth of 33 ft. and the depth to upper deck is 19 ft.

As the vessel will serve in the North Atlantic regions, the accommodation is so arranged that, when the vessel is in heavy weather, access can be gained from one part of the ship to another without crossing an open deck.

The vessel is fitted with electro-hydraulic steering gear, two hydraulically operated vertical capstans and a 20-ton hydraulic towing winch aft. Both the capstans and the towing winch can be remotely controlled from the wheelhouse. A television system is installed on the vessel whereby a view of the towing winch and quarterdeck is displayed on a monitor on the bridge. A helicopter hangar is positioned aft of midships between the twin funnels ad there is a large helicopter landing deck at the aft end of the vessel. Provision is made for mounting a cannon on the boat deck forward of the bridge and on the forecastle there is a mast with a five ton derrick for salvage purposes.

Propulsion machinery comprises two M.A.N. trunk-piston type turbocharged and aftercooled Diesel engines each driving a three-bladed KaMeWa c.p. propeller. The auxiliary machinery consists of a M.A.N. Diesel engine directly coupled to an alternator having an output of about 315 KVA at 1000 rev/min. In addition there is a shaft driven alternator having the same output and speed. Both alternators supply current at three-phase, 380 V, 50 c/s and are fitted with a fully automatic synchronizing system.—*Shipbuilding International, September 1968, Vol. 11, pp. 38–39.*

Dutch-built Cargo Liners for Royal Inter Ocean Lines

The first two ships of a series of six cargo liners, built in Holland for the services of Koninklijke Java-China-Paketvaart Lijnen N.V. (Royal Interocean Lines), Amsterdam, have now entered service. They are *Straat Amsterdam*, built by Verolme United Shipyards, and *Straat Adelaide*, built by Van der Giessen-De Noord. The ships have been constructed to the design and specifications of R.I.L.

Straat Amsterdam has been constructed to Lloyd's Register of Shipping ✠ 100 A1, LMC, RMC "edible oil or latex in deeptanks". She also fulfils the requirements of the Dutch Shipping Inspection Bureau and the Port Labour Inspection Service. The hull has been constructed for minimum freeboard, with four cargo holds forward and one aft of the engine room. The vessel has a long forecastle and long bridge, the aftermost parts of which are utilized as cargo holds. The entire accommodation is disposed over various decks in the bridge deckhouse.

Deeptanks are situated in No. 1 lower tweendeck, No. 1 lower hold and No. 2 lower hold. Three of the tanks are suitable for the carriage of liquid cargo, dry cargo or water ballast. Of the cargo holds, the Nos 3 and 4 hold are open, having three hatchways abreast in all the decks. The four refrigerated compartments in No. 4 hold and in No. 5 bridge space can be loaded entirely separately from the other holds via insulated cargo hatches.

The hull of the ship is constructed with longitudinal framing in the double bottom, with the B deck forming the strength deck. Two passive anti-rolling tanks of the U-type which were designed by the Dutch TNO Research Organization, are installed one above the other at the afterend of No. 4 hold.

The ship is provided with an Oertz rudder and steering is effected by a four-ram Hastie and Co. steering engine. The rudder carrier is of the same make. Steering from the bridge is entirely electric by means of an Arkas all-electric autopilot.

The main propulsion machinery of the ship consists of a Stork Diesel engine, type SW6x80/160, having a maximum service output of 13 500 shp at 117 rev/min. The engine is turbocharged by means of two exhaust-gas turbines, type BBC VTR630, provided with sleeve bearings. The inlet and outlet housings of the turbines are provided with a protective lead coating. A scavenging system is fitted for the cleaning during operation of the air and exhaust gas parts of the exhaust gas turbines. On failure at sea of one of the exhaust gas turbines, an electrically driven emergency blower can be used, as a result of which an output of 6000 shp can still be developed, so that the ship's speed will be 16 knots. The main engine is suitable for burning heavy fuel with a viscosity of 3500 sec Redwood No. 1, at 100°F.—*Holland Shipbuilding, October 1968, Vol. 17, pp. 44–52.*

Steam Turbines for LASH Vessels

The main propulsion turbines and gearing for the 11 revolutionary LASH cargo vessels are to be built by De Laval Turbine Inc.

Instead of loading cargo directly into a LASH vessel's hold, containers are first loaded into individual lighters, a step normally completed in advance of the vessel's arrival. Upon reaching port, the LASH ship need not be docked at a quay, but can anchor at a convenient location where pre-loaded lighters are embarked over the stern and transferred to the hold by means of a travelling deck crane. Using this cargo-handling arrangement, LASH vessels can reduce their port time to a minimum. A full complement of lighters can be loaded in less than 24 hours.

When a LASH vessel arrives at its destination the shallow-draught lighters are lifted out and may then proceed to berths unsuitable for conventional ships, or even miles up shallow creeks. The LASH vessels have obvious advantages in utilizing under-developed ports which may not have adequate dockside facilities.

The turbines which De Laval are supplying are from the new DLT-M range which covers powers from 18 000 to 40 000 shp. They are designed for modern regenerative cycles using steam conditions of 850 lb/in^2 and 950°F total temperature at the throttles with an exhaust pressure of $1\frac{1}{2}$ in abs. The number of frame sizes required for the power range has been minimized in order to achieve maximum interchangeability of parts. There is only one H.P. turbine (two if reheat is considered) and four L.P. frames, two with axial and two with down-flow exhaust. The new turbines are combined with De Laval's latest reduction gear designs of both the locked-train and articulated types and include co-ordinated condenser designs to give a total propulsion package.

The ahead bar-lift valves and the astern throttle are controlled remotely by electro-hydraulic actuators. The L.P. turbines have tapered, twisted nozzles and blades designed according to the modern constant mass circulation principle with mechanical strength to ensure high efficiency and reliability.

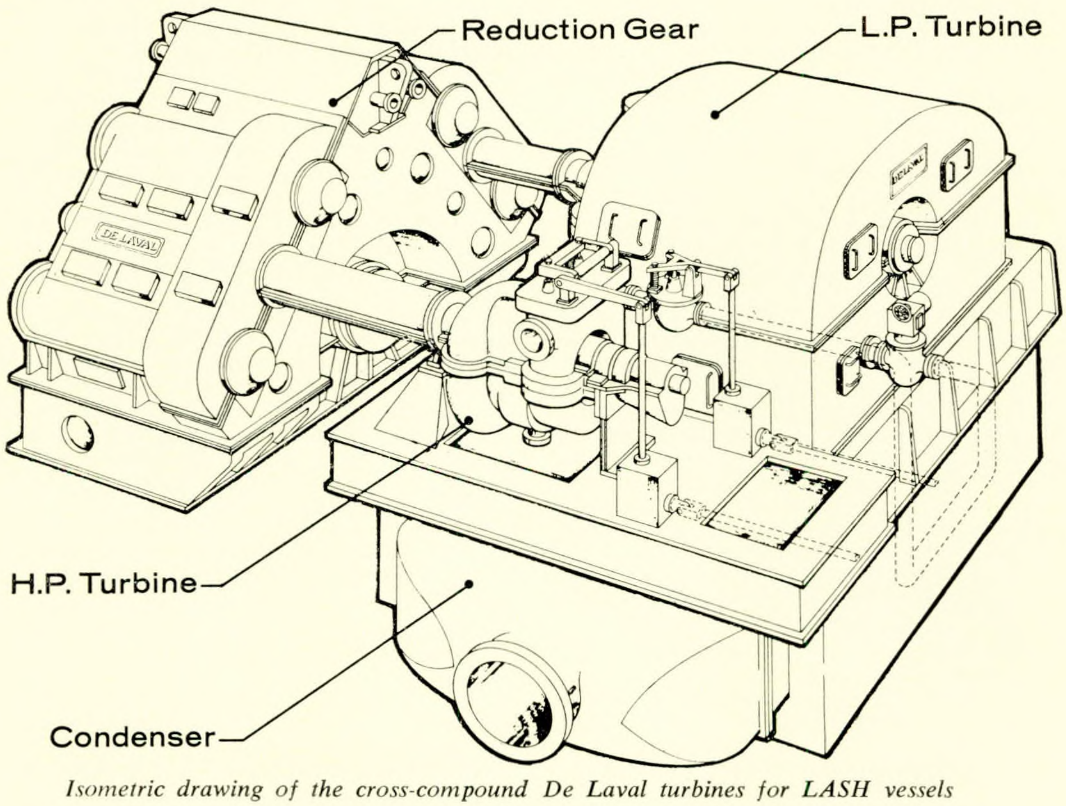
The LASH vessels will each have one DLT-M-40 main propulsion unit. This has a nine-stage impulse turbine with an integral steam chest designed to give maximum freedom and minimum thermal distortion with 950°F inlet steam temperature, thus ensuring reliable joints and minimum seal wear. The L.P. turbine also has nine stages and the thrust bearing is at the after end, to minimize the effect of thermal expansion upon the axial clearances. The exhaust is downward into the condenser. The turbines have centreline-supported diaphragms that allow for expansion in any direction without affecting the clearances at the steam seals. Labyrinth-type spring-backed seals are used throughout.

Both H.P. and L.P. turbine rotors are solid forgings and the journal bearings are of De Laval tilting-pad type which provides for self-alignment while ensuring a more stable bearing than the common sleeve type, thus eliminating problems of instability and oil whip. The rotor thrust bearings are of the six-shoe segmental self-aligning type. Although rated for 32 000 shp in this application, the M-40 frame size is capable of producing up to 40 000 shp.—*Marine Engineer and Naval Architect, October 1968, Vol. 91, pp. 408–410.*

Centrifugal Pump Priming System

The PrimaVac automatic priming system for cargo and other centrifugal pumps, developed in the U.S.A. has now been introduced into the United Kingdom.

The system utilizes the ship's main cargo pumps for the



Isometric drawing of the cross-compound De Laval turbines for LASH vessels stripping of tanks and therefore does not require any separate stripping pumps or pipework. Fully automatic in operation, the PrimaVac System makes conventional cargo pumps completely self-flooding and eliminates the need for back-flooding, vacuum pumps and separate priming equipment. The pump never runs dry and can operate against a shut discharge valve or with a completely dry tank without any risk of damage to the pump. Air and gases are evacuated with the fluid being pumped.

Basically, the PrimaVac System consists of a sensitive and fast-acting re-priming mechanism located in the pump discharge line. Unlike other systems, it reacts to changes in velocity of the liquid through the pump. During normal operation the velocity is high and the re-priming mechanism is in balance, the re-circulation valves are closed and thus the liquid pumped flows through the discharge lines. When the suction is lost the velocity of the fluid drops, this causing unbalance in the system which opens the re-circulation valves. A predetermined volume of liquid flows back into a re-circulation tank on the suction side of the pump, thus re-priming it.

Re-circulation valves are replaceable poppet valve cartridges having a number of advantages over other types of valve. The poppet valve can easily be held in balance by a simple sensing mechanism, it does not project into the flow path of the liquid and does not offer physical resistance or back pressure.—*Shipbuilding International, October 1968, Vol. 11, pp. 10–11.*

Principal particulars are:

Length, o.a.	700 ft 6 in
Length, b.p.	670 ft 0 in
Beam, moulded:	
at W.L. (approx.)	85 ft 0 in
extreme	90 ft 0 in
Draught, loaded	31 ft 6 in
Depth, keel to main deck ...	50 ft 6 in
Full load displacement	31 987 tons
Net tonnage	13 262
Gross tonnage	18 765
Service speed	23 knots
Range at service speed	10 000 miles

The main machinery comprises a General Electric marine steam turbine developing 26 000 shp and driving the single propeller at 117 rev/min. Steam conditions are 850 lb/in^2 at 940°F. A Central Operations System (COS), has also been supplied by G.E.C. which provides the monitoring and control functions for the main machinery. A bridge control system, complementing the COS engine room control, gives the officer on watch the ability to control propeller revolutions, ahead and astern, from the bridge by direct control of the throttle system or by the conventional telegraph relay system to the engine room. General Electric are also providing electrical drive systems for cargo handling and constant tension winch electrical drive systems for automatic mooring.—*Shipping World and Shipbuilder, August 1968, Vol. 161, pp. 1374–1375.*

American-built Fast Container Ship

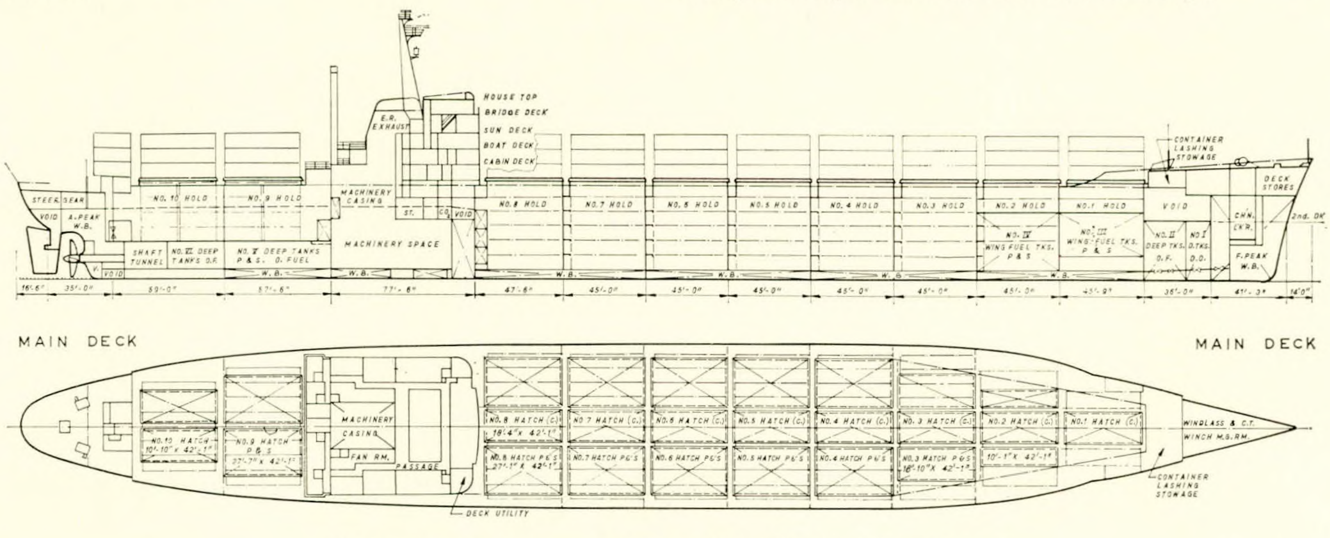
Built by the Sun Shipbuilding and Dry Dock Co., for the United States Lines, *American Lancer* is the first of a series of six ships building for the same owner. In company with *American Legion* and *American Liberty*, will be used on the container service between Rotterdam, Tilbury, Hamburg, Norfolk, Va. and New York.

The ship has ten holds, Nos 1 to 8 are situated forward of the bridge supersturcture and Nos 9 and 10 holds aft. Containers are stowed in cells below deck, standing one on top of the other, six deep. Container capacity is 200 standard ISO 8 ft by 8 ft, 20-ft containers and 220 40-ft containers. On deck, they can be stacked two high when 364 20-ft containers or 174 40-ft and eight 20-ft containers can be accommodated. When stacked three high, the maximum permissible, 534 20-ft containers or 261 40-ft with 12 20-ft containers can be carried. Provision is also made for the carriage of refrigerated containers.

Steam v Diesel — Up to 3000 hp

The controversies of the late twenties and early thirties have subsided. The post-war verdict favours the internal combustion engine. A triumph indeed for the Diesel, but perhaps a Pyrrhic victory. Does the internal combustion engine retain its reliability ten years after it has taken its first revolution? Do not running costs and the formidable planned maintenance schedule prejudice its continued existence?

Comparative advantages claimed for the Diesel are well known, and include low fuel consumption, consequential increase in dwt carrying capacity and no standby charges in port; but against these advantages the steam reciprocating engine scores in respect of lower fuel prices, lower lubricating oil consumption, smoother running and better torque, absence of noise and vibration, negligible maintenance, cheap spares and easy repairs and positive speed control. The present neglect of steam units stems perhaps from a lack of opportunity to compare units of more modern design.



General arrangement of the container ship American Lancer

Liner wear is negligible in steam engines and most ships can survive their span of life without a rebore. Renewals every 25 000 or 30 000 h are commonplace with Diesels and the replacement of eight or ten sleeves every third or fourth year is a heavy expense. Add to this the cost of pistons and gudgeon pins, numerous compression rings and scrapers and the overall maintenance is a heavy budgetry item. It may be less apparent because it is spread evenly over a period of years, but its total cost is often staggering.

However, the main concern with Diesels is crankshaft deflexion and bearing damage. Re-choking of an actively vibrating engine must be envisaged periodically and this is both time consuming and difficult. Yet it has to be done to avoid serious breakdowns. The steam engine hardly ever has crankshaft trouble and bedplate holding-down bolts need no attention between surveys.

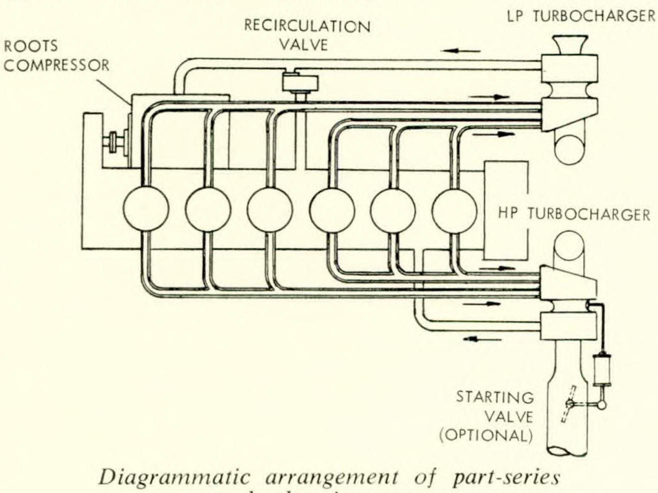
Adjustment of revolutions is considerably easier with steam engines and speed can be more precisely controlled from a genuine slow ahead to full astern. Often this fine measure of adjustment can help to dispense with the need for a variable-pitch propeller. When coupled to an exhaust turbine, racing in rough water is much reduced, the risk of breakage minimized and the smooth torque of expanding steam is effective in eliminating fatigue fractures of shafts or the failure of main hull structural members in the vicinity.

In the past, the Diesel manufacturers have striven hard to improve, develop and progress, while steam has remained a Cinderella. Had a small portion of the inventiveness bestowed on motors been spent in perfecting boiler plants, the situation might have been different. It may still not be too late, although a lot of prejudice will have to be overcome and training methods altered to suit.—*Barclay, C., Shipping World and Shipbuilder, July, 1968, Vol. 161, 1062–1064.*

Part-series Turbocharging

Sir W. G. Armstrong Whitworth and Co. (Engineers) Ltd., is a research and development company which does not itself manufacture but which sells licences.

A new system of turbocharging developed by the company and applied to an opposed-piston engine has resulted in the following figures: a fuel consumption of 0·335 lb/bhp-h at a top specific output of 4·25 hp/in^2 of piston area 0·66 hp/cm^2, and 2000 ft/min piston speed.



Diagrammatic arrangement of part-series turbocharging system

The economy of fuel at fractional loads is equally noteworthy. At the same 2000 ft/min piston speed the specific fuel consumption was only 0·41 lb/bhp-h at quarter load. Optimum efficiency ocurred at 1500 ft/min piston speed at which speed, half the power range was covered without the specific fuel consumption rising above 0·33 lb/bhp-h.

In simple turbocharging the exhaust gas driven turbocharger supplies all the air for the engine scavenging and charging: i.e. there is no engine-driven compressor or scavenge pump.

In series turbocharging the turbocharger delivers the air in series to the inlet of an engine-driven blower (usually of the Roots type) which in turn delivers it to the engine.

Effort was aimed towards finding a way in which both systems could be used on the same engine, at the same time, as shown diagrammatically in the figure. Normal turbocharging methods could not be used with such a system since it would lead to violent surge or blow-back through the simple turbocharger. It was found that this can be avoided by using two turbochargers having different internal flows.

A conventional turbocharger has equal flow through turbine and compressor, and this is its accepted method of operation for balance of expansion and compression work. However, when two turbochargers are used this balance need not apply and for the new system one turbocharger is given a larger than normal turbine nozzle ring and a smaller than normal compressor diffuser. The larger nozzle ring passes more exhaust gas and so makes the turbine run faster, while the smaller diffuser makes the compressor handle less air, which, to absorb the higher turbine power, it then compresses to higher than normal pressure. The other turbocharger, running in parallel, has a correspondingly smaller turbine and larger compressor and so runs more slowly, giving a lower pressure but handling a larger volume of air. This air is then compressed by the engine-driven Roots blower to a pressure equal to that of the high pressure turbocharger, and so both systems can feed the same air chest.

By these means the engine-driven compressor is halved in size and the parasitic losses of the engine materially reduced.

The part-series turbocharging system has two dissimilar turbochargers, running in parallel, one delivering air direct to the manifold and the other delivering at a lower pressure in series to an engine-driven compressor, which further compresses the charge and delivers it to the manifold.—*Gas and Oil Power, December 1968, Vol. 64, pp. 302–304.*

Minimizing Hydrodynamic Blade Spindle Torque by Ventilation

The engineering effort and the costs needed for the realization of a c.p. propeller installation are directly dependent on the magnitude of the actuating forces. These are the forces to be produced by the actuating mechanism for the changing of pitch during operation. The actuating forces are proportional to the blade spindle torque, that is the torque with respect to the spindle axis required to change pitch. In modern large c.p. propeller installations, with over 25 000 hp which is no longer exceptional, the blade spindle torque can easily raise the actuating force (e.g. in the actuating rod) to over 200 tons. The immense engineering problems of such heavy loadings put a limit on the realization of the shipowners' wishes concerning the c.p. propeller. A drastic decrease of the blade spindle torque, without affecting other aspects, could therefore mean an important step forward. How can this be achieved?

The blade spindle torque is made up of three parts:

1) a torque due to hydrodynamic forces;
2) a torque due to mechanical friction;
3) an inertia torque due to centrifugal forces.

The inertia torque is much smaller than the hydrodynamic and the frictional torque, although it should be taken into account in quantitative calculations of blade spindle torque. The friction torque depends directly on the hydrodynamic torque which itself is roughly half the total blade spindle torque. Significant gains are thus only to be expected from a reduction in the hydrodynamic blade spindle torque.

Of course, many attempts have been made to reduce the hydrodynamic spindle torque, but only the application of skew back has no significant additional disadvantages. It is not possible, however, to choose the skew in such a way, that a low hydrodynamic spindle torque is obtained over the whole range of operation. A useful improvement is therefore only to be expected from hydrodynamic means, which can be temporarily used and do not affect the performance of the propeller in the design condition. Such a solution may be found in movable flaps, which involves, however, serious mechanical problems.

A more practical means for the control of the pressure distribution over the blades may be the injection of air (or other gases) through holes in the blades, the air being supplied through tubes in the shaft and actuating mechanism. It is the aim of this paper to investigate and discuss this idea.

The following conclusions can be drawn from the investigation. The calculation of actuating forces of c.p. propellers with a digital computer seems to be sufficiently accurate, provided that proper allowances are made for:

a) the effect of heavy loading in the three dimensional analysis;
b) the distortion of mean lines due to pitch changing;
c) the effect of cavitation on the chord-wise pressure distribution.

Ventilation through holes on the face of the blade can lower the actuating forces by more than 50 per cent. This seems therefore to be worth further research. In order to check the promising results of this paper, model tests should be carried out, including the aspects of ventilation inception, air requirements and optimum location of air inlets.—*van Gunsteren, L. A., Holland Shipbuilding, October 1968, Vol. 17, pp. 90–96; 98.*

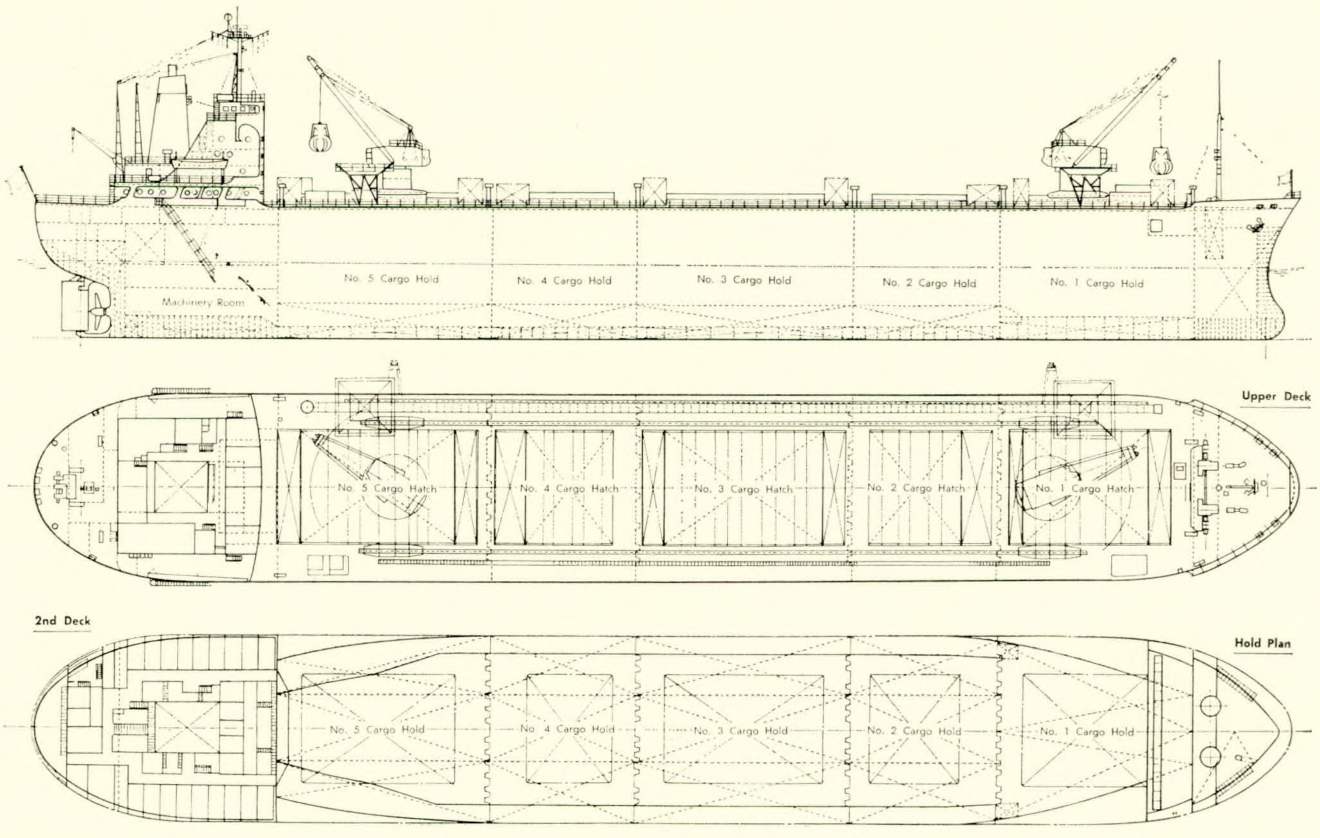
Analysing Stress Data from Ships in Service

In order to rationalize strength requirements in respect to wave loads, full scale statistical data have been collected on a number of ships over the last ten years. The range and scope of these investigations have varied considerably and, as a result, a number of techniques have had to be developed to make the best use of the various types of data recorded by the wide selection of instrumentation employed. B.S.R.A. activities in this field began in 1955 and since that date service stress data have been collected on 35 ships. Some of this work has already been reported and this present article deals with a more advanced data processing technique which B.S.R.A. has recently developed for these data.

Initially, only one ship's data was investigated representing a period of some 4000 watches although the position now is such that most of the collected data are stored in a form which can be readily analysed by computer.

It is possible to utilize the data in two quite unique ways. Firstly, it is possible by judicial selection of weighting factors to create hypothetical voyages in which realistic and unrealistic proportions of severe weather are encountered. The resulting theoretical distribution may be derived to determine the most likely maximum wave induced stress.

It is thought that by subjecting the ship to unrealistic lengths of time in severe conditions, the predicted maximum stress would be a useful guide to a design stress since the chances of this level ever being exceeded are extremely rare. Secondly, when data are available for several ships, the final distribution obtained from each ship may be used to compare their structural performance. By adopting a similar method to that above, the comparison may be even more weighty by forcing each ship to follow an identical voyage pattern. This procedure achieves something which in practice would be



General arrangement of Kathryn Maru

impracticable, that is sailing the ship on identical routes in identical sea conditions and headings.—*Taylor, K. V. and Larkin, E. W., Shipping World and Shipbuilder, August 1968, Vol. 161, pp. 1364–1366.*

Japanese Chip Carrier

The 27 597 dwt chip carrier *Kathryn Maru* is the latest addition to the chip carrier fleet of Yamashita-Shinnihon Steamship Co. She is provided with special cargo handling facilities, developed and supplied by Tsuji Industries, which constitute an important feature.

Principal particulars are:

Length, o.a.	176·20 m
Length, b.p.	165·00 m
Breadth, moulded	25·00 m
Depth, moulded	17·50 m
Draught, summer full load ...	10·023 m
Gross tonnage	20 686·14
Net tonnage	15 071
Class NK	
Main engine, Hitachi B and W 862VT2BF140 type	
Output (m.c.r.) ...	9600 bhp at 139 rev/min
Output (n.c.r.)	8160 bhp at 132 rev/min
Maximum trial speed	16·98 knots
Service speed (full load)	14·5 knots
Cargo holds capacity (grain) ...	48 879 m^3

—*Zosen, August 1968, Vol. 13, pp. 32–33; 35.*

Acoustic Position Sensing System

Edo Western's acoustic position sensing system is a high resolution, deviation measurement system providing precise data of the position of a barge or vessel relative to an ocean floor reference position. It provides deviation distance directly in ft, resolved to fore/aft and port/starboard components to an overall accuracy of one per cent of water depth.

A single ocean floor sound source and an array of receiving elements on the surface vessel or barge are used. This type of positioning system offers minimum set-up time, maximum simplicity for the computation sub-system and a readout directly in terms of ship co-ordinates. The bottom mounted reference emits a continuous series of pulses timed by a precision internal clock. Precise measurement of arrival time of the pulse at each of the receiving stations on the vessel is converted to deviation information in the digital computer. Position deviation is displayed in numerical form resolved to port/starboard and fore/aft directions and a polar display.

By adding a second hydrophone array with a very short baseline in a collar on the end of the riser, the system can be used for re-entry.—*Undersea Technique, September 1968, p. 53.*

L.P.G. Carrier

The 15 000 m^3 capacity gas carrier *Wiltshire*, recently completed by Hawthorn Leslie for the Bibby Line, is now on long-term charter to George Gibson and Co., Leith.

Principal particulars are:

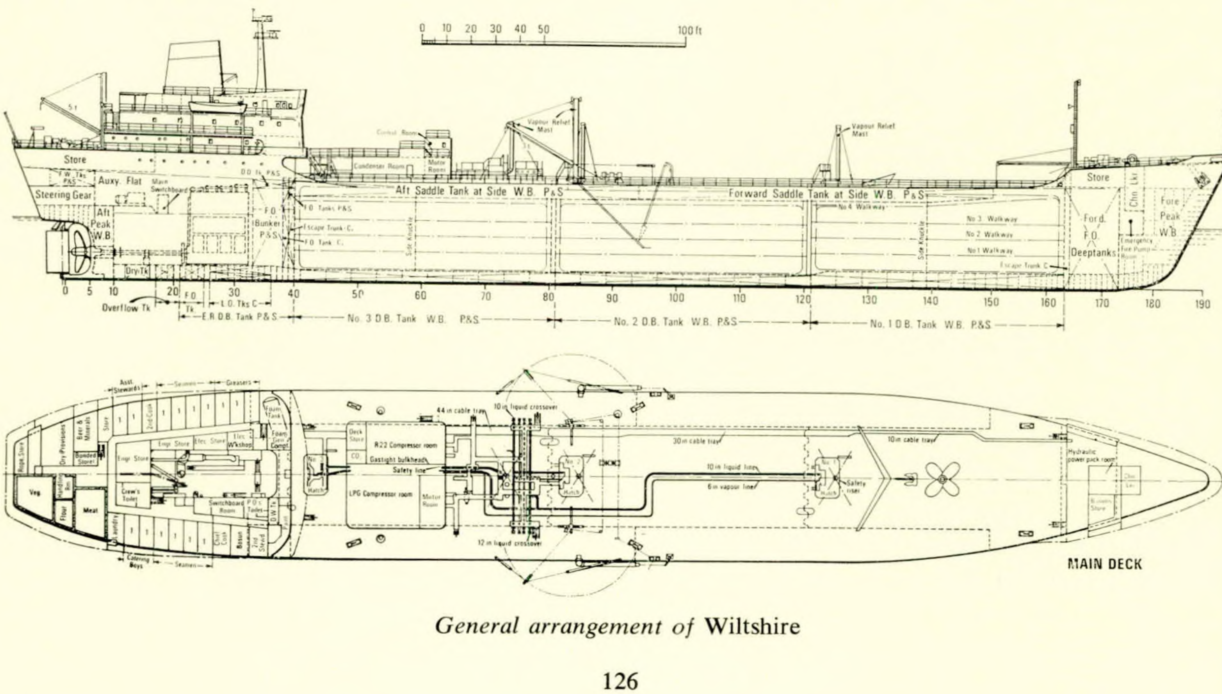
Length, o.a.	497 ft 8 in
Length, b.p.	465 ft 0 in
Breadth, moulded	70 ft 0 in
Depth, moulded to upper deck ...	41 ft 0 in
Draught	27 ft 0 in
Tonnage, gross	10 036
Tonnage, net	5630
Cargo capacity	15 502 m^3
Classification	Lloyd's Register ✠ 100 A1/Ice class 3 liquefied gas carrier
Speed, service	16 knots

The main machinery consists of a Hawthorn-Doxford 76J4, four-cylinder turbocharged oil engine, designed in this instance for a maximum propulsion power of 8600 bhp at 113 rev/min. Electric power is generated by four 345 kW Paxman Diesel alternator sets running at 1200 rev/min.

Steam raising plant comprises an oil-fired unit having an evaporation of 1440 kg/h available at any time with an exhaust gas boiler capable of producing an additional 800 kg/h when the vessel is at sea.

The vessel has three cargo tanks with a total capacity of 15 000 m^3. Two separate piping systems allow the carriage of two cargoes at one time. The ship has been designed to load and discharge her cargoes with a vapour return line if none is available and is fitted with direct, oil-free reliquefaction plant.

Wiltshire has been built to Lloyd's Register Class ✠ 100 A1/Ice Class 3 liquefied gas carrier (propane, butane, anhy-



General arrangement of Wiltshire

drous ammonia and butadiene) in independent tanks, maximum vapour pressure 4 lb/in^2, minimum temperature $-58°F$ ($-50°C$) and to U.S. Coast Guard requirements in respect of the cargo.

Cargo is carried in three tanks independent of the ship's structure and insulated with expanded polyurethane foam. The hull structure surrounding the cargo tanks is constructed in tank quality steel to meet the requirement for a secondary containment of the cargo if the primary tanks should leak. Manual and automatic welding procedures developed by the builders were used in the fabrication of the low temperature structure to meet the U.S. Coast Guard rquirement of 30 ft lb Charpy V impact test at $-68°F$ ($-55°C$) in longitudinal oriented specimens. The cargo tank shell butts and seams were 100 per cent X-rayed.—*Shipbuilding and Shipping Record, 18th October 1968, Vol. 112, pp. 504–507.*

Modern Deck Cranes

The introduction of the thyristor enables a good speed control of a.c. motors to be achieved as well. ASEA use, at present, two different thyristor speed control systems for cranes, one with slip-ring motors and the other with squirrel-cage motors. The latter system has been selected for deck cranes.

A characteristic feature of the thyristor-controlled deck crane is that its control characteristics are comparable with those of Ward-Leonard control, while at the same time maintenance has been reduced to a minimum and the heavy and bulky converter has been eliminated. This system is based on the use of a special squirrel-cage induction motor developed for the purpose, whose driving and braking are controlled by thyristors in a closed-loop control system.

Three thyristors are connected up in the primary circuit of the motor, while three diodes are connected in anti-parallel with them for the return currents. This combination enables the motor voltage, and thus also the torque, to be controlled in the conventional manner by means of phase-angle control. The use of a closed-loop control system with a tacho-generator and a control amplifier, which in its turn is controlled by the difference between the motor voltage and a signal quantity, ensures that the motor speed will be relatively independent of the motor load.

The heat developed in the rotor will be relatively large when the crane is run at low speed and full torque. The motor has therefore been designed so that this heat can be dissipated. In addition, it has a high starting factor, which ensures that the starting current will be low.

It is desirable that the braking should be accomplished electrically so as to obtain smooth retardation and thus reduce the wear on the mechanical brake. The diagram shows how this is achieved by means of a specially wound stator. This has two star-connected parallel windings, which means that the motor has two neutrals. The feeding of a variable direct current between these by means of thyristors will result in a stationary magnetic field with a different number of poles than the main flux of the motor. This stationary flux impresses a braking torque on the motor.—*Rudelius, P., ASEA Jnl, 1968, Vol. 41, No. 6/7, pp. 91–94.*

Liquid Phosphorus Carrier

Albright Pioneer is a single decker with raised forecastle, seven cargo holds and machinery aft. The vessel complies with the requirements of the Board of Trade Class 7, Canadian Department of Transport, the St. Lawrence Seaway Regulations and Lloyd's Register of Shipping. The scantlings of the vessel are based on a draught of 29 ft and the class notation is "To carry liquid phosphorus in independent tanks. Strengthened for heavy cargoes. Specified holds empty." The dimensions are:

Length, o.a.	414 ft 0 in
Length, b.p.	390 ft 0 in
Breadth, moulded	56 ft 0 in
Depth to upper deck, moulded ...	38 ft 6 in
Service speed	14 knots

Four cylindrical tanks in Nos 2, 3, 5 and 6 holds carry the liquid phosphorous and the remaining holds, Nos 1, 4 and 7, are suitable for carrying other cargo such as rock phosphate. Water ballast tanks are arranged in the wings of Nos 2 to 6 spaces.

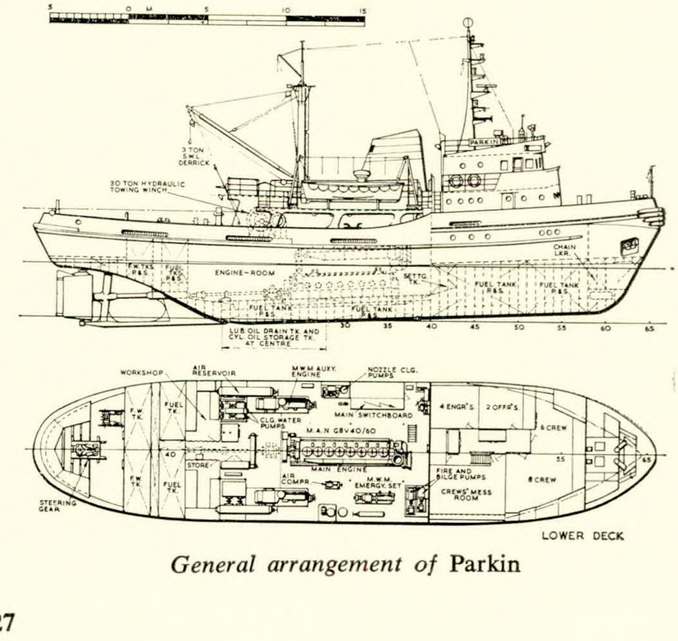
The four cargo tanks are each designed to carry 1210 tons of elemental white or yellow liquid phosphorus at a temperature of 60°C (140°F). The lower cylindrical portion is 36 ft in diameter, on top of which is a 6 ft diameter cylindrical neck with a domed top. The lower part of each tank is surrounded by a water jacket through which hot water, at a maximum temperature of 80°C (176°F), is circulated so as to ensure that the cargo is kept in a liquid state. All the exposed external surfaces of the tanks are clad with Newall's thermal insulation.

When the tanks are loaded with phosphorus, a water seal is maintained in the neck to prevent the phosphorus coming into contact with the atmosphere, and the space above the water seal is filled with nitrogen. When sailing back to Newfoundland in ballast the tanks will be kept filled with water, maintained at 60°C (140°F). The tanks are constructed of Grade B mild steel, the necks being lined with stainless steel to minimize corrosion at the phosphorus/water interface. The water jacket is constructed of mild steel split pipe and carries a pressure greater than the static pressure at the bottom of the tank, to ensure that any leakage is inward and not outwards into the heating circuit. Except for bolted connexions the tanks are independent of the ship's structure and in no way contribute to the strength of the ship.

The propelling machinery consists of two Deutz RBV8M358 eight-cylinder in-line 400 mm bore by 580 mm piston stroke turbocharged and intercooled direct-reversing Diesel engines each developing 2416 bhp at 433 rev/min and geared to a single c.p. propeller.—*Marine Engineer and Naval Architect, November 1968, Vol. 91, pp. 445–447.*

Long Range Motor Tug

The latest addition to the already extensive fleet of more than 100 tugs owned by the Manila-based Luzon Stevedoring Corp.—the long-range motor tug *Parkin*—was recently delivered from D. W. Kremer Sohn, of Elmshorn, West Ger-



General arrangement of Parkin

many. Built to the ✠ A1 (E) Towing Service AND and AMS classification of the American Bureau of Shipping, the principal particulars are:

Length, o.a.	39·2 m
Length, b.p.	35·0 m
Breadth, moulded	8·9 m
Depth	4·7 m
Freeboard draught	4·29 m
Gross register	357·3 tons
Maximum speed	13·4 knots
Cruising range	9600 miles
Bollard pull	28 tons

The main engine of *Parkin* is an eight-cylinder M.A.N. unit of the G8V 40/60 m.A.L. turbocharged type, capable of developing 2260 bhp at 300 rev/min. It is provided with a separate thrust block and is coupled directly to a three-bladed, nickel-bronze propeller of Schaffran manufacture, working in a Kort steering nozzle. The turbocharger is fitted with its axis parallel to that of the engine and is arranged for forced lubrication of the rotor bearings from the main engine system through a bleed valve and separate filter.

The main engine is arranged for bridge control and this is accomplished through a pneumatic system of Westinghouse design, coupled to the engine telegraph in the wheelhouse. The master control for this system is fitted on the pneumatic manoeuvring block situated behind the normal engine hand controls. The position of the changeover handle is indicated by lights on the mode and running light panel in the wheelhouse. Although the engine full power turning speed is 300 rev/min the bridge telegraph can be placed in an astern position while the engine is still turning at 60 rev/min and starting air will be admitted to the cylinders. Remote reading of the engine exhaust temperatures is allowed for by a selective, multi-point indicator arranged on a pillar adjacent to the engine manoeuvring station. An alarm panel for the main and auxiliary engines' oil pressures and cooling water temperatures is also situated in the engine room on the forward bulkhead.—*Motor Ship, September 1968, Vol. 49, pp. 305–306.*

Nuclear Ship Refuelling

The world's first nuclear-powered merchant ship *Savannah,* recently returned to Galveston, Texas, for her first refuelling after travelling 330 000 miles during which about 120 lb of nuclear material were used. This compares with 96 000 tons of fuel that would have been burned by a conventionally-powered vessel of about the same size.

During the past three years the 21 000 ton *Savannah* has been in commercial cargo service and latterly has been engaged in a U.S.–Eastern Mediterranean run. She carries 9400 tons at a cruising speed of 20–21 knots and has called at 76 different ports in the U.S., Africa, Asia and Europe. Even after this length of service her nuclear fuel is not fully exhausted and only the centre four of the 32 fuel elements in the nuclear core will be replaced. The others will be rearranged in an operation designated a "fuel shuffle" by the vessel's owners, the Maritime Administration, U.S. Department of Commerce and Todd Shipyard's Nuclear Division, which maintains a servicing facility for the vessel at Galveston. It is believed that this rearrangement of the fuel elements will permit a further two years of cruising before a complete refuelling is necessary.

In this rearrangement, the reactor head with the assembled control rod drives is removed from the ship and placed in a building near Pier E at Todd's Galveston shipyard. Early in the operation a pair of standardized neutron sources manufactured by Monsanto Research Corp., are inserted in new fuel elements located centrally in the reactor vessel. Similar neutron sources were also used during the vessel's original fuelling operation late in 1961. These sources will provide enough continuous neutron radiation to keep all instrument and control systems operating safely and dependably during the rearrangement. Remaining elements are then rearranged.—*Shipbuilding International, November 1968, Vol. 11, p. 28.*

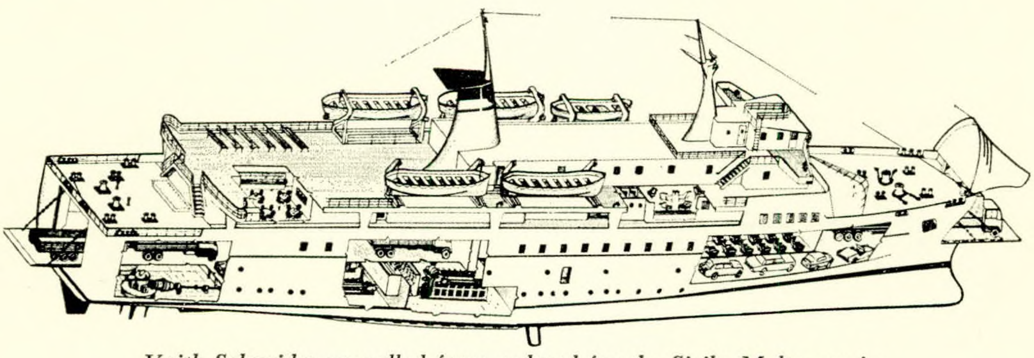
Voith Schneider Propelled Ship in Italian Ferry Building Programme

A major new building programme has been instituted by Tirrenia S.p.A., Naples, one of the leading Italian ferry operators. Orders have recently been placed for no fewer than seven roll-on/roll-off ferries: six in a series of 6500 gross ton ships for both new and established services in the Mediterranean and one 2100 gross ton 18-knot vessel for service between Sicily and Malta. The latter vessel is of particular interest on account of its propulsion arrangements, comprising two Voith Schneider stern propellers. These are to be five-bladed units driven by extended shafting from a pair of Ansaldo Diesel engines of the B3212 type, each rated at 3150 bhp at 514 rev/min.

Fin-type stabilizers will be fitted. The order for this vessel has been placed at the Cantieri Navali Pellegrino, in Naples, but the bulk of the new Tirrenia orders has been placed with Italcantieri yards; four of the 6500 gross ton vessels will be built by the Castellammare, Naples, yard and two by the Cantieri Navali del Tirreno e Riuniti, Palermo.

These larger ships have been designed by Italcantieri in conjunction with the owners' technical staff. Each vessel will carry 1000 passengers, in two classes of accommodation. Vehicles will be carried on two levels; the main deck will be suitable for 24 heavy commercial vehicles while up to 100 cars may be stowed on an upper garage deck. Vehicles will be loaded and discharged through two large side ports and a stern door but there will be no bow door in this case. A bow thruster and fin-type stabilizers will be fitted.

Propulsion will be by two direct-coupled Fiat turbocharged two-stroke engines of the B609S design, each rated at 8280 bhp at 220 rev/min and driving c.p. propellers.—*Motor Ship, September 1968, Vol. 49, p. 298.*



Voith Schneider propelled ferry ordered for the Sicily–Malta service

Semiconductor Control Circuits for Automated Auxiliary Drives

The use of contactors in automated systems involving the control of electric motors makes the equipment cumbersome, the large number of contacts reduces reliability and necessitates careful maintenance. There is consequently a tendency to adopt contactless devices; this is quite feasible on the basis of modern developments in semiconductor techniques and results in more compact equipment giving better protection and control. Several such semiconductor systems are described in some detail; circuit diagrams are given in each case.

SOLAS requirements state that steering gear electric motors should have short-circuit but not overload protection. It is therefore necessary to provide visual and acoustic overload alarms. A semiconductor scheme, as used for the three-phase motors of electro-hydraulic steering gears in some harbour icebreakers and in vessels of the *Amguema* class (436 ft o.a.), is described. Each indicating lamp shines steadily to show that the motor concerned is on line and is switched over to a flashing circuit when a thermal alloy gives an overload signal. A variable resistance allows the brightness of the lamps to be adapted to the illumination conditions in the wheelhouse.

Ships now have numerous automated pump and compressor drives controlled by impulses from pressure relays, level relays, etc. As a rule they operate for long periods unattended. It is generally considered that, for motor overload protection in such cases, a thermal relay on the starters is adequate and that in practice overloading cannot occur because the motors usually have a certain power reserve. A conventional control circuit for such a drive, with the addition of overload warning devices, is shown. Such a scheme will only work reliably if the thermal relays are not self-resetting. Three modified circuits which avoid this drawback are then described. The variant which should be used in a given case depends on the application of the motor, on whether or not a watch is kept nearby and on other operating conditions.—*Avik, Yu. N. and Serzhantov, V. V., 1967, Sudostroenie, No. 5, pp. 29–32; Jnl Abstracts B.S.R.A., July 1968, Vol. 23, Abstract No. 26 571.*

Investigation of New Stirling Engine

United Stirling (Sweden) AB states that in the course of the next few years at least £3 000 000 will be spent on the development of the Stirling engine for marine and automotive applications.

United Stirling, a consortium of the National Swedish Defence Factories, Kockums and Husqvarna, starts its development programme on the basis of know-how and patents acquired under the licensing agreement recently signed with Philips Gloeilampenfabrieken in Holland.

The engine is of the hot gas type in which helium is the working medium. The gas, in a closed system, is alternately heated and cooled on the opposed sides of a working piston. External combustion permits perfect control of the combustion process and it is claimed that the exhaust gases contain only about one per cent of the unburnt carbon products found in ordinary Diesel exhaust. Furthermore, the exhaust is continuous, not pulsating, hence the low noise level.

Current disadvantages of the Stirling engine are weight, which is greater than in other types, and cost, both of which factors will receive attention by the Swedish consortium.—*Shipping World and Shipbuilder, November 1968, Vol. 161, p. 1744.*

Italian-built Refrigerator Vessels for Soviet Ownership

The 4400 dwt *Vasily Chapayev* is the first of five refrigerated cargo ships to be delivered from the Cantiere Navale Breda, Venice, to the Soviet Union.

All the vessels in the series are propelled by Fiat-type machinery and Fiat Diesel engines are also employed to power the generating sets.

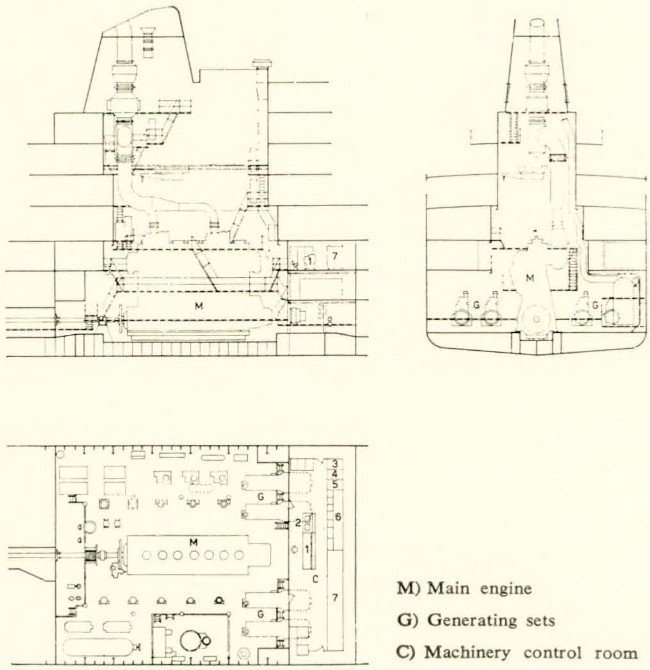
Principal particulars are:

Length, o.a.	121·85 m
Length, b.p.	110·00 m
Breadth	17·00 m
Depth	11·30 m
Depth to main deck	9·05 m
Depth to tonnage deck	6·56 m
Full load draught	7·50 m
Correspoding deadweight ...	4400 tons
Gross register	5000 tons
Draught at tonnage mark (banana load)	6·30 m
Corresponding deadweight ...	2600 tons
Hold capacity	5770 m³
Trial speed at 6·30 m draught ...	19·3 knots

Each ship has a total of eight refrigerated spaces and these are cooled by air chilled in brine batteries employing Freon 22 refrigeration in the primary circuit. The air in the holds is recirculated at rates of up to 80 cyl/h: separate ventilation arrangements are provided to give up to three fresh air changes per hour. Insulation of the hold spaces is by mineral wool covered with aluminium alloy sheathing.

The Fiat two-stroke engine of the B680S-type is employed for propulsion in each ship of the series. The seven-cylinder version selected develops 8400 bhp at 150 rev/min.

Four Fiat engines of the four-stroke LA236ES type power the electrical generators, each producing 375 kVA. This type of Fiat engine is of 230 mm-cylinder bore and 350 mm-stroke and is built with six or eight cylinders.—*Motor Ship, October 1968, Vol. 49, p. 342.*



M) Main engine
G) Generating sets
C) Machinery control room

1) Manoeuvring console
2) Printer
3) Main engine data logger
4) Main and auxiliary machinery automation
5) Auxiliary machinery data logger
6) Synoptic panel
7) Main switchboard

Machinery arrangement drawing of the Fiat-engined, automated ship Kotovsky

Fiat Marine Automated Plant for Russian Cargo Ship "Kotovsky"

The main engine is automatically controlled both from the engine control room as well as from the bridge.

This manoeuvre is performed by the engine telegraph of the conventional type, with four positions for ahead and reverse running.

On an auxiliary panel (close to the telegraph), two selectors have been added for the following operations:

a) Manoeuvring mode selector with a three position switch for:
   1) Manoeuvre. In this position the engine is adjusted by the engine telegraph for fixed and predetermined speeds corresponding to each position of the engine telegraph.
   2) Navigation. In this position the engine is adjusted with constant delivery of the fuel pump, corresponding to each position of the engine telegraph.
   3) Emergency. The engine can be manoeuvred also in case of main failure when the automation would stop it (i.e. lubeoil failure).

b) Fine adjustment selector
   The engine speed and the position of the fuel throttle corresponding to the above "manoeuvre" and "navigation" positions, can be continuously varied within a wide margin by means of an auxiliary knob in order to adapt them to the actual conditions of the moment (e.g. reducing speed when the sea is rough).

A data logger constantly keeps under control 122 points of measurement.—*Fiat G.M. Technical Bulletin, 1968, Vol. 21, No. 1, pp. 27–32.*

Air Conditioning System

It is commonly known that because of different densities, hot air rises and cold air falls. Many air conditioning systems on board ship have their distribution system located at deckhead level or else direct the airflow upwards and, whilst this is a satisfactory arrangement in summer, it has inherent disadvantages in winter when the air is heated. This warmer air remains at deckhead level and the cold air collects at deck level thereby causing discomfort to cabin occupants.

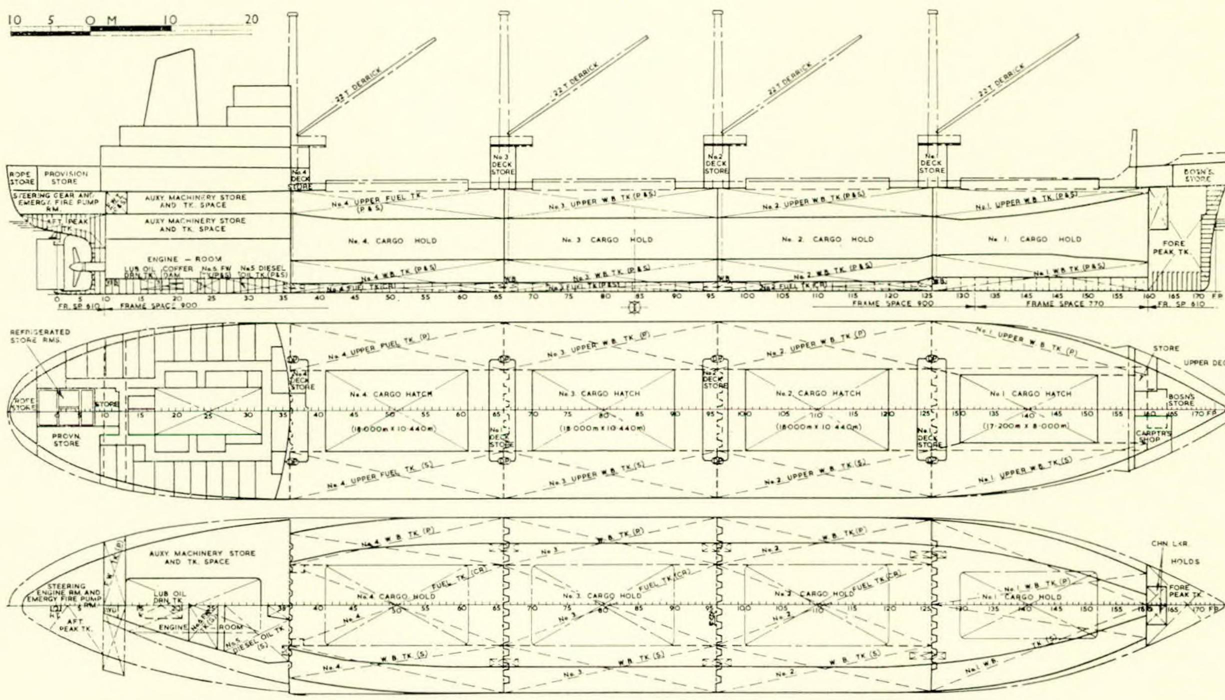
To overcome this problem, Svenska Flatfabriken has recently introduced two new bulkhead-mounted cabin distribution units suitable for use with the SF Regovent or Duovent ship air conditioning systems. Designed to allow the air supply to be directed upwards or downwards, the units thus give optimum air and heat distribution during summer or winter conditions. In summer when outside temperatures are high the cooled air is blown upwards against the bulkhead through the grille incorporated in the top of the unit and flows along the deckhead before subsiding to the deck. During cold weather the inlet air is heated and, by opening a flap at the bottom of the unit, most of the air is directed downwards to disperse across the deck before rising. A certain proportion of the warm air escapes through the grille to rise immediately to the deckhead thus ensuring optimum temperature conditions throughout the cabin.—*Shipbuilding International January 1969, Vol. 11, p. 38.*

Japanese 18 000-dwt Standard Bulk Carrier

Maritime Queen, recently delivered to International Maritime Carriers Ltd., Hong Kong, is the fourth of the type to be built by Hitachi Zosen, Japan.

Principal particulars are:

Length, b.p.	146·00 m
Breadth	22·60 m
Depth	12·90 m
Draught, loaded	9·29 m
Corresponding deadweight ...	18 302
Corresponding displacement ...	23 465
Light ship	5163
Light draught	2·3 m
Gross register	11 434 tons
Net register	6844 tons
Complement	55
Capacities	
Cargo (grain)	842 393 ft^3
centre of gravity (forward amidships) ...	24·37 ft



General arrangement drawings of the Hitachi *18 000-dwt standard bulk carrier*

centre of gravity (above base)	23·68 ft
W.B.	4960·68 tons
F.W.	294·19 tons
Fuel	1539·82 tons
Diesel oil	117·79 tons
Lubricating oil	15·71 tons

The vessel has four holds each 88 ft 7 in in length. Upper and lower wing tanks are fitted in each hold, all lower tanks being for water ballast, while fuel is carried in No. 4 upper wing tank and in the double bottom.

The three aftermost holds each have a single hatch measuring 18 m × 10·44 m and the hatch to the forward hold is 17·2 m × 8 m. The hatches are served by single 22-ton derricks, stepped from the top of small deckhouses located between goal-post type derrick posts placed aft of each hatch.

Main propulsion is by a Hitachi-B. and W. 762-VT2BF-140 type engine developing 8400 bhp.

The following sea trial data were recorded:

Weather		fine	
Wind		4–5	
Sea		smooth	
Draught, forward		5 ft 7½ in	
Draught, aft		19 ft 11 in	
Displacement		8990 long tons	
Output	rev/min	bhp	speed (knots)
½	114·5	4170	14·764
Continuous service	140·8	7946	17·336
Maximum continuous	144·2	8579	17·748

—*Motor Ship, October 1968, Vol. 49, p. 347.*

Integrated Computer Ship Design System

The shipbuilding division of Nippon Kokan has developed a sophisticated initial ship design system based on integrated, rather than the usual piecemeal, use of computers.

The system computes the optimum design from computer calculations of various factors including draught, trim and hull and longitudinal strength.

Advantages offered by the system include rapid preparation of initial designs, speedy formulation of required design changes to original owner specifications, improved accuracy of preliminary designing and facilitating the subsequent design work.

Owner's specifications are applied to a type vessel on receipt. The specifications of the type vessel are previously fed into the computer, which then calculates various data for the new design including longitudinal strength factors such as bending moment and shearing force, trim and draught, vessel stability, displacement and deadweight.

Following the above procedure, several initial designs are prepared, each including slight alterations of configuration. The computer system enables N.K.K. to present multiple initial designs to owners with detailed explanations, backed by computer-prepared statistics, and recommendation of the optimum design.

After the initial design is selected, further detailed designs are completed by each section of the basic ship design department.

Conventionally it requires about one month to complete the required calculations and prepare a single initial design. The new system reduces the time needed to only one or two days.—*Shipbuilding and Shipping Record, 22nd November 1968, Vol. 112, p. 676.*

Winch/Windlass with Fluid Drive

A new type of winch/anchor windlass equipped with fluid drive has recently been introduced by Friedrich Kocks G.m.b.H. At present these units are being manufactured with maximum pulls of from 10 to 40 tons and with slack rope speeds from 200 to 260 ft/min. An explosion proof version is available for use on tankers. These winch/windlasses meet the requirements for units which are as rugged and flexible as the well-proven steam-driven deck machinery, yet simply require an electric feeding cable instead of steam piping.

Basically the fluid drive system consists of a constant-speed squirrel cage motor, a torque converter giving an infinitely variable fluid drive, and a reversing gear wheel unit. The torque converter is of standard design and provides a stepless control of the speed or the pull for standstill operation. Should there be a power failure, the winch/windlass is automatically stopped, thus preventing rope damage.

When compared with winch/windlasses with steam, conventional electric or hydraulic drive, the winch/windlass with fluid drive has, for the same capacity, a higher maximum pull and, with the exception of the steam-driven type, a higher slack rope speed. Other features of the fluid-drive units are that they are easy to operate, require little maintenance, all components have a long working life, the motor starts off load and operates continuously and no piping or external switchgear is required.—*Shipbuilding International, December 1968, Vol. 11, p. 35.*

Damage Stability Investigations

Damage stability calculations are assuming increasing importance and, in addition to the requirements for passenger vessels, it is becoming more and more a routine job for any kind of ship. The availability of computer programmes covering these calculations has increased the amount and complexity of the investigations which can be carried out and at the same time has reduced the burden on design offices. The present article, besides a short summary of the hand calculations today in practice and of the computer calculations which are available, gives a description of several damage stability investigations. Suggestions for areas of future development in this field are given.

One of the main limitations of hand calculations of damage stability data lies in the fact that independent calculations are performed for each cargo condition and corresponding GMo value. Usually, out of all cargo conditions previously calculated, the one having the least metacentric height is selected and investigated. This means that, at the first stage of the investigation, a source of error is introduced owing to the fact that it is not always the condition having the minimum GMo value which is the most dangerous so far as damage stability is concerned.

Computer programmes available for carrying out stability calculations provide for calculation of the damage stability with symmetrical flooding, asymmetrical flooding, damage stability during the period of flooding and cross-flooding examinations. These programmes, which use the added water method, can be broadly subdivided into two categories. The former gives the minimum initial metacentric height GMo required before flooding in order to fulfil the requirements of the authorities after flooding has taken place, or stricter ones (e.g. maximum heeling angle, modified margin lines, limiting bow and/or stern draughts, etc.) imposed by the circumstances. In addition, data concerning draught, trim, co-ordinates of the centre of buoyancy of the damaged hull, moment of inertia of waterline plane and particulars of shipped water are supplied. The latter provides details of the floating position and hydrostatic data of the vessel after damage, assuming it had a certain GMo value beforehand.—*Dicovi, G. R., Shipping World and Shipbuilder, October 1968, Vol. 161, pp. 1627–1630.*

Conversion of Free-piston-engined Ship to Diesel Propulsion

The conversion of the machinery in m.s. *Galini* (formerly *Rembrandt)* from a five-gasifier free-piston installation to an S.E.M.T.-Pielstick engine, marks the final stages in the extinction of free-piston gas turbine machinery for marine purposes. Around 1958 and the early 1960s high hopes were held for such machinery; it was claimed to be simple, to offer flexibility in engine room planning and the advantage of independent power units each of about 1000 hp, which could be operated to supply the gas turbine or put on standby or overhauled as required.

But however attractive the theoretical advantages appeared—and several free-piston-engined ships were put into service—operating experience emphasized many shortcomings, principally unreliability.

Rembrandt, when she was delivered in 1960, was described as the highest-powered, newly built free-piston-engined cargo vessel and the first British-built ship with such machinery to have a c.p. propeller—of Stone-Kamewa construction. The obvious advantage of the c.p. propeller with a gas turbine-propelled ship was the elimination of astern blading in the turbine with a corresponding reduction in windage losses.

The original free-piston plant had apparently shown itself capable of acceptable performance with good overall reliability when well serviced, but was dependent on the continued availability of trained staff and spares to achieve this. The ship itself was, however, built as an open shelterdecker with rather low power and when it was thought to exploit the advantages now obtainable by "closing" the vessel, it became immediately obvious that increased power would be required to make this a practical proposition.

Accordingly, a feasibility study was carried out by the owner's consultant which revealed that a single 6000 bhp geared Pielstick engine could be installed in the space available and adapted to drive the existing Kamewa propeller with modified blades.

The 14-cylinder Pielstick engine, type PC2V, is connected to the engine propeller shafting via a Spiroflex elastic coupling and Lohmann and Stolterfoht single-reduction gearing. It was determined that the orginal propeller mechanism would absorb 6000 bhp at 120 rev/min when fitted with new blades, designed to suit this power and the new speed predicted.—*Motor Ship, November 1968, Vol. 49, pp. 369–372.*

Marine Engine Bedplate Repair

An unusual repair to the starboard main engine of the twin-screw, 11 400 dwt *Empire Star,* owned by the Blue Star Line, was recently carried out by the Belfast shiprepair department of Harland and Wolff Ltd. The vessel, powered by two eight-cylinder Harland and Wolff-B. and W., 550 mm-bore, double-acting engines, suffered a broken piston rod during service, and although this was replaced, subsequent operation of the engine produced a noticeable "shuttling" of the exhaust piston gear.

On examination it was found that the aft web on the No. 6 unit crank had been forced out of alignment and that a number of cracks existed in the adjacent No. 8 main bearing bedplate cross-girder. Although the bedplate had a long history of cracks, it was felt that the operation of the engine with a misaligned crankshaft had aggravated the cracking, and Harland and Wolff was, therefore, asked to repair all cracks on this cross-girder in addition to the realignment of the crank web.

This latter operation was carried out *in situ* in a fairly straightforward operation by shrinking the crankpin with the aid of liquid nitrogen and expanding the web by heating with oxy-propane torches: hydraulic jacks were employed for re-alignment purposes. The fact that this particular crankshaft carried no chisel location marks was a complication which was overcome by lining up the lubricating oil holes in the crankpin and web to ensure the correct relocation of the parts.

The cracks, however, presented a different problem. Although most of them could have been repaired by welding *in situ,* one of them, in the housing below the journal, could only be welded by completely dismantling the whole engine and removing the crankshaft. To avoid this costly and time-consuming operation it was suggested to the owners that a repair could be effected by cutting away that part of the cross-girder which carried the bearing housing—and the crack—and replacing it with a specially manufactured insert.

This was agreed, and the section of the cross-girder containing the main bearing pocket was, therefore, cut away and a new insert manufactured and welded into position. To prevent any movement of this insert during welding operations a dummy bottom-half main bearing was designed so that the crankshaft journal would act as a mandrel. The pocket insert was held in position by fitting a beam across the main bearing studs and held from underneath with fox-wedges. After welding, the whole assembly was stress-relieved.—*Motor Ship, December 1968, Vol. 49, p. 438.*

South African-built Fisheries Research Ships

The second of two advanced fisheries research vessels, *Benguela,* was recently handed over to the South West Africa Administration. Built by the Barens Shipbuilding and Engineering Corp. Ltd., Durban, she is a 145-ft $13\frac{1}{2}$-knot vessel, equipped to undertake such functions as gate-trawling on the white fish grounds and experimental tuna catching; the ship will also sample environmental factors by meteorological, hydrological and biological observations, in addition to undertaking electronic research.

Equipped with a roll-stabilizing tank and a lateral-thrust unit, *Benguela* has a round-nosed stem, a continuous main deck and a long forecastle deck, the latter containing laboratories, accommodation, catering and toilet spaces. Accommodation is provided for 23 persons, including a team of five scientists.

Benguela has an o.a. length of 145 ft (123 ft b.p.), a moulded breadth of 31 ft, a draught of 12 ft and a range of 6000 miles. The gross tonnage is 486·17. Her two main engines are Burmeister and Wain-Alpha Diesel units, which together develop 1300 hp at 400 rev/min; they are coupled through a twin-input, single-output Brevo co-axial reduction gear-box to a single shaft turning a KaMeWa c.p. propeller, working in a Kort steering nozzle. All machinery is vibro-insulated.

In addition to the main controls in the engine room, there are two control consoles in the wheelhouse—one facing forward and the other aft—each of which is also a steering position.—*Motor Ship, December 1968, Vol. 49, p. 456.*

Use of Free Scavenge Air in the Development of Two-stroke Loop Scavenged Engines for Turbocharging

A brief historical description of the special characteristics of Crossley two-stroke c.1. engines is employed to justify the use of an independently driven and controlled source of compressed scavenge air in an extensive series of engine tests, from the analysis of which novel techniques were evolved for assessing:

1) scavenge air pressure drop and flow;

2) ihp from brake test results;

3) bhp using a special turbocharger performance chart.

—*Wood, J. H., I.Mech.E., 9th January 1969, Paper No. P13/69.*

B.S.R.A. Trawler Series

The paper describes experiments carried out in still water to assess the effect of varying the shape of the forebody and afterbody sections of the parent form used in the breadth/draught ratio and length/displacement ratio series. The experiments were carried out for the British Ship Research Association by Vickers Ltd., Shipbuilding Group. All the models were tested for resistance and propulsive efficiency in the trimmed design load condition.—*Thomson, G. R. and Pattullo, R. N., 1968, R.I.N.A., Paper No. W9.*

Effect of Peening and Grinding on the Fatigue Strength of Fillet Welded Joints

This report contains details of fatigue tests carried out on specimens containing longitudinal and transverse non-load-carrying fillet welded joints which had been either ground or hammer peened. Both types of treatment resulted in considerable increases in fatigue strength. The increase due to grinding was approximately independent of life over the endurance range considered, but that from peening was dependant on life.—*British Welding Jnl, December 1968, Vol. 15, pp. 601–609.*

Automated System for Optimum Ship Routing

The salient features of an automated system recently developed for routing ships in the North Atlantic Ocean are presented. The goal of the routing procedure is for the ship to reach its destination in minimum time. A mathematical model for the optimum route, based on Pontryagin's maximum principle, is presented. Methods are described for specifying and forecasting directional wave spectra at a large number of points in the ocean, once certain meteorological data are available.—*Marks, W., Goldman, T. R., Pierson, W. J., Tick, L. J. and Vassilopoulos, L. A., 13th–16th November 1968, S.N.A.M.E. Paper No. 1.*

Diesel Engine in Association with Gas Turbine

The future development of the Diesel engine will inevitably depend on its association with a gas turbine. This paper explores three known methods by which such an association may be achieved, viz., turbocharging, compounding and the use of a gasifier. The performance obtainable from these three arrangements for both two-stroke and four-stroke cycle engines, with and without charge cooler, considering various compression ratios, adiabatic efficiencies and trapped air-fuel ratios are derived.—*Loiwal, A. S., Dhingra, D. M. and Patel, R. C., Jnl Institution of Engineers, India, January 1967, Vol. 47, pp. 84–99; Applied Mechanics Reviews, August 1968, Vol. 21, p. 859.*

High-speed, High Frequency Electrical Machinery for Marine Applications

High speed pumps and compressors for marine applications offer significant reductions in size and weight. Major innovations in the design of high speed motors and frequency conversion devices prompt a review of the current status of these devices and their potential application to marine auxiliaries. This paper describes the current status of high-frequency motors and high frequency conversion devices in terms of what is available, what is under development and how the performance of these devices compares with the performance of more conventional electric machinery.—*Shapiro, H., 13th–16th November 1968, S.N.A.M.E. Paper No. 3.*

Marine Boiler Design Today

As marine boilers grow larger for increased powers, and as steam conditions are raised for increased efficiency, there are more facets of design that must be carefully considered. These include the furnace rating and arrangement, the superheater design, circulation, steam quality, and water level variation during manoeuvres. Of serious concern in the superheater is the probability of slagging and high temperature corrosion as metal temperatures rise.—*Signell, W. I., 13th–16th November 1968, S.N.A.M.E. Paper No. 2.*

Resistance of Large Powered Catamarans

This paper presents hull form and model resistance data for two large powered catamaran designs. One is for a 700-ft special cargo ship, which remained only as a study. The other is for a 230-ft submarine rescue ship, which is now under construction. In each case, both symmetrical and asymmetrical hulls were tested, and the spacing between the demihulls was varied. Principal emphasis is on the wave-making interference effects.—*Turner, H. and Taplin, A., 13th–16th November 1968, S.N.A.M.E. Paper No. 6.*

Some Aspects of Hydrodynamic Design of High-speed Merchant Ships

Modern cargo-handling techniques have encouraged the design of transom-stern ships. The authors have examined the resistance characteristics of several unrelated transom-stern ships and have designed and tested four related models which are representative of modern designs. Effects of longitudinal position of centre of buoyancy, maximum transverse section coefficient, operation in off-design conditions and bulbous bows are discussed.—*Michelsen, F. C., Moss, J. L. and Young, B. J., 13th–16th November 1968, S.N.A.M.E. Paper No. 7.*

Hull-deckhouse Interaction by Finite-element Calculations

This paper presents an extension of earlier work by the first author in which an analysis of ship structures based on the finite element technique was described. A procedure has been developed whereby the investigator may examine closely, by a succession of refinements, small regions of interest in a large structure. In the first step, the forces exerted by the remainder of the structure on a small segment in the region of interest are computed. In the next step, the segment alone is examined under the loading just computed.—*Pauling, J. R. and Payer, H. G., 13th–16th November 1968, S.N.A.M.E. Paper No. 9.*

Influence of Fabrication on Reliability of High Temperature Superheater Tubes

The authors describe two investigations organized to provide data on the influence of fabrication factors on the reliability of high temperature superheaters for steam power generation. Particular consideration is given to heat treatment, forming operations and superheater design in influencing service performance. Discussions include:

a) results of a series of stress-rupture tests on pre-strained and heat treated specimens and superheater alloys;
b) the performance of specific types of support fixtures during studies and laboratory experiments.

—*Burghard, H. C., Lautzenheiser, E. and Wyllie, R. D., 22nd–26th September 1968, A.S.M.E. Petroleum Mechanical Engineering and 1st PVP Conference. Paper No. 68-PVP-25.*

Aluminium/Steel Joint

A new type of aluminium/steel joint has recently been developed to eliminate corrosion problems when these metals are joined and exposed in a salt water environment. The new system, which will be especially useful in securing aluminium superstructure elements to steel decks, consists of fabricated strips of steel and aluminium joined by the Du Pont Co's explosion bonding process.

The steel face of the composite strip is welded to the deck or other steel structional member and aluminium components are welded to the aluminium element of the bimetallic strip. Conventional welding methods are employed. With this system, the two dissimilar metals come together only at the metallurgical bond created by the explosion process. The corrosion problem that usually results when the two metals are in contact does not occur.—*Shipbuilding International, August 1968, Vol. 11, pp. 56–57.*

Marine Diesel Engine Exhaust Noise: Exhaust Sound Criteria for Bridge Wings

Data are given on the acceptability of exhause noise on bridge wings due to the gas pulses of large low speed marine Diesel engines, directly coupled to a constant pitch propeller. I.S.O. noise rating numbers are used. For full speed ahead N75 is the upper limit of acceptability. The background noise due to auxiliaries is not treated in detail but should be substantially less.— *Janssen, J. H. and Buiten, J., International Shipbuilding Progress, October 1968, Vol. 15, pp. 348–352.*

Marine Diesel Engine Exhaust Noise: Scale Models of Exhaust Systems

Simplified scale models are shown to be useful for determining the low frequency sound transfer properties of an exhaust system of a large low speed marine Diesel engine. Experimental results of a full scale system including silencers closely agree with model results. Ways of constructing, simplifying and investigating the models are indicated.—*Buiten, J. and Janssen, J. H., International Shipbuilding Progress, October 1968, Vol. 15, pp. 353–370.*

Some Features in Underwater Welding, Cutting and Tools

Several new techniques for underwater cutting have been proposed but these are still in the development stage as far as marine use is concerned. Electron beams have been used to cut granite and metals underwater and plasma arc cutting has been used for underwater repair of components of nuclear reactors. Electrical shock hazards will be a major handicap to be overcome in using these processes. Development has also been undertaken to perfect a cutting torch operating on liquid fuel and oxidizer.—*Mischler, H. W., Jnl Ocean Technology, July 1968, Vol. 2, pp. 46–48.*

Introduction to Finite Element Methods of Structural Analysis

In order to make the underlying theory and broad applicability of the finite element method more widely understood, a unified introductory description of the three basic phases of the method—structural idealization, determination of element stiffness and solution of the idealized structure—is presented. In addition to a detailed discussion of the fundamentals, an attempt is made to make the reader cognizant of the critical features of the method.—*Tolefson, D. C. and Brand, L., Marine Technology, October 1968, Vol. 5, pp. 331–346.*

Some Remarks on Model Tests with Floating Platforms in Waves

This paper reviews briefly the results and problems associated with model tests in waves on moored, floating drilling platforms. It considers the analysis of the observed phenomena which is based on experience gained from several tests on these structures carried out at the Netherlands Ship Model Basin. Most parameters which affect the motions of the platform are interrelated and not independent of one another.—*Wahab, R. and van Sluijs, M. F., Marine Technology, October 1968, Vol. 5, pp. 382–391.*

Incipient Failure Detection in Bearings

The detection of incipient failure, in general, is related to the detection of the basic causes of failure. The detection of the basic causes at an early stage will allow action to be taken which can, in some cases, prevent the failure from occurring and, in others, allow scheduled replacement of the affected part on a convenience basis. In this paper, the problem of detecting incipient failure in bearings is detailed.—*Balderston, H. L., 17th October 1968, 28th National Conference American Society Nondestructive Testing, Materials Evaluation, September 1968, Vol. 26, p. 39a.*

Predictions on Low-cycle Fatigue Life of Specimens with Fabrication Flaws

A modification of the notch stress procedure for fatigue life analysis is presented by the author. The importance of considering the mechanics of the specimen and the effects of the notch on specimen mechanics is illustrated by example. The procedure is applied to correlate the results of small specimen tests with large weld defect specimen tests. The significance of crack-initiation life, and crack-propagation life and the dependance of these portions of total fatigue life on specimen geometry and loading is developed.—*Pickett, A. G., 22nd–26th September 1968, A.S.M.E. Petroleum Mech.E. and First PVP Conference. Paper No. 68-PVP-15.*

Effect of Repeated Loads on the Low Temperature Fracture Behaviour of Notched and Welded Plates

The influence of repeated loadings on the susceptibility of weldments to fracture in a brittle manner is studied for an A.B.S.-Class C steel. The repeated loads or loading history are found to affect the fracture behaviour of the weldments. In all but one instance, the fracture stresses obtained for the notched-and-welded wide plates were greater than the stresses to which the members had been subjected during the repeated loadings.—*Munse, W. H., Cannon, J. P. and Kiefner, J. F., October 1968, Ship Structure Committee, Rep. SSC-188.*

Calculation of Leakage Between Metallic Sealing Surfaces

A procedure for calculating leakage flow through the seal interface of a fluid connector, the initial step in ultimately developing a sufficiently accurate and economic calculation procedure that will qualify as a design tool, has been developed and evaluated. The authors measured surface roughness, calculated surface statistics, generated and pressed the surface together mathematically, generated the interfacial gap map and flow map and calculated flow parameters and the flow.—*Wallach, J., Moore, H. B., Tahbun, F. V., Glizendanner, L. G. and Hawley, J. K., 17th–20th June 1968, A.S.M.E. Symposium on Gas Lubrication and Exhibition. Paper No. 68-Lub-15.*

Methods and Significance of Base Determinations in Marine Cylinder Lubricating Oils

A study was made of important factors in analysing the base content, i.e. acid neutralization capacity, of additive concentrates used to compound highly basic marine cylinder lubricants and of the lubricants themselves. The standard analytical method now in general use, ASTM D 644-58 is not entirely satisfactory. Variations of this method were also found to have disadvantages. The method found to be most satisfactory in this study and which is preferred, is based on perchloric acid titration.—*Abbot, A. D. and Farley, L. L., Lubrication Engineering, September 1968, Vol. 24, pp. 422–429.*

Analysis of Three-fluid, Crossflow Heat Exchangers

In this study, the performance of three-fluid, crossflow heat exchangers is determined and presented graphically in terms of the temperature effectiveness of two of the fluids. The effectiveness is determined as a function of heat exchanger size for sets of fixed operating conditions. The introduction of nondimensional operating variables reduces the volume of data required to represent a practical range of operating conditions.—*Willis, N. C. and Chapman, A. J., Trans. A.S.M.E. Jnl Heat Transfer, August 1968, Vol. 90, pp. 333–339.*

Investigation of Vibrational Stress Relief in Steel

Vibrational stress relief in steel has been investigated. Two tests were made: in the first, a small steel specimen being shot peened to induce residual stresses. Cyclic stressing in bending was imposed by a standard fatigue testing machine and residual stresses before and after cycling were determined. It was found that stress relief could be produced under certain conditions and that the magnitude of stress relief could be predicted from the cyclic stress-strain. —*Wozney, G. P. and Crawmer, G. R., Welding Jnl, September 1968, Vol. 47, pp. 411s–419s.*

Thermoplastic Syntactic Foam for Structural Void Fillers

The purpose of this investigation was to determine the feasibility of using thermoplastic matrices for syntactic foams. The results of this exploratory investigation clearly indicate that syntactic foams compounded from polyolefin, polyamide or phenolic resins and glass phenolic or silica microballoons yield materials which are mechanically adequate for many purposes.—*Waite, W. A., Waldron, M. L. and Nahabedian, A., Undersea Technology, June 1968, Vol. 9, pp. 33; 40–43.*

Resistance Butt Welding of 0·8 per cent Carbon Steel

The object of this work was to investigate the flash welding of $\frac{7}{16}$-in 0·8 per cent carbon steel and its subsequent heat treatment in the machine, so as to compare flash welds with resistance butt welds. The inter-relation between the many variables during flashing and upsetting has been investigated and settings are recommended.—*Squires, I. F., British Welding Jnl., December 1968, Vol. 15, pp. 610–620.*

Patent Specifications

Burning Crude Oil on a Marine Tanker

This invention envisages a novel apparatus for safely utilizing crude oil as a fuel for the power plant of a tanker.

Fig. 1 illustrates the relative locations of oil compartments, pump room and power plant area, i.e. boiler room and engine room, on a typical one to three tank vessel. The aft oil tank (1) with starboard and port compartments (starboard compartment shown in the drawing) is separated from the pump room (2) by an oiltight bulkhead (3) extending from the upper deck (4) to the keel of the vessel. Likewise, the pump room (2) is separated from the engine room (5) by bulkhead (6), while the engine room is separated from the boiler room (7) by bulkhead (8) extending from the upper deck (4) to the flat deck (9), as well as by bulkhead (10) extending from flat deck (9) to the tank top of the vessel.

Located above the pump room area (2) is a cross-bunker tank (11) which is separated from the crude compartment (1) by a cofferdam (12). The cross-bunker tank (11) is normally used for carrying bunker oil fuel.

Referring to Fig. 2, when it is desired to burn low flash point crude oil, i.e. crude having a flash point below 150°F, for instance below 120°F, as a fuel for generating steam, the crude may be withdrawn from a cargo tank, for instance the starboard compartment of the aftermost tank (1), through conduit (13) provided with a valve (14) and then passed through strainer (15) on the suction side of pump (16). A second strainer may be provided on the discharge side of pump (16). The crude oil supply conduit (13) is provided with a main crude oil shut-off valve (18), which is preferably air-operated. Crude oil supply conduit (13) extends through the engine room (5) and supplies the burners of port and starboard boilers (19) and (20).

An essential feature of the invention is that all crude oil-carrying pipes and valves located in the power plant area are arranged in a manner such that free oil vapours and

FIG. 1

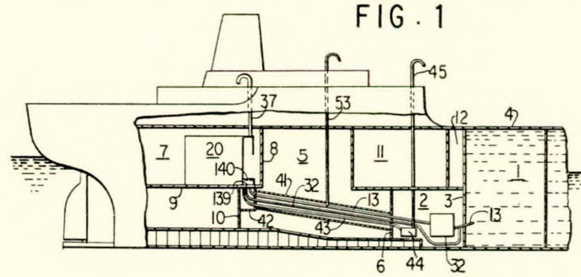
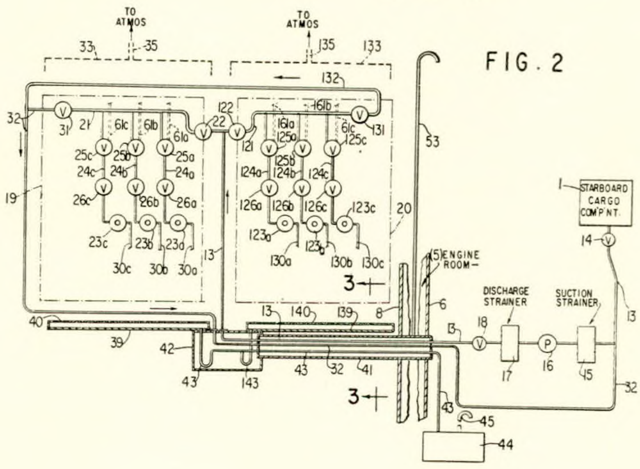


FIG. 2



drippings are not permitted to come into contact with any source of ignition in the power plant.—*British Patent No. 1 138 930 issued to Cities Service Tankers Corp. Complete specification published 1st January 1969.*

Device for Removal of Marine Growth from Ships

This patent proposes a new device by the aid of which a ship can have its sides and part of its bottom cleaned without any need of docking. The cleaning operation can thus be carried out by letting the ship pass through the dock gates, the cleaning device being stationarily arranged ashore.

In Figs 1 and 2 a dock gate is shown with the usual fenders (1′) on the sides. Ship (2) is shown passing through the dock opening. At each side of this opening a cleaning device is arranged, comprising an elongated vertically arranged brush (3) which is rotatably suspended on the free end of an arm (5) swinging from (6) on the structure (1). The brush (3) is suspended so that it can rotate about its longitudinal axis by driving power transmitted through the arm from a source of power (not shown) ashore. Furthermore the brush (3) is mounted in such a way that it can be swung around a bearing (4) so as to be able to swing towards the ship's side, e.g. into position (3′). Thus the brush can be brought into effective contact with the more curved portions at the bow and stern of the ship.

FIG.1.

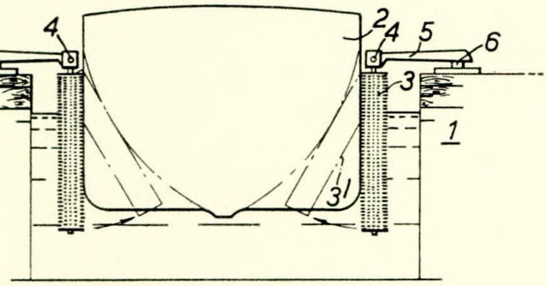


FIG.2.

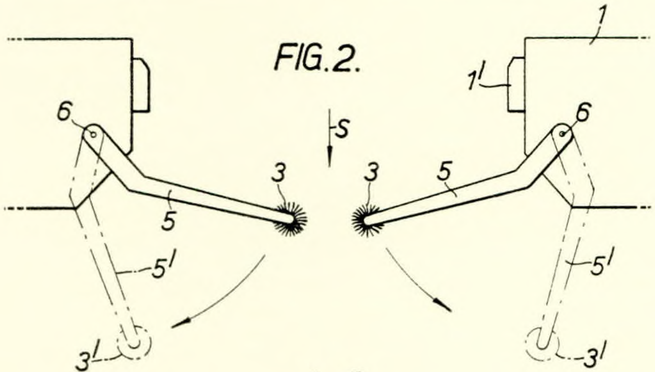
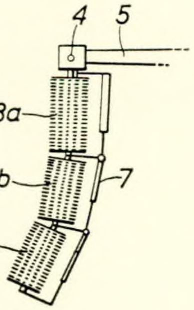


FIG.3.



The vertical brushes may be divided into sections, which are to some degree movable in relation to each other, as shown in Fig. 3. The sections (3a), (3b) and (3c) are here rotatably hinged to each other and can be controlled with respect to the curvature by drawing rods or pressure cylinders (7).—*British Patent No. 1 136 186 issued to J. Vadseth. Complete specification published 11th December 1968.*

Surface-sweeping Vessel

This invention envisages a sweeping ship propelled by the outboard motor (22) (see Figs 1 and 2) to a desired stretch of water with raking blades (19) positioned in close contact with the respective outsides of the pontoons (1a), (1b) as shown by chain lines in Fig. 2. Water which enters the middle hollow space between the pontoons from the front end opening of the ship body flows through the openings (21) in the stern plate (3). After the ship reaches the stretch of water to be swept, the raking plates (19) are swung forward to extend angularly outward and are fixed in this position by the struts (20) as shown in Fig. 2.

FIG. I

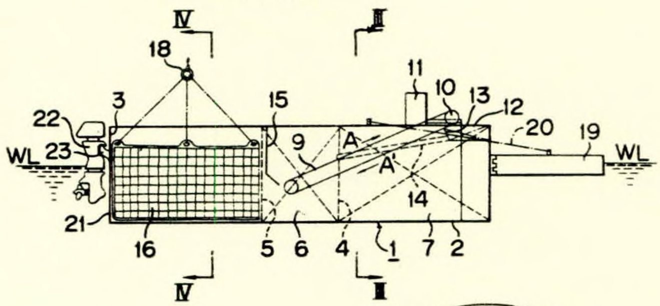
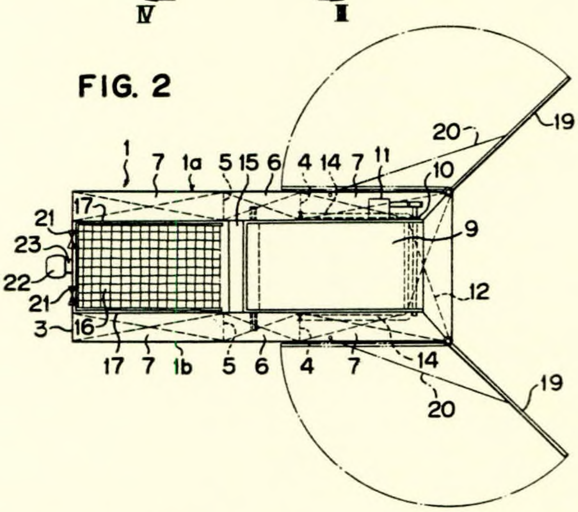


FIG. 2



The prime mover (11) is then started to move the conveyor belt (9) in the direction shown by arrows (A) (A′) in Fig. 1, i.e. with the lower run travelling rearward, while the ship is moved forward slowly by running the outboard motor (22) at low speed. Thus debris and oil floating on or suspended in the water in front of the ship are gathered by the raking plates (19) and fed into the space between the pontoons so that they encounter the underside of the lower run of the conveyor belt (9) which transports them rearwards.

Wood pieces and other debris float up again behind the conveyor (9) and are collected in the wire rope net (16). Oil which encounters the underside of the lower run of the belt adheres to it and is scraped by the upper edge of rear wall (13) of the oil sump (12) to be collected in the sump from which the oil flows through pipes (14) into the oil tanks (6).—*British Patent No. 1 131 513 issued to Mitsubishi Jukogyo Kabushiki Kaisha. Complete specification published 23rd October 1968.*