There is no doubt that these assumed values could be definitely fixed from proper progressive trials, in which the vessel would be run with one propeller only, used both pushing and pulling, and so the necessary quantities for analysis obtained under known conditions. It is in the hope that sometime

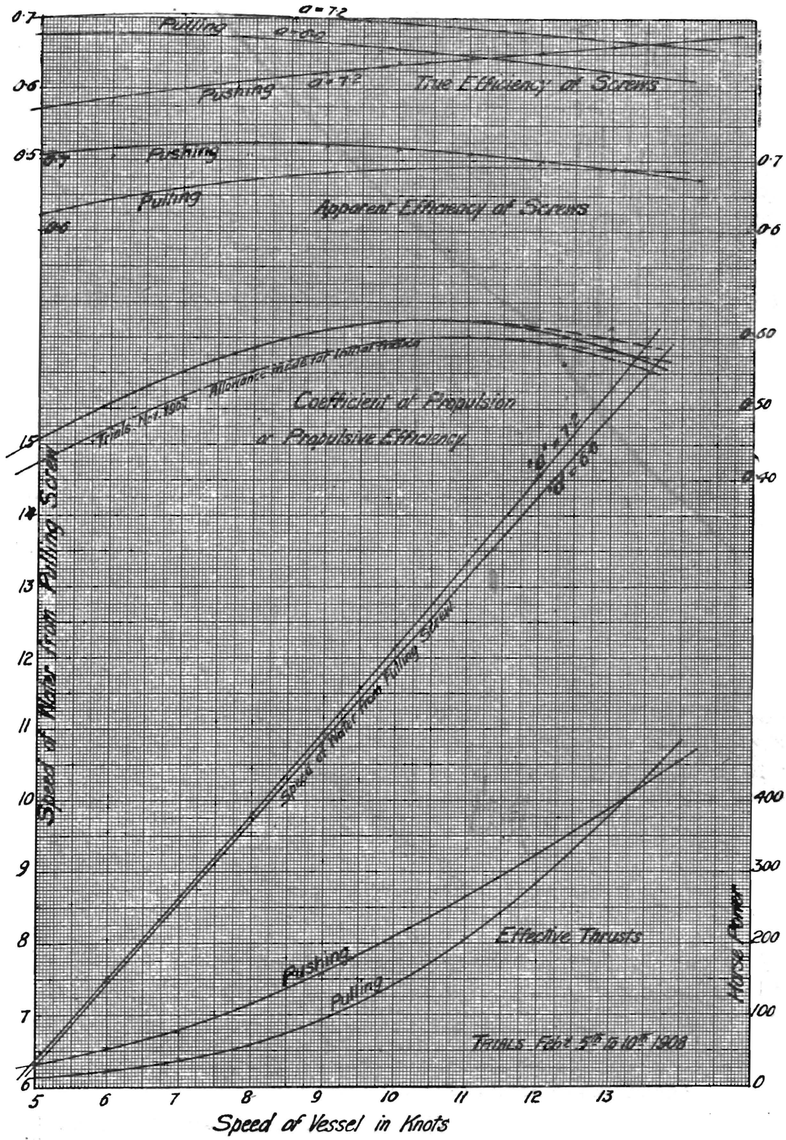

Fig. 8.

in the future the opportunity of conducting a full series of trials will be given, that the writer analysed the data at command as far as possible.

From the foregoing the true slips can now be plotted, these are merely the values of a.s. shown plotted in Fig. 7, divided by 7.2. These slips are also shown in Fig. 7. From these can be computed the efficiency (true and apparent) of the screws, using the formulæ—

$$e = (1 - s)\frac{s - \dfrac{f\,Y_c}{a\,X_c}}{s + \dfrac{f\,Z_c}{a\,X_c}}$$

$$e^1 = (1 - s^1)\frac{s - \dfrac{f\,Y_c}{a\,X_c}}{s + \dfrac{f\,Z_c}{a\,X_c}}$$

in which—

e^1—apparent efficiency.

e—true efficiency.

s^1—apparent slip.

s—true slip.

and the other symbols as before.

These are shown plotted in Fig. 8, in which is also given the effective thrust horse power of each screw, this being simply the product of the apparent efficiency of the screw, by the B.H.P. absorbed by it.

The propulsive efficiency is of course the $\dfrac{\text{Thrust H.P.}}{\text{Ind. H.P.}}$

and is shown plotted for both series of trials, assuming that in the case of the November trials, the propeller losses are the same as computed for the February trials.

This, then, is the power available for propelling the vessel, and it is necessary to determine how it is used.

DISTRIBUTION OF POWER AVAILABLE FOR PROPULSION.

The ship resistances may be considered generally to consist of skin friction and wave resistance. Of these the wave resistance is the more uncertain to deal with. It is usually obtained by computing the skin friction H.P. from Froude's or Tideman's co-efficients, the balance of the H.P. available for propulsion being considered as overcoming wave, or wave and eddy resistance. If this is done for the case in hand, it is at once seen that the residuary resistance is abnormal for such a vessel.

The vessel is very fine, and is not driven at an excessive speed for its length; accordingly, the wave resistance should not be so great a proportion of the available power.

This leads the writer to look for some other cause for the high resistance shown, and he has come to the conclusion that the abnormally high resistance is due to the action of the pulling propeller in throwing a stream of water back against the hull of the vessel. Ordinarily the skin friction depends on the speed of the vessel as observed, but in this case it can be readily understood that the vessel is entering water which already has motion relative to it. This effect is shown in the trials of the s.s. "Lady Northcote," where about 40 per cent. more power is developed when the propeller is pulling than when pushing.

Before the skin friction can be properly estimated, it is necessary to know the speed of the water from the pulling screw.

In Fig. 7 are given curves of true slip and virtual wake factors on the assumed values of "a."

If $V' =$ the observed speed of the ship

$w' =$ the virtual wake factor

the rate of feed of water to the pulling screw is

$$V' - (V' \times w')$$

The increase of velocity of the feed after passing through the screw will be—

$$s \left\{ V' - (V' \times w'') \right\}$$

where s is the true slip.

The actual speed of the water impinging on the vessel is—

$$(s + 1) \left\{ V' - (V \times w') \right\} = V''$$

This speed V'' is shown in Fig. 8 for values of "a" equal to 6 and 7.2.

It can be safely assumed that the greater amount of the resistance of a vessel of the form and speed under consideration is that due to skin friction. The wave and eddy resistance may be affected to some extent by the action of the pulling screw, but the writer has preferred to consider the whole of the increased resistance to be due to the stream of water from the pulling screw, acting over the full draft of the vessel. The wave résistancse H.P. may be computed by the following formula due to Tavlor—

$$R_w = 0{\cdot}00307 \text{ b } V^5 \frac{D^{\frac{2}{3}}}{L}$$

in which b may be given a value for this case of 0.4.

This may be plotted as shown in Fig. 9, and the balance of the propulsive power is absorbed bv the skin resistance and its augmentation due to the pulling screw. If this augmentation be considered as all skin resistance (which will be taken to cover the augmentation due to eddies, as the stream of water is not the full draft of the vessel) a basis of comparison is readily obtained, which is sufficient for the practical purposes of design.

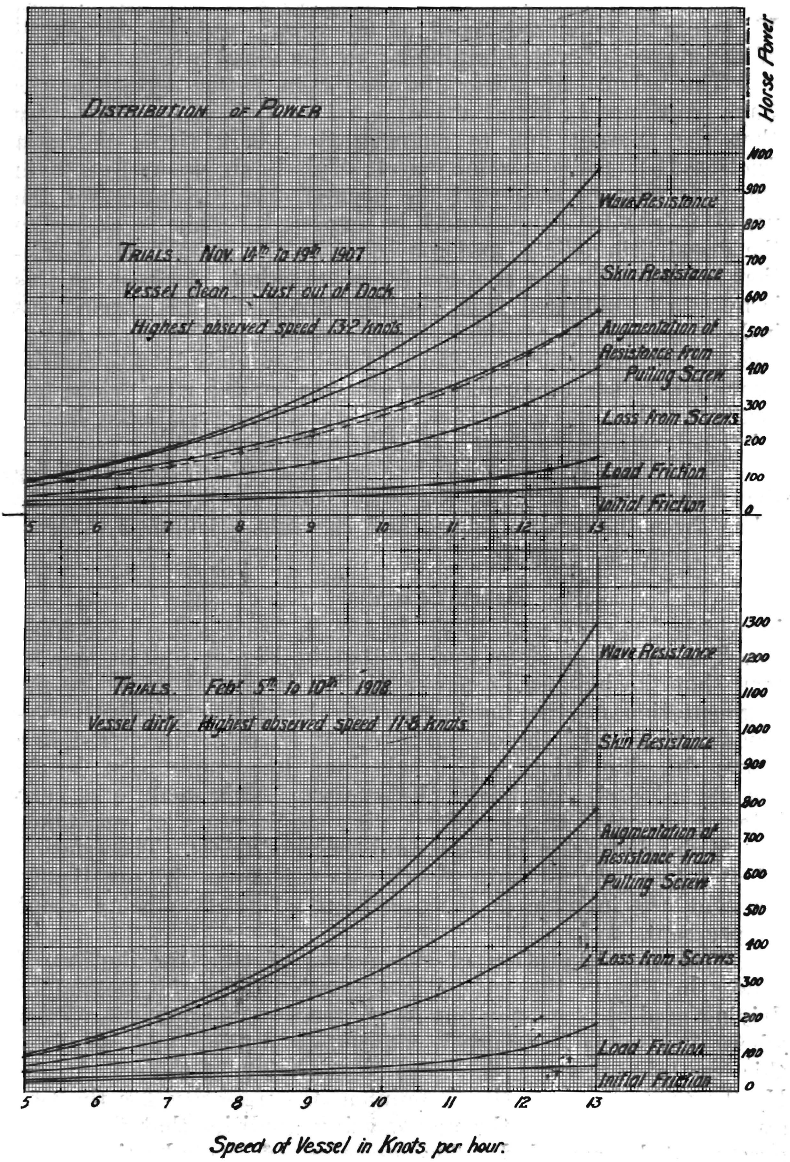

FIG. 9.

The skin resistance H.P. may be taken to vary as $V^{2.83}$ and so an expression for the augmentation of skin resistance for a vessel of this type, over that necessary for a vessel without a pulling screw, may be written:—

$$\text{Augmentation factor } k = \frac{V''^{\,2\cdot83}}{V'^{\,2\cdot83}} - 1$$

V' being the observed speed of the vessel, and V'' the speed of the stream of water impinging on the vessel's bow.

The balance of power absorbed then (after deducting the wave resistance as found above) must be divided in such a way that the amount due to augmentation, is k times the skin friction H.P. for a normal vessel. This is shown in Fig. 9, in which it may be seen that the normal skin resistance for the vessel is greater than would be expected. This is owing to the vessel being dirty, and consequently the co-efficient of friction is higher than is adopted for clean skins. A check on this is now to be obtained from the first series of trials carried out in November, 1907, at which time the vessel had just come out of dock. The propulsive co-efficient for these trials is obtained by assuming the same propeller efficiency as for the later trials, but modifying the engine efficiency in accordance with the initial friction losses shown by the M.E.P. curve in Fig. 1.

The wave resistance will be the same in both cases, but for the first series of trials the co-efficient of friction may be taken from Tideman's tables, and the skin friction H.P. computed from the formula,

$$R^1 = 0\cdot00307 \text{ f S } V^{1\,2\cdot83}$$

in which f for a vessel 190ft. in length has a value of 0.0094.

This has been done and plotted in Fig. 9. and it can be seen that the balance of power (which is now that due to the augmentation of the pulling screw) bears a very close comparison to that of the second series of trials. The dotted line shows the skin friction H.P. obtained by computation, as in the second series of trials (February). This enables a very interesting point to be noticed, viz., the increase of resistance due to the dirty skin in the second series of trials. The relation of the two normal skin frictional resistances is practically constant, the increase in the co-efficient of friction being about 60 per cent.

From an inspection of the curves of the Distribution of Power in Fig. 9, the serious loss occasioned by the method of driving used at present in our ferry steamers, is apparent. Even where one screw is adopted, alternately pulling and pushing, this loss takes place, as is shown in the trials of the s.s. "Lady Northcote." In the writer's opinion, the necessity for the two screws, on account of the manœuvring qualities that they possess, must be conceded. The question of the adoption of a different form of drive, must be decided by further investigation, having for its object the more definite fixing of the quantities dealt with in this paper, in order that an estimate of the first cost and running economies to be effected by any change, may be obtained.

It is with the idea of calling the attention of the engineers engaged in the running of the ferries of Sydney Harbour, to this question of the most economical form of drive for the type of vessel that has been evolved to suit the conditions of the Harbour traffic, that the writer has brought forward these results, knowing that they are derived from data which were far from complete, but from which he considers it possible to obtain some useful ideas of the conditions under which these vessels are working, and also in the hope that in future trial trips of these vessels the opportunity will be taken to obtain full data on the lines here laid down.

The experiments formed part of the Laboratory work, arising in connection with the Mechanical Engineering Seminary, conducted by Prof. S. H. Barraclough and the writer, who desires to thank Prof. Barraclough for much assistance and advice, and also Prof. Warren, by whose courtesy the apparatus was constructed in the Worshops of the P. N. Russell School of Engineering.

The writer wishes to thank Messrs. Flashman and Carter, who, as students in Engineering at the University of Sydney, designed and made the torsion-meter, and carried out the trials, and also those students who assisted them.

Sincere thanks are also given to the Directors of the Ferry Companies concerned, and to Mr. Brown, Superintendent Engineer to the Sydney Ferries, Ltd., and to Mr. Hopkins, Superintendent Engineer to the Port Jackson Steam Navigation Coy., Ltd., whose hearty co-operation made the trials so far carried out, possible.

APPENDIX.

Diagrams, taken from Mr. H. G. Carter's Thesis on the trials on the s.s. "Bingarra," are shown in Figs. 10 and 11.

These show the torsional vibrations of the shaft at different speeds of revolution. The range of speed is from 62 to 130 revolutions per minute, each diagram differing by about 10 revolutions within that range. The curves from both shafts are given, but those from the forward end are very flat (due probably to there being a node at the point of measurement), and show the vibrations very little, and so do not lend themselves to comment.

At 61.7 revolutions per minute, the curve is very flat, but the ratio of maximum torque to mean torque is as much as 2.6. It is difficult to see how many periods there are to one revolution, but there are probably eight or more. At 70.4 revolutions the curve shows eight periods very distinctly. The zero line is crossed at one point, and the ratio of T max. to T mean is 1.77.

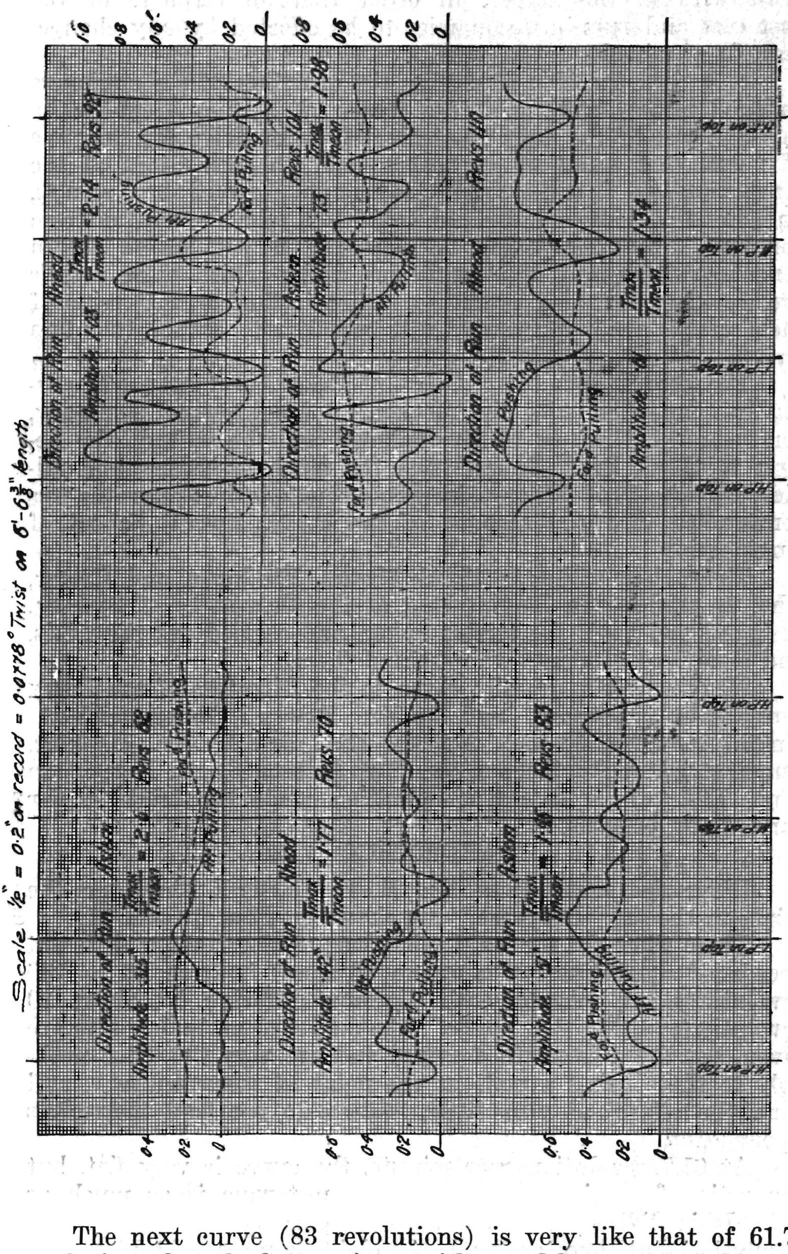

FIG. 10.

The next curve (83 revolutions) is very like that of 61.7 revolutions, but the harmonics at either end have a much bigger amplitude. Eight harmonics are still evident, but some are getting smoothed out considerably.

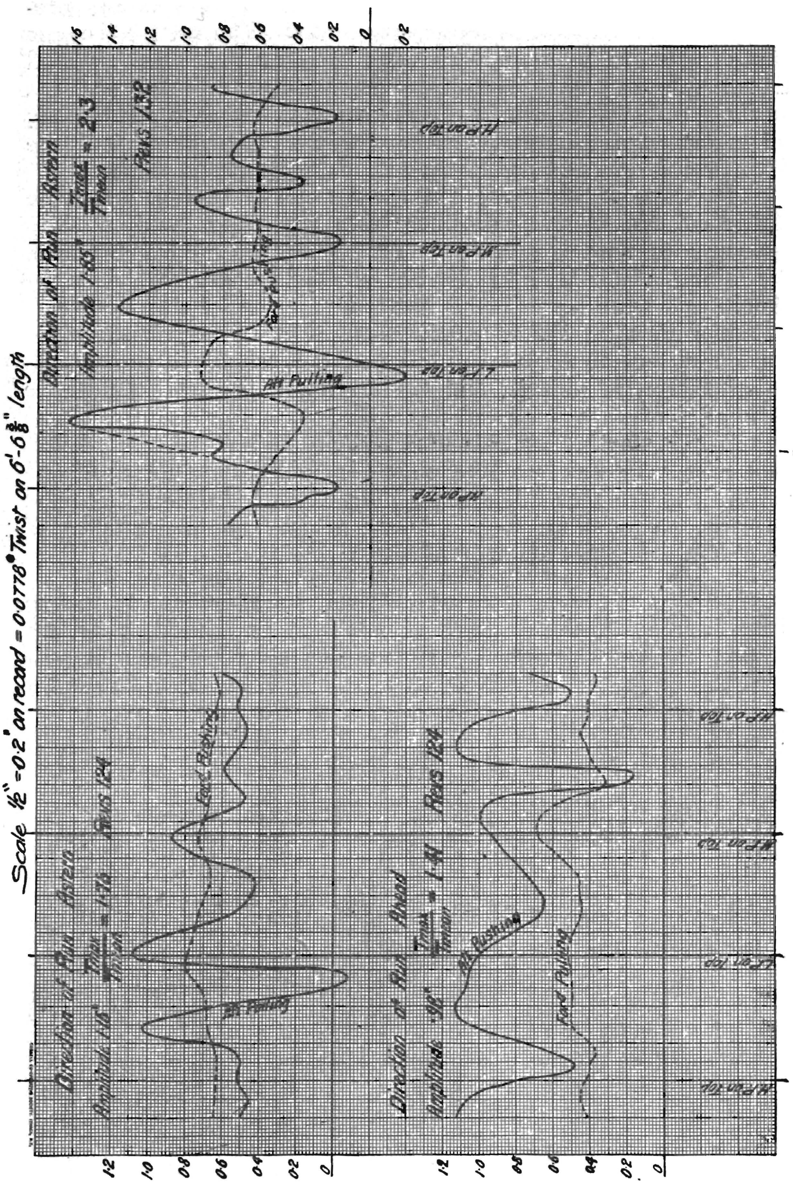

FIG. 11.

At 92.3 revolutions, the sixth harmonic is very prominent and the vibrations are distinctly bad. The ratio of T max. to T mean is 2.14, and the amplitude is 1.03in. The minimum torque is — 0.04in.

The curve at 101.3 revolutions is not so bad as the previous one, the amplitude being only 0.73in. The sixth harmonic is still prominent, but it is gradually coming to a fourth period vibration. At 110.5 revolutions the amplitude is only 0.61in., and the vibrations are not very serious, and are not nearly so sudden as in the two previous cases. The curve is of the fifth period, but the five crests are not very pronounced.

Two sets of curves at 124 revolutions are given, one from the ahead record and the other from the astern, and it is interesting to note the different shapes of the curves for each direction. The sixth harmonic can still be observed, but the fourth is beginning to make itself felt.

The curve at 132 revolutions shows the worst vibration recorded on any of the trials. The period is wholly of the fourth, being 1.85in. amplitude, and the minimum twist — 0.2in., the ratio of T max. to T mean being 2.3.

In the starting records it is possible to see these vibrations changing very easily. The curve is fairly flat till about 45 revolutions are reached, when a vibration of the twelfth period appears, diminishing to the lower periods as the number of revolutions per minute increases.