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Further Funnel Design Investigations

A. D. THIRD, B.Sc., Ph.D., A.R.C.S.T., A.M.I.Mech.E.*, and
W. McCLIMONT, B.Sc., M.I.C.E., M.I.Mech.E., M.Inst.F., (Member)†

The application to the design of funnels of the data presented in a previous paper led to suggestions being made for an extension of the investigation, to include certain conditions not fully covered in the earlier paper.

For a number of the funnel designs, no record had been made of the behaviour of the smoke plume at 10-deg. yaw. Since then, studies of certain ships have shown that the smoke conditions at this angle are more likely to cause trouble than greater angles at which the smoke passes quickly off the ship. In the present investigation, tests have been carried out to include this angle for all designs.

It was suggested in discussion of the previous paper that the possible influence of a variation in the ratio of the uptake diameter to the funnel-casing breadth was not sufficiently considered. Arrangements were therefore made to carry out tests on an orthodox funnel, covering a wide range of this ratio. The results do not show any appreciable difference in performance.

Although a representative range of funnel shapes was included in the previous paper, the opportunity has been taken to test a few more. Among these is a top with a rounded rim, which is characteristic of many funnels today.

In the earlier paper, simple rules for the design of funnels were formulated and a sequence of calculations to be used in the applications of these rules was given. At that stage it was not easy to reach exact values for the interpenetration that could be allowed of the turbulent zone above the superstructure and the smoke plume before a major breakdown of the plume occurred. Experience in the application of the data has enabled the interpenetration to be defined in equation form. This in turn has permitted a correlation of the height of the funnel, the height of the turbulence boundary and the level of the lower boundary of the smoke plume in the form of a chart which should facilitate the application of the funnel design data.

It was previously stated that the buoyant effect of the hot gases might be neglected, which implied acceptance of the fact that the model performance is a little on the pessimistic side. Some further tests were carried out with heated gas and very low wind-tunnel velocities to allow for scale effects; these confirm that the influence of buoyancy is small except under special circumstances.

INTRODUCTION

This paper describes further investigations on the behaviour of the smoke plume, prompted by the application of the rules formulated in a previous paper [1] to full-scale practice.

The experimental arrangements and method were fully described in the previous paper, and the same techniques have been adopted. The nomenclature adopted has also been retained and is repeated below.

Nomenclature

- b' = Breadth of funnel outer casing at base.
- e = Extension of uptake above funnel casing top, in terms of b' $(=1)$.
- H = Height of funnel to casing rim above datum deck level, in terms of b' $(=1)$.
- h' = Vertical height of extreme lowest point of smoke plume above level of casing rim, in terms of b' $(=1)$.
- h_t = Maximum height of turbulence boundary above datum deck level, in terms of b' $(=1)$.
- p = Interpenetration ratio.
- s = Mean speed of efflux of gases.
- v = Relative wind velocity over funnel (same units as s).
- Angle of yaw = Angle of relative wind off the bow.

VELOCITY DISTRIBUTION ACROSS UPTAKE OUTLET

Before proceeding with the tests, it was decided to investigate more closely the rather surprising results noted in the earlier paper for the "domed" top with an angled rectangular uptake. The plume rises considerably higher than for any of the other funnels over the lower range of s/v ratios, while at the higher ratios, wisps of smoke are seen to be shed from the rising main plume.

It was thought that the velocity distribution across the outlet might have something to do with these results, since the angling of the uptake necessitated a bend just below the outlet. It is well known that a bend in a duct in a plane parallel to its wider side causes unequal velocity across the duct section beyond the bend, and this difference could be quite appreciable for the low "aspect ratio" of 1 : 5, even with the generous radius provided at the bend in the model. It may be concluded that the higher gas velocity towards the forward edge of the outlet resulting from this, provides a shield for the lower velocity stream towards the aft edge, thus lifting the plume to a higher

* Director of Research of Thermotank Ltd.

† Formerly Principal Assistant Chief Marine Engineer, British Ship Research Association.

level than it would be if the velocity is uniform across the section. The higher the s/v ratio, the more chance there is of some of the smoke in the aft portion of the jet being carried away by the wind.

In any investigation of this effect, it is of course essential to know how the velocity is distributed across the outlet. It was therefore decided to provide an outlet flow distribution such as would be obtained in undisturbed pipe flow, by arranging a plenum within the funnel casing, followed by a bell-mouth entry into a straight and uniform final length of uptake. This arrangement was applied to the angled rectangular uptakes of both the domed and the shaped tops. Since the velocity distribution even in a vertical discharge might be affected to some extent by the conversion from circular to rectangular uptake section, it was decided to include the vertical uptake of the shaped top in these tests. The results are shown in Figs. 1 and 2.

It is observed that the slope of the curves now follows more closely the general trend of the cylindrical uptakes, and that the "turn-over" at the high s/v ratios has disappeared, as was anticipated. Both tops still give a decidedly better showing at the lower than at the higher angles of yaw, since they were designed to give their best performance under the former conditions. The characteristic is due in large measure to the rectangular shape of the uptake section; with the cylindrical uptake in the domed funnel, the effect of yaw is not nearly so great.

It must be pointed out that the ratio of length to breadth of the rectangular uptake section in the models is rather extreme at 5 : 1. In practice, the ratio is likely to be a good deal lower, say 2·5 : 1 or less, so it is rational to assume that the effects of velocity distribution across the outlet will be much less marked.

EXTENSION OF TESTS ON EXISTING MODELS

In the earlier investigations, some of the funnels were tested at angles of 0, 20 and 30-deg. yaw only, although in a few cases 30-deg. yaw was omitted. It is well known that the smoke plume is very sensitive to yaw and that when the critical angle of around 15 or 20 deg. is reached, the suction in the lee of the funnel casing can cause the smoke to be drawn strongly towards the decks. However, as the angle is increased, the smoke has a shorter length of ship to pass over before going over the side, hence there is a particular angle which, by combination of downwash and length-of-path effects, results in the most unfavourable smoke conditions.

Since the earlier work was completed, investigations of a number of ships have shown the profiles to be such that the worst conditions could well be at an angle of around 10-deg. yaw, so it was decided to carry out further tests to include this angle in the tests of all the funnels.

The curves representing 10 and 30-deg. yaw where applicable have been added to the figures prepared for the previous paper, and are given in Figs. 1 to 9, which correspond to Figs. 18, 19, 8-11 and 15-17 respectively, of the earlier paper(1).

The previous tests which include the 10-deg. yaw angle gave results roughly midway between the 0 and 20-deg. curves and seemed to indicate that the deterioration in performance over the range of angles was progressive rather than sudden. The same may be said of most of the additional tests, although there

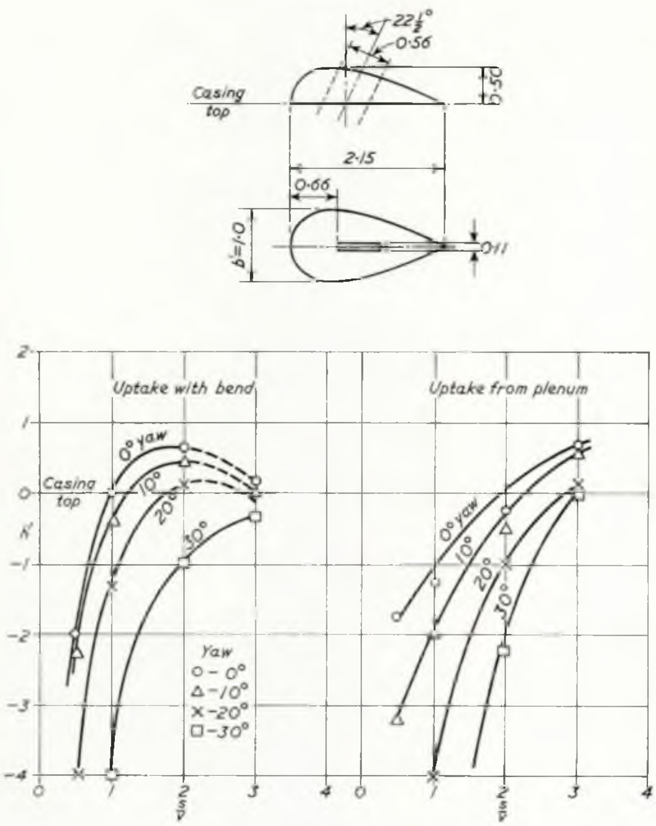


FIG. 1—*Domed top with rectangular uptake*

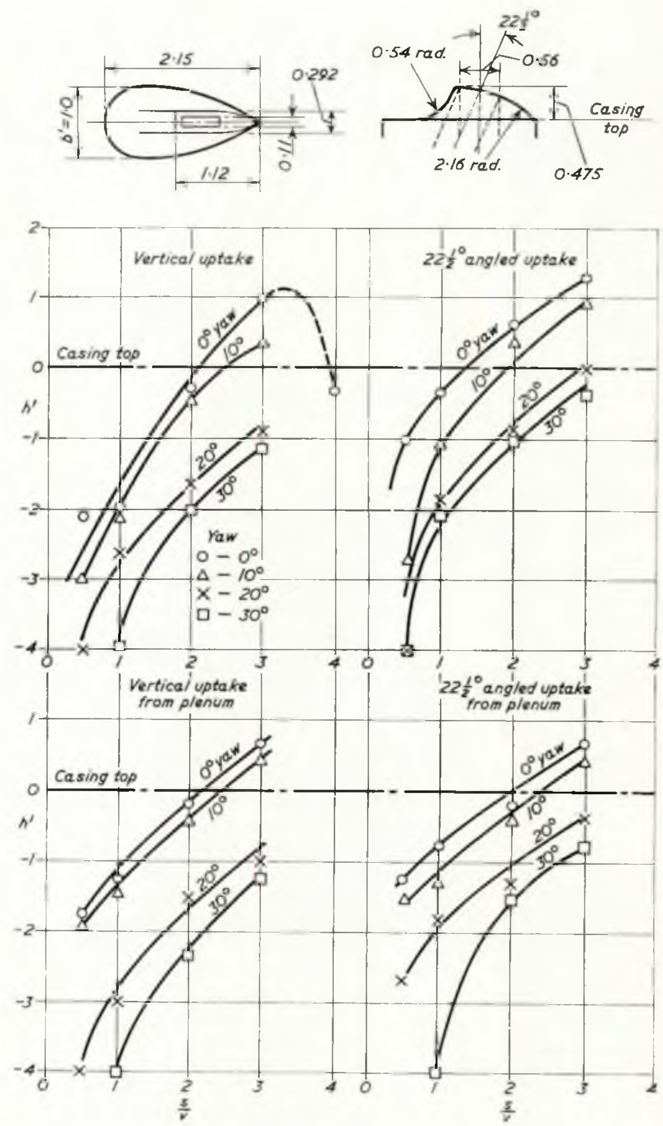


FIG. 2—*Shaped top*

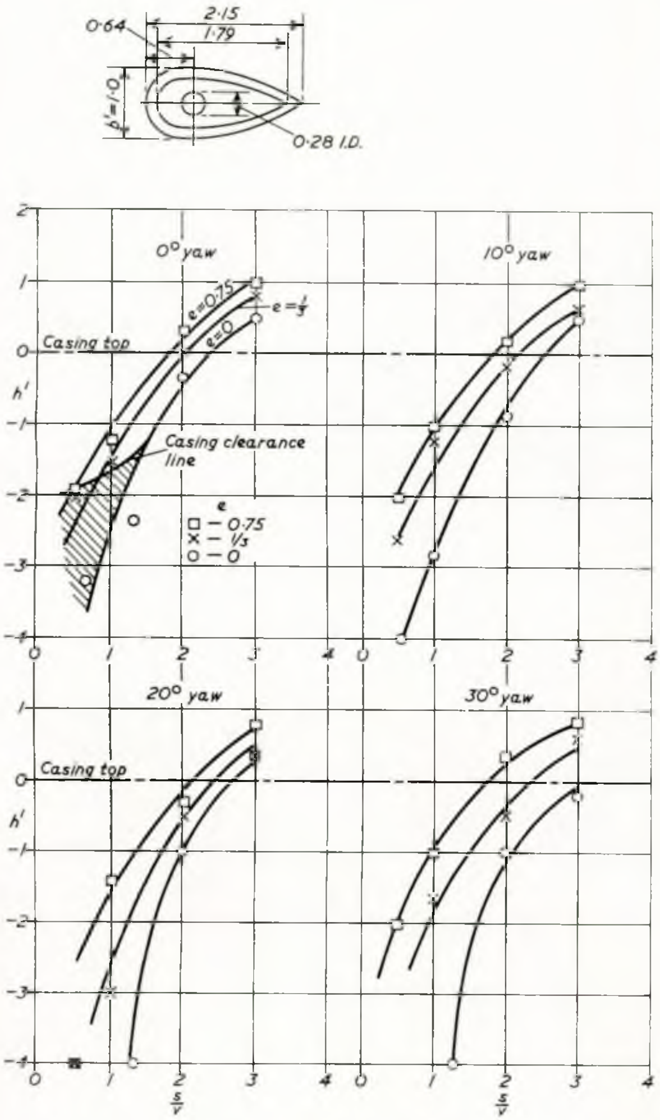


FIG. 3—*Casing B*

is a not unexpected tendency for the 10-deg. curve to lie rather closer to the 0-deg. curve. An extreme case is the fine streamlined casing C, which, as reported in the earlier paper, gives a first-class performance at 0-deg. yaw, but shows up very poorly at 20-deg. yaw, owing to the pronounced effect of the critical angle for this profile shape. The results now recorded for 10-deg. yaw are nearly as good as for 0 deg., showing that the critical angle has not been reached. A further test in which the funnel was slowly rotated indicated that the critical angle for this profile is between 20 and 25 deg.

EFFECT OF UPTAKE SIZE IN RELATION TO CASING

The number of variables entering into the effect of uptake size in relation to casing makes it impossible to cover all the possible combinations. However, it had been suggested that the effect of the diameter of the uptake or its equivalent in terms of casing breadth b' could be of greater signficance than might be inferred from the earlier study and might warrant further investigation.

In most of the earlier tests, the ratio of uptake diameter to casing breadth b' was 0·28, but it was reduced to 0·21 when the same uptake was introduced into a larger casing in one test. The latter gave a flatter curve of plume height h' on a base of s/v than did the equivalent condition for the smaller casing, which could lead to some doubts in applying the results to actual examples, particularly where the ratio falls between these values. Arrangements were therefore made to investigate the effect of this variable more fully.

On this occasion casing A (see Fig. 10 for dimensions) was used for all the tests, and the uptake diameter was varied. An uptake of 0·44-in. diameter was first tested, followed by one of 1·69-in. diameter, so that ratios of 0·14, 0·28 and 0·56 are represented, when the standard uptake of 0·875-in. diameter is included.

The results are shown together for comparison in Fig. 10. It will be observed that, even for this very wide range of the ratio there is no appreciable difference in the plume heights; certainly not enough to suggest that it should be taken into account in calculations.

TESTS ON FURTHER SPECIAL FORMS OF TOP

Since the earlier paper was presented, a number of special forms of top has been featured in current shipbuilding. Models of a few of these were made and tested, to supplement the range already investigated. These new forms have been classified as follows:

Special forms of top, mounted on streamlined casing A
Rounded (Fig. 11):
with rectangular uptake angled 22½ deg.

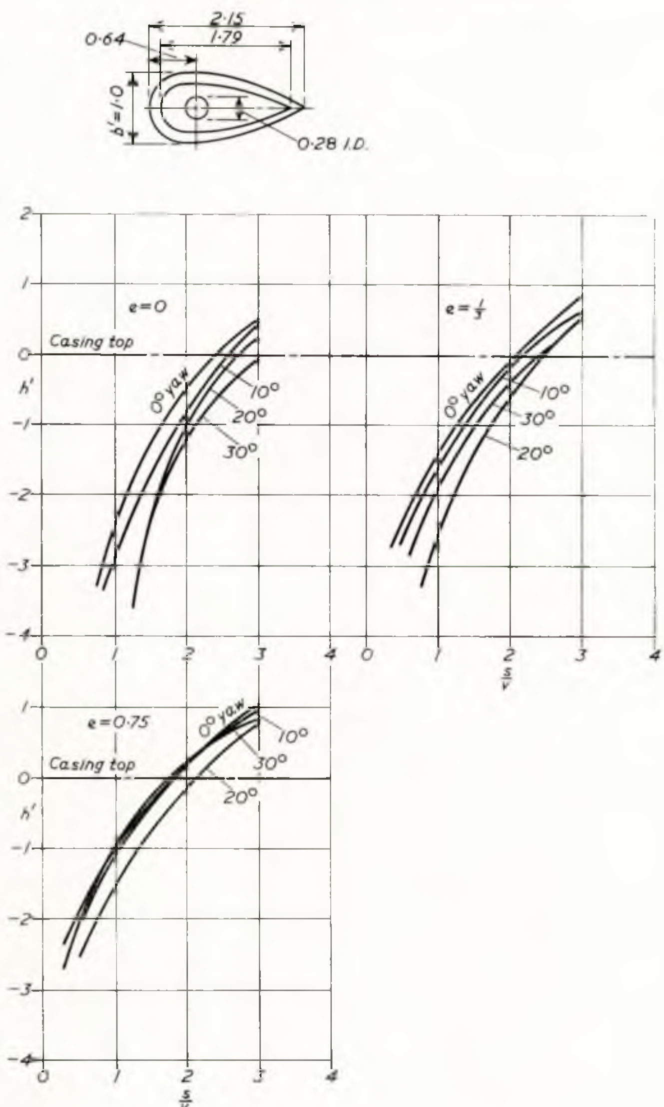


FIG. 4—*Casing B*

Winged (Fig. 12):
- a) with wings (or ailerons) at 0-deg. incidence;
- b) with wings at 15-deg. incidence and 15-deg. dihedral angle;
- c) as (b) but with discharge through lee wing only;

Shaped projection (Fig. 13):
- i) with stepped vane arrangement;
- ii) with vanes removed.

It will be appreciated that the construction of a fully domed top involves a considerable amount of high-class sheet-metal work, which would be contemplated only for first-class liners. A compromise in the form of a flat top with a rounded rim suggests itself, and is indeed quite common practice in all classes of vessels. A top of this type, described as a rounded top, was therefore included. The outlet, of rectangular shape, was led up from a plenum chamber within the casing.

From time to time suggestions have been put forward for a funnel having the gas outlets brought out through wings or ailerons some distance to port and to starboard, near the top of the main casing. Publicity has been engendered by the recent entry into Atlantic service of a large vessel equipped with funnels of this type. There are, of course, many variations possible in the form of the components of such a funnel, which would call for a great number of tests in a complete analysis. Fortunately, however, the smoke plume is influenced very little or not at all by most of the characteristic design features. It would be reasonable to claim therefore that representative performance could be obtained from a few tests including variables which clearly do affect the plume. One of these is the angle of incidence or tilt of the ailerons on an athwartship axis. If the angle is made negative, i.e. with the trailing edges higher than the leading edges, the effect is to deflect the airstream upwards, and so assist in lifting the smoke plume clear of the suction in the lee of the main casing. Further assistance in this direction can be obtained by tilting the axes of the ailerons upwards, i.e. giving them a "dihedral" angle, so that the outlets are brought up as high as possible relative to the main casing.

It is clear that in yaw, the gas from the windward-side outlet must come under the influence of the turbulence created by the ailerons and the main casing, but if all the gas is diverted through the lee aileron, the smoke is less likely to be disturbed, and it goes off the ship very quickly in any case. It is reasonable to expect a very good performance, provided that arrangements can be made for the correct aileron to be in use, according to the wind direction.

The tests on this funnel were run first with no incidence and no dihedral angle on the ailerons, then with 15 deg. incidence and 15 deg. dihedral angle, and finally with these angles and a single aileron in use.

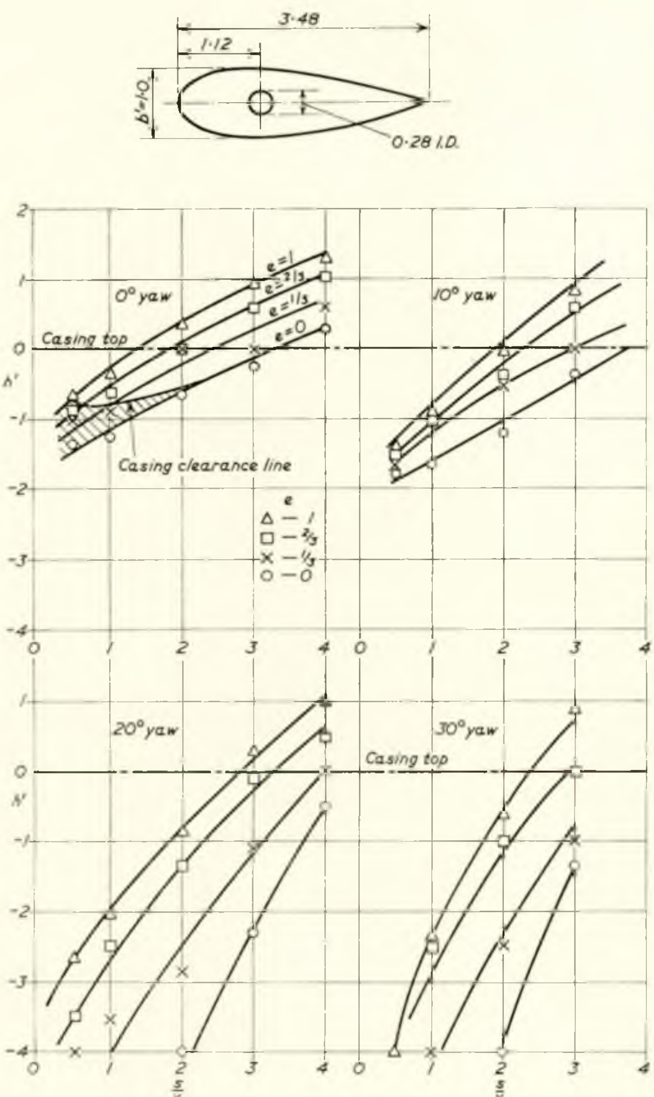


FIG. 5—*Casing C*

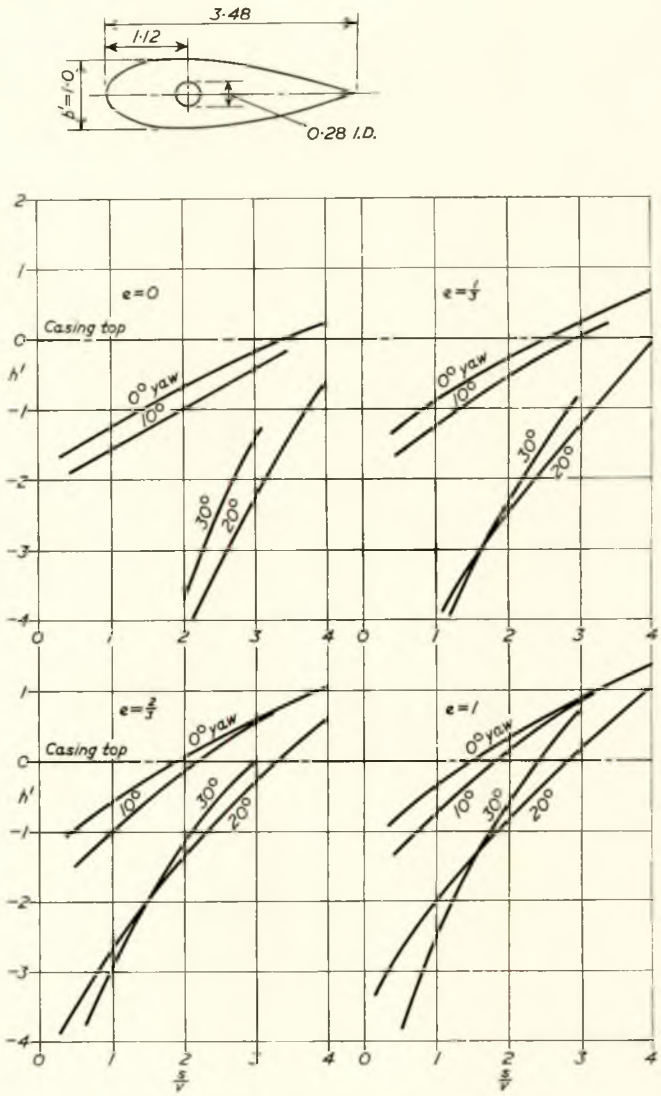


FIG. 6—*Casing C*

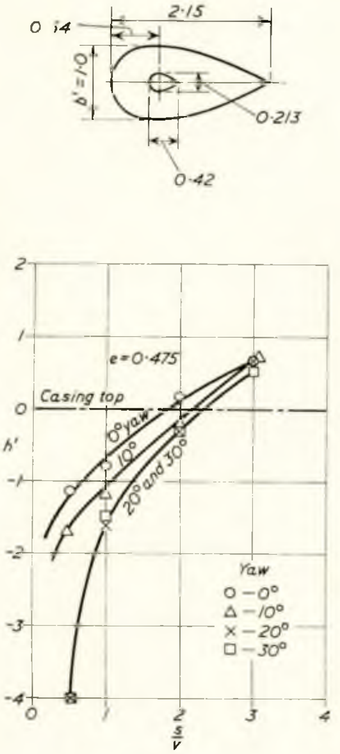


FIG. 7—*Casing A—Streamlined uptake*

The last addition to the models is of a type incorporating a fine streamlined top portion added to the orthodox casing A, details of which were kindly supplied by the Shaw Savill Line. Just below the top was mounted an arrangement of guide vanes: a forward vane was turned slightly downwards from the horizontal, and a rear vane was turned slightly upwards. The latter vane was attached a little below the forward one, thus providing a slot by way of which a vigorous stream of air was led along the top of the rear vane. Both vanes were flanged upwards along their sides. This design was developed by Professor Wille of the Hermann Föttinger Institute in Berlin.

This top combines the beneficial effect of three features:

a) an increase in the height of the funnel;
b) a fine streamlined top;
c) guide vanes, whose function is to prevent the smoke filling up the low pressure region immediately abaft the funnel.

RESULTS OF TESTS ON SPECIAL FORMS OF TOP

(1) *Rounded Top* (Fig. 11)

This top may be compared with the modified domed-top funnel having a rectangular outlet leading from a plenum chamber within the funnel, Fig. 1. At 0-deg. yaw the performance of the rounded top is not quite so good over the whole range of s/v ratios. At the higher angles of yaw, the domed top is rather poor, but the rounded top is poorer still. It must be stated, however, that the relatively poor performance of both tops in yaw is due to the rectangular shape of the uptake. Earlier tests on the domed top with both a cylindrical and a rectangular uptake make this quite clear.

(2) *Winged Top* (Fig. 12)

The results from the various arrangements of the winged

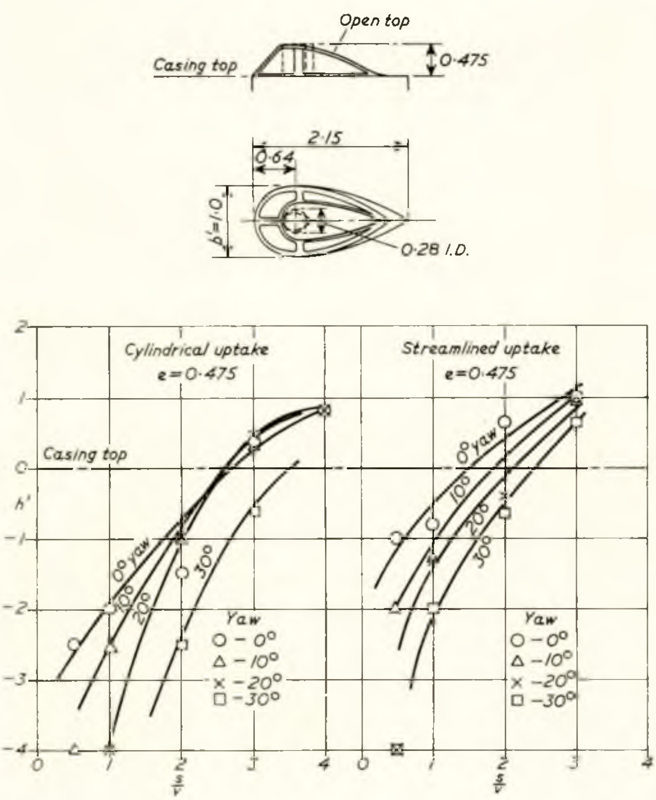


FIG. 8—*Caged top*

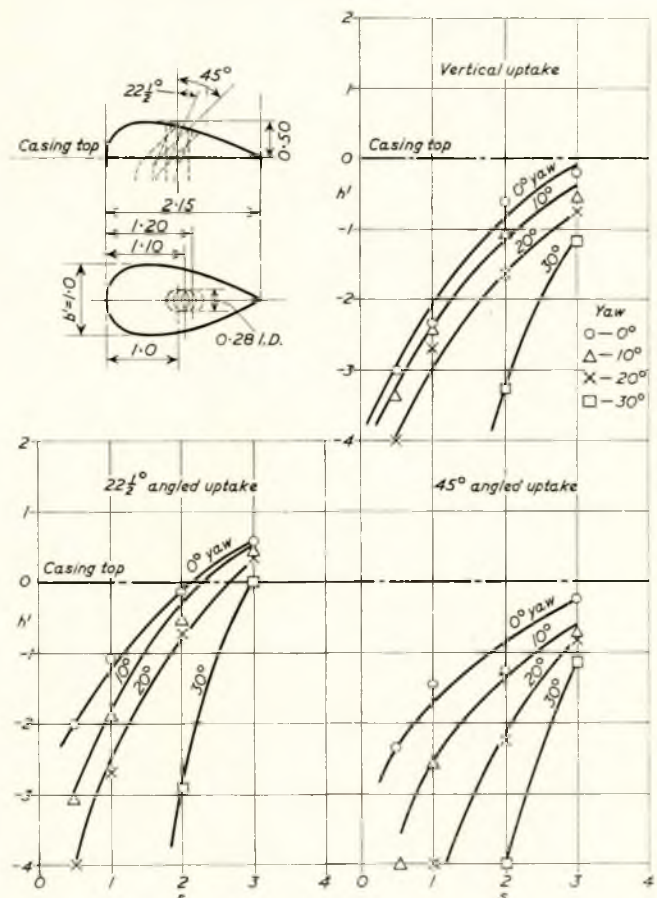


FIG. 9—*Domed top with cylindrical uptake*

top are much as was to be expected. With the ailerons at zero incidence, the characteristic initial rise of the plume with upward discharge funnels is absent. However, at low angles of yaw the plume does not show any tendency to fall, even at the lowest s/v ratios. Only at angles of the order of 20 deg. or more does the plume begin to be affected by the suction in the lee of the funnel casing.

With the ailerons swung round to give a lift to the airstream, and also tilted upwards, the performance is notably better. The funnel casing has very little influence up to yaw angles of about 30 deg., so that this arrangement of funnel should give a satisfactory performance under most conditions in service. Of course there are vessels such as tankers, where the behaviour of the smoke plume in astern winds is of great importance. For such vessels it might not be advisable to swing the ailerons and so impair the performance in astern-wind conditions.

The provision of a three-way valve to direct the smoke through the lee aileron only when the ship is in yaw, results in a uniformly good performance through all s/v ratios and yaw angles up to 30 deg.

(3) *Shaped Projection* (Fig. 13)

The results for 0 deg. and small angles of yaw for the

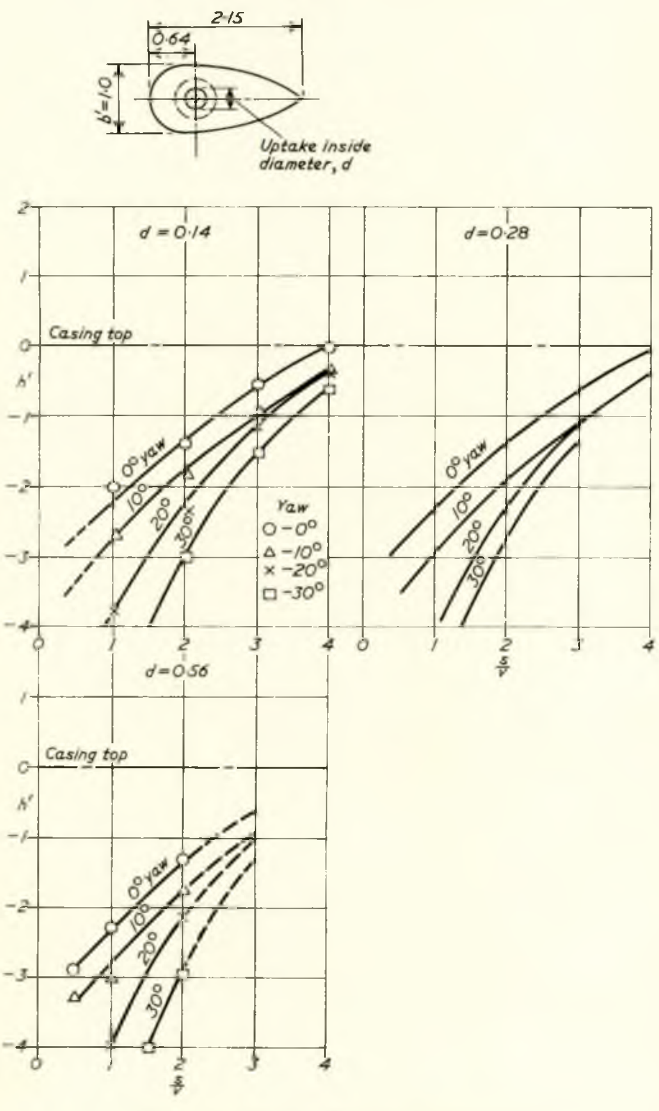


FIG. 10—*Casing A uptakes forward*
$e = 0$

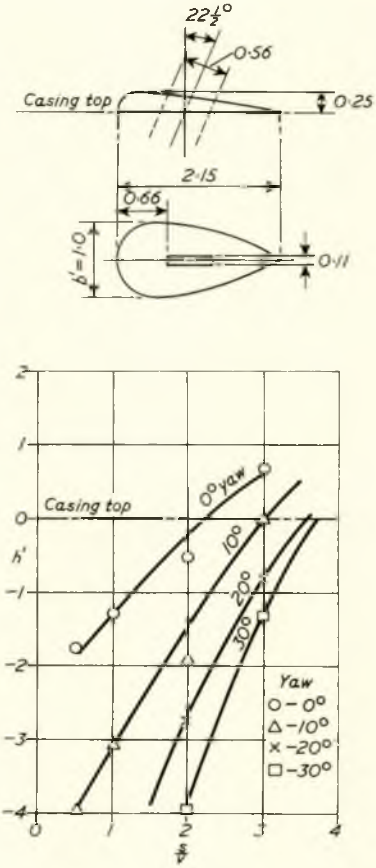


FIG. 11—*Rounded top*

funnel with a shaped projection are very good, mainly as a result of the fine streamlining of the top. The effect of the vanes themselves on the plume height downstream is not very pronounced, any advantage being limited to the lowest s/v ratios. The real benefit of the vanes is to be found in the prevention of smoke eddies coming down the lee side of the funnel, so that the paintwork keeps clean for a longer period. This is demonstrated clearly from photographs taken with and without the vanes mounted in place. The photographs have not been included, since the eddies would not appear clearly in a reproduction.

The results from the modified uptake arrangements on the existing tops and on these new tops are compared with some of the earlier results in Figs. 14 and 15.

Discussion of Results

It has been observed that an unequal velocity distribution across the uptake outlet can affect the pattern of the smoke plume to quite an appreciable extent. Since the marine engineer is faced with many practical problems when arranging the layout of the uptakes within the funnel casing, and may tend to sacrifice good gas-flow conditions to the exigencies of space and economy of construction, some comment on the factors involved will not be out of place.

At first sight it may appear that something is to be gained by having a backward bend in the uptake to give a non-uniform velocity across the outlet, since the plume is found to rise to a higher level under certain conditions of operation. But with the same outlet area and volume of gas flowing, the kinetic energy in the non-uniform stream is obviously greater, so that any improvement in performance is secured only at the expense of extra fan power. Whether it is better to use power in this manner, or to expend the same power in a higher average velocity with a uniform distribution across the outlet, will de-

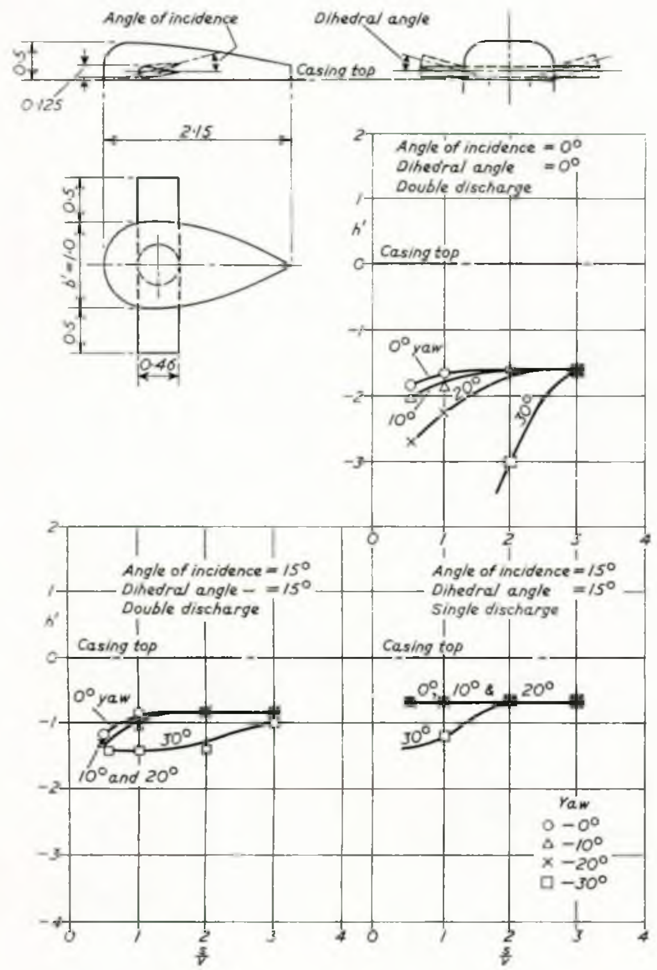


FIG. 12—*Winged top*

pend upon details of the uptake form. A complete analysis would be very involved, but a fair conclusion from the tests is that it is better to design for as uniform a velocity as possible.

The later results show that when the velocity is uniform, there is not a great deal of difference between a vertical discharge and one at say 22½ deg. to the vertical. From a practical standpoint, the angle of discharge may be dictated by the position of the uptakes in the plan arrangement of the casing; sometimes a vertical discharge may be more easily arranged.

To sum up, due attention should be given to the following points when the uptakes are being designed:

- i) changes from circular to rectangular section, or any reduction in section area, should be carried out in tapered conversion pieces;
- ii) sharp bends should be avoided;
- iii) if dampers are to be fitted to adjust the velocity of discharge or to shut off the uptake, they should be so designed that any disturbance to the gas flow is kept to a minimum;
- iv) the final straight length should be as long as is practicable;
- v) the discharge should preferably be raked aft at an angle of from 0 deg. to not more than 20 or 25 deg.

Since it is desirable to make a quick assessment of the effect of changes in outlet velocity and of bends, etc. in the uptakes on the fan pressure required, a note on the calculations involved is given later in the paper.

The tests with the various diameters of uptake in the same funnel casing have confirmed the suitability of choosing the breadth b′ of the casing as a basis for the comparison of results, for this arrangement of uptakes at least. It would appear logical to infer that this will apply whenever the plume comes under the influence of the funnel casing, and the designer is normally concerned only with such cases. Only when the plume is projected clear of the casing altogether, does another parameter, that of the uptake diameter or its equivalent, begin to have its own influence.

The Use of the Interpenetration Fraction

Since the completion in 1959 of the main funnel design investigation, the British Ship Research Association has been asked on a number of occasions for advice on funnel smoke problems and this has provided opportunities to examine the validity of the data and also to note the ease with which they may be employed. The results achieved by use of the data have been good and have shown that the design rules, including the allowable interpenetration fractions, appear to be sound. Application of Rule 1 has, however, been cumbersome on certain occasions owing to the discontinuous relationship between the interpenetration fraction, p, and the vertical height, h', of the extreme lowest point of the smoke plume above the level of the casing rim. Anomalous results may, in fact, be obtained in certain circumstances and it has been felt that discretion has had to be used which could only be obtained by experience in the use of the data. This might well discourage designers from making use of these design rules and some simplification has, therefore, been sought.

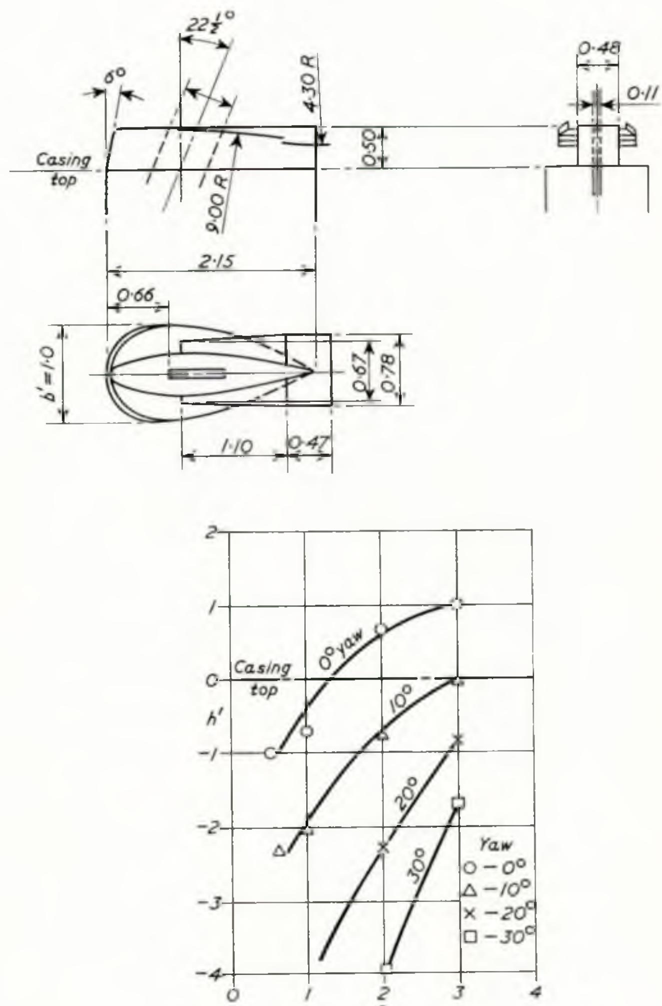


FIG. 13—*Shaped projection (with vanes)*

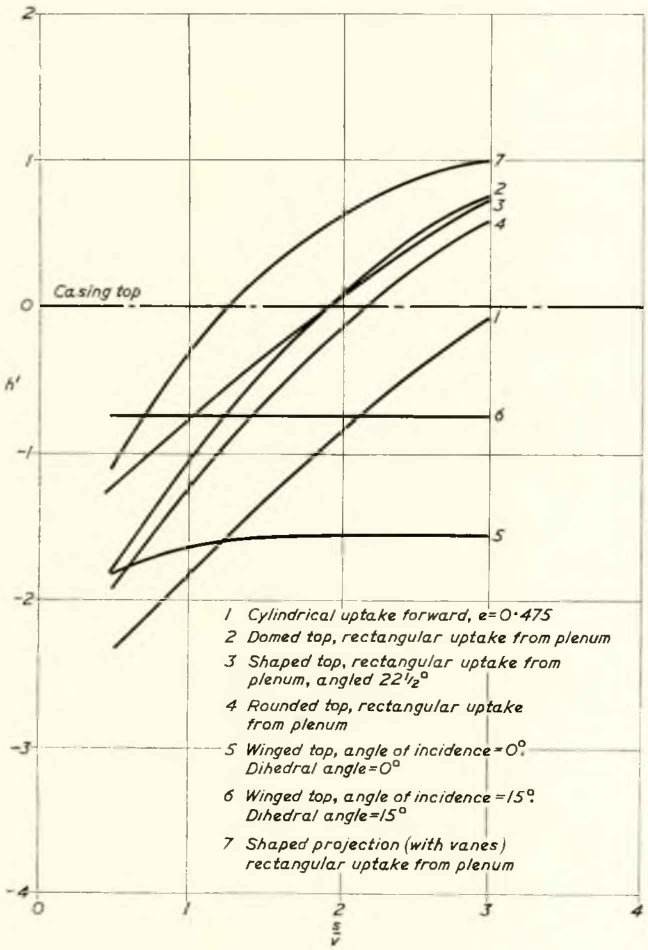


FIG. 14—*Influence of design on plume height h′, casing A, 0 degrees yaw*

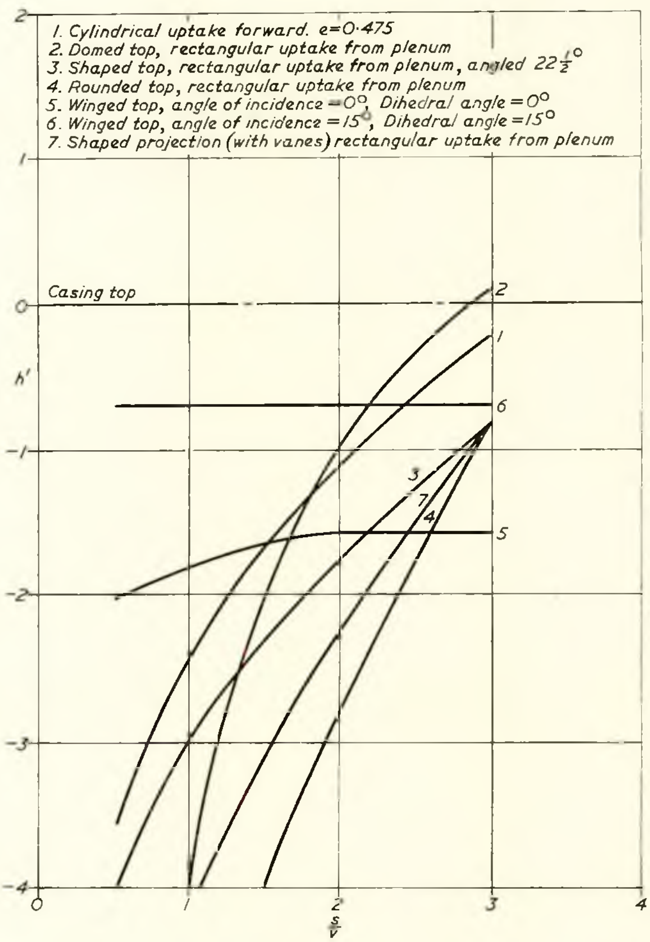


FIG. 15—*Influence of design on plume height h′, casing A, 20 degrees yaw*

Definition of the Allowable Interpenetration

Rule 1, as set out in the previous paper, read:
"The lower boundary of the smoke plume may be allowed to penetrate the zone of turbulence created by the ship's superstructure to a vertical depth in accordance with the following table, in headwinds and winds up to a maximum of 20 deg. angle of yaw".

TABLE I.

h'	Interpenetration fraction allowed
Above $-0{\cdot}5$	0·35
From $-0{\cdot}5$ to $-1{\cdot}5$	0·50
Below $-1{\cdot}5$	0·70

The dotted lines in Fig. 16 are a graphical expression of this relationship and it will be seen that it is discontinuous.

It is now proposed that the relationship between p and h' should be defined by the full line in Fig. 16, which is, of course, a continuous one, and can be expressed by the equation:

$$p = 0{\cdot}3 - 0{\cdot}2\ h' \qquad (1)$$

This relationship has been found to work well in practice and obviates the anomalies found when using the earlier relationship.

APPLICATION OF RESULTS TO FUNNEL DESIGN

Under this heading in the previous paper[1], a sequence of calculations to be followed in applying the rules to the design of a funnel were given, following a discussion on the conditions under which either Rule 2, or both Rules 1 and 2, should be applied.

The designer will probably look at the problem from one of two aspects. He may either: (a) decide upon the limiting

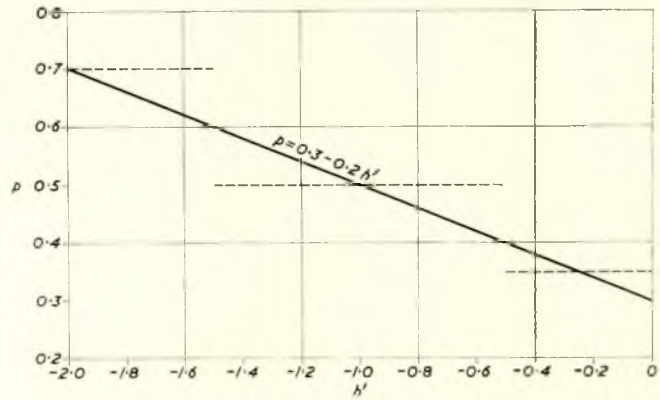


FIG. 16—*Relationship between allowable interpenetration factor and the height of the lower boundary of the smoke plume*

value of the gas speed he can allow and proceed to calculate the minimum required funnel height for different funnel arrangements or, (b) decide upon the funnel height he wishes to have and proceed to determine the lowest value of the gas speed he must allow. Certain preliminary calculations and decisions which are common to both methods, have first to be made from a study of the drawings of the ship. These are repeated here.

Determine the height of the turbulence boundary h for the superstructure by the methods described in an earlier paper by Ower and Third[(2)]. This is given relative to the beam width $b(=1.0)$ of the ship, and is measured from the superstructure top or the imaginary top, as defined in column 2, p. 117 of that paper. Since the rules now to be applied are based on the funnel dimensions, it is necessary to convert the units in which h is measured from hull-beam widths b into funnel widths b'. Denote the height of the turbulence boundary so expressed by h_t, the measurement being taken from the same datum as h in reference 2. Note that in a tanker or similar vessel with navigation bridge well forward, the height of the boundary of the turbulence caused by the after superstructure should be taken.

Classify the form of the funnel casing with one of those tested. It will generally be possible to compromise when there is not exact agreement. Thus, it has been noted that a length/breadth ratio of from 1·5 to, say, 2·5 can be represented by casing A with sufficient accuracy. Again, on many vessels, the uptake is not circular or there is more than one uptake. If these do not project much above the casing top, they can be represented by a single circular uptake of the same total cross-sectional area, unless the length/breadth ratio of the uptake or group of uptakes is greater than about 2·5.

The plume is very sensitive to yaw in most funnel designs, so it is imperative to study the plan of the ship to find the maximum angle for which Rule 1 should be applied. On some ships, only a poop deck may have to be considered, in which case the maximum angle could be less than 10 deg. On others, it might reach 20 deg. At greater angles, the smoke plume passes off most ships' decks before penetrating the turbulent zone. It was stated earlier that Rule 1 should be observed when a line drawn from the funnel-casing top at an angle of 20 deg. below the horizontal does not clear the profile of the decks aft. This construction may also be used to determine the maximum angle in plan. If this downward-sloping line is swung round in plan so that it sweeps out a cone whose apex is at the funnel, an angle is reached when it clears the ship altogether (Fig. 17). This angle is taken as the maximum yaw in applying Rule 1.

Decide upon the relative wind speed v to be used in determining the velocity ratio s/v. It is clear that head winds will reduce the value of this ratio, and so impair the performance of the funnel, but it would hardly be logical to take into account the possible maximum wind speeds in evaluating the ratio. In high winds, the decks of passenger ships would probably be deserted, and in any case the smoke would be very much diluted, so it would seem reasonable to design for a wind of say 15 to 20 knots, depending on the prevailing winds on the regular service of the ship. However, it is necessary to make sure that no fumes or smoke enter ventilation intakes under any condition of working, so a higher wind speed would have to be taken where there is a possibility of this happening. On the other hand, the ventilation intakes are usually not more than a few degrees off the fore-and-aft line, except when near the funnel base, so that in such cases it would be sufficient to limit the calculation to 0 deg. yaw at these higher wind speeds. For some ships therefore, to comply with Rule 1, an evaluation of the plume height h' may have to be carried out for two conditions namely 0 deg. yaw at a relatively high wind speed and an agreed angle of yaw at a moderate wind speed. The lower of the two values of h' is taken in estimating the necessary gas velocity. The wind speed is added to the ship speed to determine v for head-wind conditions, but of course the velocities are added vectorially for yaw conditions. The ship speed should be the normal or service speed of the vessel, upon which the gas speed s is assessed.

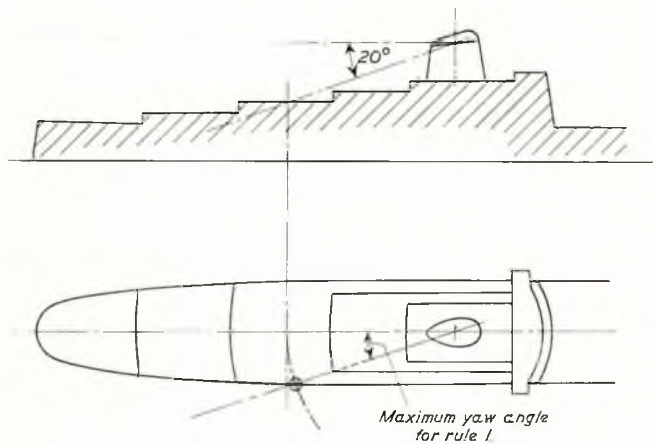


FIG. 17—*Maximum yaw angle for Rule 1*

Case (a). Maximum Gas Speed Fixed by Design: Calculation of Minimum Funnel Height

To determine h' from the s/v value classify the funnel with one of the forms tested, then refer to the appropriate figure of the present paper or reference 1, which have been compiled from the test results. Note that the ***negative sign*** must be used with h' when the lower boundary of the plume comes below the funnel-casing top, as it usually does.

Rule 2 should be applied first in order to make sure that the performance of the funnel itself is satisfactory at angles of yaw up to say 30 deg. To meet this rule, h' must not fall below -2 for all angles of yaw up to say 30 deg., although this angle may be relaxed somewhat if it is clear that the descent of smoke at the higher angles of yaw would not constitute an unacceptable nuisance. Table III of reference 1 will be of assistance in many cases as it gives the minimum s/v values necessary to comply with the rule for many of the funnels tested; when using this table, calculations for the relative wind velocity should be based on 30 deg. angle of yaw. If it is found that the s/v value originally fixed is not high enough, and cannot be raised, then the design of the funnel top must be improved before proceeding to apply Rule 1.

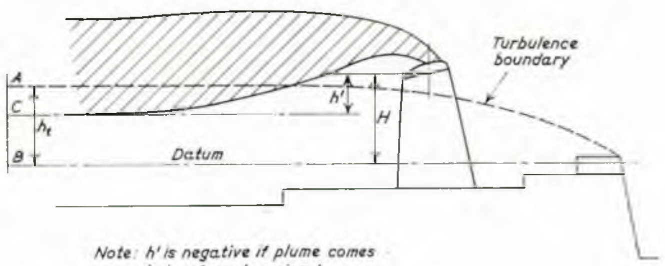


FIG. 18

Rule 1 is based on the degree of interpenetration of the smoke plume and the zone of turbulence, which is expressed as a fraction of the turbulent-zone height. Denoting this fraction by p, then from Fig. 18, its value is given by:

$$p = \frac{AC}{AB} = \frac{AB - BC}{AB} = 1 - \frac{BC}{AB} = 1 - \frac{H + h'}{h_t}$$

also equation 1 gives $p = 0{\cdot}3 - 0{\cdot}2\,h'$.

Therefore the minimum funnel height H is given by

$$H = h_t\,(0{\cdot}7 + 0{\cdot}2\,h') - h' \qquad (2)$$

Note that H is the height of the funnel measured above the same datum as the turbulence boundary.

Equation 2 has been expressed in chart form in Fig. 19 so

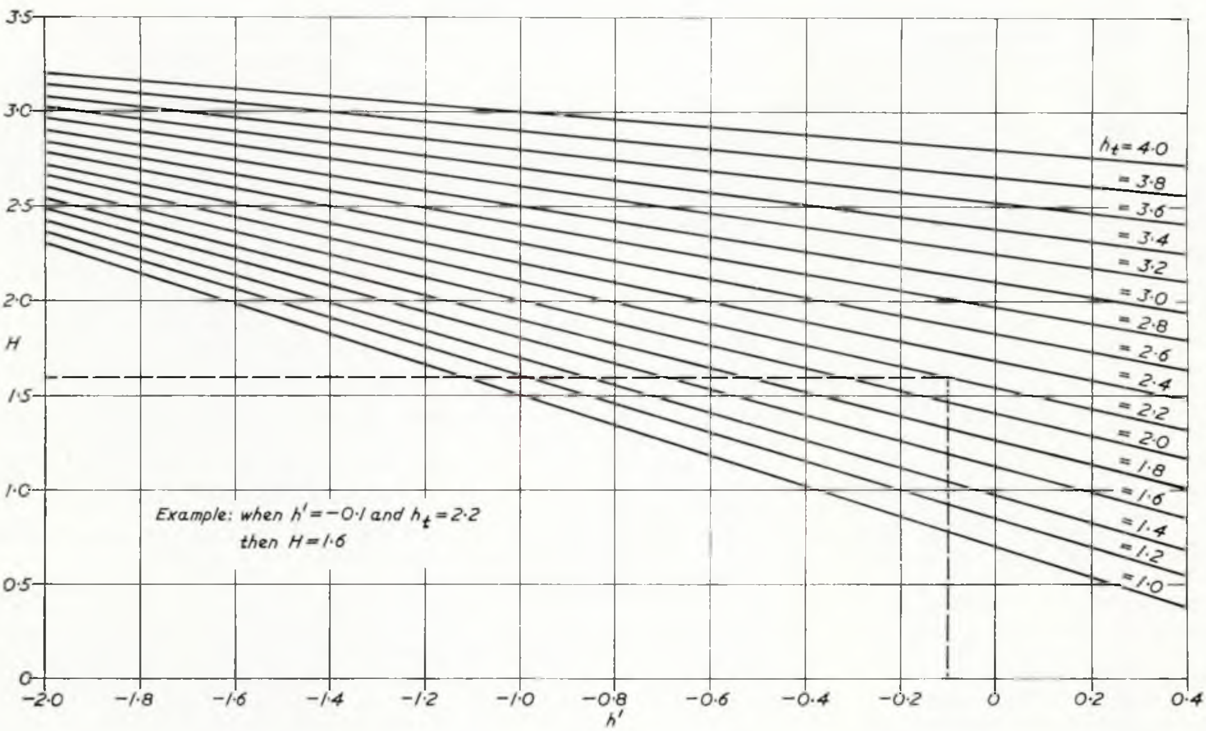


FIG. 19—*Values of H, h_t and h′ which satisfy Rule 1*

that by the use of the known values of h_t and h', H can be read off directly.

If h' was negative in the application of Rule 2, it is now necessary to check that H is not less than the numerical value of h'. This is to conform to the section of Rule 2 which states that when the height of the funnel is less than a distance equivalent to twice the breadth of the casing, the allowable descent of the plume is reduced accordingly. If H proves to be less than the numerical value of h', the minimum funnel height allowable must be increased to the numerical value of h'.

Case (b). Funnel Height Fixed by Design: Determination of Minimum Gas Speed

At the minimum gas speed, the velocity ratio s/v must be such that the plume height h' is sufficient to meet the requirements of Rule 1. The minimum acceptable value of h' must therefore be determined. Since the values of H and h_t are known, the minimum acceptable value of h' can be read directly from Fig. 19.

Classify the funnel with one of the forms tested, as for Case (a), then refer to the appropriate figure of the present paper or reference 1 to find s/v for this value of h'. Assess the value of v to be taken, and so determine the gas speed s.

Rule 2 is then applied by using the value of s just determined in association with the relative wind velocity v for 30 deg. angle of yaw. The resulting s/v value is used to obtain the value of h' for the relevant funnel design at 30 deg. angle of yaw. To comply with Rule 2, the value of h' must not fall lower than -2; if h' is negative, its numerical value must not be greater than H. If this Rule is not complied with by the value of s determined by Rule 1, then the minimum gas speed must be increased until both Rules are complied with, or an alternative design of funnel top must be adopted.

Note that in many cases the cross-section of an uptake will have to be reduced at the top to give the calculated discharge velocity. The final portion should preferably take the form of a converging taper, followed by a parallel section of at least one diameter in length. This parallel section gives time for any smuts entrained by the gases to accelerate with them and to have a better chance of clearing the ship. Every effort should be made, however, to prevent smuts from reaching this point at all.

FUNNEL PERFORMANCE

A further opportunity to apply the design rules arose when difficulty was experienced on a 50,000 d.w.t. tanker due to entry of fumes to the ventilation intakes and B.S.R.A. was asked to recommend modifications to the funnel design.

The existing funnel had a domed top with a circular uptake angled back at 15 deg. to the vertical and terminating flush with the top of the dome. Examination showed that the worst conditions for the intake of fumes occurred when the relative wind direction was at between 15 and 20 deg. off the ahead position. Calculations made, using the earlier papers[1,2] confirmed that prediction and service experience were in good agreement. Further calculations showed that to correct the defect, without any funnel modifications, a gas velocity 2½ times the existing value would have to be used; from the point of view of back pressure on the boilers this was out of the question.

A combination of modifications was therefore carried out, each one giving a proportion of the required improvement; the data in reference 1 were used to calculate the effects of these modifications:

a) the section of the uptake was changed at the upper end from circular to rectangular with an aspect ratio of 3;
b) the angle of the upper end of the uptake was increased from 15 to 22½ deg. to the vertical;
c) the height of the uptake was increased by 3ft., which gave a projection of this amount above the existing dome, which was unaltered except to modify the aperture to suit the rectangular section of the uptake;
d) the gas velocity was increased by approximately 56 per cent, by reducing the outlet area by 36 per cent. This restriction at the funnel top could be tolerated without upsetting conditions in the combustion chambers.

These modifications appear to have been entirely successful in overcoming the trouble reported.

Note on Power Required to Discharge Gases

It is obvious that at the higher gas speeds, an appreciable fraction of the fan pressure is absorbed in creating the necessary velocity in the effluent gas stream. This pressure loss may be calculated readily from the velocity and temperature of the gases at the outlet. If the velocity is s ft./sec. and the absolute temperature is T deg. F., then

$$\text{pressure loss} = \frac{520}{T}\left(\frac{s}{64\cdot5}\right)^2 \text{ in. w.g. approximately}$$

An allowance has been made for the fact that the gases are denser than air at normal temperature and pressure.

If an existing outlet is to be reduced in area at the top to accelerate the gases, the additional pressure required will be the difference between the pressure loss calculated for the higher velocity and that for the original velocity.

When a bend is included near the top of the uptake, the velocity across the discharge is not uniform, as has already been noted. It is not possible to make an exact estimate of the losses under these circumstances, but for practical purposes it should be satisfactory to take the total losses in the final section of the uptake as being the sum of twice the bend loss (as obtained from published data for a bend followed by a length of duct) and the loss due to the outlet velocity, calculated as shown on an average velocity basis.

It may be repeated here that any bend, change of shape, or obstruction that seriously upsets the uniformity of the flow at discharge can result in very considerable pressure loss, owing to the production of excessive local velocity at this point.

THE SCALE EFFECT

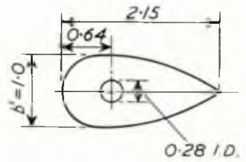
In the Appendix to reference 1 it was concluded that the buoyant effect on the plume of the relatively high gas temperature might be neglected, which implies acceptance of the fact that the model performance is a little on the pessimistic side at the moderate and high gas speeds with which this investigation is mainly concerned. The subject gave rise to some discussion during the presentation of reference 1 so it was decided to investigate it a little more fully.

The tests have been based solely on the criterion of equal velocity ratio s/v on model and full scale. This procedure was justified by the fact that years of experience, by many experimenters, had shown that it did produce data from which reliable full-scale prediction could be made.

Actually, the criterion, apart from buoyancy, was equality of $\rho_g s^2/\rho_a v^2$, the ratio of the momentum per second in the efflux gases to that in the wind. If wind-tunnel tests were made at the full-scale value of s/v instead of $\rho_g s^2/\rho_a v^2$ the results would be too favourable as a rule*, in other words, the wind tunnel would show the smoke plume as higher than it would be on the ship. But, on the other hand, there was the favourable buoyancy effect on the ship, which was not present in the wind tunnel if, as was usual, the efflux was not heated. It had been found by experience that these two effects pretty well cancelled out, since wind-tunnel tests made at the full-scale value of s/v did give results that accorded satisfactorily with the full-scale behaviour of the smoke plume. The same conclusion had been reached by another experimenter reporting recently in a German journal[3].

The buoyancy effect is approximately proportional to the diameter of the uptake and also to the temperature difference between the effluent gas and the ambient air, and is inversely proportional to the square of the relative wind velocity. For a typical modern vessel these relationships are satisfied on the model scale of the tests when the wind-tunnel speed is reduced to 8ft./sec., so it was decided to carry out a few tests as a check on the effect of buoyancy on the smoke plume.

An electric heater inserted in the supply pipe to the funnel raised the temperature of the effluent gas to 350 deg. F., which is representative of efficient boiler practice. Tests were carried out at this tunnel speed and gas temperature for a range of s/v ratios. The results are shown in Fig. 20 alongside those for the cold plume. Over most of the range, the hot plume did rise above the cold plume, but only by a height equivalent to not more than about $\frac{1}{4}$ b'. This confirms the earlier assumption, that the buoyancy effect may be neglected in most cases, to give results which are a little on the pessimistic side when referred to the full-scale ship.



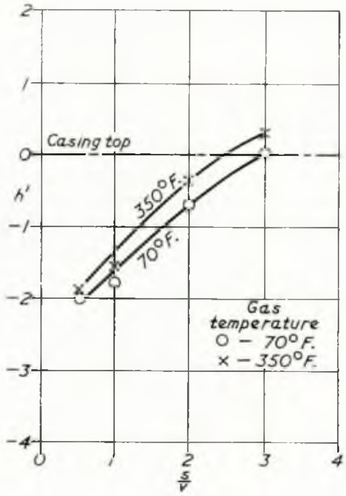


FIG. 20—*Casing A uptake forward—e = 0·475—Influence of gas temperature on plume height*

Most modern ships have relatively small uptakes, with fairly low gas temperatures, and relative wind velocities tend to be higher, owing to increased ship speeds. These factors, particularly the lower gas temperatures, tend to minimise the buoyancy effect. At the other end of the scale there is a class of vessel, (now diminishing in numbers) which has a low effluent-gas speed, requiring a relatively large uptake section, and a very high gas temperature, while the low speed of the vessel results in a low relative wind velocity. Since each of the three factors accentuates the buoyancy effect, the design rules should not be applied to ships of this class.

ACKNOWLEDGMENT

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* ρ_a and ρ_g represent respectively the density of the ambient air and the funnel gases.

Aspects of the Mechanical Propulsion of Tugs*

C. R. FORD, M.Sc., A.M.I.Mech.E. (Graduate)†

GENERAL

The extremely poor manœuvrability of large vessels, particularly at low speeds, renders tugs vital to all docking operations throughout the world in all but the most favourable tide and wind conditions. They perform many other functions including ocean towing, salvage and rescue work, harbour, coastal, river and canal towing. Each of these demands different performance characteristics and the resulting machinery arrangements are legion.

The layout of Southampton Docks, the extensive oil refinery facilities at Fawley, the large area covered by the port and the prevailing south-west wind produce a combination of tug requirements peculiar to this port. These require that both ahead and astern static pulls are of great importance and that the time spent travelling free between the various parts of the docks, particularly to and from Fawley, should be a minimum.

The basic problem to be met in the design of any tug machinery is to provide ample power for both ahead and astern towing and the free speed conditions. The old reciprocating steam engine was ideal for this purpose and held its own long after the Diesel engine was adopted for other vessels. The large size and poor efficiency of reciprocating steam engine installations eventually rendered them obsolete some 10 to 15 years ago and they have now been replaced almost universally by the Diesel engine. The Diesel engine has many advantages so far as economics are concerned but three inherent disadvantages: a flat torque characteristic, high idling speed and poor manoeuvrability. Many of the devices to be discussed in this paper are ingenious methods of circumventing these shortcomings.

For the purposes of this paper, it is fortunate that the two most modern classes of tug in use in this port are very different in design and thus provide a wide basis upon which this paper has been founded. The tugs of the Alexandra Company fleet are single-screw whilst those of the Red Funnel Company are twin-screw. Their leading particulars are given in Table I.

TABLE I.

	Single-screw	Twin-screw
	Alexandra Towing Co., Ltd.	Red Funnel Steamers Ltd.
	M.T. *Romsey* and class	M.T. *Dunnose* and class
Length, b.p.	93ft. 0in.	100ft. 0in.
Breadth, moulded	25ft. 6in.	27ft. 0in.
Depth, moulded	12ft. 6in.	13ft. 3in.
Draught, aft	13ft. 6in.	13ft. 6in.
Displacement	365 tons	407 tons
Main engines—maker	Crossley	Crossley
Type	HRP 8/42	HGN 6/50
b.h.p.	1,200	700 × 2
r.p.m.	420	500
Gearing	Plain reduction	Reverse reduction
	direct reversing	uni-directional

* Paper presented to the Southern Joint Branch, R.I.N.A. and I.Mar.E. on Tuesday, 16th March 1965, at Southampton.

† Chief Technical Engineer, John I. Thornycroft and Co. Ltd.

Single-screw Tug Propeller Arrangements

Fig. 1 shows a comparison of three propeller arrangements suitable for a single-screw tug similar to the Alexandra Towing Company's tug *Romsey*.

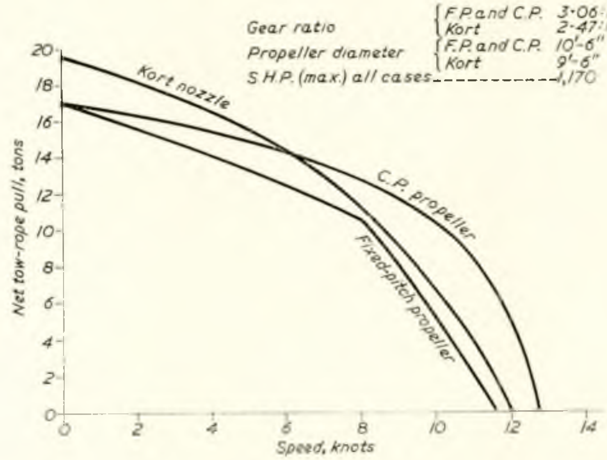


FIG. 1—*Towing performance of single-screw tugs with various propeller arrangements—Vessels powered by Crossley HRP 8/42 engines*

The production of this diagram and the one to follow should be viewed strictly subject to the conditions stated. They are related to particular hull forms, machinery, propeller diameters and various other fixed parameters. No claim is made that these curves are necessarily the optimum but they represent, on as fair a basis as possible, the performance that may reasonably be expected from these different propeller arrangements when fitted to these vessels. This has been done in order to obtain a valid comparison of the various modes of propulsion available for local tugs. No account has been taken of using engine overload conditions at the bollard as is sometimes done.

The fixed-pitch propeller curve has been calculated knowing the trial bollard pull, free speed and engine performance of the *Romsey*. Horsepower is constant and propeller r.p.m. rising from 0 to 8 knots; thereafter propeller revolutions are constant and horsepower decreasing. This curve is governed by the designed performance of the turbo-charged two-stroke engine fitted to this vessel. This propeller has been designed for the bollard pull condition.

The variable-pitch propeller curve assumes constant horsepower and revolutions throughout, at the rated conditions, whilst the change in pitch between the bollard and free speed condition is assumed to have negligible effect on performance.

The Kort nozzle curve assumes the same engine performance, but in order to achieve comparable results the gear ratio has been altered and the propeller designed for six knots. Thus further increase in bollard pull could be obtained at the expense of some loss in free speed. This Kort nozzle curve does not affect the machinery inboard to any large extent and is only shown for completeness.

Twin-screw Tug Propeller Arrangements

Fig. 2 shows the towing performance of the single-speed Red Funnel tugs *Dunnose* and *Thorness*, the two-speed tug now in course of construction for the same company, and a hypothetical case of a similar tug fitted with controllable-pitch propellers.

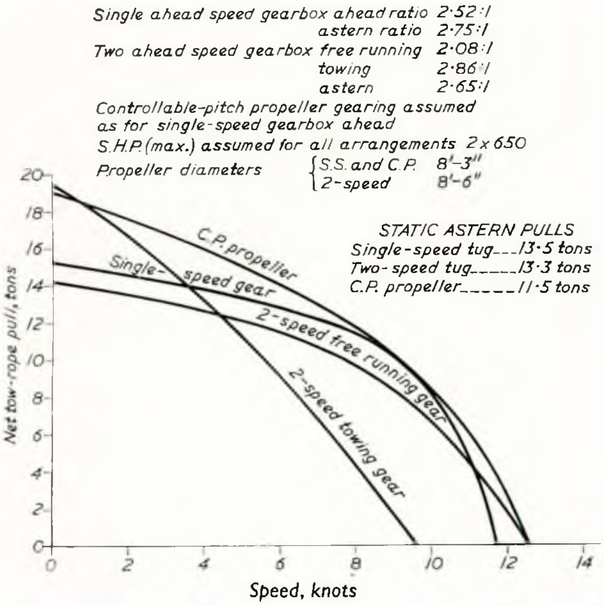


FIG. 2—*Towing performance of twin-screw tugs with various propeller arrangements—Vessels powered by Crossley HGN 6/50 engines*

The fixed-pitch propeller arrangements all show clearly the various compromises necessary to approach the required combination of ahead pull and free speed.

The single-speed tugs are optimum in the region of nine knots owing to the emphasis placed on free speed. This optimum point could have been chosen to occur at a lower speed in which case a higher pull and lower free speed would result. Similarly, an increase of free speed could have been obtained at the expense of pull.

The advantage of the two-speed gearbox is clearly shown at the bollard where an additional 4·2 tons pull is obtained and when running free where 0·7 knot is gained. It is also clear that, in the range $3\frac{1}{2}$-11 knots, a penalty is paid in tow-rope pull, but fortunately this speed range is not frequently used for towing in this port. In harbours where this towing speed range is important, and multi-speed gears required, this diagram shows that the three-speed gearbox would be advisable.

The controllable-pitch propeller shows to distinct advantage throughout the whole range, as it did in the previous figure. In the ahead direction it approaches the optimum, since the pitch can be adjusted to absorb full power at any operating condition. Since the pitch angle of these blades is varying from a high value at the root to a small value at the tip, changing the pitch adds or subtracts a constant angle at all radii. Thus efficiency is lost at all points away from the design condition. This effect is particularly severe when going astern and accounts for the poor astern performance of this propeller compared with the fixed-pitch alternative.

Having very briefly highlighted the tug owners' and propeller designers' problems in selecting suitable propeller arrangements for a vessel, consideration will now be given first to the major machinery arrangements available for each propeller system and then to the machinery components and controls.

MACHINERY ARRANGEMENTS

The simplest of all machinery arrangements is the direct-reversing, direct-coupled engine. This has largely been superseded by the very common direct-reversing geared arrangement for many reasons including:

1) lower cost;
2) reduced engine size;
3) free choice of propeller r.p.m.;
4) reliability of gearing;
5) reduced starting air requirements.

The main disadvantages of this arrangement are high wear due to the reversing operation, relatively large starting air requirements and complicated reversing mechanisms on four-stroke engines. These reasons and particularly the last have led to the adoption of reverse-reduction geared systems utilizing unidirectional engines.

This system is predominant on most twin-screw British tugs since it eliminates the above disadvantages and is very suited to the high degree of manœuvrability demanded from such vessels. Its acceptance has been due to the development of simple and reliable clutches.

The gearboxes used on these arrangements permit a free choice of both ahead and astern r.p.m. Multi-speed gearboxes are also available to obtain the effects shown in Fig. 2.

A growing number of single-screw tugs, particularly the more powerful ocean-going types, are using multi-engine arrangements. In these, two or more Diesel engines are geared through flexible or hydraulic couplings into a multi-input single-output reverse-reduction gearbox. This enables a higher power to be produced in the same engine room space than would be possible with a single engine and provides some degree of standby in the event of failure of one engine. Engine disconnecting clutches are always provided. The major advantage of this arrangement is that it permits tugs operating over wide areas, e.g. ocean-going tugs, to cruise at a low free speed for long periods, without detriment to the engines through light loads, and with improved economy. It should be appreciated that with a fixed-pitch propeller it is not possible to shut one engine down and use the others at full power. If this facility is required a controllable-pitch propeller should be fitted.

Probably the most rapidly growing class of tugs at the moment are those incorporating controllable-pitch propellers. These are particularly common on the Continent but have not yet found great favour amongst British owners due to early difficulties associated with the mechanism outboard of the hull. This mechanism can only be attended to during slipping of the vessel. It must however be conceded that great advances have been made with this type of propeller over the last ten years and their record during this period is sufficient recommendation of their current reliability. The usual machinery arrangement selected for use with controllable-pitch propellers is indicated in Fig. 3. This shows a unidirectional Diesel engine driving through an isolating clutch and plain reduction gearbox. The clutch disengages whenever the engine dwells at idling speed thus enabling the propeller to cease rotating during periods of rope handling.

Despite the slight drop in astern pull which is experienced with controllable-pitch propellers due to the incorrect pitching of the propeller when operating in this direction, it is considered

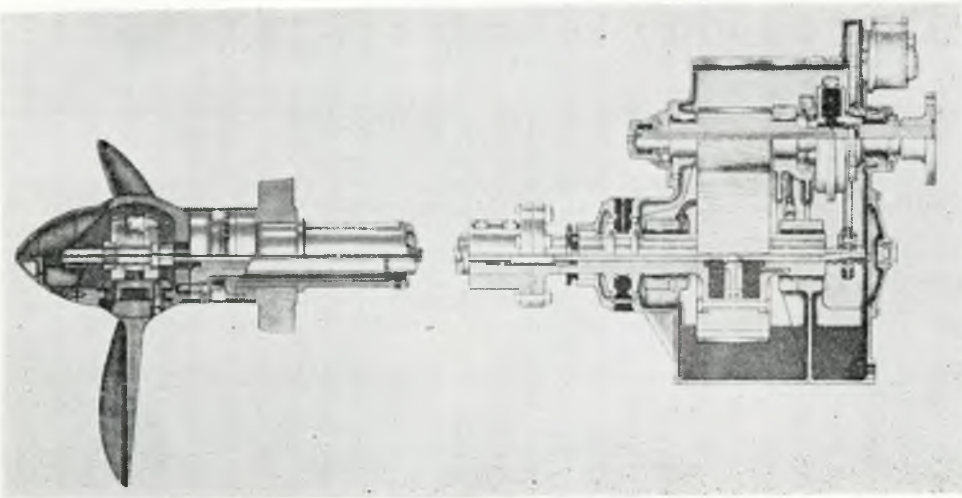


FIG. 3—*C.P. propeller drive via an isolating clutch and plain reduction gearbox*

that this type of machinery will continue to grow in numbers due to the high ahead performance, excellent manœuvrability and simplicity of bridge control.

In addition to these propulsion arrangements both Diesel-electric and Voith-Schneider propeller systems are occasionally encountered.

The Diesel-electric transmission has excellent manoeuvrability, control and versatility but a high cost and poor efficiency—less than 85 per cent. Thus it is only used on tugs for special purposes i.e. when large pumping equipment or winches are also installed.

The Voith-Schneider water tractor is a most interesting form of tug. Some small tugs of this type have been operating at Middlesbrough for some years now and the Port of London Authority have just accepted the first of four larger tugs into service. It is hoped that after some service experience the P.L.A. will publish their performance details. These tugs have the Voith-Schneider propeller fitted forward of amidships and the towing hook aft. This arrangement produces an extremely manœuvrable tug which is inherently stable when towing since it always aligns itself with the tow. The tow rope pull is not as high as with conventional propellers.

MAIN MACHINERY COMPONENTS

Gearing

Gearing as normally fitted to tugs allows the Diesel engine to run at speeds between 250 and 750 r.p.m. with a consequent reduction in size, weight and generally in overall cost. A very small increase in fuel consumption is incurred. The reduction in engine size makes high powered twin-screw or multi-engined single-screw arrangements possible with their attendant increased flexibility, reliability and ease of overhaul.

Where maximum bollard pull and free speed are simultaneously important, but towing ability at intermediate speeds is of less importance, then the two-speed gearbox is now being adopted. The complication of these gears together with the control system has to be weighed against the merits and demerits of the c.p. propeller outlined earlier. With multi-speed gearboxes the efficiency and lubricating oil pump power are factors which must also be considered. The efficiency of a typical two-speed ahead, one-speed astern gearbox is of the order of 93 per cent. This represents a large loss of engine power.

Occasionally a third speed is added to these gearboxes to provide either a very low propeller speed when taking the tow, thus avoiding tow rope snatch, or more commonly an intermediate speed to improve the towing performance at 3 to 6 knots.

The gearbox itself represents a good example of the steady progress in component design towards smaller and lighter units. Naval requirements have produced a much greater understanding of the problems involved and in the latest naval applications, using case-hardened and ground gears, K factors in continuous use have reached as high as 460 with very satisfactory results. To achieve this a large amount of research has been necessary into gear materials, gear cutting machines and methods, lubrication, bearings etc. Not all of this has yet been applied to the commercial quality gearing in use in tugs owing mainly to the additional costs involved—K factors remain below the Lloyd's Register limit of 110 for hobbed and shaved gears. The industry has however drawn steadily on this research to improve the quality and reliability of commercial gears to a point where old problems such as pitting and scuffing arising from poor material selection or incorrect gear cutting are now very unusual. Perhaps the most immediate results as applied to tugs are the weight savings from fabricated gearboxes, the more accurately cut gears, the use of E.P. oils, reduction in bearing size and the use of steel backed bearings. It can be anticipated that more and more of this knowledge will enter into normal good commercial practice in the next few years.

Representative gearbox designs by two different manufacturers are included to illustrate some of the above principles.

Fig. 4 illustrates a single-input, single-output tug gearbox having two ahead speeds and one astern. It transmits 660 b.h.p. at 750 r.p.m. with a 3:1 and 3·45:1 reduction. The input shafting is on the left of the diagram and carries the three input

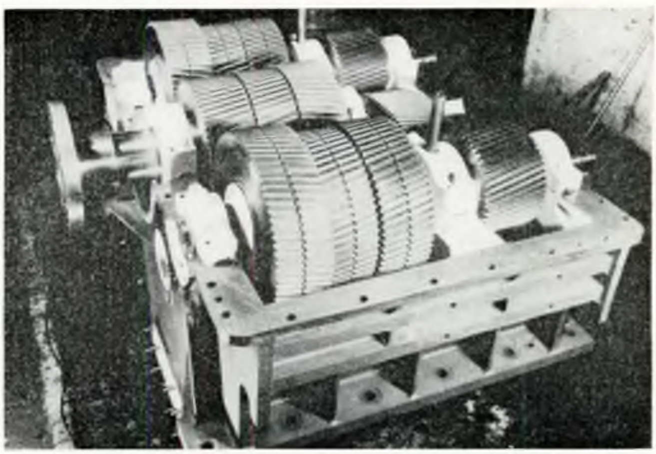


FIG. 4—*Single-input, single-output tug gearbox*

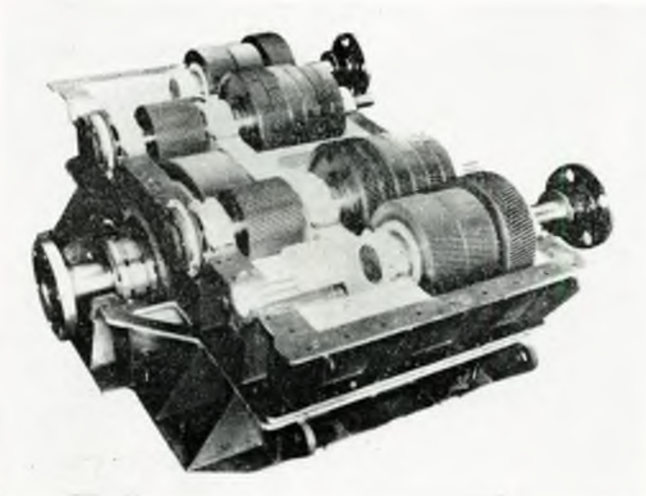


FIG. 5—*Twin-input, single-output gearbox as used in multi-engined arrangements*

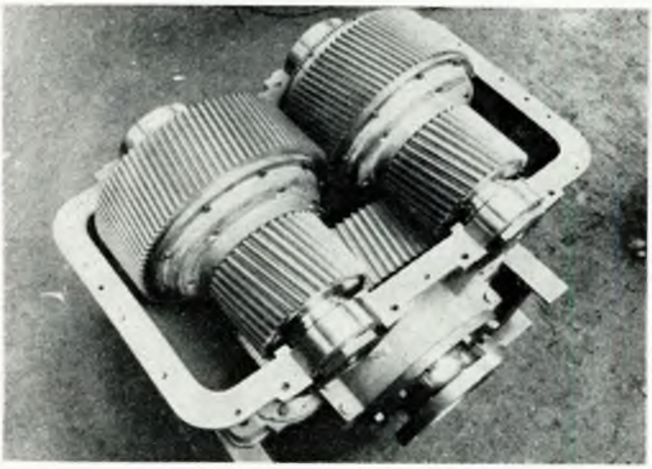


FIG. 6—*Continental design for a reverse-reduction gearbox*

pinions. The outer two are the ahead gears and the centre one is the astern. The ahead pinions are in constant mesh with gear wheels containing oil-operated clutches and supported on splined clutch shafts. These shafts which are duplicated port and starboard, also carry the astern gear wheels and clutches which are driven through idlers from the input astern pinion. The idlers are out of sight below the other shafts. The final reduction pinions are mounted on the clutch shafts and are in constant mesh with the final wheel which is carried on the output shaft. Oil input to the clutches is achieved through the grooves shown at the aft end of the clutch shafts. On designs incorporating disc brakes, these grooved oil inlets are moved to the forward end of the shaft and the brakes fitted at the aft end adjacent to the output pinion and wheel. It is of interest to note the thicknesses of the casing of this cast gearbox and to compare it with Fig. 5 which shows a much larger fabricated gearcase. This gearbox is a twin-input, single-output unit as used for multi-engined arrangements.

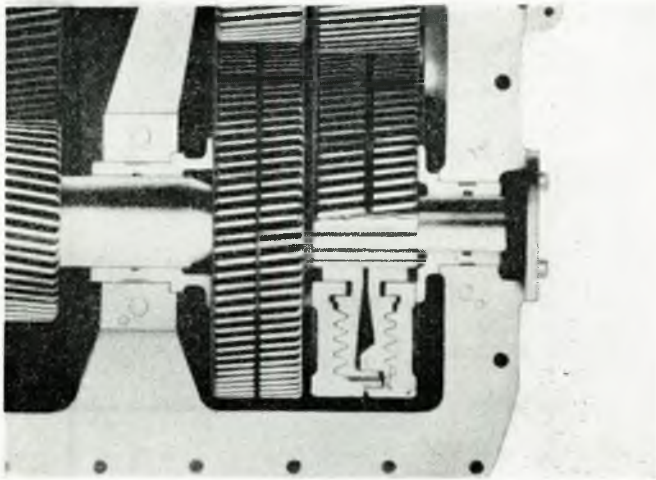


FIG. 7—*Principle of a clutch*

Fig. 6 is a Continental design for a reverse-reduction gearbox. This particular box is suitable for one engine of 750 b.h.p. at 435 r.p.m. with $2\frac{1}{2}:1$ reduction. Apart from the usual gearing arrangement the complete use of ball and roller bearings can be seen, which have been most satisfactory in service.

Clutches

The growth of reverse-reduction geared tugs can be attributed almost entirely to the successful development of reliable, small clutches. There are several makes available operating either hydraulically, pneumatically or electrically, and for illustration two very different hydraulic clutches are included.

Fig. 7 shows the principle of a clutch which consists of two steel outer members screwed together and locked, but free

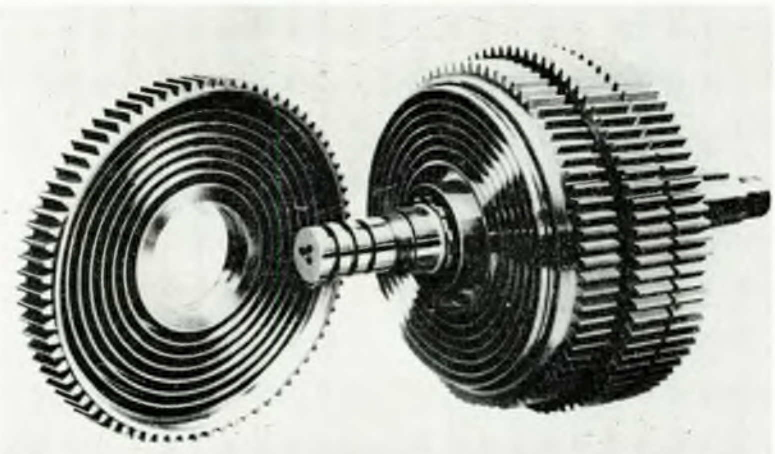


FIG. 8—*Close-up of the inner members of the clutch shown in Fig. 7*

to rotate on the hubs of the inner members. The two bronze inner members are fitted to splines on the clutch shafts so that they can transmit the torque, but are free to slide axially. The outer member also carries the first reduction gear teeth which are cut on the outer periphery. With the control cock in the stop or neutral position the inner members are held together by oil under pressure which is fed along a port in the shaft into the annulus between the inner and outer members. When the control cock is moved to the appropriate "engaged" position, oil is fed along another port into the space between the inner members. Thus so long as the input shaft is rotating, oil pressure maintains the concentric, conical mating surfaces of the clutch members positively engaged or disengaged.

Fig. 8 shows a close-up of the inner members of the same clutch, clearly illustrating the principle of concentric conical rings.

An alternative form of clutch, this time the more usual friction plate type, is illustrated in Fig. 9. This is the type used in the reverse-reduction gearbox illustrated in Fig. 6. The clutch plates are of cast iron, being plain discs with a spiral and four radial grooves on the surface to dispense and spread the oil between the plate surfaces during engagement and disengagement. The clutch is hydraulically-operated by oil entering the piston annulus via the shaft ports. When disengaged, the small springs separate the clutch plates thus preventing any wear.

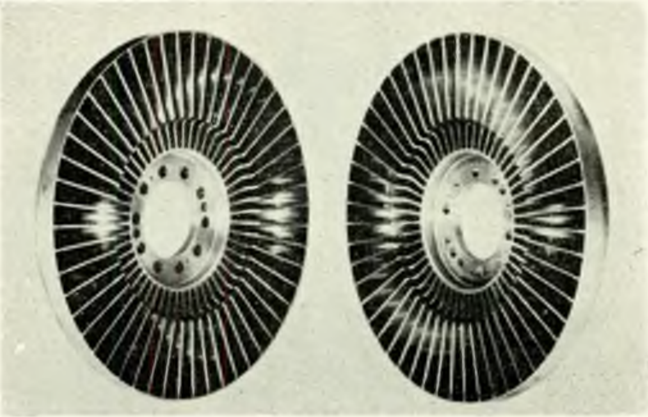


FIG. 10—*The impeller and runner of a fluid coupling*

Couplings

There are basically two types of coupling in use on tugs, the flexible rubber type coupling and the hydraulic slip coupling.

The former introduces low stiffness and flexibility between

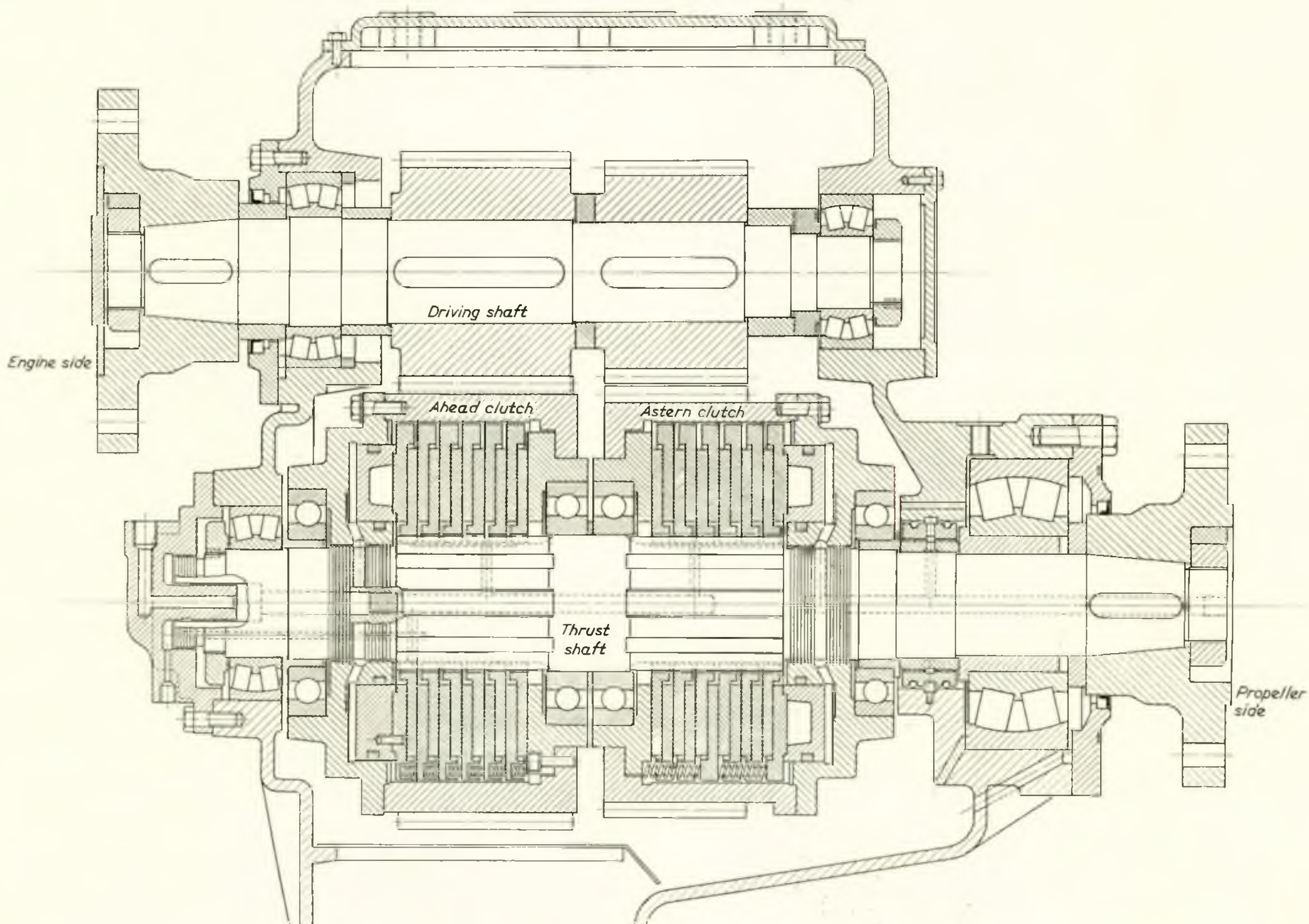


FIG. 9—*Brevo reverse-reduction gearbox*

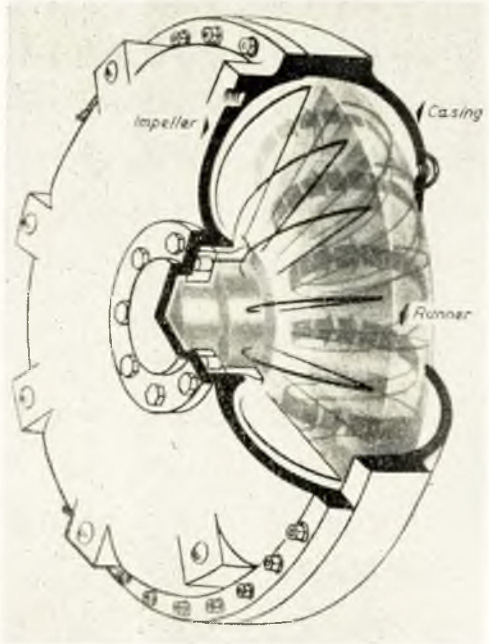


FIG. 11—*Vortex flow in fluid between impeller vanes*

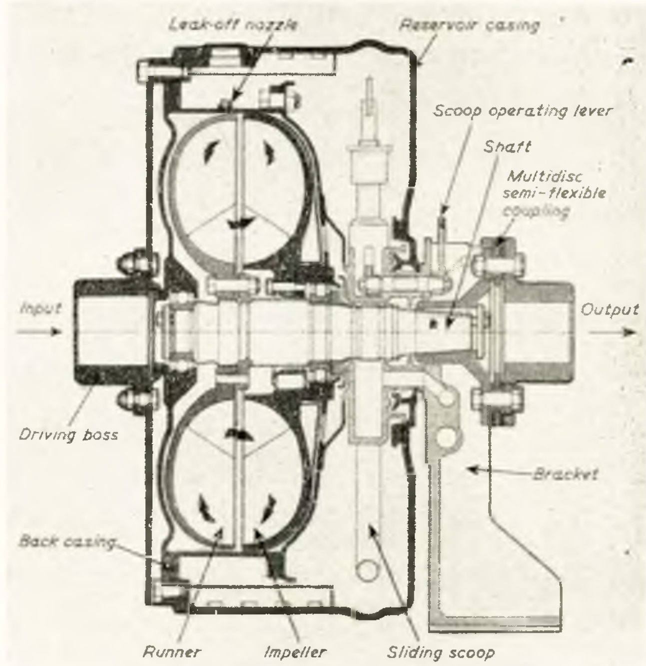


FIG. 12—*Sliding-scoop coupling*

engine and gearbox by means of rubber. This coupling is used to alter the torsional characteristics of the system and to allow a minor degree of misalignment. Its theory is well understood and amply covered elsewhere.

The hydraulic slip coupling can serve additional purposes to those just mentioned, stemming from the fact that there is no mechanical contact between the two halves. Thus torsionally the propeller shaft output and engine input sides of the coupling are completely separate and by variation of the quantity of fluid an infinitely variable "gear" effect is obtained. Essentially the fluid coupling consists of two almost identical components, the impeller and runner (see Fig. 10). These contain a large number of accurately produced, well balanced, straight radial vanes. As the impeller, which is attached to the input shaft, is rotated, the fluid between the vanes (see Fig. 11) is subject to centrifugal force and a vortex flow is initiated. It flows from the inside to the outside diameter of the impeller and returns to the centre via the runner, thus transmitting torque to the runner or output shaft. It is very suited to marine use since its basic equation

$$\text{h.p.} = KD^5 \left(\frac{N}{100}\right)^3$$

is similar in form to that of a fixed-blade propeller. K is a constant depending on slip. If slip is held constant then the h.p. which can be transmitted is proportional to $(\text{speed})^3$: in practice the slip is set for about 3 per cent.

These fluid couplings are not commonly encountered on the size of tug which is the main consideration of this paper, although a considerable number exists. These utilize the scoop-controlled type of coupling (see Fig. 12) in which a sliding scoop tube operates to control the fluid level in the casing and thus the slip. This type of coupling permits propeller r.p.m. down to about one-tenth of full r.p.m. to be used, thus giving extremely smooth manœuvring characteristics. The normal method of operation is to vary the engine speed between full and half speed and below this to vary the coupling slip.

On larger ocean-going tugs and particularly on multi-engined, single-screw tugs, these slip couplings are very common and in many cases a necessity. In this instance they serve to give flexibility in the following ways:

i) elimination of torsionals;
ii) speed control as previously described;
iii) the complete disconnexion of one engine from the drive.

When speed control is not required, the sliding scoop mechanism is removed, and if it is required to retain the ability to disconnect an engine a quick emptying ring can be fitted.

The major problem associated with installing these couplings is concerned with their diameter. This tends to make them very large when used with slow to medium r.p.m. engines. Installation problems are eased when higher engine r.p.m. are selected.

Diesel Engines—General

The features of the various well tried and reliable Diesel engines in use within this port are well known and consequently little needs to be said regarding these. Attention will therefore be given to recent developments.

In the past ten years engine makers have been both designing and producing new engines and improving and up-rating their existing designs. The results of these changes have been the production of more power per unit volume and per unit weight of engine, a reduction in price per horsepower and maintained or improved reliability.

Four-stroke Marine Diesel Engines

The up-rating of engines in use on tugs has been most spectacular within the four-stroke engine range. The application of turbocharging to these engines has become universal and few manufacturers now offer naturally aspirated engines unless specifically requested. Initially turbocharging was achieved simply by adding the turbocharger to existing naturally aspirated engines. This enabled the brake mean effective pressure to be raised from 90 to 120lb./sq. in. Subsequent addition of intercoolers permitted a further increase of b.m.e.p. to about 150 lb./sq. in. Thus an increase of some 50-60 per cent in power was reached without much alteration to the original design. This is the state of many well-known engines today. This rating of about 150lb./sq. in., appears to be approximately the limit of development of these engines due to factors such as cylinder head, liner, piston and particularly crankshaft design.

Within the last two years new engines have been introduced into the tug market designed specifically for high ratings. Fig. 13 illustrates a Ruston ATCM engine which is typical of these. These engines are available in the pressure-charged and inter-

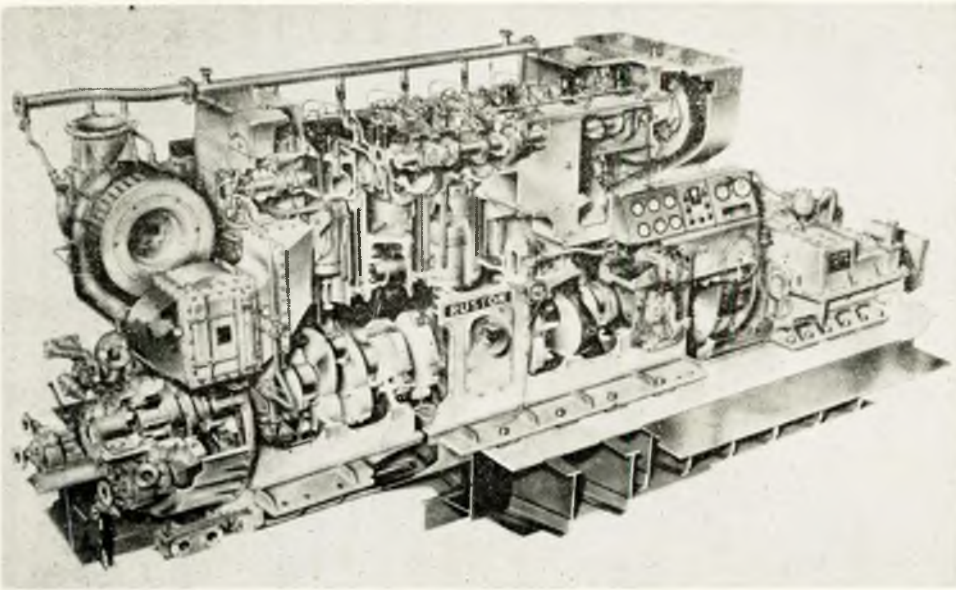


FIG. 13—*Ruston ATCM engine*

cooled versions only, and at present are offered with a b.m.e.p of 200lb./sq. in. This is 100 per cent in excess of naturally aspirated power ratings. Development of these engines within the engine maker's works has reached a rating of 250lb./sq. in. and figures of 300 and 400lb./sq. in. are said to be feasible. It is, however, considered that the rate of introduction of these ratings into service will not be as rapid as previous increases since several component parts of the engine are reaching the limits of strength of existing materials simultaneously.

The achievement of a 100 per cent increase in power has involved the application of many new techniques to cater for the increased loads and temperatures. As a result these engines are more sophisticated than their predecessors and require a better understanding for the reliable achievement of the high powers of which they are capable.

In designing completely new engines it has, of course, been possible to greatly improve the limiting features of earlier standard ranges. This is clearly shown by the crankshaft problem, where new designs are shorter and stiffer, and forging processes produce a stronger and sounder crankshaft; similarly crankcases are stiffer and provide a more rigid housing for the highly loaded main bearings.

The common white metal on bronze bearing has proved quite inadequate for high powered Diesel engine main and large end bearings and progress has advanced through steel-backed copper-lead-indium flashed bearings to aluminium-tin steel-backed bearings. The load-carrying capacity of the latter is some $2\frac{1}{2}$ times greater than white metal.

The most important limitation to further up-rating is that due to thermal stress failure, to which cylinder heads, pistons and liners are prone. Fig. 14 shows this unusual form of failure between the exhaust and injector ports of a flake graphite cylinder head of a medium-bore Diesel engine. This is the hottest zone of the cylinder head. Two smaller cracks are also forming on the opposite lip of the injector port and are extending towards the inlet port.

The cracks do not form during the running of the engine. since it can be seen that they occur at hot spots where the running stress is compressive. It is therefore apparent that residual or thermal stress failures occur as an after-effect of running conditions. The simplified stress system is shown in Fig. 15. The curve *a-b* represents the development of a compressive stress during the first running of a new cylinder head. Continuous running under compressive stress at elevated temperature induces creep at the combustion face *b-c*. On cooling when the engine is stopped the lowered compressive stress is converted to a residual tensile stress *c-d* instead of reverting to zero. Subse-

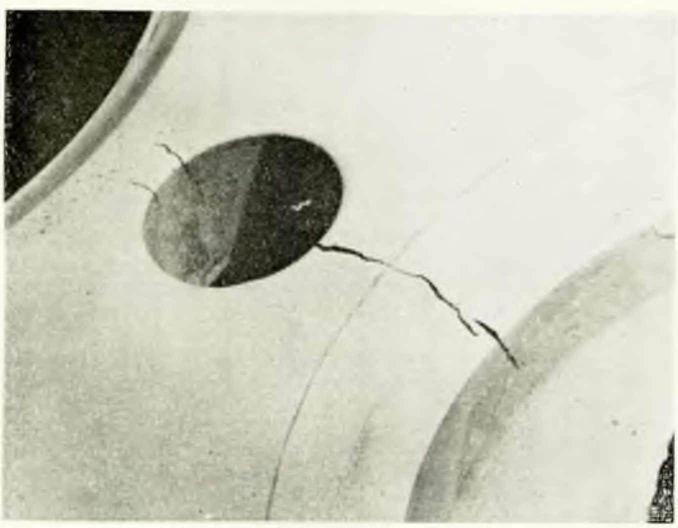


FIG. 14—*Thermal stress failure between the exhaust and injector ports of a flake graphite cylinder head*

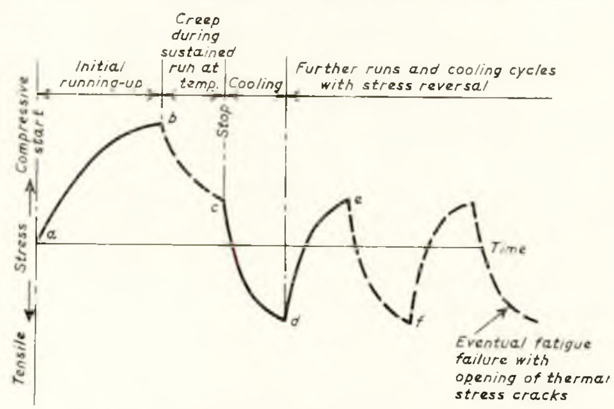


FIG. 15—*Thermal stress cycle in a cylinder head*

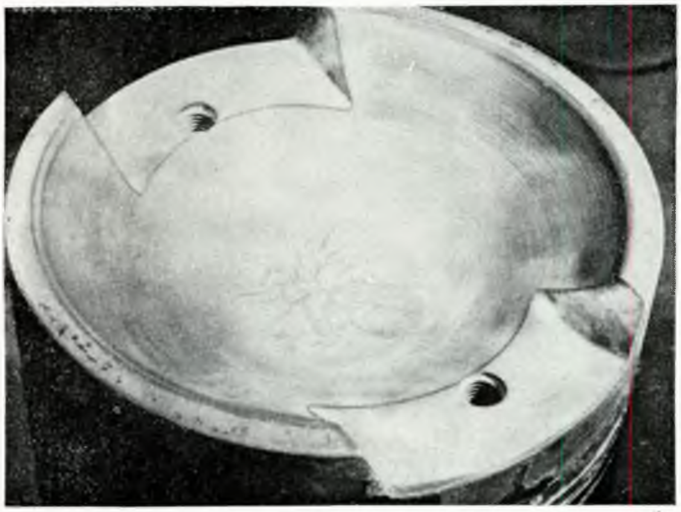


FIG. 16—*Thermal stress failure in a piston*

quent starting and stopping will result in a tensile fatigue failure if the stress range *d-e* is too high.

Pistons are prone to the same form of failure, as illustrated in Fig. 16, and in spite of their simpler geometry are more complex to design than cylinder heads. This is due to the need to use greater material thicknesses to withstand pressure loading of the large unsupported piston crown and the limitation on the position and temperature (230 deg. C.) of the top piston ring to avoid breakdown of lubricating oil and consequent ring sticking. Special attention is also necessary to position and design the gudgeon pin bosses to ensure that the minimum heat transfer takes place along this path away from the piston crown. The use of oil cooling within the space under the piston crown is universal in one form or another on these engines. This form of failure has been the subject of much research[1] and has resulted in the use of new materials, e.g., flake or nodular cast iron together with high heat transfer rates. To achieve the latter it is necessary to ensure:

1) minimum material thickness consistent with firing pressures;
2) increased velocity and quantity of cooling water particularly in way of "hot spots";
3) elimination of air in cooling water to avoid air locks in small water passages;
4) water treatment to avoid deposits.

Two-stroke Loop Scavenge Engines

The loop scavenge two-stroke engine has for many years held a very prominent position in tug work.

Fig. 17 illustrates clearly the low height and simplicity of this engine. The absence of valve gear makes this engine ideal for direct-reversing installations and the consequent low cylinder heads provide ample headroom for maintenance purposes which is always a limiting feature of tug engine rooms. This low height also helps to produce the low centre of gravity required for tug machinery installations.

Fig. 18 shows a section of the same loop scavenge two-stroke engine from which some impression of the relative ease of maintenance can be obtained. The inherent simplicity of the engine is also apparent.

This simplicity, which the operator appreciates so much, is in fact deceptive. The absence of a positive scavenging stroke presents the designer with many problems. Consequently this form of engine has not been up-rated so rapidly as its four-stroke rivals. Pressure-charged engines with a b.m.e.p. 30 to 40 per cent greater than the naturally aspirated engine have been marketed for some time but intercooled engines of high power outputs have not yet been sufficiently developed for general marine use.

The component developments of the two-stroke engine have followed broadly the same course as that described for the four-stroke engine, but the difficulties encountered have been very different in origin. The three major obstacles are low r.p.m., turbocharger matching and a variation of the thermal stress failure problem examined earlier.

The r.p.m. of the medium-bore loop scavenge engine is limited to 600. This is a direct consequence of the absence of a positive scavenging stroke. Intensive research has not been able to achieve efficient scavenging at speeds above this figure.

Exhaust-driven turbochargers are readily added to the loop scavenge engine but, due to starting and low speed running requirements, some form of scavenge pump must be retained. The scavenge air pressure must remain adequately above the exhaust pressure to achieve efficient scavenging and thus the inlet pressure to the exhaust-driven turbine is less than the outlet pressure from the compressor. The two-stroke engine also requires an excess of scavenge air above the swept volume of the cylinder which lowers the exhaust temperature. These two points create a very difficult situation for the turbocharger to

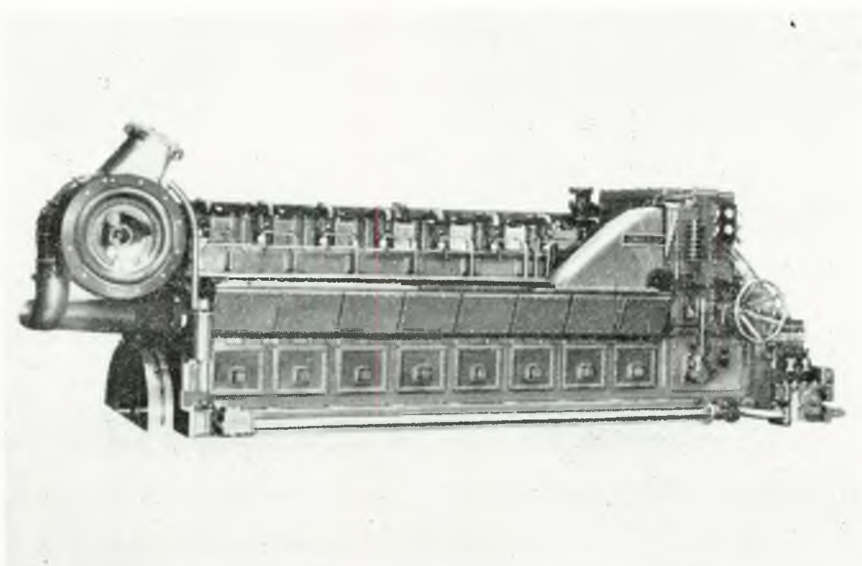
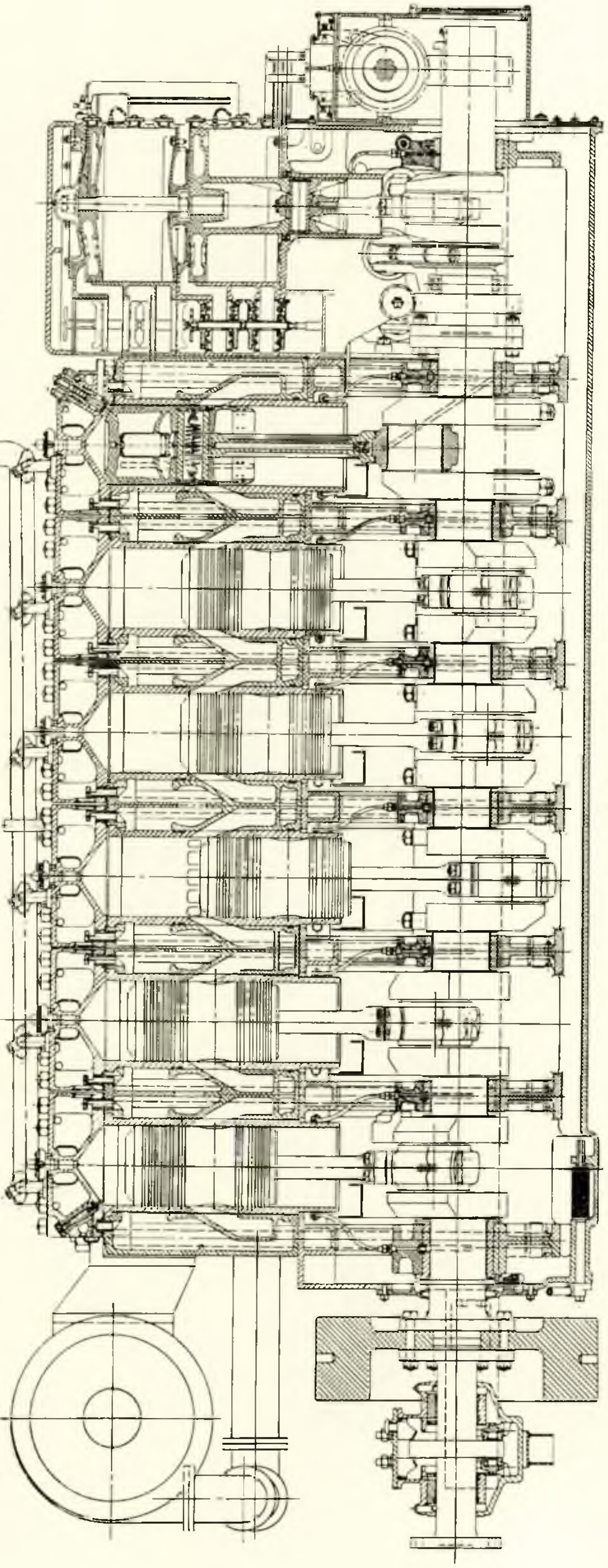
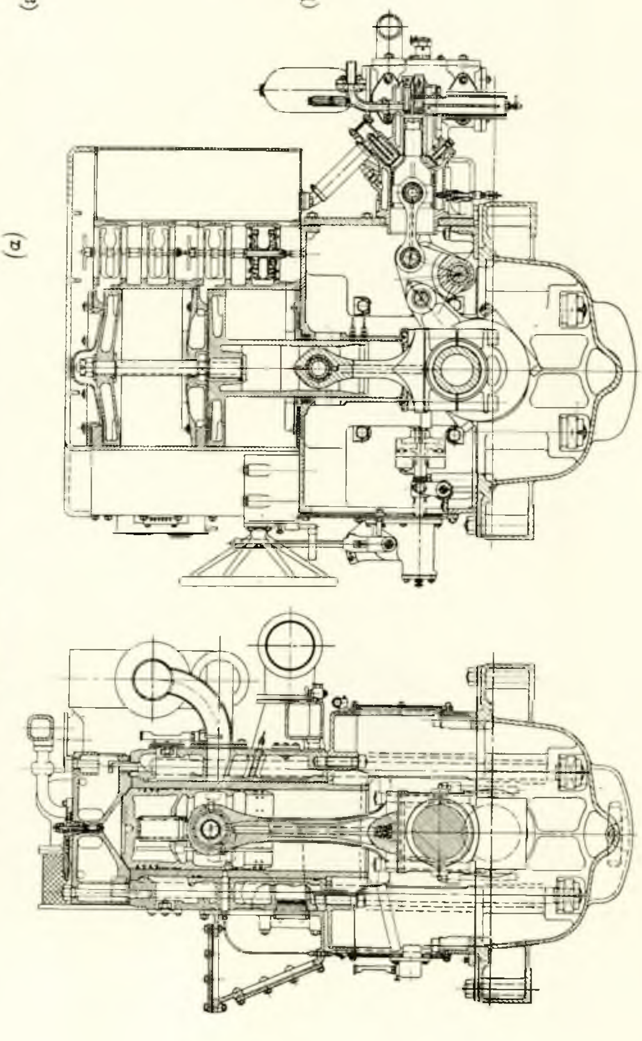


FIG. 17—*Two-stroke loop scavenge engine—Crossley HRP8*



(a) Longitudinal sectional arrangements



(b) Cross-sectional arrangements

FIG. 18—*Section through two-stroke loop scavenge engine—Crossley HRP6*

meet efficiently. To minimize this it is essential that minimum inlet and exhaust port pressure drops and excess air quantities are used.

The interval between successive firings in a cylinder is shorter in a two-stroke engine than in a four-stroke engine running at similar r.p.m. Thus the heat absorbed by the piston, liner and cylinder heads must be transmitted away to the cooling system more rapidly. Also, since there is no positive scavenging stroke, the working surfaces within the cylinder are not scoured and cooled as effectively by the fresh inlet scavenge air and remain at a higher temperature throughout the cycle. The onset of thermal stress failure therefore occurs at a lower b.m.e.p.

Further up-rating of the four-stroke engine is expected to become less rapid for the reasons stated. This will now provide an opportunity for the two-stroke loop scavenge engine makers to produce new designs. It can be presumed that these designs will be based on acceptance of a lower b.m.e.p. and r.p.m. than the four-stroke engine, whilst increasing power per unit volume and weight ratios and simultaneously reducing costs. It seems likely that the approach will include replacement of the reciprocating scavenge pumps by a smaller and lighter form of mechanical blower, increased use of fabrication and increased use of specialist manufacture of component parts. Providing these new designs retain the very attractive features of the existing loop scavenge engines—simplicity, low height and ease of maintenance—they are certain to remain popular with tug owners.

CONTROL SYSTEMS

Automatic Control Systems—General

Much has been said and written in the past few years about automation of ships, but in nearly all cases this has referred to large vessels, supertankers and the like. Small vessels tend to be overlooked and yet this is the class of vessel where the greatest strides have already been made.

In this instance the term automated ship is used in the usual marine engineer's interpretation, i.e. a ship on which no engine room watchkeeping is necessary, all controls being handled by the bridge or in an engineer's special control room.

Many tugs and other vessels have now been fitted with direct automatic bridge control of the main propulsion machinery and automatic control and shut-down systems exist in profusion for each separate item of auxiliary machinery usually fitted to tugs. The acceptance of these innovations is seen as a symptom of a rapidly growing concept that routine engine room functions can safely be performed by automatic means.

This concept receives added impetus from the reliability of modern machinery which can be achieved providing a strict maintenance routine is followed. This in turn leads to the adoption of preventive maintenance procedures which all manufacturers recommend. Preventive maintenance, and the initial installation of machinery in the shipyard, must be carried out under strict conditions of cleanliness. This is necessitated at all times due to the severe damage to machinery which can occur by the ingress of dirt to lubricating oil, fuel, fresh or salt water systems. Thus the maintenance requirements of modern engines now demand a high skill, combined with a thorough and up-to-the minute knowledge of the state of the machinery.

The only logical outcome of the adoption of both automatic controls and preventive maintenance on short voyage craft, especially harbour tugs, is to operate without trained engineers on board.

One such tug already exists on the Manchester Ship Canal and the author's company has already completed a design to the requirements of a tug owner based on a total crew of three. In this instance one of the deck crew would be given a rudimentary knowledge only of the standby machinery manual control arrangements. The general adoption of this principle demands:

i) a simple and reliable installation;
ii) courage and foresight of tug owners;
iii) acceptance of the principle by the various inspecting authorities;
iv) co-operation of the trade unions.

TABLE II.—MANŒUVRING TIMES (SECONDS) FOR VARIOUS CONTROL SYSTEMS FROM AN 850 S.H.P. TUG

	Four-stroke Diesel			Reciprocating steam
	Fixed delays	Variable delays	Fly-wheel brake	One manœuvre only repeated changes take longer times
Full ahead to shaft turning astern	10	10	5½	5
Half ahead to shaft turning astern	10	8	5½	—
Slow ahead to shaft turning astern	10	7	5½	—
Full ahead r.p.m. to full astern r.p.m. (Ship just moving ahead).	26	26	20	15
Full ahead way to ship just moving astern	37	37	31	20

The passing of the steam tug and the introduction of the Diesel tug had a serious effect on tug manœuvring times. This is well illustrated by Table II. All subsequent improvements in ship's telegraph and engine control systems stem from the times given in the right hand column of the table.

It can be seen that the manœuvrability of the old reciprocating steam engine far excelled that of even the most modern bridge-controlled, geared Diesel installation. Only a variable-pitch or Voith-Schneider propeller arrangement rivals the steam tug in this respect.

Control of Reverse-reduction Geared Installations

1) *Conventional Telegraphs with Manual Control of Engines*

The great majority of tugs in service today employ the conventional telegraph, but it is significant that not one tug having this control for normal use has been completed by the author's company in the past five years. It is now fitted for standby use only.

Propeller reversal is a matter of skill and judgment and the protection of the valuable machinery is in the hands of the operator. Manœuvring is slow and inconsistent. Its main merit is that of cost.

2) *Dashpot-controlled Bridge/Engine Control Systems*

This is a common and inexpensive system which gives an adequate response for tolerant operators. The dashpot or equivalent device involves an arbitrary resistance or delay to movement of the telegraph handle whilst passing through the neutral position. Thus protection to gears, clutches and engine are ensured during this period. Many tug masters are critical of this resistance to the telegraph movement and, since a long manœuvring time is unavoidable, this method is now being superseded by fully automatic systems.

3) *Fully Automatic Bridge Control of Main Engines*

To obtain the optimum manœuvring performance from modern unidirectional Diesel engines and reverse-reduction gearboxes, fully automatic bridge control is necessary. The system must be designed to provide positive protection to the machinery and the ship under all conditions of service and a typical control sequence is shown in Fig. 19 for a "full ahead-full astern" manoeuvre. In this system two time delays are necessary:

1) to enable the engine speed to fall to the safe clutch engagement speed;

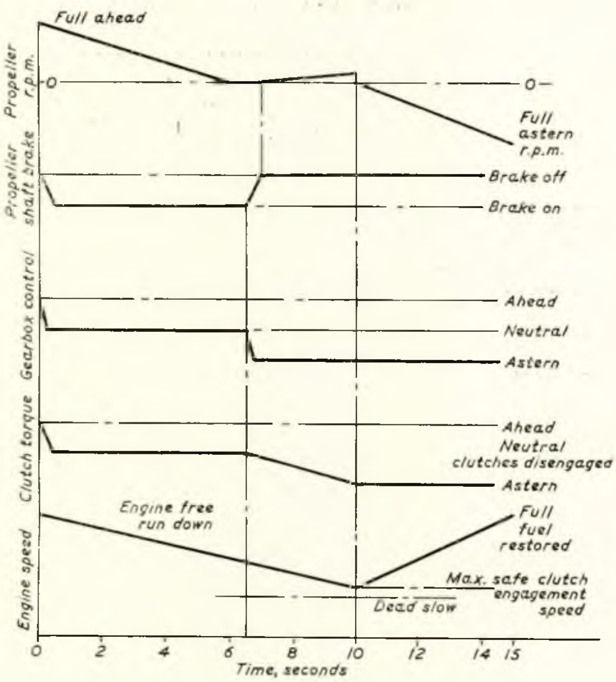


FIG. 19—*Sequence of operations for a full ahead-full astern manoeuvre using fully automatic pneumatic bridge controls*

2) to allow time for the clutches to engage and the brakes to operate in correct sequence (these may overlap).

A third time delay is also advisable with turbocharged engines to prevent full fuel being applied when the turbocharger is idling, since this produces smoky exhaust and dirty combustion chambers. The governor control is therefore moved slowly to give full fuel in about five seconds. The time taken to reach full engine speed after clutch engagement depends on the conditions under which the manœuvre is carried out and this particular delay does not in any case adversely affect the reversal of the ship.

It should be noted that propeller shaft brakes are now a universal fitting. These are set to stop the shaft rotation immediately a "change direction" order is given and should remain on until the engaging clutch is about to transmit power. Prior release which is indicated on the diagram should be

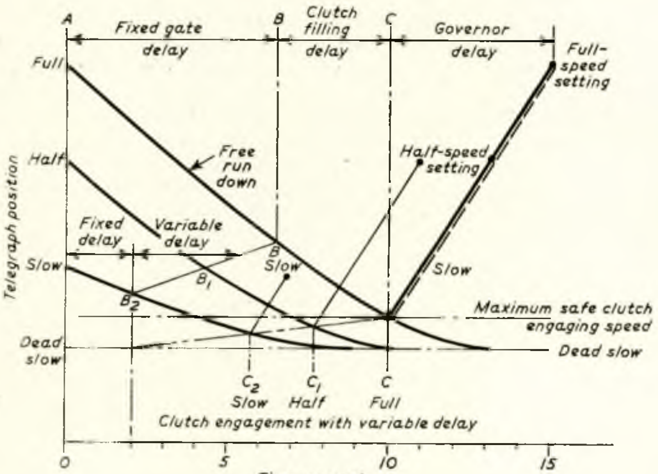


FIG. 20—*Pneumatic controls with fixed and variable delays*

avoided since, if there is way on the ship, the feed back of energy from the water to propeller accelerates the shaft rapidly in the original direction and on clutch engagement the engine is liable to stall.

The early systems incorporated fixed time delays for clutch engagement, the delay being set to suit a "full ahead-full astern" manœuvre. Such a system results in unnecessary delays for a slow or half ahead change, as can be seen from Fig. 20. The curves in this diagram show engine run down in speed for ahead-astern changes from "full", "half", "slow" and "dead slow" ahead conditions. The clutches, indicated here, require $3\frac{1}{2}$ seconds filling time so that the start of the operation is timed for $3\frac{1}{2}$ seconds before the engine speed is due to reach a safe engaging speed. If the gate delay AB is constant, clutch engagement in astern gear will occur ten seconds after a telegraph movement whether the movement is from "slow ahead" or from "full ahead" (at C). For the "slow ahead" to "astern" change, clutch engagement is thus held off for six seconds longer than is necessary to protect the clutches.

To improve the response, a variable delay system is now in use giving delay times in neutral, proportional to telegraph movement. The effect of this system is shown in Fig. 20. A saving of $4\frac{1}{2}$ seconds is achieved when working from the "slow ahead" setting. For movements from "half ahead" and "slow ahead", clutch filling begins at B_1 and B_2 instead of at B and clutch engagement is complete at C_1 and C_2 instead of C. It will be noted that with this system a small fixed delay of about 2 seconds is still required on reversals from dead slow ahead to astern to ensure that the clutch has time to empty between quickly repeated small movements such as when "inching" to take up the tow.

As will be realized from the diagram, a large part of the manœuvring time is taken up by the free run-down time of the engine, particularly in installations with a large flywheel, i.e. four-stroke. It is possible to reduce this time by applying a brake to the engine flywheel as well as to the propeller shaft and a system has been developed in which the run-down time has been halved. The diagrams of Fig. 21 show the effect of

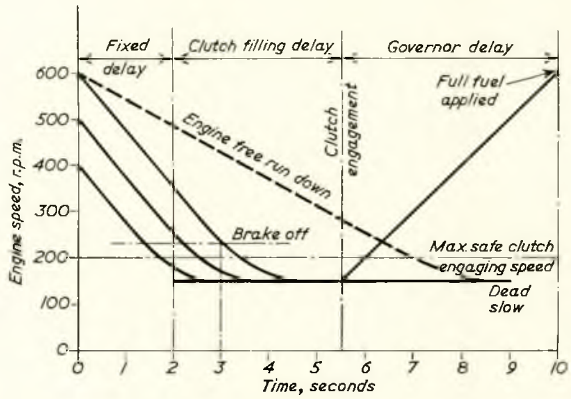


FIG. 21—*Improved operating times for a pneumatic control system using a flywheel brake*

the flywheel brake. The flywheel brake is arranged to lock out at speeds below 350 r.p.m. for which run-down time is less than 5 seconds. Thus the overall time is the same for any ahead-astern manœuvre and is two thirds of the time taken without a brake. It will be noted that with this arrangement a fixed gate delay has been used since variation of this short period would produce only very slight further savings.

As has been stated the automatic type of control system is logically the only one to be considered for use on highly developed modern reverse-reduction geared Diesel installations. Unfortunately experience has shown that several of the early vessels completed with this system, whilst working quite well, failed to give complete satisfaction. It is considered that this situation has been caused basically by too many firms being concerned in the control system and insufficient liaison between them. Some of the troubles experienced have been:

a) *Braking Effect*: The shaft brakes fitted for many years on tugs with conventional telegraph or dashpot bridge control were designed to withstand 30 per cent full ahead torque at standstill. This has proved incapable of stopping the propeller shaft in the time available during a "full ahead-full astern" manœuvre using full automatic controls. 60 per cent brakes are now being fitted and further experience is awaited. The problem of heat dissipation is one difficulty already indicated.

b) *Brake Release*: The stalling effect of early brake release has already been noted. This has occurred on gearboxes with integral shaft brakes where brake controls are operated simultaneously with clutch controls. The short brake release time of ½ second compared with the clutch engagement time of three to four seconds leaves a long unacceptable period during which the shaft is free to accelerate ahead due to the way of the ship. (This effect has been shown in Fig. 19). It is therefore necessary to delay brake release to the last possible moment before transmission of astern power begins.

c) *Engine Boost*: Engine boost has often been fitted to overcome stalling during clutch engagement. This boost applies more fuel to the engine for a short period during clutch engagement. Not only is the principle fundamentally wrong but it results in the tug surging forward at the instant when the tow is being taken. The presence of boost indicates that other parts of the control system are not operating in a suitable manner. The complete control system includes such parts as engine fuel system, governor, flywheel, clutch operating mechanism and brakes, in addition to the bridge to engine components.

d) *Gearbox Temperature*: The time constants of the system must remain constant under all conditions of operation. If this is not strictly obeyed then the settings of the control system will wander. The commonest forms of clutch are those illustrated earlier, i.e. oil operated. Main gearbox lubricating oil is normally in the range S.A.E. 60-90. The service temperature of this oil on local tugs is in the range 50-100 deg. F. even with the lubricating oil coolers completely bypassed, whilst the recommended temperature range for these oils lies in the range 140 deg. to 170 deg. F. depending on the oil used. Thus in practice the clutch response is sluggish due to thick oil and inconsistent due to the temperature range. This situation does not appear to have any serious effect on the gears and previous correction of this condition on manually-controlled vessels has not been necessary.

It is considered that if the best response is to be obtained from these automatic systems then the gearbox must be brought rapidly to its normal operating condition and maintained at this temperature by thermostatic control throughout the operating period.

One solution of this problem now being considered is to use main engine fresh water which is controlled at 140 deg. F. to heat the oil initially, should the temperature rise above 140 deg. F., then the fresh water will automatically act as a coolant.

Control of Controllable-pitch Propellers

There are many methods used for the bridge control of controllable-pitch propellers, but basically these can be resolved into three main groups only two of which are suited to tug work.

Two Lever Control

The first group utilizes two lever control giving separate manual setting of propeller pitch and engine revolutions. This is frequently used on single-screw tugs but it does allow a wide choice of pitch, speed and power combinations which inevit-

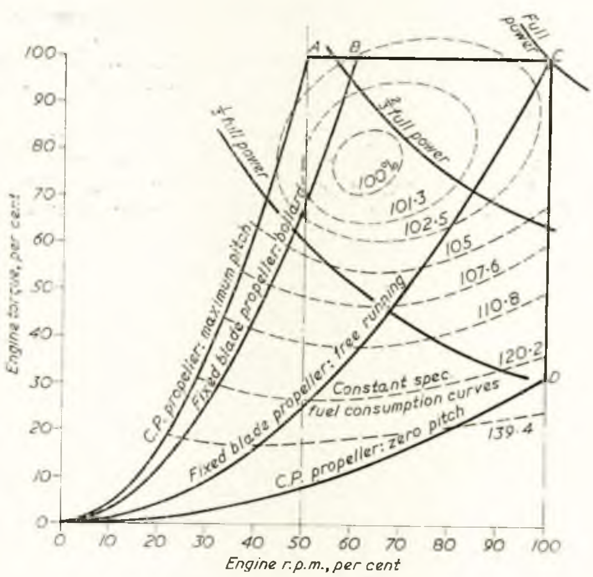


FIG. 22—*Hypothetical propeller characteristics versus Diesel engine performance*

ably results in poor engine operating conditions. Any point between curves OA and OD of Fig. 22 may be selected. Operation in the region of high r.p.m. and low torque, which is very likely, results not only in high fuel consumption as shown, but also in high wear rates and carbon formation. Also at the other extreme of the diagram the engine can easily be overloaded unless adequate instrumentation is fitted to indicate that the maximum continuous torque setting has been reached.

Single Lever Control

In order to simplify bridge control, particularly for twin-screw vessels, obtain rapid response and eliminate undesirable engine operating conditions, single lever control is frequently used.

The system best suited to tug work involves a fixed relationship between engine speed and torque. With this system the propeller pitch is automatically adjusted to absorb the available torque at the set r.p.m. thus the power of the engine remains constant at the set position irrespective of sea, load or tow conditions. The resulting operating line approximates to OAC which limits the engine to better loading conditions throughout the power range.

This form of control employs an hydraulic engine speed governor embodying a control valve. This valve is actuated by both engine speed and torque (fuel rack position). The equilibrium condition of this valve is obtained only when both speed and torque signals are balanced according to the set relationship (OAC). With any departure from this relationship the valve causes the propeller pitch to be adjusted in such a manner as to restore the original balance of torque and speed.

One major advantage of this system is that it has been in use for many years for the control of Diesel electric locomotives and the governor is therefore well tried.

ACKNOWLEDGEMENTS

In conclusion, several important acknowledgements must be made.

The permission of Messrs. John I. Thornycroft and Co. Ltd. for publication is gratefully acknowledged although it is emphasized that any opinions expressed are solely the author's responsibility and in no way indicate the official views of the company.

The ready co-operation of several companies including Crossley Brothers, Ruston and Hornsby Ltd., Fluidrive Engineering Ltd., Modern Wheel Drive, Permain Engineering Ltd., (Brevo Gearboxes) and Richard Dunstans Ltd., is very much appreciated and the author only regrets that time does not allow him to cover adequately the material which these firms provided.

The author would also like to thank Dr. R. V. Hughes of Ruston and Hornsby for allowing him to use an amended extract from an excellent paper of his as the basis of the section on the control of reverse-reduction gearboxes. Finally Mr. A. L. Dorey and many of the author's colleagues at Thornycrofts are thanked for their help.

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INSTITUTE ACTIVITIES

Branch Meetings

Devon and Cornwall

A meeting of the Devon and Cornwall Branch was held on Tuesday, 7th December 1965, at the South Western Gas Board Lecture Theatre, Derry's Cross, Plymouth, at 7.00 p.m., when a paper entitled "30,000 s.h.p. Unitized Reheat Steam Turbine Propulsion" by T. B. Hutchison (Member) was presented by the author to an audience of forty.

Prior to the reading of the paper the presentation of an Honorary Life Membership scroll was made to Mr. N. L. Wright (Member), by the Chairman, Mr. R. E. Pritchard (Chairman of the Branch).

This was an unique occasion in the five-year history of the Branch. Mr. Pritchard spoke of the devotion and service shown by Mr. Wright to his profession and said that it was an added honour that he, the Chairman, was able to make a presentation to a former apprentice of his own firm.

Mr. Wright was educated at Plymouth Public School and later apprenticed to Messrs. Willoughby Bros. Ltd. until going to sea in 1905 with the Royal Steam Packet Company of Southampton. Having obtained his motor endorsement at the age of 56, four years later he was appointed Commodore Engineer of the Royal Mail Line. In 1946 he represented the Merchant Navy as a Vice-President of the Institute, and the following year retired at the age of 63 having served for the last three years in the flagship of the line R.M.S. *Andes*. Now 82 years of age, Mr. Wright has given long and faithful service to the Royal Mail Steam Line and to his profession, and the Chairman expressed the hope that he might live long to enjoy his retirement as much as he had enjoyed his life at sea. Mr. Pritchard then presented the scroll to Mr. Wright on behalf of the Institute.

In reply Mr. Wright said how pleased he was to receive the scroll. It was a great occasion for him and he could look back over the years to a very happy life in the profession. He thanked the Chairman for his kind remarks and asked him to convey his thanks to the Institute for this great honour.

The Chairman then introduced Mr. Hutchison who presented his paper and with the aid of slides gave a very clear and concise presentation of his subject which was well received by the audience.

The paper aroused much interest as was shown by the many questions ably answered by the author, which continued for an hour until at 9.25 p.m. the meeting closed.

A vote of thanks to Mr. Hutchison was proposed by Mr. N. W. Jephcott, M.A. (Corresponding Member for Falmouth), and was endorsed in the usual way.

Merseyside and North Western

A general meeting of the Merseyside and North Western Branch was held on Monday, 6th December 1965, in the Conference Room of the Mersey Docks and Harbour Board, Dock Board Building, Pier Head, Liverpool, 3, at 6.00 p.m., when a

Photograph by permission of the Western Morning News Co. Ltd., Plymouth

The Chairman of the Devon and Cornwall Branch, Mr. R. E. Pritchard (left), seen presenting the scroll of Honorary Life Membership to eighty-two-year-old Mr. N. L. Wright, while Commander W. Farrell, M.B.E., R.N. (Honorary Secretary) looks on paper entitled "The Design and Application of Paxman Marine Engines" by W. R. Dingle and N. J. Wadley, was presented by the authors.

Chairman of the Branch, Mr. T. E. Jones (Member of Council) was in the Chair and approximately seventy-five members and guests were present.

In introducing their paper the authors said they had selected for consideration those engines in the 7-in. bore range used as marine auxiliaries in large numbers in the power range 100-1,045 b.h.p., comprising four, six, eight and twelve cylinders either normally-aspirated, pressure charged, or intercooled.

Acceptance of these faster-running engines in the speed range of 1,000/1,500 r.p.m. had grown rapidly over the past four or five years and experience in service had proved that these sets, whether for main generating services, cargo pumping and the like, could be relied upon to give long and dependable service with the minimum of attention at least equal to their slower-speed counterparts in what was generally accepted as being a very demanding and exacting duty.

In mentioning ease of overhaul, helped by the fact that components were light and easy to handle, the authors also made the point that even with underslung crankshafts it was possible to make simple arrangements for inspection of main bearings and crankshafts.

Northern Ireland Panel

A meeting of the Panel was held on Tuesday, 7th December 1965, in the Millfield Building of the College of Technology, Belfast, at 7.00 p.m., when a paper entitled "Fire Protection and Fire Fighting in Ships" by D. R. Murray Smith and A. T. Willens (Member of Council), was presented by the authors.

Chairman of the Panel, Mr. D. H. Alexander, O.B.E., F.C.G.I., M.Sc., Wh.Sc. (Local Vice-President) was in the Chair and approximately eighty members and visitors attended the meeting.

A vote of thanks to the speakers was proposed by Mr. R. R. G. Cameron, O.B.E., B.Sc., and seconded by Mr. J. W. Bull (Member).

The meeting closed at 9.15 p.m.

Scottish

Ninth Annual Dinner and Dance

The Ninth Annual Dinner and Dance of the Scottish Branch was held on Saturday, 11th December 1965, at the Grosvenor Restaurant, Glasgow.

Mr. H. Brady (Chairman of the Branch) presided and with Mrs. Brady received over two hundred members and guests.

Following the Loyal Toast the Chairman extended a warm welcome to all present and especially the ladies. He emphasized that it was a night for dancing and not for speeches and that he did not intend to make a speech.

Mr. Brady was pleased to welcome Mr. R. Beattie (Vice-President), and Mrs. Beattie, also the Immediate Past-Chairman of the Branch, Commander A. J. H. Goodwin, O.B.E., R.N., and Mrs. Goodwin.

David Sibbald proved himself to be an excellent Master of Ceremonies and his Orchestra kept everyone on their feet throughout the evening.

The function was a great success.

General Meeting

A general meeting of the Scottish Branch was held on Wednesday, 15th December 1965, at the Institution of Engineers and Shipbuilders in Scotland, 39 Elmbank Crescent, Glasgow, C.2., at 6.15 p.m., when a paper entitled "Air Consumption Data and Practical Performance of the Stork Uniflow-scavenged Two-stroke Marine Engine" by E. A. van der Molen and Ir. H. van der Wal, was presented by the authors.

Mr. H. Brady (Chairman of the Branch) presided at the meeting and extended a welcome to the fifty-eight members and visitors present.

In presenting the paper Mr. van der Molen opened with an introduction to the principles and design features concerned with the efficient operation of Diesel engines in general and the Stork engine in particular. He defined the development work and the procedures which had been adopted in their endeavour to obtain a perfect mixing of air and fuel in the cylinder.

Mr. van der Wal then took over the meeting and after

At the Ninth Annual Dinner and Dance of the Scottish Branch. From left to right: Mr. H. Brady (Chairman of the Branch), Mrs. R. Beattie, Mrs. H. Brady, and Mr. R. Beattie (Vice-President) giving a brief description of the engine, gave details of the practical performance of the engine. He highlighted design features which had been included to facilitate maintenance giving such details as accessibility to the cylinder liner and access to piston rings by simply lowering the bottom half of the cylinder liner. Slides showed ready access to the combustion face of the cylinder cover without removing the cover.

A stimulating discussion followed and it was with some regret that the meeting was called to a close.

A vote of thanks was proposed by Mr. Henderson who congratulated the speakers on the high standard of their paper and the very capable way in which they had handled the discussion. The vote of thanks was carried unanimously.

The meeting closed at 8.00 p.m.

South East England

Annual Dinner Dance

The Annual Dinner Dance of the South East England Branch was held on Friday, 15th October 1965, at the Masonic Hall, Gravesend, at 7.30 p.m.

Photograph with acknowledgements to the Kent Messenger, Maidstone

At the Annual Dinner Dance of the South East England Branch. Standing from left to right: Mr. R. H. Cadle (Honorary Treasurer), and Mrs. Cadle, Mr. J. S. Allan (Vice-Chairman of the Branch), and Mrs. Allan, Mr. R. R. Strachan (Vice-Chairman of Council), Mrs. J. R. Richardson and Mr. J. R. Richardson (Honorary Secretary). Seated from left to right: Mr. A. E. Franklin (Assistant Secretary of the Institute), and Mrs. Franklin, Chairman of the Branch, Mr. A. H. Stobbs (Member of Council) and Mrs. Stobbs, Mr. A. T. Willens (Member of Council) and Mrs. Willens

Chairman of the Branch, Mr. A. H. Stobbs (Member of Council), was in the Chair and with Mrs. Stobbs received the principal guests, Mr. R. R. Strachan (Vice-Chairman of Council), Mr. A. T. Willens (Member of Council), and Mrs. Willens, and the Assistant Secretary of the Institute, Mr. A. E. Franklin, and Mrs. Franklin.

Dinner was followed by a cabaret and dancing and was attended by 150 members and guests.

General Meeting

A general meeting of the Branch was held on Tuesday, 16th November, at the Clarendon Royal Hotel, Gravesend, at 7.30 p.m. when a paper entitled "30,000 s.h.p. Unitized Reheat Steam Turbine Propulsion" by T. B. Hutchison (Member) was presented by the author.

The paper was followed by a general discussion in which Mr. Hutchison answered questions put to him by the members.

Mr. A. H. Stobbs (Chairman of the Branch) was in the Chair and in closing the meeting thanked Mr. Hutchison for his most interesting and informative paper.

South Wales

Junior Meeting

A junior meeting of the South Wales Branch was held on Friday, 22nd October 1965, at the Technical College, Swansea, at 6.30 p.m. when a lecture on "Petroleum Refining" was given by Mr. B. L. Wright.

Mr. A. R. Simpson (Chairman of the Branch) was in the Chair and forty-seven members and guests were present.

The showing of two technical films, kindly loaned by the BP Oil Company Limited, followed an interesting talk.

After the film show, a lively discussion took place which was closed by the Chairman at 8.00 p.m.

Mr. Simpson paid tribute to the BP Oil Refinery, Llandarcy, and the BP Oil Company, for their helpful contributions to a most successful evening.

Mr. J. Wormald, B.Sc. (Member of Committee) proposed a vote of thanks to Mr. Wright which was warmly received by those present.

General Meeting

A general meeting of the Branch was held on Monday, 29th November 1965, at the South Wales Institute of Engineers, Park Place, Cardiff, at 6.00 p.m., when a paper entitled "Some Factors Affecting the Selection of Systems for Automatic Control of Marine Machinery" by Ll. Young (Member) and

P. J. Wheeler, B.Sc. (Eng.) was presented by the authors.

Mr. T. W. Major (Vice-Chairman of the Branch) presided at the meeting in the absence of the Chairman of the Branch, Mr. T. C. Bishop, from whom apologies were received, and after introducing Mr. Young and Mr. Wheeler to the meeting invited them to deliver their lecture.

The paper aroused much interested discussion, reluctantly terminated by the Chairman at 8.00 p.m.

A vote of thanks to the speakers, proposed by Mr. D. Skae (Vice-President), was warmly seconded by those present.

West of England

On Tuesday, 14th December 1965, the West of England Branch held a joint meeting with the Western Division of the Institution of Mechanical Engineers, to which members of the Institution of Electrical Engineers were invited. The meeting took place in the Small Engineering Lecture Theatre, Queens Buildings, University of Bristol, at 7.30 p.m.

Chairman of the Branch, Mr. J. P. Vickery, was in the Chair and extended a warm welcome to the seventy members of the participating institutions, and their guests. He said how gratifying it was to see such a good response to the meeting and hoped that the joint meeting would be the forerunner of many more between different engineering institutions in the area.

The Chairman introduced the speakers, Mr. Ll. Young (Member) and Mr. P. J. Wheeler, B.Sc. (Eng.), who presented their paper "Some Factors Affecting the Selection of Systems for Automatic Control of Marine Machinery".

The paper dealt primarily with a comparison between pneumatic and electronic devices for the automatic control of boiler plant in steam-driven vessels. The authors considered that electronic control would eventually be more widely used and it was essential that the component parts in an electronic system should be completely reliable, especially as conditions aboard ship were more severe than on shore. To this end improvements were being made all the time, one of the biggest features of their manufacture being an elaborate inspection system on the factory floor.

The paper was most interesting and enlightening and in a discussion opened by Mr. T. A. Mogg (Member of the Branch Committee) no less than ten speakers took part.

A vote of thanks to the authors was given by the Chairman of the Western Division of the Institution of Mechanical Engineers, Mr. G. H. Beauchamp, B.Sc., who also thanked the West of England Branch for organizing such a pleasant evening.

The meeting closed at 9.30 p.m.

Election of Members

Elected on the 13th December 1965

MEMBERS

Colin Allison
Raymond John Sidney Bailes
Robert Henry Blenkinsopp
Ronald Blundell
Neil Jack Bradley
Samuel Stephen Chadwick
Alfred Walter Clark
Richard Edward Roland Crick
Harry Bertram Davies
Roy Dixon
Charles Davie Galbraith
John Campbell Hart, B.A. (Oxon)
Thomas Cyril Johnson
Keith James King
Hugh McIlmurray
George Robert Barton Pattison, Capt. R.N., B.Sc. (Elec. Eng.) (Durham)
James Arthur Taylor
Arthur Charles Vogt

ASSOCIATE MEMBERS

William Archer
Robin Keith Avery
Alan Edward Bell
Barry James Butterworth
Francis Cooper
George Moir Douglas
Thomas Farrell
Anthonypillai Francis
John Giles
John Albert Houghton
Clifford Ellis Huestis, Lieut.(L.D.), C.D., R.C.N.
Jeffrey John Hunking
John Whitney Hurlow
Stewart Hutton
William Benjamin Hyde
Jon Eladio Legorburu, B.Sc. (Nav. Arch.) (Durham)
Clifford Alfred Le Quelenec, Lieut., R.N.
John Charles Lester
Ramesh Vishwanath Modak, Lieut., I.N., B.E. (Baroda)
William Macleod Morum
Thomas Stewart Neilson
Donald John Gustave Olufson
Stanley Robson
Dipak Kumar Roy
N. D. Salwan, Lieut., (E) I.N.
Alexander Shaw
John Dixon Brunton Shaw
Alexander William Stevenson
Robert Thomson
David Graham Todd
Frank Guy Travers, Cdr., R.N.
John Alfred Randel Tutte
Thomas George Wallace, Eng. Lieut., R.N.
Gerald Townshend Watts
Charles Whiteside
Peter Alexander Wight

ASSOCIATES

Arthur Malcolm Bennett, B.E.M.
James Alexander Cowie
John Leslie Denny
Robert Osborne Fleming
Norman Graveson
Hermanus Johannes Cornelis Henderson
Robert Hamer Jones
Lam Yu-kee
Alexander Thomas Lucas
Robert Gordon Murdoch
Albert O'Connor
Brian Sinclair

GRADUATES

Malcolm Annand
John Robert Board, Lieut., R.N.
Lawrence Brownson
John Boyd Orr Fullarton
Peter Henry Gee
John Grgich
Kenneth Harrison
Derek Richard Hoare
David Kenneth Martyn
Krishan Dev Mehta
Alistair Peter Charles Taylor

STUDENTS

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Barend Bal
Roger Nicholas Cooper
John Cameron Cullen
Anthony James De'ara
Philip Brian Dunham
John Gordan Timothy Epsom
Andrew John Gallaway

Robert Wilson Gardner
David Greenshields
Peter Herring
Noel Harold Hogg
Richard John Ingham-Berry
Charles Patrick Irvine
John Anthony Legg
Lim Chee Onn
Michael John Luther
John MacAlasdair Milne
Jack Ormiston
Alexander John Pace
John Martin Palmer-Felgate
James Millar Paton
Thomas Peebles
Peter Henry Seymour Robinson
Peter Sidney Culpin Rogers
Kenneth Charles Lynton Ross
Edward Neil Thompson
Austin James West
Raymond Paul White
Richard Wright

PROBATIONER STUDENTS
Kamarul Baharin
George Frederick Blacker
Michael Byrne
John Anthony Coates
David John Cox
James Robert Cron
Christopher Paul Denman
Jack David Dyson
Colin Hoskins
James Ainsworth Knott
James Legard
Nicholas Peter Lowe
Simon John MacAllan
Nicholas Dexter Peet
Ralph Henley Richards
Richard Allen Scurrey
David Richard Taylor
Michael Watts
Allen Brian Webb
David Webb
David Crankshaw Wise

TRANSFERRED FROM ASSOCIATE MEMBER TO MEMBER
Henry Berchmans Azevedo, Lt. Cdr.(E), I.N.
Wilfred Donald Beard
Peter Broadway
John Bernard Cowan
Frank Green
Joseph Frederick Hainen
John Eric Kingham
Neil Macleod
Ronald Graham Moscrop
Francis Norman Pulford
Francis Henry Stephenson Scriven
John Weston

TRANSFERRED FROM ASSOCIATE TO MEMBER
Philip Walter Dean
Craig Kilgour Stevenson
David Watson

TRANSFERRED FROM ASSOCIATE TO ASSOCIATE MEMBER
Sardar Taffil Uddin Ahmed, Lieut., P.N.

TRANSFERRED FROM GRADUATE TO ASSOCIATE MEMBER
George S. Armstrong, Lieut., R.C.N.
John Dudley Bassett
Francis William Cockburn
John Walter Dalrymple
Emmanuel Debono
Ian Elder
Peter Henry Edward Head
Gavin McGregor
Qaiser Mirza Rizqi
Colin Gregory Stonebridge
David Victor Taylor

TRANSFERRED FROM STUDENT TO ASSOCIATE MEMBER
Ross A. Goodman, B.Sc. (Hons.) (Sunderland)
Owen Clive Whiteaker

TRANSFERRED FROM PROBATIONER STUDENT TO ASSOCIATE MEMBER
David Alexander Charles Williams

TRANSFERRED FROM STUDENT TO GRADUATE
Ronald Ellison
Raphael Okechuku Morah
William Robert John Webb
Peter John Wyld, B.Sc. (Hons.) (Durham)

TRANSFERRED FROM PROBATIONER STUDENT TO GRADUATE
Godwin Layiwola Layole

TRANSFERRED FROM PROBATIONER STUDENT TO STUDENT
Nicholas Michael Pope

Institute Awards

Members are reminded that the following awards are now made:

The Denny Gold Medal for the best paper read by a member during the session.

The Institute Silver Medal for the best paper read by a non-member during the session.

The Junior Silver Medal and Premium of £5 for the best paper by a Graduate or Student read before the Junior Section during a session.

The W. W. Marriner Memorial Prize, value £5, given annually to the candidate who submits the Engineering Knowledge paper (Steam or Motor) of the highest merit in the Board of Trade examinations for the Second Class Certificate of Competency.

The Extra First Class Engineers' Certificate Examination—Institute Award of a Silver medal for the candidate obtaining the highest marks in the Board of Trade examination.

The Herbert Akroyd Stuart Award, value £50, available biennially, open to members of all grades and non-members for the best paper read at the Institute on "The Origin and Development of Heavy Oil Engines".

The Yorkshire Award, value £40, available biennially for the writer of an essay or the author of a paper read before the Institute dealing with any development related to any aspect of marine engineering or a product applicable to marine engineering.

A cash prize of £25 awarded annually from the interest on the John I. Jacobs, W. Murdoch, D. F. Robertson and A. Girdwood funds for the best essay on the technical advantages to be gained by taking the Extra First Class Engineers' Certificate course—available to engineers taking such a course at a technical college.

Awards, value £4 4s., are given annually to students of technical colleges in marine centres for the best year's work in the study of heat engines.

Prizes for students taking the Ordinary National Diploma Course under the alternative scheme for the training of seagoing engineers. Two prizes are given each year to each technical college operating the scheme, a prize of two guineas being awarded to the best first year student and a prize of six guineas to the best second year student.

The Frank Roberts Award of books or instruments to the value of £7 10s., given annually to the Student or Probationer Student member of the Institute gaining the highest aggregate marks in the courses and examinations in Phase III of the alternative scheme for the training of seagoing engineers.

Administered by the Institute

The William Theodore Barker Award—£100 annually to the candidate who gains the highest marks in the Board of Trade examinations for the First Class Certificate of Competency, provided that such candidate takes the course for the Extra First Class Engineers' Certificate at a technical college.

OBITUARY

JOHN PATERSON CAMPBELL (Vice-President)

An appreciation by G. M. Kennedy (Local Vice-President)

It is with regret that we record the death on 1st November, 1965, of Mr. J. P. Campbell, one of the most respected and widely known members of the Institute of Marine Engineers.

Mr. Campbell served at sea with the Port Line Limited from 1918 to 1929. He then came ashore to act in various capacities in shipyards at Jarrow and Hebburn-on-Tyne. In 1940 he was appointed Assistant Works Manager with Messrs. Green and Silley Weir at their Blackwall Works, remaining there until June 1945 when he accepted the post of Superintendent Engineer, Southern Railway, Southampton Docks.

From then until he moved with the Marine department to Victoria in 1956, much of his time was given up to the furtherance in Southampton of the aims of the Institute, particularly in endeavouring to encourage younger members to take an active interest in our proceedings.

Mr. Campbell was associated with the Southern Branch in the early days when it was a Junior Branch and when the Southern Joint Branch was inaugurated in 1953 he was nominated a Vice-President. His enthusiastic support was valued so much that despite his move to London in 1956 he continued to be closely associated with the Branch as President in 1957 and then as an Honorary Vice-President.

Mr. Campbell did not confine his activities only to Southampton. He was elected a member of the Council of the Institute in 1949 remaining on the Council until 1952. He served again on the Council from 1953 to 1958 and also was re-elected in 1960. In 1954 he was Chairman of Council and in 1961 was elected a Vice-President. During the above periods he was a member of various Institute committees.

There were few of our functions that were not graced by the presence of Mr. Campbell and his wife, and his passing is a source of sorrow to all who knew this friendly and generous man.

COMMANDER SAVIOUR LAWRENCE AGIUS, R.N.R. (Member 4732) was born in Malta on 16th November 1897. After being apprenticed with the Government Railway and at the Royal Naval Dockyard on the island, he went to sea, at the age of seventeen, with the Prince Line. Whilst with this company he gained his First Class Board of Trade Certificate and, at the age of thirty-two, was promoted to chief engineer. He continued to serve with the Prince Line until the outbreak of the Second World War, at which time his ship was m.s. *Southern Prince*. When this vessel was taken over by the Royal Navy, at the end of 1939, and converted to a mine-layer, he joined the Royal Naval Reserve and served in her with the rank of Commander.

Commander Agius was discharged from the Navy through ill-health and this eventually caused his early retirement. In the meantime he held appointments as a surveyor with the Ministry of War Transport and as marine and engineer superintendent with C. Rowbotham and Sons, of London. His death occurred in March 1964, after a prolonged illness due to lung cancer.

Commander Agius was elected a Member of the Institute in 1929.

CHARLES WESTON FREDERICK BASS, D.S.C. (Member 12156) died suddenly on 8th December 1965 at the age of sixty-five.

From 1915 to 1919, Mr. Bass served an apprenticeship at H.M.S. *Fisgard* and for the following seven years served both at sea as an engine room artificer and on instructional duties in marine engineering at H.M.S. *Fisgard*. By 1928 he had been promoted to commissioned rank and was serving as a watch-keeping officer in H.M.S. *Furious*. From 1931 to 1944 he was chief engineer officer in H.M.S. *Bridgewater, Vega, Wanderer, Vanoc, Eskdale* and *Scorpion*. In the latter year he was appointed to the staff of the C.-in-C., Portsmouth, where he served for a year, after which, until 1948, he was senior engineer officer in H.M.S. *Vernon* and *Ceylon*. In 1948, he received the Distinguished Service Cross and a First Class Service Certificate; he held the rank of Lieutenant (E), R.N., at that time.

Retiring from the Royal Navy early in 1950, he was later that year appointed engineering manager by Camper and Nicholsons, shipbuilders and engineers of Gosport, with whom he remained until December 1957, when he finally retired on the advice of his doctor.

Mr. Bass was elected a Member of this Institute on 13th December, 1948.

SIR GEORGE RIDDOCH CAMPBELL, K.C.I.E. (Companion 11670) died on 8th July 1965 at the age of seventy-eight

Educated at Hamilton Academy, he commenced his career with Thomas Law and Co., a Glasgow shipping firm, in 1903, and six years later joined the firm of Mackinon, Mackenzie and Company in Calcutta, with whom he remained until 1949. He served on the Western Front during the First World War and returned to India afterwards to resume his business career.

In 1935-36, he was a member of the Council of State in Delhi and, from 1937 to 1939 was Leader of the European Party in the Bengal Legislative Assembly. He was also, at that time, president of the Bengal Chamber of Commerce and of the Associated Chamber of Commerce of India.

He represented the Ministry of War Transport in India, Burma and Ceylon during the first two years of the Second World War and was also Shipping Controller in India. He returned to the United Kingdom to take up the appointment as Commercial Director of Shipping at the Ministry of War Transport and, in 1943, became Regional Port Director for the South East Region, including the Port of London.

He remained in the United Kingdom after the war and worked in the City of London, where he was managing director of the P. & O. Steam Navigation Company and the British Indian Steam Navigation Co. Ltd.

He was knighted in 1936 and made a K.C.I.E. in 1942.

Sir George was elected a Companion of the Institute in January 1948; he was also a member of the Royal Society of Arts and of the Caledonian Society of London.

Charles Edward Hall (Member 6301), formerly a ship and engineer surveyor for Lloyd's Register of Shipping at Port Adelaide, died on 21st October 1965, aged sixty-seven years.

After receiving his education at Barrow Secondary School, Mr. Hall served an apprenticeship with Vickers Ltd. at Barrow-in-Furness. He also attended Barrow Technical School, where he gained the City and Guilds Bronze Medal for mechanical engineering. He went to sea in 1920 and, during the course of his seagoing career, gained a First Class Board of Trade Steam Certificate with Motor Endorsement. He held the grade of chief engineer during the latter years of his sea service.

In 1939, Mr. Hall was an engine and ship surveyor and examiner of engineers at Singapore for the Malayan Government. Two years later he was Senior Government Ship Surveyor at Penang and Supervisor of War Contracts; he also acted as a non-exclusive surveyor for Lloyd's Register at this time. He left Malaya at the time of the Japanese invasion and was appointed Planning Engineer with the Ministry of Munitions at Sydney in 1942. He was appointed marine superintendent to the firm of W. E. Smith Pty. Ltd. in 1944, having complete control of contracts for fitting out landing craft for the United States Army.

Mr. Hall's first contact with Lloyd's Register of Shipping in Australia was in November 1944, when he was assigned to the Sydney office for training prior to taking up the appointment of non-exclusive surveyor at Port Adelaide on 1st April 1945. He served as non-exclusive surveyor until 1st January 1958, when he took charge of the office which was placed on an exclusive basis.

He suffered a heart attack in February of last year and, being unable to return to duty, retired from the Society's service at the end of June.

Mr. Hall was elected a Member of the Institute on 4th November 1929. He is survived by his wife, a daughter and a son.

George Edward Kerr (Member 12819) died on 25th October 1965, at the age of sixty-three.

Mr. Kerr was apprenticed from 1918 to 1923 with William Denny and Brothers, Ltd., Dumbarton, and for five years attended the Royal Technical College, Glasgow.

In 1924 he first went to sea as a junior engineer with the Royal Mail Lines, and in 1928 he joined the China Navigation Co. Ltd., with whom he served until 1936, rising to the rank of acting assistant superintendent engineer. During this time, he gained a First Class Steam Certificate.

Mr. Kerr left the sea to become a foreman engineer, and later chief assistant, dock accounts, at the Singapore Harbour Board. From 1942 to 1945 he was cost accountant at the Mazagon Dock Ltd., Bombay, and in 1947 he returned to the Singapore Harbour Board as assistant dockyard manager.

Elected a Member of the Institute in 1950, Mr. Kerr was in 1952 appointed general manager of the Mazagon Dock Ltd. While resident in Bombay, Mr. Kerr played a large part in the formation of the Bombay Section, and was elected Local Vice-President for Bombay in 1957, being re-elected twice as Local Vice-President and also serving as Chairman.

In 1960 he returned to the United Kingdom and became managing director of Alexander Shanks and Son Ltd., Arbroath, where he remained until the time of his death.

George William Kitching (Member 4223), a director of the firm of Kitching and Carmichael, died at his home in Surbiton on 22nd October 1965. He was in his seventy-eighth year.

Mr. Kitching served a six-year apprenticeship with Wm. Gray and Co. Ltd., after which he joined John F. Kitching and Son, the firm of consulting marine engineers founded by his father. He travelled widely on ship surveying duties and had considerable sea experience in running builders' trials and tests.

He was elected a Member of this Institute in March 1921 and was also a Member of the Royal Institution of Naval Architects and of the North East Coast Institution of Engineers and Shipbuilders. In 1951, he was elected to the Freedom and Livery of the Worshipful Company of Shipwrights.

Mr. Kitching leaves a widow, three sons and a daughter.

John Smith Thompson (Member 8272) was born in Newcastle upon Tyne in 1895.

After apprenticeships with Crawhall and Sons, Gateshead, the Sunderland Forge, and short periods with two other firms, Mr. Thompson went to sea, firstly with a local line, and then, in 1915, with the Clan Line.

He gained his First Class Board of Trade Certificate in 1918, and during his twenty years service with the Clan Line, he was chief engineer in several of their vessels, including the ship which brought the first refrigerated cargo of lamb from New Zealand. He was elected a member of the Institute in 1936.

He left the sea in 1938 to become the Scottish representative of George Kent Ltd., Luton, and went to live at Newton Mearns, Glasgow. In 1961 he retired from this post, and in July 1965 he moved to Whalton, Northumberland.

Mr. Thompson was always keenly interested in the activities of the Mission to Seamen, and for many years was on the management committee of the Flying Angel Mission in Glasgow. He was also a member of the North East Coast Institution of Engineers and Shipbuilders.

Mr. Thompson, who died on 25th September, 1965, is survived by his wife, a son and a daughter.