# CONTROLLABLE PITCH PROPELLERS IN LARGE STERN TRAWLERS

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A study has been carried out into the performance of controllable pitch propellers on distant water stern trawlers. Propeller chart analysis has been substantiated by full-scale trials on existing vessels, in both the free-running and towing conditions. From these trials wake fractions and thrust deduction factors have been obtained.

It has been shown that over the range of propeller diameters considered there is a continuous economic improvement as the diameter increases. The use of propellers of the maximum size allowed by current design practice would achieve a net annual saving, taking account of capital depreciation, between 10 per cent and 15 per cent of the annual fuel costs of typical vessels now in service. On some designs the higher efficiency could introduce additional savings in capital cost by allowing the choice of a smaller engine.

The higher efficiency of larger propellers is fairly common knowledge. The significance of this study is that the improvement has been expressed in economic terms, leading to design recommendations.

Much more basic work is needed in this field, in the form of model testing and ship design, since there is a case for re-considering the design limitations on propeller diameter for this class of vessel.

## INTRODUCTION

Over the last four years or so, a series of investigations has been carried out into the performance of commercial fishing vessels, and their main auxiliary machinery. These investigations have been made under commercial fishing conditions.

In some cases the information obtained in these investigations is relatively simple design data, for example on the maximum expected loading on winches, maximum power required for shooting a particular type of trawl, etc., etc. In other cases the data are required for much more complex studies concerned with the optimization of ship and machinery design, and these studies frequently employ operational research techniques. One important feature of this continuing programme has been the study of the performance of controllable pitch propellers on large stern trawlers.

A deep sea trawler has several régimes of operation: steaming, dodging weather, fishing and laying-to; the fishing cycle comprises a complicated series of manœuvres and includes towing the trawl at speeds of advance of three or four knots for two to three hours at a time. Fig. 1(a) shows the variation in power demand over a typical winter voyage of a large stern trawler to the Davis Straits. The breakdown of power requirements during a typical fishing cycle of shooting, towing, hauling and gear handling is shown in Fig. 1(b). The propeller pitch variations (if any) and the rev/min variations associated with these fluctuations in power demand will depend upon the type of propulsion and control system fitted. In spite of this varying pattern of power requirements and speeds of advance, most distant water trawlers built up to ten years ago were fitted with propellers designed, by and large, for the maximum power, free-running condition, because of the overriding need for speed on passage with a steadily spoiling catch. The traditional trawlers, preserving their catch in crushed ice, are now being replaced by stern trawlers, most of which deep-freeze their entire catch at sea. These freezer trawlers are able to spend a much larger proportion of their time on the fishing grounds and, at the same time, there is much less need for a high free-running speed.

There is a very strong case, therefore, for a reappraisal of propeller design. Controllable pitch (c.p.) propellers have been fitted to the majority of new freezer trawlers in order that the very large auxiliary generators required to supply power for the freezing plant can be driven off the main engine and so allow a neat, economical propulsion machinery and generator arrangement of which Fig. 2 is typical.

## 1) THE VESSELS AND THEIR PATTERN OF OPERATION

With the exception of three large factory ships, the stern trawlers built to date for British owners are between 1000 and 2000 tons displacement and from 145 ft to 215 ft length b.p. Shaft horsepowers of between 1200 and 2200 have been installed, giving calm water free-running speeds of between 12 and 16 knots, and maximum free-running propeller speeds of 175 to 300 rev/min have been employed.

For the most part the ships operate in the North West Atlantic, fishing the grounds in the Davis Straits, off Newfoundland and off Labrador. The average voyage length is, currently, about 48 days. From data logger results the following typical activity breakdown has been calculated:

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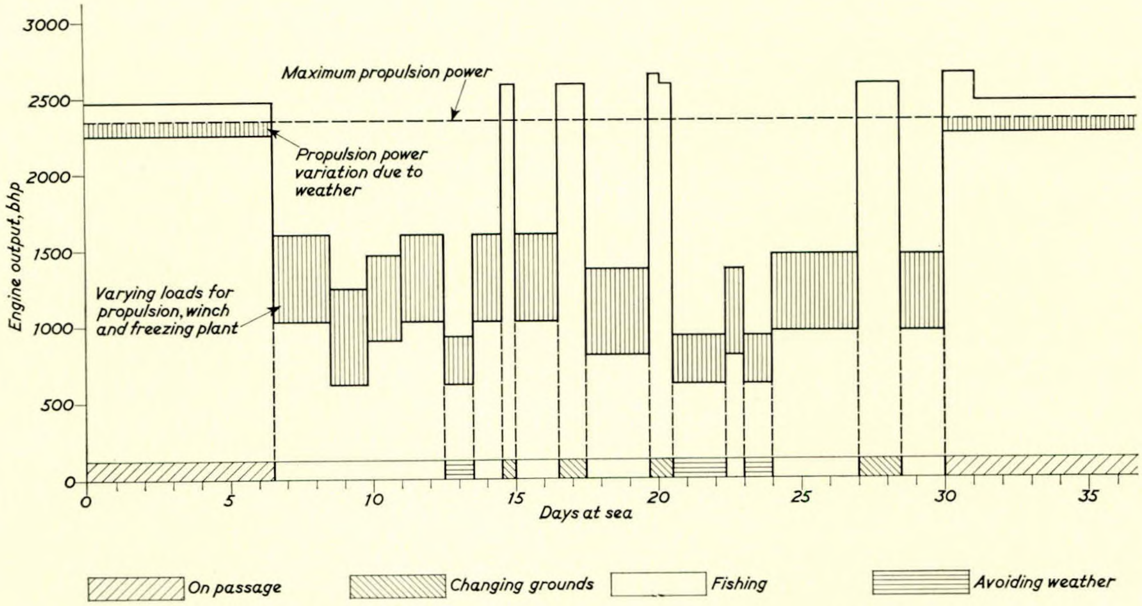



FIG. 1(a)—*Power demand throughout a voyage to Davis Straits of a large stern trawler*

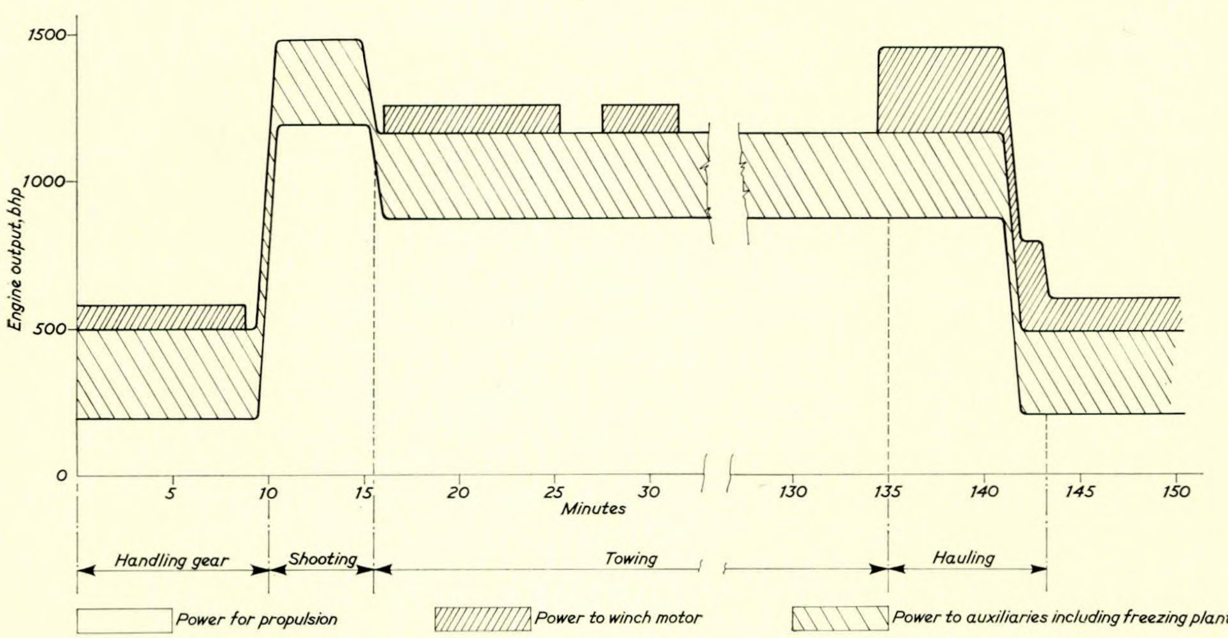



FIG. 1(b)—*Power demand and distribution during one typical fishing cycle in a freezer trawler*

TABLE I

| | Free running | Towing | Shooting the trawl | Hauling the trawl | Dodging or laid |
|---|---|---|---|---|---|
| Mean shp 190 ft vessel | 1450 | 750 | 1100 | 300 | 200 |
| Mean speed (knots) 190 ft vessel | 12·75 | 4·0 | 9·0 | 2·0 | — |
| Mean shp 212 ft vessel | 2000 | 870 | 1460 | 600 | 300 |
| Mean speed (knots) 212 ft vessel | 13·10 | 4·0 | 8·25 | 2·0 | — |

| | | |
|---|---|---|
| Free running (either to or from the grounds or changing grounds) | 100 | days p.a. |
| Towing the trawl ... ... ... | 165 | |
| Handling the gear (shooting, hauling, handling on deck) ... ... | 32 | |
| "Dodging" bad weather or laid for mending or gutting ... ... | 27 | |
| In port ... ... ... ... | 41 | |
| | 365 | |

Table I shows the average propeller powers and ship speeds associated with various activities for two of the vessels on which data-logging programmes have been conducted. One of these vessels is 190 ft/long, b.p., with a maximum propeller

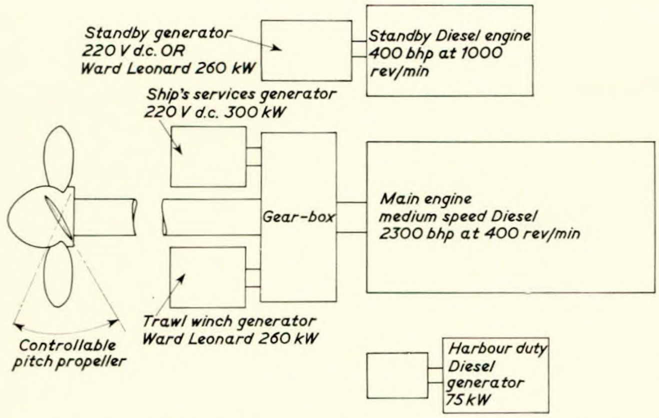



FIG. 2—*Schematic diagram of machinery layout for stern freezer trawler with medium speed Diesel engine driving controllable pitch propeller through a gear-box*

shaft horsepower of 1600. The other, 212 ft/long b.p. has a maximum of 2200 shp to the propeller.

The data logger results also give the statistical distributions of powers, and, of course, other quantities in addition to the mean values quoted in Table I.

There are wide variations in the design of different ships for essentially similar duties. Even for ships of a given size, different owners have different views on the best free-running speed. The problem of choosing the optimum design parameters for the ship as a whole, for a given duty, is particularly complex and it is necessary to take into account a large number of variables in attempting to arrive at a solution. Operations Research techniques are being employed on this particular problem.

The work discussed in this paper has a more limited objective. Its principal aim is to provide design information that will enable the naval architect to select the propeller giving the best economic results, for any ship within the likely range of size, speed, etc.

2) GENERAL METHOD OF APPROACH

In the case of more conventional types of vessel such as cargo ships and the like, a study of this type could be carried out almost entirely by reference to the relevant propeller charts, substantiated, perhaps, by model tests for new types of hull form. In the case of the trawler, however, this was found to be impossible at the outset since there is an almost total lack of adequate propeller chart data for the towing condition, namely high thrust at low forward speeds. Some propeller charts were found to be capable of extrapolation into the area in question. It was considered essential, however, to substantiate such extrapolations by full-scale sea trials on typical existing vessels, and to assess wake fractions, a knowledge of which is necessary to the use of the propeller charts, from model tests.

3) INSTRUMENTATION FOR SEA TRIALS

Before discussing the performance calculations and the trials, it is worth considering the instrumentation employed in the full scale experiments, at sea. This has formed the subject of a separate paper[1].

To quite a large extent the instrumentation used in fishing vessels is identical to that employed in other applications, including other types of ship. Some special problems have been met with, associated with the environment, poor voltage regulation etc. and a process of development and improvement has been in hand, more or less continuously, over a period of four years.

In all cases the instrumentation systems consist of a series of transducers, with electrical outputs, feeding a recorder, usually a multi-channel ultra violet trace recorder. Analysis of the records is eased by the use of a semi-automatic trace analyser.

For performance measurements taken regularly in service, over many months of operation, automatic data-logging equipment is employed. These data loggers are used with the same transducers as those for trials purposes. They give a punched-tape output which is processed on a digital computer. A detailed description of these loggers, and the various purposes for which they have been installed, is given in reference (2).

To measure propeller shaft torque and thrust it was necessary to develop a type of transducer which seems to be novel in marine applications. This need arose out of the difficulties experienced with commercial torsionmeters clamped to the shaft, including serious zero drift errors which occurred when using them on direct-coupled motor ships subject to frequent manœuvring on voyages lasting several weeks. The most successful trawler installation was on a Diesel-electric ship, in which negligible zero drift was experienced on a six week voyage. It is at least probable that the firing order torque oscillations on the shafts of direct-coupled motor ships (i.e. the vast majority of fishing vessels) were responsible for the poor results. This may have been interesting to follow up, but in any event, no torsionmeters were commercially available for shafts smaller than about 6 inches diameter. Since it was the intention to carry out work on small vessels, with shafts down to 3 inches diameter and less, and since there remained the problem of measuring thrust, it was decided to develop a system capable of meeting these various requirements.

A description of the system adopted, and its development, is given in reference (3). Basically the transducer consists of a hollow insert shaft, usually installed in the vessel's intermediate shafting. It is strain gauged for torque and for thrust. In each case a ring of gauges is installed round a diameter, to integrate out any effects of maldistribution of loading on the coupling bolts. The signals are taken off sliprings formed by wrapping silver strip round the shaft, on rubber insulation. The shaft is hollow in order to raise the axial stress to an adequate value for the measurement of thrust. To comply with stressing requirements, therefore, the shaft has to be made in steel of 40 tons U.T.S., or greater.

Calibration of these transducers in torque is carried out

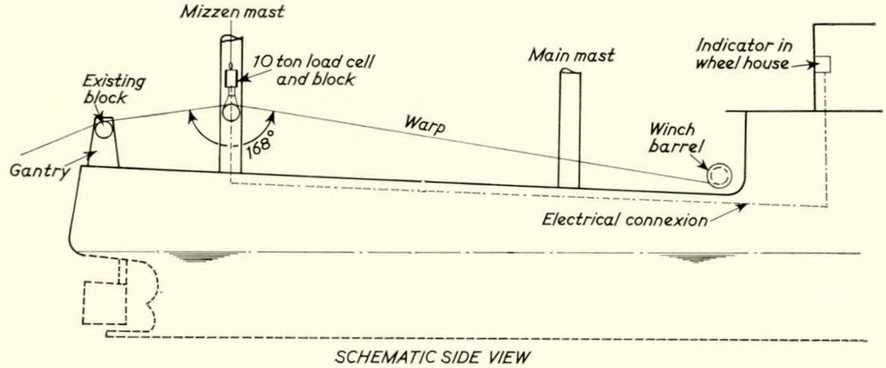



FIG. 3—*Layout of trawl warp load meter*

in a specially designed torsion test rig[2] and in thrust in a standard compression test machine.

For the measurement of tension in the towing warps another strain gauge system was developed; in this case the system is now used commercially as a fishing aid. The principle is shown in Fig. 3. Referring to this diagram it can be seen that the warp, which would normally have passed from the winch to the rear gantry pulley, has been deflected through a fixed angle by passing it over another pulley suspended from the mizzen-mast. The load in the support for this pulley is thus proportional to the warp tension and is measured on a strain gauge load cell.

Table II shows a typical list of measurements which would be taken on a given performance trial. It also lists the transducers used and indicates the method of calibration.

TABLE II

| Measurement | Transducer | Calibration |
|---|---|---|
| Torque | Hollow insert shaft | Torsion test rig |
| Thrust | Hollow insert shaft | Compression test machine |
| Ship speed | (1) Signal from ship's instrument | On measured mile |
| | (2) Towed log (low speeds only) | Towing tank |
| Propeller rev/min | Pulse switch on shaft | Against stop watch |
| Propeller pitch | Potentiometer on mechanism | Against maker's graduations. Checked in dry dock. |
| Warp loads (2) | See Fig. 3 | Spring balance |
| Pitch and roll | Gyros with electrical output | Tilting table |
| Wind speed | Anemometer | Maker's calibration |
| Relative wind Direction | Wind vane with electrical output | Manual setting |

4) PRELIMINARY INVESTIGATIONS

The first British-built distant water stern trawler to be fitted with a controllable pitch propeller was completed in late 1964 by Ferguson Brothers of Port Glasgow for Thos. Hamling and Co. Ltd. of Hull. The ship, *Saint Finbarr,* was 185 ft length b.p. and had a displacement of 1432 tons. Unlike most of the subsequent ships of this type to be built, her engine was direct-coupled to the propeller. The engine, a Mirrlees type KLSSM8, was capable of delivering 1592 bhp at 300 rev/min maximum. The propeller was a Stone-Manganese controllable pitch type with the following main features:

| | |
|---|---|
| Diameter ... ... ... ... | 7·4 ft |
| Expanded blade area ratio ... | 0·494 |
| Pitch/diameter ratio ... ... | 0·827 |
| *Four blades* | |
| Blade thickness ratio ... ... | 0·038 |
| Boss diameter ratio ... ... | 0·344 |

Measured mile trials were carried out on this vessel and two series of towing trials in the Farne Deeps off the coast of Northumberland in 1965 and early 1966. There was one serious limitation in the use of this vessel to establish correlations with extrapolated propeller charts. This arose from the fact that at shaft speeds below the designed free-running value, the maximum permitted torque from the engine was limited along a curve which dropped off sharply as the rev/min was reduced. The net result of this was that it was only possible to cover a relatively small rev/min range when towing the small Granton trawl used by the bulk of the British distant water fleet. The information obtained was therefore somewhat limited for the purpose of producing general design recommendations but the results were reasonable, and striking enough to form an inducement to further work.

These earlier results confirmed, for example, that for a similar vessel equipped with the machinery layout of Fig. 2, that is with a gear-box between engine and propeller, there would be a steady improvement in annual fuel consumption with increase in propeller diameter from the 7·4 ft diameter *Saint Finbarr* to about 10·25 ft which was the maximum propeller diameter which one shipbuilder considered feasible for that size and type of vessel, and indicated what the value of such an alteration in choice would be. The 10·25 ft diameter propeller would result in a reduction in annual fuel costs of approximately £2000 by comparison with the smaller one at 7·4 ft diameter. This improvement, which amounts to about 12 per cent of the annual fuel bill, would be achieved for a capital expenditure of approximately £4500 which represents the difference in price of the larger propeller. Quite apart from the incidental benefits of smaller fuel bunkers, or, alternatively, greater fuel endurance, this is a worthwhile saving in itself and it was decided that the whole question of choice of propeller for this class of vessel warranted a detailed investigation.

In the course of the *Saint Finbarr* trials there were strong indications that for a given thrust there was a fairly sharp optimum combination of propeller pitch and rev/min to give minimum power and fuel consumption. There was furthermore the possibility that fuel consumption might be further reduced by suitable matching of the engine and propeller by tuning for minimum fuel consumption at the optimum rev/min. The rev/min/torque limitations of the engine again reduced the scope of the investigations.

A larger stern trawler, *Arctic Freebooter,* with an engine configuration which produced a much flatter rev/min/torque relationship, came into service, and it was decided to concentrate the next phase of the studies on this vessel.

5) TRIALS ON ARCTIC FREEBOOTER

This vessel is an all-freezing stern trawler built by Goole Shipbuilding Co. for Boyd Line Ltd. of Hull. She was delivered in January 1966. She is 212 ft length b.p. and has a displacement of 1910 tons. The main machinery is a Mirrlees National Monarch engine type ALSSM6 delivering 2380 bhp at the maximum designed shaft speed of 275 rev/min. The machinery is of the same general configuration as in Fig. 2 but the gear ratio between engine and propeller shaft is 1:1.

The ship is fitted with a Liaaen controllable pitch propeller, the main features of which are:

| | |
|---|---|
| Diameter ... ... ... | 9·187 ft |
| Expanded blade area ratio | 0·45 |
| Pitch/diameter ratio ... | 0·70 full power running free in calm water |
| *Three blades* | |
| Blade thickness ratio ... | 0·056 |
| Boss diameter ratio ... | 0·286 |

Propeller rev/min and pitch can be controlled from the bridge or from the engine room. Full range of pitch/diameter ratio can be obtained from 0 to $\pm 0{\cdot}7$. In the engine room the rev/min can be controlled from 180 to 275 rev/min but bridge control of rev/min is limited to 230 to 275 rev/min because of the voltage control limitations in the electrical equipment. By running standby sets it was possible, for trials purposes, to use the 180 to 275 rev/min range available on engine room control.

*Trials instrumentation*

The measurements made during the trials were those listed in Table II.

From the point of view of correlation with propeller charts the accuracy of certain of these instruments is of considerable importance. Shaft rev/min is considered to be accurate to better than ½ per cent. With calibrations just before the trial, torque and ship's speed (by the towed log) are considered to be within $\pm$ 1½ and $\pm$ 2¼ per cent depending upon the level of thrust stress levels; thrust is somewhat less accurate, probably between $\pm$ 1½ and $\pm$ 2¼ per cent depending upon the level of thrust being measured.

During the trials an instrumentation fault developed which eliminated the thrust measurement on certain records.

*Trials*

Preliminary trials and data-logging work were carried out on *Arctic Freebooter* throughout 1966. During much of this time, however, the vessel was limited in maximum rev/min because of an axial vibration problem which was corrected in late 1966. A complete trials programme in both the free-running and towing condition was carried out in January 1967 at the Newbiggin measured mile and in the nearby Farne Deeps.

Six double runs were carried out on the Newbiggin measured mile in the free-running condition. Propeller pitch/diameter ratio was held constant at the design value of 0·70. In addition, two double runs were carried out at very low speed to calibrate the ship's speed log for the subsequent towing trials.

The towing trials consisted of groups of four settings of rev/min at each of six different propeller pitch settings. Runs were made in one direction only.

*Trials Results*

The results of the measured mile trials reduced to calm air conditions are shown in Fig. 4.

In the towing condition about 90 per cent of the resistance is due to the drag of the fishing gear. Consequently the thrust required depends primarily on the speed at which the skipper wishes to tow the gear. The results have, therefore, been analysed in such a way as to produce Fig. 5 which relates horizontal pull to propeller shaft horsepower, pitch and rev/min. The speed of the gear is indicated against horizontal pull for tide free conditions.

It is important to note that the effect of current differences between the surface and the bottom will be largely reflected as changes in ship speed with little effect on gear speed. However because of the very small rate of change of the thrust coefficient $K_T$ with propeller advance coefficient $J$ the thrust is fairly constant for a given rev/min over a wide range of ship speed.

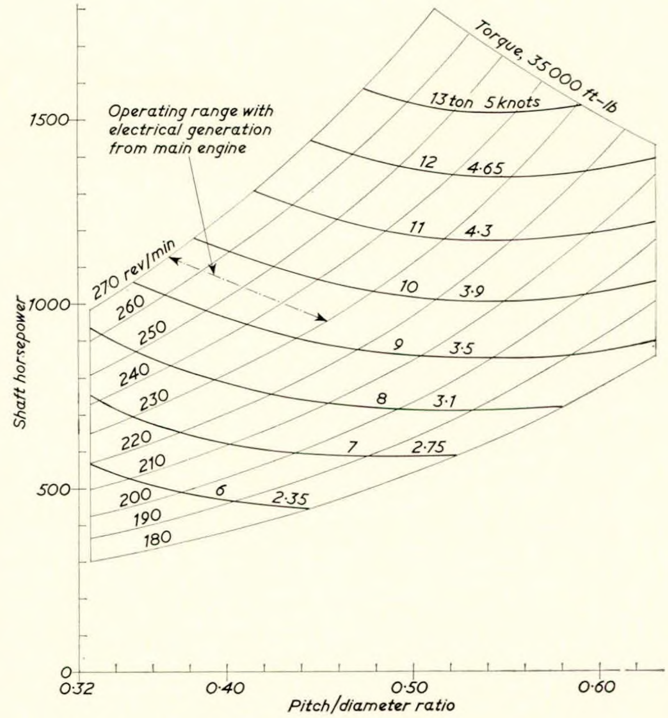



FIG. 5—*Measured towing parameters for controllable pitch propeller on* Arctic Freebooter

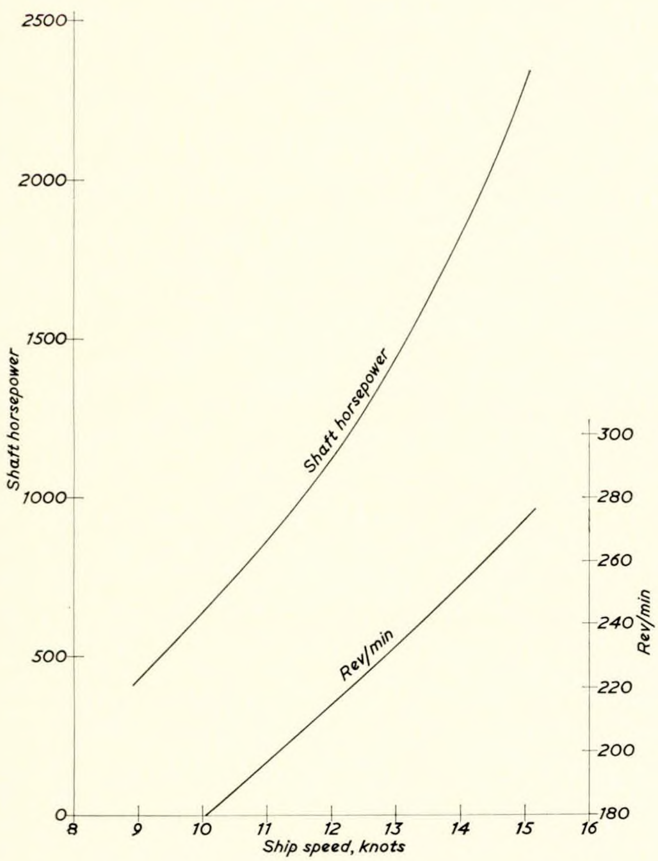



FIG. 4—*Free running measured mile trials results*—Arctic Freebooter

6) OPERATION OF EXISTING PROPELLER FOR MINIMUM FUEL CONSUMPTION

A glance at Fig. 5 will show that for all values of horizontal pull required there is an optimum propeller pitch corresponding to a pitch/diameter ratio of 0·527. On this particular ship the optimum pitch setting cannot be used below a horizontal pull of about 11·5 tons because of the limitation, mentioned earlier, upon minimum rev/min due to the design of the ship's electrical equipment. The range of values required in service lies between seven and ten tons so that for operation at the minimum power condition the rev/min required would be in the range 180 to 215. If it were possible to operate in that region then the power required during towing would, on average, be of the order of100 horsepower less than the value now being achieved at 250 rev/min, which is the average shaft speed currently being employed. This would represent an annual saving of about £550 in fuel costs. The additional capital cost involved in providing slightly larger generators to allow operation at the optimum rev/min range is quite small, between £500 and £900.

Ships have been built, and are being built, in which the auxiliaries are separately driven. For example, some owners are turning to a.c. electrical power and for this purpose are installing constant speed auxiliaries. For vessels of this type the sort of results shown in Fig. 5 could be used directly as a guide to the best method of operation for minimum fuel consumption.

It can be seen from the foregoing that whether the choice of rev/min is wide, or is restricted by electrical control limitations, the skipper or engineer has a noticeable influence on the fuel costs of his vessel through the way in which he uses his pitch and rev/min controls. On a vessel of the size of *Arctic Freebooter,* if the rev/min variation allowed is more than 33 per cent of maximum, the difference between best and worst practice would amount to £1000 per annum. The various requirements of towing, free running, and manœuvring make it somewhat difficult to provide a single lever control system. In

TABLE IIIA—ANALYSIS OF FULL SCALE DATA—FREE RUNNING
P/D ship=0·70 P/D chart=0·714

| Run No. | Rev/min | Torque ft lb | Thrust tons | Ship speed knots | $K_Q$ ship | $K_T$ ship | $K_Q$ chart | $K_T$ chart | From $K_Q$ and P/D $W_t$ | From $K_T$ and P/D $W_t$ | From $K_T$ and $K_Q$ $W_t$ |
|---|---|---|---|---|---|---|---|---|---|---|---|
| 1 | 230 | 32 000 | * | 12·30 | 0·01688 | | 0·01772 | | 0·247 | | |
| 2 | 227 | 29 900 | * | 12·65 | 0·01620 | | 0·01700 | | 0·241 | | |
| 3 | 267 | 43 000 | 16·95 | 14·35 | 0·01683 | 0·1365 | 0·01767 | 0·1421 | 0·246 | 0·253 | 0·262 |
| 4 | 266 | 41 850 | 16·27 | 14·61 | 0·01650 | 0·1320 | 0·01733 | 0·1374 | 0·244 | 0·247 | 0·247 |
| 5 | 205 | 25 500 | 10·28 | 11·03 | 0·01693 | 0·1418 | 0·01778 | 0·1462 | 0·252 | 0·271 | 0·296 |
| 6 | 204 | 24 100 | * | 11·42 | 0·01615 | | 0·01696 | | 0·241 | | |
| 7 | 182 | 20 350 | * | 9·71 | 0·01714 | | 0·01800 | | 0·255 | | |
| 8 | 180 | 19 050 | 7·52 | 10·20 | 0·01640 | 0·1332 | 0·01722 | 0·1387 | 0·264 | 0·277 | 0·293 |
| 9 | 248 | 37 650 | 14·87 | 13·23 | 0·01708 | 0·1397 | 0·01793 | 0·1454 | 0·252 | 0·261 | 0·267 |
| 10 | 248 | 35 850 | 14·35 | 13·71 | 0·01626 | 0·1339 | 0·01707 | 0·1384 | 0·237 | 0·257 | 0·280 |
| 11 | 235 | 32 100 | * | 13·05 | 0·01621 | | 0·01702 | | 0·239 | | |
| 12 | 237 | 34 600 | 13·40 | 12·50 | 0·01719 | 0·1369 | 0·01805 | 0·1425 | 0·250 | 0·240 | 0·222 |

* Not recorded

TABLE IIIB—ANALYSIS OF FULL SCALE DATA—TOWING

| Run No. | Rev/ min | Torque ft lb | Thrust tons | Ship speed knots | P/D ship | $K_Q$ ship | $K_T$ ship and chart | P/D chart | $K_Q$ chart | From $K_Q$ and P/D $W_t$ | From $K_T$ and P/D $W_t$ | From $K_T$ and $K_Q$ $W_t$ |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 13 | 182 | 9400 | 6·10 | 1·85 | 0·322 | 0·00792 | 0·1057 | 0·327 | 0·00752 | | | |
| 14 | 210 | 12 500 | 7·25 | 2·42 | 0·322 | 0·00790 | 0·0946 | 0·327 | 0·00751 | | | |
| 15 | 240 | 15 900 | 8·90 | 3·00 | 0·322 | 0·00770 | 0·0887 | 0·327 | 0·00732 | | | |
| 16 | 264 | 19 000 | 10·60 | 3·48 | 0·322 | 0·00760 | 0·0873 | 0·327 | 0·00722 | | | |
| 17 | 182 | 11 700 | 6·70 | 2·28 | 0·402 | 0·00985 | 0·1161 | 0·408 | 0·00936 | 0·276 | 0·270 | 0·248 |
| 18 | 212 | 15 600 | 9·10 | 2·95 | 0·402 | 0·00968 | 0·1162 | 0·408 | 0·00920 | 0·296 | 0·348 | 0·414 |
| 19 | 242 | 20 100 | 11·30 | 3·75 | 0·402 | 0·00958 | 0·1108 | 0·408 | 0·00910 | 0·334 | 0·292 | 0·240 |
| 20 | 265 | 23 350 | 13·10 | 4·32 | 0·402 | 0·00928 | 0·1071 | 0·408 | 0·00882 | 0·283 | 0·261 | 0·232 |
| 21 | 264 | 29 500 | * | 5·31 | 0·464 | 0·01182 | | 0·471 | 0·01122 | 0·284 | | |
| 22 | 242 | 25 100 | 13·25 | 4·58 | 0·464 | 0·01196 | 0·1299 | 0·471 | 0·01136 | 0·272 | 0·263 | 0·258 |
| 23 | 212 | 19 700 | 10·30 | 3·65 | 0·464 | 0·01223 | 0·1315 | 0·471 | 0·01161 | 0·274 | 0·221 | 0·153 |
| 24 | 180 | 14 500 | 7·71 | 2·85 | 0·464 | 0·01248 | 0·1366 | 0·471 | 0·01186 | 0·245 | 0·256 | 0·221 |
| 25 | 240 | 24 800 | 13·16 | 4·60 | 0·464 | 0·01202 | 0·1311 | 0·471 | 0·01142 | 0·296 | 0·296 | 0·296 |
| 26 | 256 | 34 700 | 17·40 | 5·60 | 0·522 | 0·01477 | 0·1524 | 0·530 | 0·01404 | 0·271 | 0·280 | 0·309 |
| 27 | 242 | 31 500 | 15·70 | 5·21 | 0·522 | 0·01501 | 0·1538 | 0·530 | 0·01426 | 0·280 | 0·280 | 0·280 |
| 28 | 210 | 24 100 | 12·20 | 4·20 | 0·522 | 0·01525 | 0·1588 | 0·530 | 0·01449 | 0·275 | 0·294 | 0·334 |
| 29 | 184 | 18 700 | 9·51 | 3·36 | 0·522 | 0·01541 | 0·1612 | 0·530 | 0·01464 | 0·257 | 0·266 | 0·320 |
| 30 | 181 | 22 100 | 10·40 | 3·80 | 0·586 | 0·01874 | 0·1824 | 0·594 | 0·01784 | 0·236 | 0·244 | 0·254 |
| 31 | 210 | 29 400 | 13·88 | 4·70 | 0·586 | 0·01859 | 0·1807 | 0·594 | 0·01767 | 0·255 | 0·271 | 0·288 |
| 32 | 228 | 34 700 | 16·32 | 5·40 | 0·586 | 0·01862 | 0·1802 | 0·594 | 0·01769 | 0·300 | 0·304 | 0·308 |
| 33 | 181 | 25 400 | 11·40 | 4·07 | 0·627 | 0·02163 | 0·1997 | 0·636 | 0·02055 | 0·305 | 0·266 | 0·211 |
| 34 | 212 | 34 400 | 15·55 | 5·16 | 0·627 | 0·02135 | 0·1986 | 0·636 | 0·02029 | 0·303 | 0·303 | 0·303 |

* Not recorded

the absence of such a system, proper instructions should be given to the captain, particularly since in the past they have been accustomed to using fixed pitch propellers and rev/min has been the only available control.

7) COMPARISON WITH PROPELLER CHART DATA

Obviously full-scale results can apply to one particular vessel only and to one particular propeller. The main purpose in carrying out work of this nature is to be able to apply the results to the design of new vessels of similar general form and method of operation but of differing size, free-running speed, etc. For this purpose use must be made of model-based data in the form of propeller charts. As mentioned earlier, one of the primary objects in carrying out full-scale trials is to give confidence in the use of such data and in particular in any extrapolations which have to be made to the existing results.

Although the data obtained were limited in quantity and scope it was decided to attempt a comparison with the Troost propeller charts, making corrections for the differences in boss/diameter ratio and the blade-thickness/diameter ratio where required.

The investigations resulted in the chart shown in Fig. 6

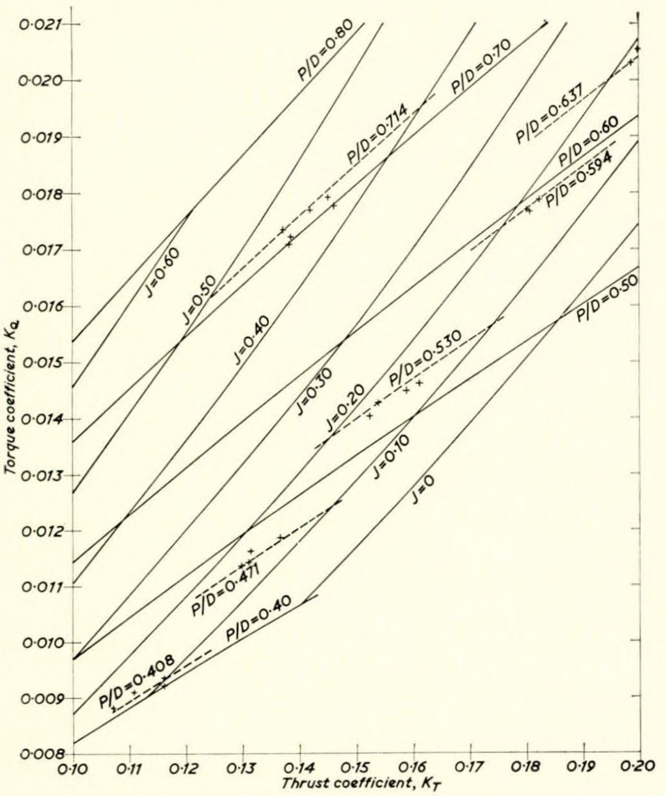



FIG. 6—Arctic Freebooter—*Trials data compared with modified Troost propeller chart data*

in which $K_Q$ is plotted against $K_T$. The reasoning behind this choice of parameters and the corrections made in arriving at the results are contained in Appendix I.

In Fig. 6 the full lines show the propeller chart data and the plotted points and dotted lines indicate the experimental results given in Table III(b). The good agreement between the experimental results and the model-based chart data can be seen from the figure. Although this degree of correlation has been obtained for one ship only, with one propeller, it is considered that the results are sufficiently encouraging for similar investigations e.g., on other sizes of ships, to proceed based upon the chart information.

It will be seen that propeller chart data for pitch/diameter ratios down to 0·4 have been given although the lowest value covered by the Troost B3 series is 0·5. The values at 0·4 have been obtained by extrapolation. This extension of the propeller chart data was felt necessary since the experimental data showed that the best pitch setting corresponded to a pitch/diameter ratio of 0·530. Using this propeller chart at the design stage to determine the best value of pitch/diameter ratio for towing would not have been altogether satisfactory because of the lack of data below a pitch/diameter ratio of 0·5. Nearly all of the towing experimental data can now be plotted and the agreement in the extrapolated region is encouraging. However there is a clear need for data at these low values of pitch/diamter ratio.

*Wake Fraction and Thrust Deduction Factor*

Wake fraction has been determined from the experimental data on the basis of thrust identity, using the chart data described earlier. The chart allows the analysis to be carried out in three different ways:

1) Using $K_Q$ and pitch/diameter ratio.
2) Using $K_T$ and pitch/diameter ratio.
3) Using $K_Q$ and $K_T$.

For the free-running trials at a pitch/diameter ratio of 0·714 the resulting wake fractions using method (1) are very consistent, falling in the range 0·237 to 0·264 with a mean value of 0·247 [see Table III (a)]. These compare very well with the model experiments carried out at the National Physical Laboratory which gave values of about 0·25. Using methods (2) and (3) resulted in a similar average figure of 0·259 but with considerably more scatter [0·24 to 0·277 for method 2) and 0·222 to 0·296 for method 3)]. Most of this additional scatter can be attributed to the fact that both these methods involved the use of thrust data which as mentioned earlier are slightly less accurate than the torque data.

For the towing trials the derived wake fractions are given in Table IIIB. Using method (1) the values fall in the range 0·245 to 0·334 with an average of 0·28. Methods 2) and 3) give considerably more scatter (0·221 to 0·348 for method 2) and 0·153 to 0·414 for method 3) but similar average values of 0·277 and 0·275 respectively. Once again the methods based on the thrust data give the greatest variation.

It will be noted that the average value of wake fraction in the towing condition is slightly higher than that for free steaming. This was also noted in the model tests where an average wake fraction of 0·314 was obtained for a pitch/diameter ratio of 0·50. For these tests the speed range was equivalent to a ship speed range of 3 to 4 knots. For this speed range the ship results gave an average value of 0·284 using method 1). However, small differences in wake fraction in the towing condition are relatively unimportant since the rate of change of $K_Q$ and $K_T$ with $J$ is small at low values of pitch/diameter ratio.

Nevertheless, it was considered imperative to obtain the necessary correlation between model test results in the towing condition and the full-scale trials, particularly as regards the question of wake fraction. For this reason a series of model tests was commissioned at the N.P.L.

Values of thrust deduction factor were obtained by comparing the measured thrust with an estimate of the total resistance of the vessel and the fishing gear for each run. Most of the resistance is due to the fishing gear and this is obtained from the measured warp pull and vertical warp angle at the towing block. The ship's resistance was estimated from N.P.L. test data for this vessel with a service allowance of 10 per cent. Finally the wind resistance was assessed using N.P.L. wind tunnel test data on a model of a similar vessel.

The results are given in Table III(c) and it will be seen that the thrust deduction factor tends to be lowest at those values of pitch/diameter ratio at which the power required to achieve a given horizontal pull is least, see Fig. 5. This suggests that if a constant thrust deduction factor is assumed at the design stage the optimum pitch setting may not be properly defined.

Tests on a model of this vessel gave values in the range 0·086 to 0·104 for a pitch/diameter ratio of 0·50 and equivalent ship speeds between 3 and 4 knots. These results are only

TABLE IIIc—ANALYSIS OF FULL SCALE DATA—THRUST DEDUCTION FACTORS

| Run No. | Ship speed knots | P/D | Ship resistance tons | Wind resistance tons | Horizontal component warp pull tons | Total resistance tons | Thrust tons | Thrust deduction factor |
|---|---|---|---|---|---|---|---|---|
| 13 | 1·85 | 0·322 | 0·25 | 0·11 | 4·52 | 4·88 | 6·10 | 0·200 |
| 14 | 2·42 | 0·322 | 0·36 | 0·14 | 5·50 | 6·00 | 7·25 | 0·173 |
| 15 | 3·00 | 0·322 | 0·52 | 0·13 | 6·72 | 7·37 | 8·9 | 0·172 |
| 16 | 3·48 | 0·322 | 0·67 | 0·22 | 7·78 | 8·67 | 10·60 | 0·181 |
| 17 | 2·28 | 0·408 | 0·34 | 0·13 | 5·49 | 5·96 | 6·70 | 0·149 |
| 18 | 2·95 | 0·408 | 0·51 | 0·12 | 7·08 | 7·71 | 9·10 | 0·153 |
| 19 | 3·75 | 0·408 | 0·77 | 0·13 | 8·87 | 9·77 | 11·30 | 0·136 |
| 20 | 4·32 | 0·408 | 0·98 | 0·09 | 10·36 | 11·43 | 13·10 | 0·128 |
| 21 | 5·31 | 0·471 | 1·40 | 0·24 | 12·26 | 13·90 | * | |
| 22 | 4·58 | 0·471 | 1·09 | 0·23 | 10·47 | 11·79 | 13·25 | 0·103 |
| 23 | 3·65 | 0·471 | 0·73 | 0·18 | 9·27 | 9·27 | 10·30 | 0·112 |
| 24 | 2·85 | 0·471 | 0·48 | 0·14 | 6·26 | 6·88 | 7·71 | 0·109 |
| 25 | 4·60 | 0·471 | 1·10 | 0·15 | 11·48 | 11·48 | 13·16 | 0·116 |
| 26 | 5·60 | 0·530 | 1·54 | 0·27 | 13·73 | 15·54 | 17·40 | 0·107 |
| 27 | 5·21 | 0·530 | 1·37 | 0·20 | 12·30 | 13·87 | 15·70 | 0·123 |
| 28 | 4·20 | 0·530 | 0·94 | 0·15 | 9·55 | 10·64 | 12·20 | 0·136 |
| 29 | 3·36 | 0·530 | 0·63 | 0·21 | 8·11 | 8·11 | 9·51 | 0·134 |
| 30 | 3·80 | 0·594 | 0·80 | 0·23 | 9·06 | 9·09 | 10·40 | 0·126 |
| 31 | 4·70 | 0·594 | 1·14 | 0·19 | 10·76 | 12·09 | 13·88 | 0·129 |
| 32 | 5·40 | 0·594 | 1·45 | 0·37 | 10·58 | 14·40 | 16·32 | 0·118 |
| 33 | 4·07 | 0·627 | 0·89 | 0·33 | 9·84 | 9·84 | 11·40 | 0·141 |
| 34 | 5·16 | 0·627 | 1·34 | 0·35 | 11·65 | 13·34 | 15·55 | 0·143 |

* Not recorded

slightly lower than those obtained on the full scale trials amounting to a difference of 2 per cent of useful thrust for a given power.

The agreement between model and full scale wake fractions and thrust deduction factors is good, but it must be emphasized that this applies to one set of trials on one ship. For the results to be applied with confidence, to stern trawler designs in general, much more model test and full-scale work is necessary.

These tests in which the drag of the trawl was represented as an external resistance were carried out upon a model of *Arctic Freebooter*. A stock propeller was used and at the time of writing the corrected results related to the controllable pitch propeller and the appropriate pitch/diameter ratios are awaited.

### 8) THE EFFECT OF PROPELLER DIAMETER ON FUEL COSTS FOR A GIVEN DUTY

Having selected a particular type of propeller for the vessel then the biggest single variable which is available to the designer is the diameter. Different shipbuilders have different views upon the maximum propeller diameter which can be employed on a given vessel without noticeably increasing the difficulties and the cost of designing the stern structure. For this and for other reasons a variety of propeller diameters has appeared in the last few years on vessels of similar size and designed for a similar duty. Accordingly, it was decided to examine the economic implications of varying propeller diameter on two large stern trawlers of basically similar designs but of different free running speeds. The following basic data were assumed for the propeller:

| | *215 ft length b.p. vessels* | |
|---|---|---|
| 1) *Free Running* | *Fast* | *Slow* |
| Trials speed | 15·8 knots | 14·25 knots |
| Average speed | 14·3 knots | 12·75 knots |
| Average thrust | 15·9 tons | 10·7 tons |
| Wake fraction | 0·22 | 0·22 |

| | *215 ft length b.p. vessels* | |
|---|---|---|
| 2) *Towing* | *Fast* | *Slow* |
| Average speed | 4·25 knots | 4·25 knots |
| Average thrust | 10·0 tons | 10·0 tons |
| Wake fraction | 0·30 | 0·30 |

Using the propeller chart data described earlier in the paper the propeller efficiency was determined for propeller diameters in the range 8 ft to 13 ft on the basis of cavitation-free operation. For this purpose the limiting value of $J$ for a given duty has been determined together with the other relevant propeller characteristics.

Three different types of propeller operation have been investigated.

a) *Free choice allowed for propeller rev/min for the free running and towing condition.*

For the present this type of operation can be considered to apply only to those vessels for which the electrical supplies are separately driven.

b) *The minimum towing rev/min limited to two-thirds of the maximum free running rev/min.*

The value of two-thirds is that which electrical manufacturers currently set as the minimum for existing voltage control equipment on shaft-driven generators to operate satisfactorily. It would be possible to design control equipment to deal with a wider range of rev/min but this would add considerably to the cost and complexity of the system. Not all manufacturers agree on the two-thirds figure but it has been assumed for the purposes of this paper as a typical figure.

c) *Constant speed operation.*

Although the results quoted earlier in the paper indicate that this is uneconomic in terms of fuel consumption, constant speed operation does have certain advantages in simplicity of the mechanical and electrical control equipment and in addition it could allow the use of a.c. auxiliaries driven directly from the main engine.

The method by which the maximum efficiency and the working pitch/diameter ratios were obtained, is illustrated in Fig. 7.

The relationship between $K_T$ and $J$ is obtained from:

$$K_T = \frac{T}{\rho n^2 D^4}$$

$$J = \frac{V_a}{nD}$$

Hence

$$K_T = \frac{T}{\rho D^2 V_a^2} J^2$$

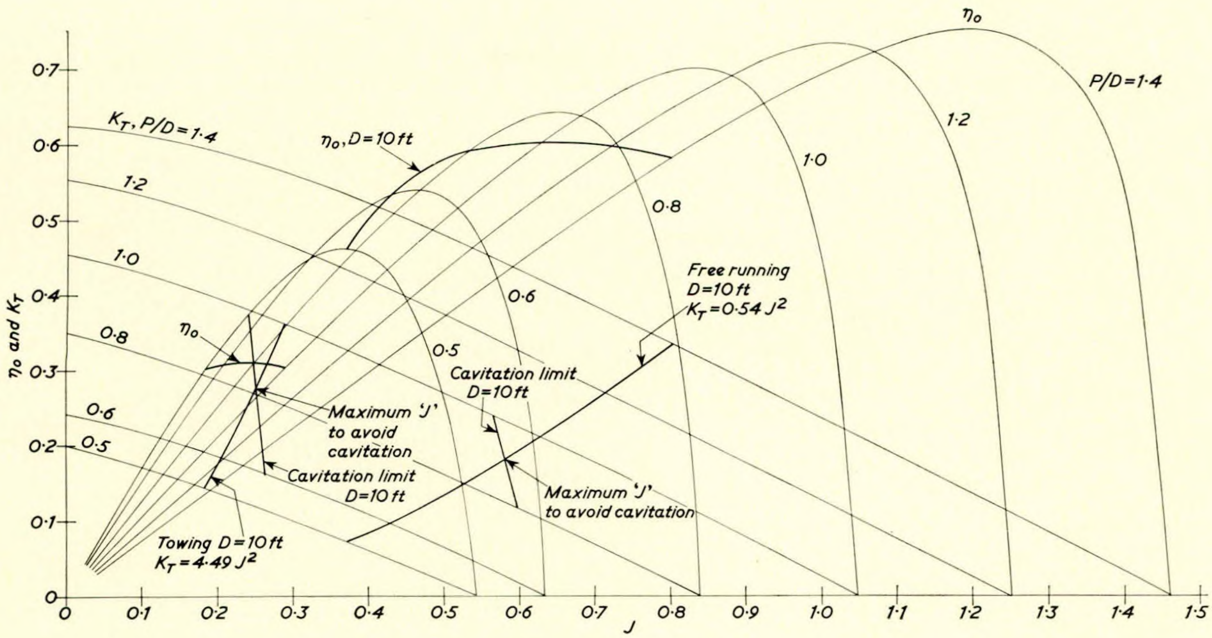



FIG. 7—*Propeller chart for trawler duties*

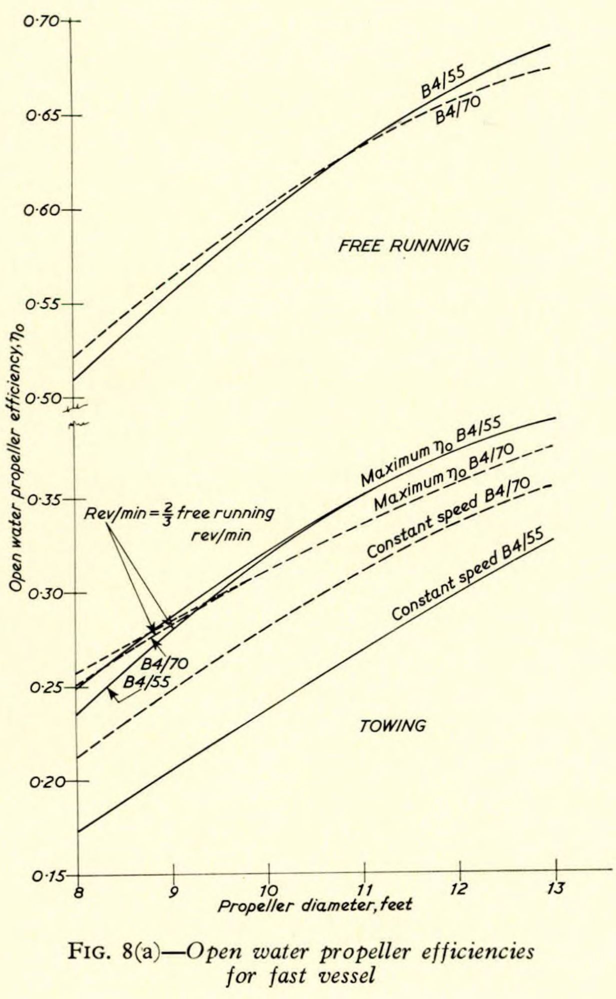



FIG. 8(a)—*Open water propeller efficiencies for fast vessel*

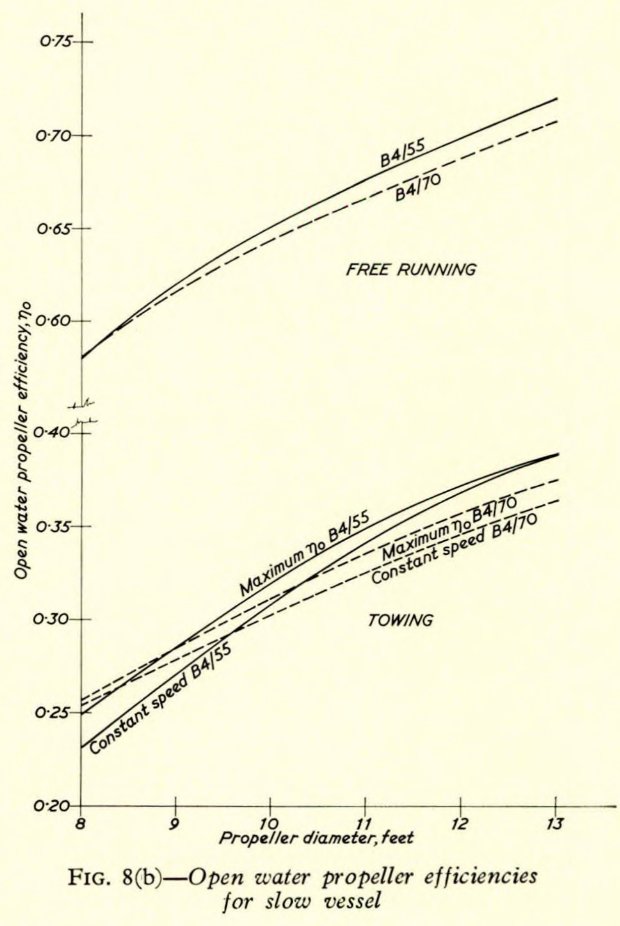



FIG. 8(b)—*Open water propeller efficiencies for slow vessel*

TABLE IV—ANNUAL POWER AND FUEL CONSUMPTION FOR TWO 215 FT LENGTH B.P. FREEZER TRAWLERS

Example of breakdown of consumptions with vessel condition when operating the engine at constant speed

| Vessel condition | Fast vessel (15·8 knots trials speed) | | | | | | Slow vessel (14·3 knots trials speed) | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | Time spent | Power | | | Annual consumption | | Time spent | Power | | | Annual consumption | |
| | | Propulsion | Auxiliary | Total | Power | Fuel | | Propulsion | Auxiliary | Total | Power | Fuel |
| | *days/* year | b.h.p. | b.h.p. | b.h.p. | *b.h.p. days/* year | *ton/* year | *days/* year | b.h.p. | b.h.p. | b.h.p. | *b.h.p. days/* year | *ton/* year |
| Free running | 100·6 | 2645 | 150 | 2795 | $281{\cdot}2\times10^3$ | 1063·4 | 102·8 | 1430 | 150 | 1580 | $162{\cdot}7\times10^3$ | 615·4 |
| Towing | 165·8 | 960 | 320 | 1280 | $211{\cdot}7\times10^3$ | 800·8 | 164·3 | 640 | 320 | 960 | $157{\cdot}5\times10^3$ | 595·8 |
| Shooting | 7·9 | 1235 | 320 | 1555 | $12{\cdot}3\times10^3$ | 46·4 | 7·8 | 670 | 320 | 990 | $7{\cdot}8\times10^3$ | 29·3 |
| Hauling | 11·8 | 300 | 670 | 970 | $11{\cdot}5\times10^3$ | 43·3 | 11·7 | 300 | 670 | 970 | $11{\cdot}4\times10^3$ | 43·0 |
| Handling | 11·8 | 400 | 420 | 820 | $9{\cdot}7\times10^3$ | 36·5 | 11·7 | 400 | 420 | 820 | $9{\cdot}6\times10^3$ | 36·4 |
| Dodging etc. | 26·3 | 500 | 320 | 820 | $21{\cdot}5\times10^3$ | 81·5 | 26·1 | 500 | 320 | 820 | $21{\cdot}3\times10^3$ | 80·7 |
| Total | 324·2 | | | | $547{\cdot}8\times10^3$ | 2072·0 | 324·4 | | | | $370{\cdot}3\times10^3$ | 1400·6 |

(N.B. 41 days/year in port)

The cavitation line is derived from Burrill's recommendations[4].

Using this method, curves of efficiency versus diameter have been constructed for two types of propeller, namely the Troost 4-55 and the 4-70 standard propellers. Fig. 8 shows the results for the three different types of operation listed above. In constructing these curves a relative flow factor of 1·0 for free running and 0·98 for towing has been assumed. In the case of the constant speed operation, it is possible that face cavitation will occur in the towing condition. This, however, has not been taken into account.

*Annual fuel consumption*

The pattern of operation discussed in section (1) of the paper has been converted to an annual breakdown of thrust requirements for the two vessels being examined.

For each type and diameter of propeller considered and for each of the three types of operation listed above, the thrust values have been converted into a breakdown of horsepower—hours and fuel consumption. A typical breakdown for one set of conditions is shown in Table IV.

In carrying out these calculations the simplification has been made that for each power level the specific fuel consumption of the engine does not vary significantly with engine rev/min. The reasoning behind this assumption is given in Appendix II. It may be that in spite of the wide scope for engine matching available on modern Diesels an engine/gear-box combination will be selected by an operator for which this does not apply. This feature would have to be checked for any specific design but it in no way invalidates the general design procedure discussed here.

The results of the calculations on annual fuel consumption for both vessels are shown in Fig. 9. In each case the consumption is plotted against propeller diameter.

A specific fuel consumption of 0·340 lb/bhp-h has been assumed for the free running condition, 0·360 lb/bhp-h for towing and 0·363 lb/bhp-h for all other conditions.

Fig. 10 shows the variation in maximum free running rev/min and maximum propulsion power with diameter, for each of the cases considered.

*Effect of propeller design on economics*

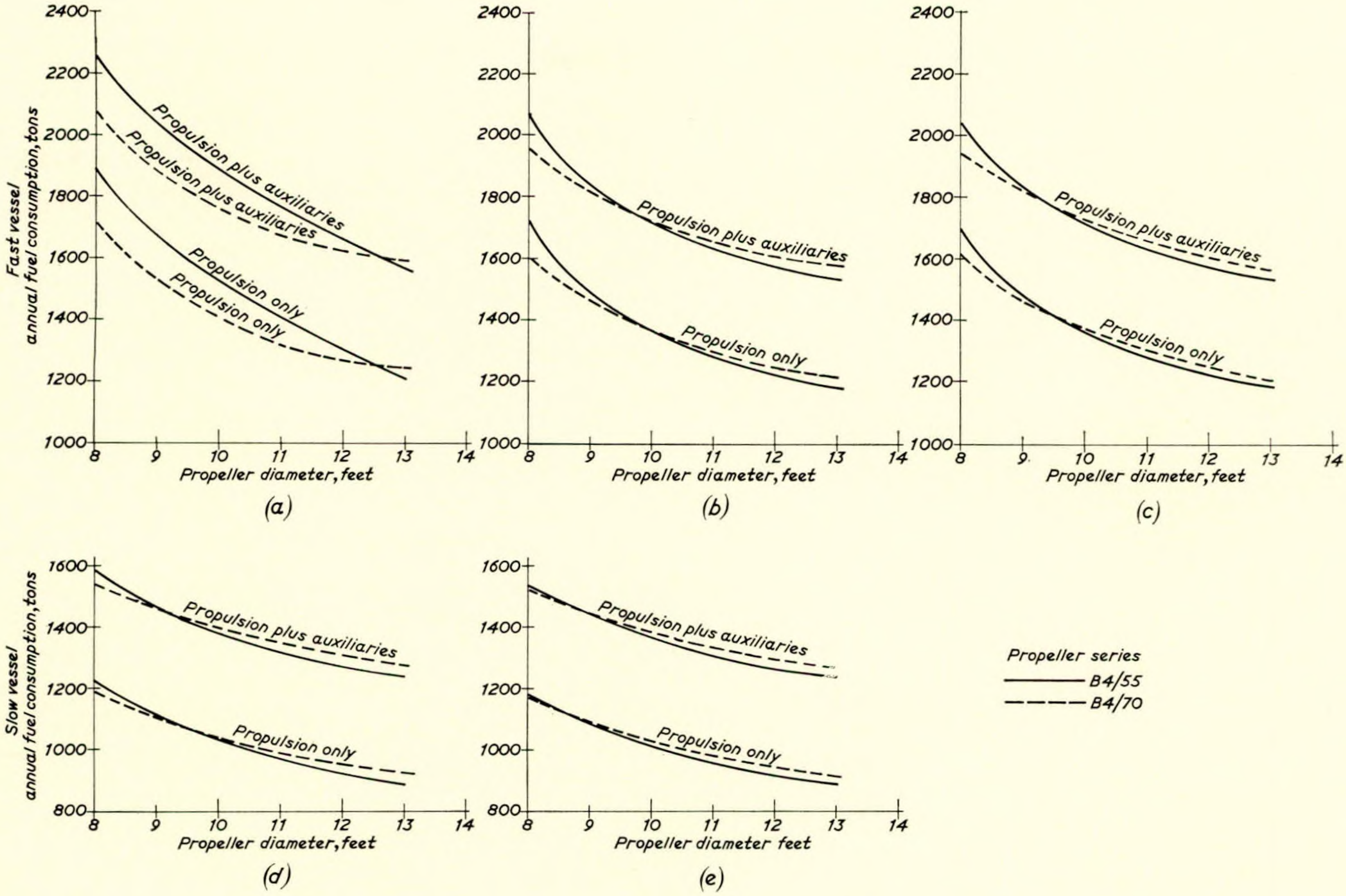
Using the above information on annual fuel consumption and the capital costs of propellers and gear-boxes the effect of propeller design on the economics of the vessels may be calculated. For the purpose of this paper it has been assumed that the capital cost of the engine is unaffected by the choice of propeller. This would not necessarily apply for all ships, since, in some cases, the choice of a more efficient propeller may allow a cheaper engine to be installed, although in many cases it would merely mean further de-rating. Similarly, improved efficiency may reduce the capital cost of some ships by reducing the size of fuel bunkers. These aspects, and others, such as reduced maintenance with the lower power absorbed by a propeller of higher efficiency, are likely to vary so much from vessel to vessel that they have been excluded from this paper. They would, however, have to be considered in the design of a specific vessel.

The annual costs have therefore been plotted against propeller diameter (see Fig. 11), on the assumption that the only varying quantities are fuel costs and the capital costs of the propeller and gear-box. Annual fuel costs have been plotted from Fig. 9 on the assumption that the cost of fuel to the operator is £8·25 per ton. The capital costs have been shown as a linear depreciation charge of 6·25 per cent per annum.

## 9) DISCUSSION

It can be seen from Fig. 11 that for all three types of operation considered, and for both types of ship, there is a continuous improvement in economics as the propellers increase in diameter, over the size range investigated, as theory indicates. However, at the largest diameter considered, the curves of total cost versus diameter are levelling off.

Obviously, for a given size of ship there are limitations on




(a) Constant speed operation
(b) Towing rev/min limited to two thirds free running rev/min
(c) Free choice of towing rev/min for minimum consumption

(d) Constant speed operation
(e) Free choice of towing rev/min for minimum consumption—Towing rev/min always greater than two thirds free running rev/min

FIG. 9—*Variation of annual fuel consumption with propeller diameter for two 215 ft (l.b.p.) stern trawlers*

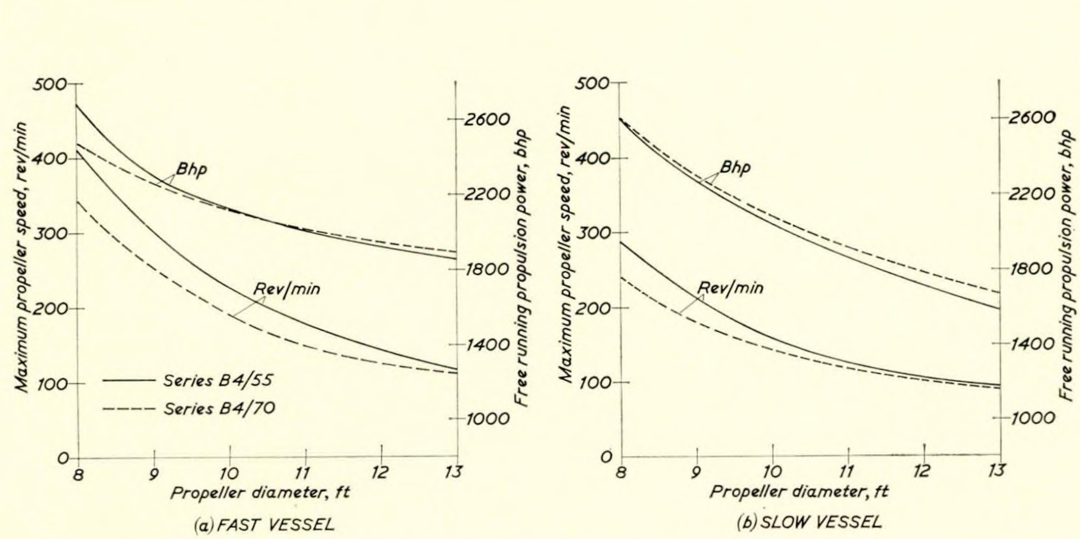
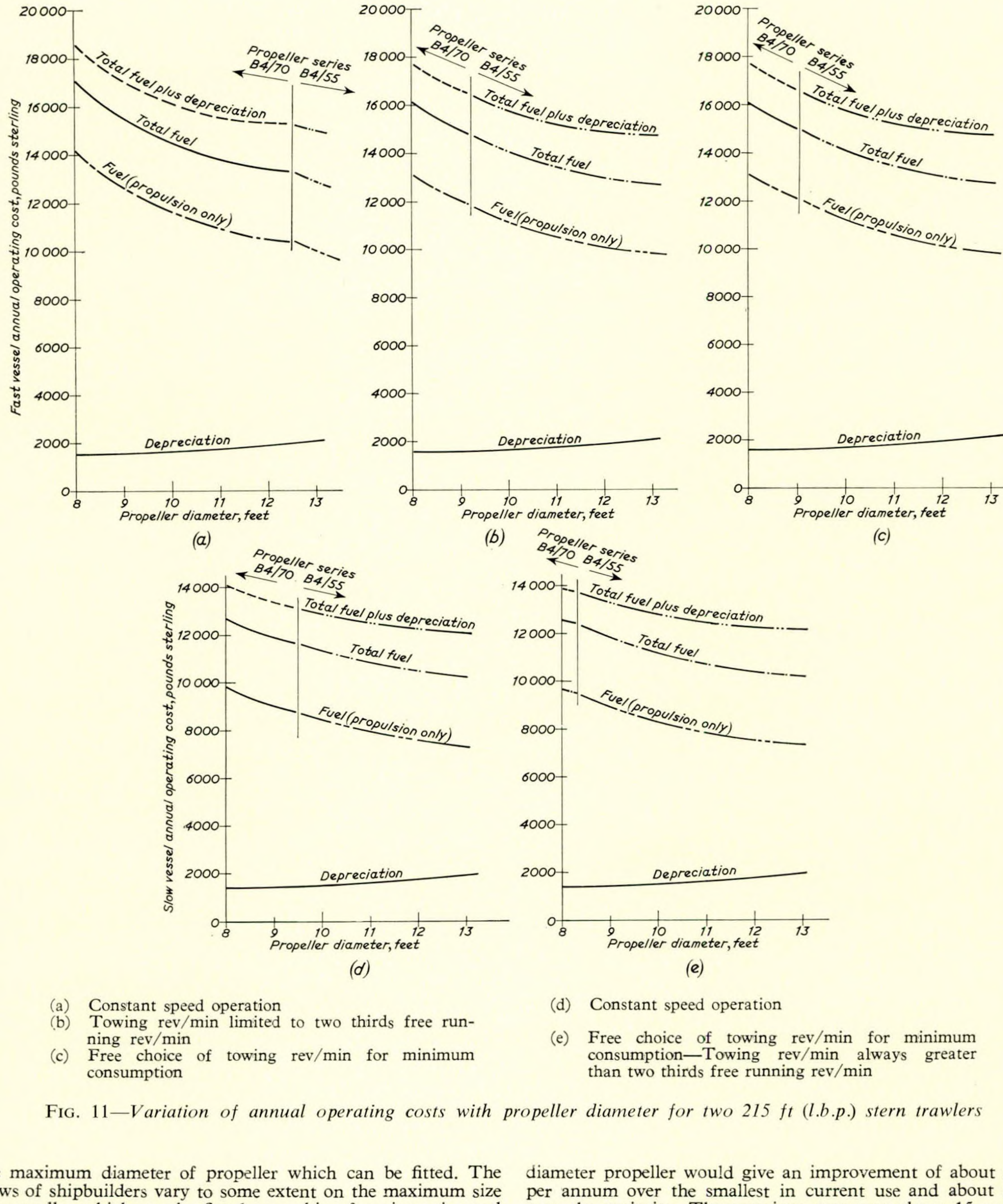



FIG. 10—*Variation of maximum propeller speed and propulsion power with propeller diameter*




(a) Constant speed operation
(b) Towing rev/min limited to two thirds free running rev/min
(c) Free choice of towing rev/min for minimum consumption
(d) Constant speed operation
(e) Free choice of towing rev/min for minimum consumption—Towing rev/min always greater than two thirds free running rev/min

FIG. 11—*Variation of annual operating costs with propeller diameter for two 215 ft (l.b.p.) stern trawlers*

the maximum diameter of propeller which can be fitted. The views of shipbuilders vary to some extent on the maximum size of propeller which can be fitted to a ship of a given size and form without seriously affecting the design and cost of the stern structure. For a stern trawler of the size being considered, namely 215 ft length b.p., the current opinion of most builders is that the maximum size of propeller is 12 ft diameter. The vessels of this size which have been built, to date, with c.p. propellers have had diameters ranging from 8½ ft to 10¼ ft. Over 80 per cent have been between 9 ft 2 in and 9 ft 4 in.

It can be seen that for the faster vessel, operating with minimum rev/min limited to two-thirds maximum, a 12 ft diameter propeller would give an improvement of about £2200 per annum over the smallest in current use and about £1600 over the majority. These savings represent about 15 per cent and 11 per cent of the annual fuel bill.

For the slower vessel the savings are slightly less, the corresponding figures being about £1400 and £1000, i.e. 12 per cent and 10 per cent of the annual fuel bill.

The significance of these figures upon annual profits is difficult to define precisely, largely because of the very great differences which exist in the abilities of individual skippers. At current catch rates, however, and current prices for sea-frozen fish, the savings of the 12 ft propeller over the majority of current designs at 9¼ ft represent an increase of between 3 per cent and 6 per cent of annual profits before taxation, but after depreciation (at 6¼ per cent per annum).

*Comparison of the three modes of operation*

The curves of Fig. 11 indicate the difference in annual costs between the three modes of operation, namely free choice of rev/min, minimum rev/min limited to two-thirds maximum, and constant speed. Since there are significant capital and maintenance cost differences implied in the overall propulsion and auxiliary systems for each of these three types, an economic comparison between them is outside the scope of this paper. One or two comments may be made, however:

i) Constant speed operation always results in a higher fuel consumption than the other types. The cost difference reduces as the vessel maximum free-running speed is reduced e.g. for a 10 ft propeller the fuel cost penalty of choosing constant speed operation is £1400 per annum with the fast vessel but only £100 per annum with the slow vessel. Since the capital cost of d.c. electrical equipment for constant speed is approximately £900 less than that for a 33 per cent speed variation there appears to be no point in designing for constant speed with d.c. electrics.

ii) However, with constant speed operation, a.c. power could be generated, with alternators driven off the gear-box. Although when towing there is a possibility of face cavitation on the propeller, the advantages offered by using a.c. power, of considerably reduced capital and maintenance costs of motors, make it well worth while considering this method of operation.

iii) There is no advantage to be gained by designing the system to make more than 33 per cent variation of rev/min available.

*Comparison of propeller series*

Of the two series considered in this study the B4/55 is better than the B4/70 for this duty, except where the avoidance of cavitation results in a significant reduction in efficiency. It is only in the case of constant speed operation with the fast vessel (Fig. 9a) that the B4/70 offers any real advantage.

## CONCLUSIONS

1) As theory would suggest, there is a continuous improvement in the economics of stern freezer trawlers, fitted with c.p. propellers, as the propeller diameter is increased, even when extra capital costs are taken into account. The size range 8 ft to 13 ft diameter, only, has been investigated in detail, but it is concluded that although the improvement will continue as the diameter is further increased, further reductions in cost will be small compared with those in the range investigated.

2) The higher efficiency of larger propellers is generally known. The significance of this study is that it has been expressed in economic terms, leading to design recommendations.

3) Current design practice sets a limit of about 12 ft on the diameter of the propeller for the larger vessels of about 215 ft length b.p. The use of a propeller of that diameter would result in an annual reduction in costs of between £1000 and £2200 (10 per cent and 15 per cent of the annual fuel bill) compared with the vessels now in operation. The actual value of the cost reduction will depend upon the design service speed of the vessel.

4) A similar design and cost analysis can be applied to larger and smaller vessels of this type, and would be expected to yield similar results.

5) The economic improvements which are shown in the paper are those due to reduced fuel consumption. In certain designs a further reduction in capital cost, may be achieved if the use of a more efficient propeller allows a reduction in engine size. In all cases, a reduction in maintenance costs should also arise.

6) The improvement with size is large enough to justify a critical examination of the factors which lead to current design limitations in diameter, to see whether or not these are real limitations.

7) The investigation has been carried out using two series of propeller charts, extrapolated where necessary. Correlation with full-scale measured data has been good over the limited range of conditions achievable, but is confined to a series of trials on one vessel only. Much more corroborative work is required, particularly model tests, including the simulation of towing conditions. These tests should cover a range of propeller types and sizes.

8) The performance analysis can be used to define the best method of operation of the propulsion system, in terms of choice of rev/min and pitch for the various operating régimes. The economic importance of this will depend on the degree of propeller shaft speed variation allowed in the particular design under consideration.

9) For the four-stroke, turbocharged engines considered, the choice of maximum engine rev/min, i.e. gear-box ratio, has a negligible effect on the economics (see Appendix II).

## ACKNOWLEDGEMENTS

The assistance given by the following organizations is gratefully acknowledged: Mirrlees National Ltd., Stone Manganese Ltd., Laurence Scott and Electromotors Ltd., W. R. Cunis Ltd. and the Ship Division of the National Physical Laboratory.

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## APPENDIX I

CORRECTIONS MADE IN CONSTRUCTION OF PROPELLER CHARTS

The Troost charts are published in three forms, $Bp$ against $\delta$ $\mu$ against $\sigma$, and $K_T$, $K_Q$ against $J$ where

$$Bp = \frac{NP_D 0{\cdot}5}{V_A^{2\cdot5}}$$

$$\delta = \frac{ND}{V_A}$$

$$\mu = n\sqrt{\frac{\rho D^5}{Q}}$$

$$\sigma = \frac{TD}{2\pi Q}$$

$$K_T = \frac{T}{\rho n^2 D^4}$$

$$K_Q = \frac{Q}{\rho n^2 D^5}$$

$$\text{so that } \mu = \frac{1}{\sqrt{K_Q}}$$

$$\text{and } \sigma = \frac{K_T}{2\pi K_Q}$$

where $P_D$ = delivered horsepower
$N$ = propeller rev/min
$n$ = propeller rev/sec
$V_A$ = propeller speed of advance in knots
$T$ = thrust in lb
$Q$ = torque in ft lb
$D$ = propeller diameter in ft
$\rho$ = mass density of water

During a trial the relevant measurements are usually thrust, torque, rev/min, ship's speed and propeller pitch in the case of a c.p. installation. Consequently the only quantities which can be formed for direct use with propeller charts are $K_T$ and $K_Q$.

It will be seen that it is not possible to enter the $B_p - \delta$ charts with either $K_T$ or $K_Q$ at a given value of pitch/diameter ratio. This also applies to the $\mu - \sigma$ charts with respect to $K_T$, it being necessary to know $J$ before thrust and torque data may be used simultaneously. There are similar objections to the $K_T$, $K_Q$ versus $J$ charts since if the data does not satisfy $K_T$ and $K_Q$ at the same value of pitch/diameter ratio it is difficult to find the effective value of that ratio.

The expanded blade area ratio of the ship propeller was 0·45 and the boss diameter ratio 0·286. According to O'Brien[4] the reduction in blade surface area for an increase in boss diameter for the Troost B3 series is:

$$\frac{\Delta a_e}{a_e} = 1{\cdot}1\ (d_b - 0{\cdot}180)$$

where $a_e$ = expanded blade area ratio of standard propeller
$\Delta_e$ = reduction in expanded blade area ratio
$d_b$ = boss diameter ratio of actual propeller

In this case $\frac{\Delta a_e}{a_e} = 0{\cdot}1145$

We have $a_e - \Delta a_e = 0{\cdot}45$
so that $a_e = 0{\cdot}506$.

The chart in Fig. 6 was constructed for this expanded blade area ratio.

In determining suitable values of $K_T$ and $K_Q$ from the full-scale data for use in this propeller chart there is a number of factors which must be considered.

Firstly the ship results are for the "propeller behind" condition whereas the chart data are for the open water condition. An allowance for the relative flow factor must be made.

From N.P.L. data it has been assumed that for free steaming $\eta_R = 1{\cdot}02$ and for towing $\eta_r = 0{\cdot}98$.

Secondly, for free steaming it is usually assumed that ship rev/min is 2 per cent greater than the corresponding chart rev/min, with the d.h.p. equal in both cases. This means that in using ship results in the propeller chart the rev/min must be reduced by 2 per cent and the torque increased by 2 per cent. These corrections are not usually applied in the towing case.

Thirdly, where the blade thickness ratio of the ship propeller differs from that of the chart propeller it is necessary to make corrections to efficiency and pitch-diameter ratio. In this case the ship propeller blade thickness ratio is 0·0564 whereas the ratio for the chart propeller is 0·045. Using the data given by O'Brien the following corrections have been made to the ship data:

| | | |
|---|---|---|
| free steaming | efficiency | −1 per cent |
| | P/D | +2 per cent |
| towing | efficiency | −1 per cent |
| | P/D | +1·5 per cent |

Fourthly, an allowance of 3 per cent has been made for shaft losses.

Using thrust identity with the above corrections the following relationships hold:

Steaming
$K_T$ chart = $K_T$ ship × 1·041
$K_Q$ chart = $K_Q$ ship × $1{\cdot}02^3$ × 0·98 × 0·99 × 1·02
= 1·050 $K_Q$ ship.

Towing
$K_T$ chart = $K_T$ ship
$K_Q$ chart = $K_Q$ ship × 0·98 × 0·98 × 0·99 × 1.02
= 0·9508 $K_Q$ ship.

## APPENDIX II

MATCHING OF THE ENGINE FOR THE TRAWLER DUTY

For the geared propulsion system of Fig. 2 there is usually a choice of gear-box ratio available to the designer for any given propeller/engine combination. The range of reduction ratios available is determined by the extent to which the engine has to be de-rated for the particular application in mind.

If the propulsion power requirement is exactly equal to the maximum power output of the engine, as in point A of Fig. 12, then only one gear ratio is available. However, even if operators were prepared to accept an engine at its maximum rating this equality of power levels would seldom occur in practice; in the marine field, especially, de-rating of engines is common practice.

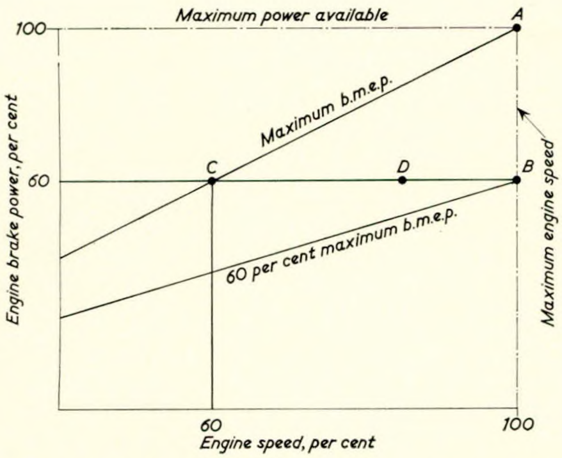



FIG. 12—*Engine power/speed diagram showing possible means of matching engine to propeller duty*

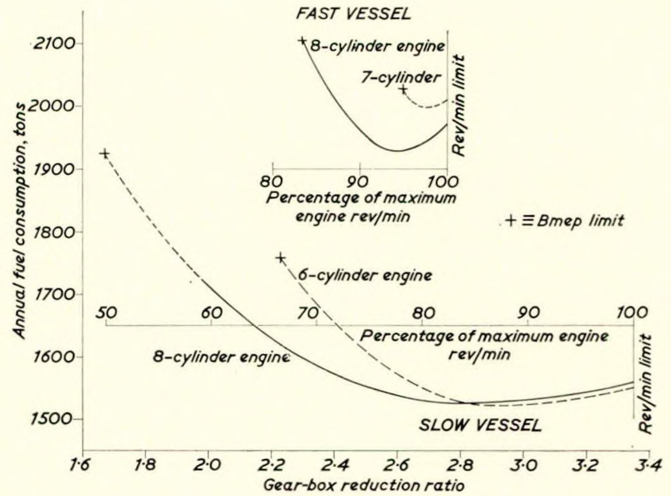



FIG. 13—*Effect of choice of gear-box ratio on annual fuel consumption for 215 ft (l.b.p.) stern trawler with main engine of 6, 7 or 8 cylinders*

As an example, the case may be considered of an engine which is to be de-rated 40 per cent on power. This could be achieved by de-rating at constant rev/min, as at point B in Fig. 12 or de-rating on rev/min at constant b.m.e.p. (point C). One or both of these methods may be unacceptable because of excessive bearing loading, poor part load performance, etc., and it is probable that both b.m.e.p. and rev/min would be lowered, as at point D, which could be anywhere on the line BC.

*Effect of engine rev/min on annual fuel consumption*

Several types of engine are currently in service in the distant water fleet and because of differences in the degree of tuning which can be achieved to alter the engine fuel consumption at different operating points it was considered, initially, that the problem of optimizing the propulsion characteristics, over a range of available gear-box ratios, would involve detailed studies by the engine manufacturers of the engine fuel isoloops, for each engine under consideration.

Before embarking on a study of this magnitude, however, it was decided to consider whether or not the variation in fuel consumption due to engine tuning, warranted this degree of examination, in light of the large variations in propeller efficiency shown in Fig. 8. Two types of ship were considered, namely a fast (16·1 knot trials speed) stern trawler of 215 ft length b.p. and an identical vessel with a trials speed of only 14·9 knots. The installed power for the first of these was 3230 bhp, and 1950 for the second. One particular type of engine now in service was selected. The available power outputs at maximum rev/min and b.m.e.p. for three configurations of this engine are as follows:

| | |
|---|---|
| 6 cylinders | 2930 bhp. |
| 7 cylinders | 3410 bhp. |
| 8 cylinders | 3950 bhp. |

Using an annual loading breakdown similar to that shown in Table IV, and the manufacturer's engine isoloop diagrams, the annual fuel consumption was calculated for the available range of gear-box ratios for the following hypothetical vessels:

Fast Vessel, 7 cylinder and 8 cylinder engines
Slow Vessel, 6 cylinder and 8 cylinder engines

The results of these calculations are shown in Fig. 13.

From this graph it is immediately obvious that high specific fuel consumption could be expected if the engine were de-rated largely on rev/min, maintaining high b.m.e.p. This practice, however, would not be recommended by engine manufacturers, who would certainly recommend a compromise solution. In the case of the slow speed vessel, a practical choice of gear-box ratio from about 2·5 to 3·35 (i.e. matching maximum propeller revs. at from 75 per cent to 100 per cent maximum rated engine rev/min) would result in an almost constant fuel consumption (within 2 per cent over the rev/min range). The lowest service b.m.e.p. is still acceptable, even at the highest ratio of 3·35 when the engine is de-rated on b.m.e.p. alone. Increasing the gear-box ratio from the lowest ratio of 2·5 to the highest ratio of 3·35 acceptable for fuel consumption increases the cost of the gear-box by approximately £2000.

For the faster vessel, the available range of rev/min is much more limited since the engines are running nearer their power limits. Once again, however, the fuel consumption is constant, within about 2 per cent, over the bulk of the available rev/min range, rising only at the low rev/min, high b.m.e.p., part of the curve.

In light of these results it was considered that the assessment of the economic effects of propeller design could proceed on the assumption that the fuel consumption would be unaffected by the choice of maximum engine rev/min, over the practical rev/min range available. Obviously, these assumptions would have to be re-checked for an engine with unusual performance characteristics.

# Discussion

MR. A. A. ADLEY (Associate) said that Mr. Chaplin in his presentation had made the point that the paper had not been laid out to show the economic comparisons between the fast and the slow vessels. He considered there to be sufficient in this however for it to be made a point, especially in view of the opening comments of the authors that freezer trawlers were able to spend a much larger proportion of their time on the fishing grounds and, at the same time, there was much less need for a high free running speed. He suggested that in order for trawlers to spend a larger proportion of their time on the fishing grounds, a high free running service speed was, for a variety of reasons, an economical necessity. He agreed that in general trawlers which froze their catch on the fishing grounds, on the assumption that their refrigeration plant remained effective, had less need for a high turn of speed on the homeward voyage because of the possibility of a spoiling catch, but did the authors consider that the main or, in fact, the only criterion on which a trawler's service speed, and hence installed propulsion power, should be determined? Should not other factors influence that decision, for instance, changes in the pattern of fishing arising out of over-fishing of the traditional grounds which led to an increase in time spent on passage to new grounds and searching for fish?

They had heard a great deal in recent years about those areas traditionally fished by the British distant water fleets in the North Atlantic, on which round trip voyages of 3000 to 5000 miles were commonplace, being consistently over-fished. That might well result in new grounds in the South Atlantic involving considerably greater on-passage times, with round trips of 11 000 to 12 000 miles having to be sought. The fact that the authors' department recently chartered a vessel to carry out just such an exploratory voyage would appear indicative of that. In such cases, he suggested, the need for speed was self-evident.

Another factor which perhaps should be considered in determining power requirement was possible future development of trawls requiring a considerably higher power to fulfill the towing duty. Several Japanese vessels were already towing a trawl necessitating a propulsion power in the order of 2500 hp. It would also seem worthwhile to anticipate a situation where an engine fault occurred necessitating one cylinder being isolated. In such a condition, the fast vessel could continue fishing, but the slow vessel would have inadequate power.

In Table IV Mr. Adley found it difficult to reconcile the comparative figures for time spent per annum in the free running and fishing conditions, or the propulsion bhp figures quoted for the shooting condition. If those figures were taken as read, it would mean that the fast vessel was having to travel 34 525·92 nautical miles per annum, compared with 31 456·8 for the slow vessel, that was 3069 miles further, an increase of 9 per cent, which at the speed of the slow vessel was equivalent to 10·04 days on passage, and yet having only 1·5 days additional fishing time, that was, 1·09 per cent longer in that condition. Regarding comparative shooting propulsion powers, it was difficult to understand why on two apparently identical vessels shooting the same gear in comparative weather conditions, virtually double the propulsion power was required on the higher powered vessel. At best gains in propeller efficiency could only account for 35 per cent of that additional 85 per cent.

Table V was similar to the slow vessel in Table IV, but had been adjusted so that both fast and slow vessels covered the same distance annually and the shooting propulsion power had been amended to give a comparative vessel speed in that condition. As mentioned, that resulted in an increase of 10·04 days in the on passage condition, and a decrease of 9·04 days in the towing condition for the slow vessel, which had thus an increased annual fuel consumption of 40·95 tons.

Tables V and VI illustrated the further savings in annual fuel consumption that could be made by use of a variable speed engine installation. The fast vessel annual fuel consumption dropped from 2072 tons to 1903·35 tons, a saving of 168·65 tons, representing a reduction of £1400 on the annual fuel bill, while

TABLE V—SLOW VESSEL 34 525·92 NAUTICAL MILES/YEAR

| Vessel condition | Time spent days/year | Power | | | Annual consumption | |
|---|---|---|---|---|---|---|
| | | Propulsion b.h.p. | Auxiliary b.h.p. | Total b.h.p. | Power b.h.p. days/year | Fuel ton/year |
| Free running | 112·84 | 1430 | 150 | 1580 | $178·3 \times 10^3$ | 674·7 |
| Towing | 155·24 | 640 | 320 | 960 | $148·8 \times 10^3$ | 569·3 |
| Shooting | 7·7 | 1000 | 320 | 1320 | $10·17 \times 10^3$ | 38·67 |
| Hauling | 11·44 | 300 | 670 | 970 | $11·11 \times 10^3$ | 42·38 |
| Handling | 11·44 | 400 | 420 | 820 | $9·38 \times 10^3$ | 36·0 |
| Dodging | 25·7 | 500 | 320 | 820 | $21·05 \times 10^3$ | 80·5 |
| Total | 324·36 | | | | $378·81 \times 10^3$ | 1441·55 |

TABLE VI—VARIABLE SPEED

| | Fast vessel 34 525·92 nautical miles/year | | | | | | Slow vessel 34 525·92 nautical miles/year | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| Vessel condition | Time spent days/year | Power: Propulsion b.h.p. | Power: Auxiliary b.h.p. | Power: Total b.h.p. | Annual consumption: Power b.h.p. days/year | Annual consumption: Fuel ton/year | Time spent days/year | Power: Propulsion b.h.p. | Power: Auxiliary b.h.p. | Power: Total b.h.p. | Annual consumption: Power b.h.p. days/year | Annual consumption: Fuel ton/year |
| Free running | 100·6 | 2645 | 150 | 2795 | $281·2 \times 10^3$ | 1063·4 | 112·84 | 1430 | 150 | 1580 | $178·3 \times 10^3$ | 674·7 |
| Towing | 165·8 | 710 | 320 | 1030 | $171·0 \times 10^3$ | 640·5 | 155·24 | 600 | 320 | 920 | $142·8 \times 10^3$ | 540·5 |
| Shooting | 7·9 | 970 | 320 | 1290 | $10·2 \times 10^3$ | 38·15 | 7·7 | 950 | 320 | 1270 | $9·78 \times 10^3$ | 35·86 |
| Hauling | 11·8 | 300 | 670 | 970 | $11·5 \times 10^3$ | 43·3 | 11·44 | 300 | 670 | 970 | $11·11 \times 10^3$ | 42·38 |
| Handling | 11·8 | 400 | 420 | 820 | $9·7 \times 10^3$ | 36·5 | 11·44 | 400 | 420 | 820 | $9·38 \times 10^3$ | 36·0 |
| Dodging | 26·3 | 500 | 320 | 820 | $21·5 \times 10^3$ | 81·5 | 25·7 | 500 | 320 | 820 | $21·05 \times 10^3$ | 80·5 |
| Total | 324·2 | | | | $505·1 \times 10^3$ | 1903·35 | 324·36 | | | | $372·42 \times 10^3$ | 1408·94 |

TABLE VII—COMPARATIVE ANNUAL COSTS AND EARNING CAPACITY

| Vessel speed | Installed power and speed variation | Distance covered, nautical miles | Fishing time, days | Catching capacity, tons | Earning capacity | Main machinery capital cost | Fuel cost | Earnings and fuel cost differential |
|---|---|---|---|---|---|---|---|---|
| Fast | 2800 bhp<br>nil | 34 525 | 197·3 | 2962 | £237 200 | £70 800 | £17 100 | nil<br>nil |
| Slow | 1600 bhp<br>nil | 31 456 | 195·5 | 2935 | £234 800 | £60 700 | £11 580 | −£2400<br>−£5520 |
| Slow | 1600 bhp<br>nil | 34 525 | 185·82 | 2788 | £223 000 | £60 700 | £11 900 | −£14 200<br>−£5200 |
| Fast | 2800 bhp<br>33⅓% | 34 525 | 197·3 | 2962 | £237 200 | £70 800 | £15 700 | nil<br>−£1400 |
| Slow | 1600 bhp<br>33⅓% | 34 525 | 185·82 | 2788 | £223 000 | £60 700 | £11 640 | −£14 200<br>−£5460 |

Fuel cost £8·25/ ton average catching rate 15 ton/ day.

Average selling price of non filleted block frozen fish 100/- per kit £80/ton

the slow vessel showed a saving of 31·61 tons, or £260. Reference to Table VII gave the following broad analysis: for an additional machinery capital cost of £10 100, the fast vessel could earn annually £14 200 more than the slow vessel, of which £4060 was absorbed in additional fuel cost, resulting in a £10 040 annual return on the additional capital investment. Mr. Adley suggested that that was analogous to a single premium comprehensive insurance policy which catered for possible development in the industry and yielded an annual dividend of 99·94 per cent.

His main reason for including the variable speed engine alternatives was that an installation of the type illustrated in Fig. 2 was not a feasible proposition as a constant speed system. Any propulsion system was liable to frequent and significant changes in torque and power demand due to variations in propeller immersion, few vessels more so than a trawler. Engine load control by pitch shedding must, of necessity be tuned so as to be insensitive to wave influences, and should operate only on condition of sustained overload. If one accepted that, then acceptance of variable engine/generator speed was automatic.

Fig. 14 showed relative changes in pitch, torque, thrust, fuel rack, engine rev/min and turbocharger boost pressure recorded during a manoeuvre on a 1600 bhp 300 propeller rev/min installation. The degree of variation was readily apparent. Greater variations could be expected if lower rev/min higher efficiency propellers were employed, and the conditions would also be further aggravated with a low inertia system that was a feature of smaller, higher speed engines. It was obviously necessary to install variable speed voltage control to cover that contingency and once installed it might as well be utilized to maximum advantage, that operational characteristic excluded the use of conventional gear-driven alternators. It was true that frequency control equipment was available, but in the kW capacity under discussion, that tended to be in the order of three-times the cost of the alternator itself.

What were the authors' views on the possibility of development of a variable speed input constant speed output hydrostatic

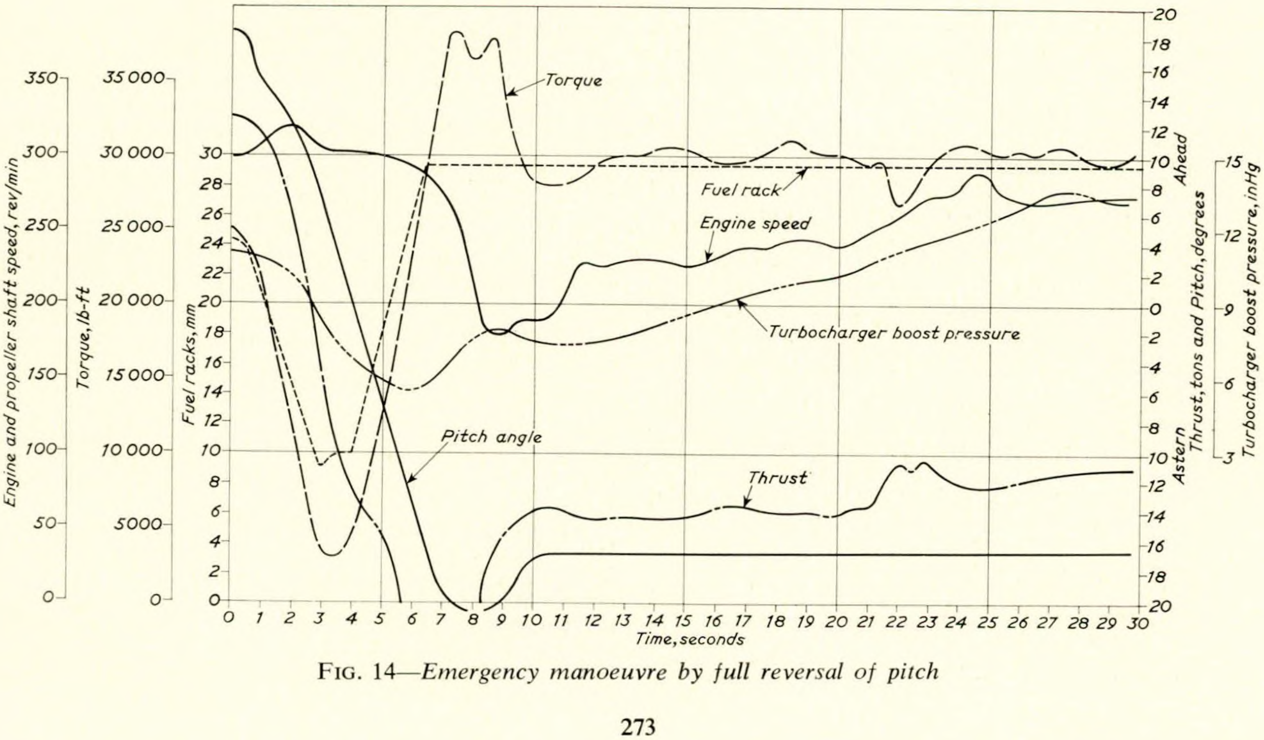



FIG. 14—*Emergency manoeuvre by full reversal of pitch*

transmission system, the pump, which could be driven from the gear-box, powering not only an alternator drive motor but also the winch?

The authors stated that "in certain designs a further reduction in capital cost might be achieved if the use of a more efficient propeller allowed a reduction in engine size. In all cases, a reduction in maintenance cost should also arise." That could be true only when related to a comparative engine range, for instance, a reduction from seven to six cylinders of a particular engine type. If, however, a reduction in power requirement was sufficient to permit consideration of an alternative engine range, that could result in higher b.m.e.p. and rev/min ratings and, consequently, a reduction in maintenance cost was less of a likelihood.

Finally, he commented on the unfortunate restrictions imposed on the authors by the engine builders relating to the torque limitations mentioned during the trials carried out on the vessels referred to in the paper. British designed and built machinery now being offered to, and accepted by, the distant water trawling industry, was capable of producing full rated torque over 50 per cent of the rated engine speed range. Therefore, a wider spectrum of trials data should be possible in the future, and that would also allow a wider range of gear-box ranges to be chosen.

CDR. A. J. H. GOODWIN, O.B.E., R.N., said that the figure of 6¼ per cent for depreciation seemed rather low compared with figures normally used for capitalization, 14–15 per cent had been used,* and 14 per cent.†‡

This 14 per cent was intended to include depreciation, insurance and interest on capital. In November 1966, his colleagues, Mr. Neumann and Mr. Carr, when making similar comparisons on the effect of different sizes of propeller, used a figure of 10 per cent in order not to overstate the case for the medium speed geared Diesel engine, and nobody challenged them on it. He believed however that 14 per cent was more generally applicable.

That in fact was the old rule of thumb that extra capital value was only justified if the savings it introduced paid off the capital value in seven years.

He was hoping the authors would state the case in favour of using c.p. propellers. In order to do that they would have had to make a comparison with fixed pitch propellers. No doubt the case was there somewhere, but he could not detect it. He was not even sure by the time they had convinced him that 0·53 was the propeller pitch to stick to, whether they were sticking to that even in the free running condition. The authors did say, in the early stages, that using c.p. propellers made a very good arrangement, and they were referring to d.c. They then said that the present arrangement did not allow sufficient change in speed, and it was necessary to add more capital value in order to allow rev/min down to two-thirds. Elsewhere in the paper they had said that there were advantages in keeping the speed constant and using a.c. In all those cases he would have liked to have seen a comparison between that and the fixed pitch propeller.

MR. P. R. CREWE said that he had listened to the paper with interest, both because of its importance for the fishing industry, and because colleagues of his were concerned with the preliminary strain gauge installation for torque and thrust measurement that was made for the Industrial Development Unit of the White Fish Authority in 1963, in *Arctic Corsair*.

He congratulated the authors on having so successfully carried through an investigation leading to such definite conclusions regarding the economic advantages of larger diameter propellers.

As had been stated, more corroborative work was required, covering a range of propeller types and sizes. Certain reservations might be felt with regard to accommodation of the propellers, aeration problems, and so forth, but those questions would undoubtedly be sorted out during practical applications of the general proposals of the paper.

MR. T. E. HANNAN (Associate Member) agreed generally with the paper. Those who worked in propeller design would not be unduly surprised that a larger diameter offered some advantage in economics.

The powers for shooting a trawl given in Table I seemed to him rather higher than he would have expected, even regarding the speed at which the trawl was shot. He thought once the ship was up to speed and the net over the stern, the speed took the net away and if necessary, power was required for breaking rather than putting the haul out. Could the authors give more information on this?

Under "General Method of Approach", there was a statement suggesting a lack of adequate propeller chart data for the towing condition, namely, high thrust at low forward speeds. That therefore led to some difficulty, apparently. Mr. Hannan would have thought that the normal propeller charts which had been available for some years, covered that area adequately. Maybe in the lower pitch ratios below 0·5 there was some dearth of data, but the rest of it was covered well. He failed to appreciate entirely the argument for re-plotting the data, as given in Fig. 6, because the $K_s$ $K_m$ chart for the N.S.N.B. data, for example, was usually adequate to analyse wakes. Nordström had published a paper "Propellers with adjustable blades" in 1945, which the authors might find of value in that connexion.

Turning to the section on instrumentation, no reference had been made to measurement and recording of helm angles when trawling. Trawlers consistently applied relatively large angles when trawling across the tide or wind, and the consequent rudder drag and drift of the hull would have an effect on pull. Could the authors comment? That would effect considerably the wake and thrust deduction figures the authors had used for comparison with model tests results where the helm presumably was amidship and not put over at an angle.

He found the paper leaning towards the general conclusion that for optimum propulsive efficiency one needed a fixed pitch, not a controllable pitch propeller. That was purely from the point of view of propulsive efficiency, and not taking into account the necessity for a.c. auxiliaries, and other things of that kind.

Those who worked in this field had suspected for quite a time that if a c.p. propeller was fitted—and Mr. Hannan was not in any way anti-c.p. propellers—it was probably better to use it as an f.p. propeller insofar as the revolution changes on the machinery would allow, than to maintain constant rev/min and change the pitch. The paper seemed to confirm that very clearly.

On the question of thrust deduction in the towing condition being lower than in the free running condition, that also confirmed something long suspected. If, for example, the vessel had a free running thrust deduction of 15 per cent then if it was a tug, and one took that to a static pull condition, one would find the thrust deduction there would be something of the order of 3 per cent or so. Very good results had been obtained with designs for the towing condition if for thrust deduction one interpolated linearly between those figures. That was to say, between 12 knots free speed with 15 per cent and 3 per cent at zero free speed. If at half free speed, six knots, one took a thrust deduction of 9 per cent, it generally gave good results, and the wake under those conditions was held constant throughout.

The authors had mentioned that they had detected a slight increase in wake. He did not challenge that, but pointed out that relatively speaking it was extremely small from the propeller design point of view, and the assumption of a constant wake for that condition did not bring any serious error.

He was sorry to have to put a little water into the wine of large diameters. One could confirm the favourable effect, but he hoped the paper would not be taken as a licence to start fitting tremendously large diameters which would fetch the pitch ratio too low, and the rev/min too high. There was a definite limit on diameter, and the effect of bad weather and so on also had to be taken into account, this required adequate immersion, so that

* Platt, E. H. W. and Strachan, G. 1962. "Post-war Developments and Future Trends of Steam Turbine Tanker Machinery". *Trans. I. Mar. E.*, Vol, 74. p. 385.

† Main, J. B. 1961. "The Design and Layout of a 22 000 shp Tanker Machinery". *Trans. I. Mar. E.*, Vol. 73, p. 277.

‡ Hutchison, T. B. 1966. "30 000 shp Unitized Reheat Steam Turbine Propulsion". *Trans. I. Mar. E.*, Vol. 78, p. 109.

efficiency was not lost through racing or through air drawing.

The paper brought out the problems facing propeller designers at the present time, not only with c.p. screws but with characteristics of the different machinery arrangements which were offered. The propeller was usually sacrificed without any hesitation if there could be a few per cent gain in the economics of the machinery. He did not necessarily challenge that, but it did add to the problem of the individual who had to design the screw.

It also underlined convincingly that oil and water, as represented by the engineer and the naval architect respectively, had to mix well. The advantage of large diameters upon fuel saving and economics was important, and the stress was rightly on that point. There was nothing in the paper, however, about the value of shrouding the propeller to obtain still further savings in that connexion. He imagined that was because neither of the vessels had a shrouded propeller. As an example, it was constructive to compare model tests results of a shrouded c.p. propeller of 7ft 9-in diameter on a stern trawler of 160ft load water line, with a length, b.p., of 145 ft. That had machinery giving 1100 shp at 250 propeller rev/min. According to Fig. 5, *Arctic Freebooter* showed at four knots a 10¼ tons pull with 1055 shp, at about 215 rev/min. Those figures were as nearly as he could read the scale of the graph. The model with the shrouded propeller showed 10¼ tons at four knots with 790 shp and 196 rev/min at a pitch ratio of 0·74. That indicated a power saving of 265 shp or 25 per cent with a shrouded propeller, of 15 per cent less in diameter. If they took, *pro rata*, the figures of section 6, that would appear to save something like £1450 p.a. in fuel alone.

The additional capital cost of a shroud of the rudder type would be pretty well offset by the saving on the propeller side, and from shafting and gear-box costs, together with the elimination of the normal rudder. The remainder would soon be recovered out of the fuel savings indicated.

That comparison could not be taken as absolute, but in view of the good agreement with model tests demonstrated, considerable savings were available by that means, without going to very large diameters and low rev/min which might prejudice operation in a sea way.

Of course Mr. Hannan knew that the authors were aware of the possibility of shrouded propellers, but since it was largely an economic paper he hoped they would forgive him for mentioning it in that context.

MR. G. FALCONAR (Associate) said that although it was well known that propeller efficiency increased with diameter, the actual results in economic terms disclosed by that study were most valuable and of such interest that one would wish the investigations carried further.

The paper dealt with an investigation into trawlers fitted with c.p. propellers, and if a trawler owner had decided to use such a propeller the findings could be most helpful. However, to assist the trawler owner still further in obtaining the most economical and efficient vessel, he wanted to see comparative performance and cost figures for a trawler fitted with a two ratio gear-box.

He wanted to draw attention to comparative curves for a c.p. propeller installation and a two ratio gear-box installation put forward by Mr. C. Ford in a paper to the Southern Joint Branch, "Aspects of the Mechanical Propulsion of Tugs". They might look upon a trawler as a tug while it was towing its gear. Those curves showed that the two ratio gear-box had a definite advantage over the c.p. propeller in pure bollard pull, and, particularly, in free running speed.

Not only was the capital cost of the two ratio gear-box a fraction of that of the c.p. propeller but it had negligible maintenance costs. If the comparatively modest savings in running costs that could be achieved by using the optimum propeller diameter were of importance to the trawler owner, the saving of thousands of pounds in capital cost of a two ratio gear-box plus the savings achieved through an appreciably higher free running speed, should be considerably more attractive.

A criticism of the two ratio gear-box with fixed pitch propeller was that control of the vessel's speed would need such a wide range of engine rev/min as to upset the electrical generation characteristics, assuming that the electrical supply would be provided from the main propulsion engine or from a p.t.o. shaft in the gear-box.

That criticism could be met by a twin-input/single-output two ratio gear-box with a p.t.o. shaft driven by one engine that could be clutched-in to a generator. Both engines were clutched-in to the propeller in high ratio to get quickly to and from the fishing grounds. The electrical requirements on passage could be met either from an auxiliary Diesel engine, or from a second p.t.o. shaft on the gear-box. On commencing fishing, one engine only would be required for propulsion, driving the propeller through the low ratio. The other engine would be isolated from the propeller and clutched-in to the winch generator.

The capital cost of such an installation would still be very considerably less than that of the c.p. propeller installation. The fishing efficiency and running costs of a trawler with such an installation could only be decided by as careful an investigation as the authors had made.

His thoughts were prompted by the many requests that they in Brevo received to put forward designs for gear-boxes with two or three ratios of the multi-input/single-output type, and with many variations of p.t.o. shaft arrangements. Many of the trawlers incorporating those gear-boxes were fishing successfully and profitably, but as careful an investigation as that described could be immensely helpful in providing even more efficient fishing vessels.

MR. B. HARVEY said that, to him, the information contained in Tables III(a) to (c) was of particular value. Evaluations of wake fractions and thrust deduction factors in the trawling condition, were very rare, and those figures provided a useful contribution to knowledge of the subject.

Again, the comparison in Table VI between c.p. propeller experimental data and the Troost standard series was encouraging and confirmed his opinion of the applicability of the latter to basic c.p. propeller calculations. However, in constant speed applications, to avoid serious face cavitation it was probable the propeller would be designed primarily for the trawling condition, and the blade rotated to higher pitch for free-going. In that case, the pitch distribution would vary from standard, and the Troost propeller efficiency prediction might prove rather optimistic. That in turn could marginally affect Figs. 9(a) and (d) and 11(a) and (d).

Regarding "Comparison of Propeller Series", he came to a more serious point. Throughout the paper, the Troost B4.55 and B4.70 propeller diagrams appeared to be regarded as separate series. That was not a true concept as the B series was a family of similar form, with the .55 and .70 referring to the blade area ratio. In the figures plotted to a base of propeller diameter, especially 8(a) and (b), he suggested the correct procedure for maximum propeller efficiency would be to select an appropriate minimum area ratio for each diameter (e.g. by reference to Burrell's cavitation criteria already mentioned) and interpolate between the diagrams for the correct value. That would halve the number of plots in Figs. 8–10, and produce continuous curves in Fig. 11.

In that respect he was also interested to know the area ratio for Fig. 7.

One item which would merit further investigation was mentioned in Appendix II "Engine Matching". Although the engine covered in the paper undoubtedly showed a considerable increase in fuel consumption, with rev/min reducing towards the b.m.e.p. limit, he would like to see the results of similar studies for different engines, and higher ratings. In that respect he thought the 40 per cent de-rated example quoted was rather a large reduction.

Comprehensive fuel isoloops were not always readily available, but he thought that not all engines conformed to the statements in the paper, "at each power level the specific fuel consumption does not vary significantly with rev/min". It was also stated, "There is no advantage to be gained by designing the system to make more than 33 per cent variation in rev/min

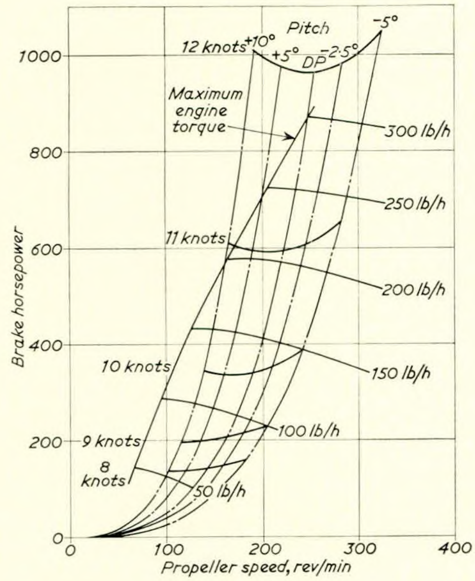



FIG. 15—*Bhp-rev/min diagram for trawler—Free running*

available" with the addition of fuel consumption curves in lb/h.

In Fig. 15, which was basically similar to Fig. 5, the vertical scale represented horsepower, the horizontal scale rev/min. The propeller manufacturer supplied curves of power absorbed by the propeller and those were plotted for various constant pitches, for example, design pitch, D.P. + 10 per cent, D.P. − 2·5 per cent, etc. Each line, in effect, applied to a separate solid propeller. For a constant rev/min it could be seen that the power demand rapidly increased as the pitch coarsened.

| | | | | |
|---|---|---|---|---|
| e.g. 250 rev/min | Pitch | −5° | −2·5° | D.P. |
| | hp | 425 | 625 | 875 |

From the ship resistance curve supplied by the shipbuilder or ship test tank, cross-plots of constant ship speed were made. The lowest point of that line represented the minimum hp required to obtain the relevant ship speed and that indicated the pitch and rev/min for maximum propeller efficiency.

The torque limit line and curves of constant fuel consumption in lb/h, as obtained from the engine builder, were then superimposed.

It was now possible for any given speed required from the vessel to select a point on the appropriate ship speed line giving minimum fuel consumption. That point would also indicate the required pitch and rev/min for that optimum combination of propeller and engine efficiencies. It was easily seen how, in practically any condition away from the full speed, a substantial economy in fuel consumption was made possible by the ability to coarsen the propeller pitch, for example at 9 knot condition:

Design pitch —165 rev/min 83 lb/h
Optimum pitch (D.P. + 10 per cent)—116 rev/min 71 lb/h

That represented a 15 per cent improvement in fuel consumption for that particular condition.

Finally he drew the authors' attention to two minor items:

i) The *St. Finbarr* boss ratio was 0·292, and thickness ratio 0·0518, and not as stated.
ii) There was a slight discrepancy in the quoted free running relative flow factor used in the calculation. This was stated to be 1·02 in Appendix I, but shown as 1·0 elsewhere. Would the authors please clarify this?

MR. J. N. EDGAR (Associate Member) said, regarding the second part of the paper, that it was gratifying to see that emphasis had been on the recurring theme of the advantage of installing a propeller which was as large as possible, running at its optimum rev/min. That was a matter for close consultation between the shipowner, shipbuilder, engine builder and propeller designer.

In Fig. 5, the authors demonstrated that for the propeller/ship combination of the type considered it would be of little use studying the effects of the choice of pitch ratios below 0·5 for obtaining the best conditions for a given thrust. Thus, in future designs, it would seem that a lower limit of, say 0·40 for a standard series chart would be sufficient.

One such chart* was a diagram constructed from cavitation tunnel tests on a c.p. model. Such a chart, based as it was on a c.p. propeller set at different blade angles in order to obtain the appropriate pitch ratios, was probably a more definitive diagram to use.

Use of correction factors as described in Appendix I was probably sufficiently accurate, within certain limits, but the case of a c.p. propeller, where the section pitch angles were fixed by the basic design condition, and then increased or decreased all by he same fixed increment, so as to obtain the required pitch ratio, meant that the hydrodynamic characteristics might vary considerably from that of the standard series propeller having the same pitch ratio. In preparing an analysis diagram it might be worth while carrying out a more rigorous study involving all the relevant characteristics. The vortex theory method described by Burrill was one useful means of carrying out such a study. Such methods could also assist in assessing cavitation criteria under differing operating conditions at different mean pitch settings.

It was hoped that such systematic studies as had been described by the authors would continue and in the future results of their investigations would be correlated to the use of standard charts, but in addition to the application of more comprehensive design/analysis methods, and also systematic experimental work, carried out in, say, a cavitation tunnel.

Finally, did the authors intend to look into the performance of c.p. propellers running in a backing condition, with the ship running either ahead or astern?

* Sinclair, L. and Emerson, A. 1968. "The Design and Development of Propellers for High-Powered Merchant Vessels". *Trans. I. Mar. E.*, Vol. 80, p. 129.

## Correspondence

MR. L. HAWDON wrote that the paper consisted, firstly of an examination of full scale performance data and comparison with model tests, and secondly of an investigation into the effects of changing the propeller diameter. The trials data were useful and confirmed the present practice in the design of propellers for trawlers, but it was difficult to see what bearing this had on the conclusions reached in the second part of the paper, which could readily be determined from existing standard series propeller data and were already well known. In the latter connexion the statement concerning the lack of adequate propeller chart data was surprising, since successful trawler propellers had been designed for many years using existing published data.

With regard to the analysis of the trial results, would it not have been more useful to have used torque identity in obtaining analysis wake data? The authors stated that the torque records were more reliable than the thrust data, and in re-designing propellers from analysis data it was more convenient to use torque values.

While it was refreshing to note that the authors had brought to light the advantages of large diameter propellers, it should be remembered that there was nothing new in this approach, but that it had been largely overlooked since the introduction of Diesel machinery into trawlers. It had, perhaps, not been made sufficiently clear, that in using a large diameter the propeller revolutions should be reduced correspondingly in order to achieve optimum efficiency.

The first Diesel engines had been introduced some 15 years ago and were initially small units of about 700 hp, turning at about 250 rev/min, the propeller being directly driven and having a diameter of the order of 7ft–8ft. While powers had been considerably increased since that time and reduction gearing had been introduced in some cases, there seemed to have been little effort to reduce propeller revolutions to the order of those of the steam engines that these Diesels have now almost completely replaced. Therefore, it might be of interest to compare the performance in terms of propulsive efficiency of *St. Finbarr* with that of one of the last steam trawlers of similar size, operating at equivalent speed. For this purpose the following details had been tabulated.

| | *St. Finbarr* | Steam trawler |
|---|---|---|
| ihp | — | 1525 |
| shp | 1592 | — |
| dhp | 1544 | 1296 |
| rev/min | 300 | 134 |
| Ship Speed | 14·05 | 13·95 |
| Wake fraction Wt | 0·20 | 0·13 |
| Propeller diameter | 7·4ft | 11·75ft |
| Pitch | — | 11·40ft |
| Blade surface area | — | $49ft^2$ |
| No. of blades | 4 | 4 |
| Propeller efficiency | 0·60 | 0·73 |
| Q.P.C. | 0·61 | 0·70 |
| Improvement | — | + 15 per cent |

This comparison showed that while the ship speeds were sensibly the same, the power required on the steam trawler was 150 dhp less than that on the Diesel ship. The improvement in actual Q.P.C. was 15 per cent, and such large propulsive efficiencies were usual with large steam trawlers having propellers of optimum diameter. The comparison was perhaps a little biased in this case, since the machinery of *St. Finbarr* had been running at unusually high revolutions, and furthermore the propeller diameter had been, for some reason, rather too small. The example, nevertheless, illustrated the kind of advantage that could be obtained by reducing shaft revolutions, and it was surprising that larger reduction ratios were not more widely used in geared Diesel installations.

Mr. Hawdon said his remarks related more to trawler propellers in general rather than to c.p. propellers, which, after all, was the subject of the paper. In this respect it would have been interesting, and refreshing, to have read a stronger case for c.p. propellers on this class of ship. Would the authors comment on the advantages or otherwise of c.p. propellers from the results of their investigation?

Finally, why was a three-bladed propeller chosen for *Arctic Freebooter* and was any advantage, either in propulsive performance or vibration characteristics, derived from this choice?

MR. M. B. F. RANKEN (Member) wrote that in the discussion Mr. Adley contested one conclusion in the paper, that high trawler speeds were unnecessary and uneconomical in freezer trawlers. This was something that he and others concerned with main engines had been saying for years, but it was not in line with almost universal practice in other countries. For example, vessels from Israel, Greece, Italy, Spain and Portugal fished in the South Atlantic and off Labrador, with round trips extending to 10 000 or 12 000 miles. Spain was now considering fishing in the Pacific which would extend the round voyage to perhaps 16 000 miles.

The price obtained for frozen fish stored at –13°F to –22°F (–25°C to –30°C) was unaffected by storing it for a few days, weeks or indeed months. In fact it often remained in a shore store for six to eight months before distribution and consumption.

The round voyage from these countries, including fishing time, usually extended to about four months nowadays, and arrangements were made to tranship the first cargo to a cold store or fish carrier near the grounds, e.g. in Capetown, so that two fishing periods could be undertaken on each round trip. Spain was now to build larger fish carriers to enable her ships to work in the Pacific, then she might keep the fishing vessels on the grounds for longer periods, revolving the crews by sea or air. This was the same pattern adopted by Japan and the U.S.S.R., operating in all parts of the world, though both these countries seemed able to keep their crews away from home for much longer periods than did other countries.

High speeds were necessary for wet fishing vessels to bring back iced fish to market as quickly as possible, so landing better quality fish which should command a higher price. For freezer vessels, however, simple arithmetic would show that there was no advantage in a knot or two extra in the context of normal lengths of voyage, methods of operation and the sizes of vessel concerned. 12·5 to 13·5 knots was plenty for free running, provided the engine power was sufficient to allow shooting and trawling under all normal conditions up to the maximum towing speeds likely to be used, perhaps 5–6 knots though 3–4 knots was normal.

## *Authors' Reply*

The authors wished to correct an error in Table IV, for the slow vessel in the shooting condition. This should have been 7·8 days, 990 bhp propulsion, 1310 bhp total, $10{\cdot}3 \times 10^3$ bhp days/year, 38·8 tons fuel/year giving totals of 372·8 bhp days/year and 1410·1 ton/year.

Referring first to some of the more general points which were raised, there was the question on alternative methods of driving ships. Mr. Falconar had mentioned the use of gearboxes, in various configurations, and Mr. Hannan had also raised the question of shrouded propellers. The short answer to all of these points was that it was highly desirable that detailed economic studies should be carried out on these and other types of propulsion equipment. To date the authors had concentrated on the c.p. propeller largely because this was employed on the bulk of the distant water fleet.

In the long term, they were certainly intending to study the sort of systems described. In particular, as regards the shrouded propeller, they had, at the time of writing this reply, just completed "before and after" performance trials on a ship with a shrouded c.p. propeller. Although this work was on a smaller trawler of 130 ft overall length, the results would provide design data capable of being extrapolated to larger and smaller vessels.

A preliminary assessment of the results of these trials suggested that the addition of the shroud had given increase in thrust of about 15 per cent for the same power at a towing speed of 4 kn. Furthermore the free running speed had been maintained. The system was still being evaluated and a report would be issued by the White Fish Authority later in 1968.

The argument put forward by Mr. Falconar, in favour of a two speed gear-box drive to a fixed pitch propeller, was valid and had much to recommend it. By designing the propeller for one of the two major duties and arranging the gear ratio to give optimum rev/min in the other condition, propulsion power consumption costs could differ from those for the c.p. propeller installation given in the paper, by as little as £100–£200 per annum. This would be a small price indeed to pay for the reduced capital and maintenance costs, if it were not for the disadvantages introduced by the fixed pitch propeller, discussed in the reply to Commander Goodwin's question.

Mr. Adley had raised a number of general points regarding the choice of ship for a particular duty and the choice of powers to be installed. In the introduction the authors had emphasized that, in the particular study described in the paper, they had been trying to develop a design method which could be applied generally to that class of vessel, namely the distant water freezer trawler, and they were not in any way advocating the adoption of the fast or slow ship used as examples in the paper.

In order to indicate the complexity of the problem of choosing the right ship and the right propulsion system for that and most types of fishing vessel, the authors introduced Fig. 16. It was a very complex job requiring a great deal more detailed consideration of many aspects, not merely the narrow field of the propeller studied in the paper. Capital cost, catch rates, running costs, the effects of weather, distance to the fishing grounds etc., all of the things which Mr. Adley had mentioned and some more had to be taken into account.

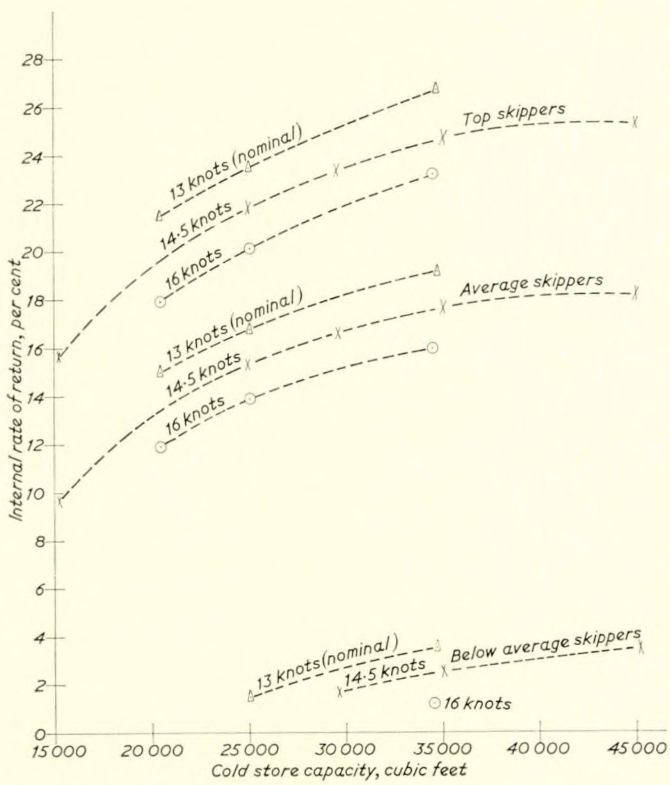



FIG. 16—*Internal rate of return compared with cold store capacity.*

Fig. 16 showed some preliminary results of a computer based study using operational research techniques into the choice of the ship and fishing system as a whole. It was a graph of the economics of the freezer trawler (in terms of rate of return on investment) against the cold store capacity of the ship, one of the main design variables. In addition the effect of free running speed and consequently the level of engine power installed was also shown. For all three different classes of skipper considered, the larger the cold store capacity, the better economic proposition the vessel became, up to the maximum of 45 000 $ft^3$, which was the maximum size studied to date. It was also shown that, for that particular duty—the North West Atlantic fishery—the slower ships were in every case the better choice. In straightforward terms it meant that, for that particular fishery, the additional capital and running cost involved in giving the skipper a faster ship was not justified in terms of catch rate and the price he got for the extra fish.

The authors agreed that if the fishing grounds changed by a very large distance from the home port or if the design of the propulsion machinery changed or if fuel costs were reduced significantly, the calculations would have to be done again.

The same sort of answer must be given to many of the other points raised by the speakers, in that they had to be studied in the context of the ship as a whole, and not merely in the context of the propeller and propeller machinery.

Commander Goodwin had queried the choice of the c.p. propeller for this application and this point had been raised, also, by Mr. Hannan and Mr. Hawdon. The c.p. propeller, of course, had been chosen for trawlers for reasons which had very little to do with the propulsive efficiency of the ship. For example, the need for accurate manoeuvring of the vessel, quick response of the ship to control changes when handling fishing gear, these were the sort of aspects which had dictated the choice of a c.p. propeller. These reasons were very difficult to quantify in economic terms, but they had been regarded as of sufficient importance by very many owners of small and large fishing vessels. The purpose of this particular paper was to study, and try to optimize the system which had been selected rather than influence the choice of system. However, to particularize a little on this subject, in the examples with which they dealt, the authors had generally found that in the towing condition the pitch/diameter ratio, which was acceptable from the cavitation point of view, was considerably lower than the acceptable ratio for free running. Thus if one accepted the lower pitch/diameter ratio for towing and used that ratio for free running there could be a serious fall in propeller efficiency in the free running condition. The extent of the loss in efficiency depended very largely on how well the towing and the free running conditions were matched. If the vessel was fast, perhaps three or even four times the towing power for the free-running condition was required and, thus, the c.p. propeller seemed to be the better suited to this vessel. On the other hand, if the towing and free-running conditions were more evenly matched, as with the slow vessel, the fixed pitch propeller would be satisfactory as far as its efficiency was concerned.

Turning now to the more detailed points there was firstly the matter of accommodating larger propellers. The studies carried out so far were fairly crude, but they indicated for the size of ship in which the authors were interested, namely 200 to 220 ft in length bp, the 12 ft propeller should be possible with reasonable stern frame designs, as existed at the present time. Above 12 ft diameter there would be certain difficulties and those would increase until one started to run into serious immersion problems, as Mr. Hannan had pointed out.

The figures the authors had given for percentage changes in costs up to anywhere between 10 or 15 per cent of fuel costs per annum applied to the recommended change of from about 9 ft diameter, which was popular at the moment, up to 12 ft, the figure which the authors had given as being a reasonable possibility.

Referring to Commander Goodwin's points about the depreciation charge, the authors emphasized that they were trying to demonstrate a design method, and those who had different ideas on depreciation would do their own sums and arrive at their own conclusions. The 6¼ per cent they used was a straight linear depreciation on a 16 year life, which was about that expected for the new run of stern freezer trawlers. The authors agreed that the figure ought to be higher if one took into account the question of interest on bank loans. To set against that, they had taken no account of the investment grant which in fact would just about pay for quite a high bank rate charge. In the opinion of the authors, somewhere between six and eight per cent was reasonably accurate for comparative purposes, for fishing vessels at the current rate of investment grant.

Regarding the auxiliaries, the authors said that the distant water section of the fishing industry was gradually turning to a.c. auxiliary power and the ships using it now were achieving very favourable maintenance costs, in comparison with d.c. systems. They were convinced, therefore, that considerable effort was justified in attempting to find solutions to the alternator speed control problems which were raised. There was the possibility of using a hydrostatic transmission, mentioned by Mr. Adley to achieve constant speeds in spite of fluctuating engine rev/min. That was one sort of solution which ought to be looked at, although, from the authors' present knowledge, hydraulic transmissions of over 250 hp with variable flow control were extremely expensive and higher powers could only be reached by duplication of systems. Certainly it was to be hoped that the electrical manufacturers might try to find even simpler and cheaper solutions to the voltage and frequency control problems caused by engine speed changes. By grouping together those loads which could tolerate a variation in frequency, a considerable proportion of the total a.c. load could be generated from the main engine. It was interesting to note that the Fisheries Research Vessel *Corella* had a 125 kW alternator driven from the main engine, with a frequency control which could accommodate a variation of $\pm 5$ c/s (which was equivalent to about ten per cent change in rev/min). It was understood that the cost of this control gear, which had to deal with the propeller variations, was about the same as that needed on a separate Diesel alternator set.

Mr. Adley had raised the question of the effect of fishing gear changes on the choice of installed propulsion power and went on to question the authors' analysis of the relative shooting powers and times spent fishing for the fast and slow vessels. It was quite possible that in the future British fishing vessels would employ fishing gears which were larger than the present Granton trawl. Since the powers required to shoot and tow this gear were 30 to 60 per cent less than the power levels currently being installed in new distant water vessels, there was a considerable margin in hand to meet future gear developments and to cover the case of operating with one cylinder out of action. Mr. Adley was quite correct to question the shooting powers given in Table IV for the slow vessel. This had been wrongly given as 670 bhp, when it should have been 990 bhp. In analysing the time spent fishing it was important to note that the number of trips per year was greater for the fast vessel than for the slow vessel. It was therefore not correct to say that both vessels covered the same distance per year as because of the greater number of trips per year, the distance covered by the fast vessel when running free would be greater. Perhaps rather surprisingly the time spent towing varied by the small margin of about one per cent. This was mainly because the free running time referred to in the paper included the time spent changing from one fishing ground to another and it was a fact that the annual time spent changing grounds did not vary noticeably with ship speed: the maximum free running speed did influence the skipper's decision of when and where to shift.

In answer to Mr. Hannan's question of how the trawl was shot, the most important feature was that during shooting the fishing gear must maintain a speed of three to four knots relative to the water. This was essential to ensure that the fishing gear was open and correctly orientated at the moment it came into contact with the ground. At the same time the relative velocity between the ship and the fishing gear should be about four to five knots to complete the shooting as rapidly as possible, which gave a ship speed of about eight to nine knots. Application of the winch brakes was essential in maintaining the correct relative speed between ship and trawl. For a large stern trawler, the net thrust required to propel the ship would be about 4·5 tons and the pull of the gear would be about seven tons. Taking a total net thrust of 11·5 tons, a ship speed of 8·5 kn and a Q.P.C. of 0·50, the shaft horsepower was approximately 1350.

Mr. Hannan and Mr. Hawdon queried the authors' statement that there was a lack of propeller chart data. The authors' remarks on this point referred to pitch/diameter ratios of less than 0·5. The results from *Arctic Freebooter* illustrated the need for this data in order to be able to define clearly the optimum towing propeller pitch. In Fig. 6, they carried out an extrapolation to a pitch/diameter ratio of 0·40, but this could only be regarded as tentative. This was necessary as more than half the trials data in the towing condition were at a pitch/diameter ratio of less than 0·50.

The authors were grateful to Mr. Edgar for pointing out the c.p. propeller chart previously published by the Institute and would endeavour to make use of this in the future. With regard to the correction factors the authors felt that the position was far from satisfactory. The existing techniques applied to fixed pitch propellers and it was unknown as to what extent they were satisfactory for c.p. propellers. Correcting for pitch on a c.p. propeller not operating at design pitch was an obvious difficulty.

The authors appreciated the need for a good analysis diagram but, since they were not experts on propeller design, they would seek expert advice as appropriate.

Mr. Edgar would appreciate that the authors' work was determined by the needs of the fishing industry. So far the problem of backing had not been raised, probably because the horsepower/ton displacement available was generally about unity, whereas for large tankers, which suffered this problem, a value of 0·25 was becoming common.

With regard to Mr. Hannan's question about the use of helm angle during the towing trials, the authors could assure him that care was taken to tow with or against the tide. In this way it was possible to operate a stern trawler with the helm amidships since no turning moment was applied by the warps. It would no doubt be appreciated that on a side trawler, where both warps were attached to the same side of the vessel, helm angle was always required to counteract the resulting turning moment.

The authors agreed with Mr. Hannan that the thrust deduction fraction could generally be taken to vary linearly with speed, although the results reported suggested that propeller pitch might be important in determining the actual value.

The point raised by Mr. Harvey concerning face cavitation was appreciated by the authors. They took the view that the main aim of the paper was to present the economic arguments in broad terms using comparatively simple analysis techniques. When it came to actual construction, the design of the propeller was a matter for experts such as Mr. Harvey and his colleagues.

Similarly the authors recognized that in the actual design of a propeller the appropriate blade area ratio would be determined. The purpose of Figs. 8, 9 and 10 was to present examples of the improvement to be expected by increasing propeller diameter so that this fundamental point would be understood.

The propeller chart reproduced in Fig. 7 was the Troost B 4–70 chart.

The authors agreed with Mr. Harvey in his comments on the importance of the shape of the engine iso-loops in determining the influence of the engine on the optimum efficiency of the complete propulsion system. The authors had for the past three years been concerned that the propeller should not be designed in isolation from the engine. In fact in a study of the *St. Finbarr* propulsion system they found exactly the same benefit of reduced fuel consumption with reduced rev/min and increased propeller pitch, as mentioned by Mr. Harvey.

In the new generation of Diesel engines now being manufactured, the designers have been applying an increasing skill in giving a larger "plateau" to the iso-loop diagram. This had meant, as in the example quoted in Appendix II, that the influence of the engine on the determination of the combined optimum was reduced from 10–15 per cent to less than two per cent of the optimum value. The authors thus felt that it was reasonable to simplify the arguments presented in the paper by confining them to the propeller, the influence of which, on the optimum value, could be as much as 30 per cent. The authors were well aware of the dangers of this simplification and had pointed this out in the last sentence of Appendix II.

With regard to the engine derating of 40 per cent shown in Figs. 12 and 13, this was merely chosen as an example to cover the existing range of derating of trawler engines, which was in some cases as high as 25 per cent, with a margin for extrapolation.

The authors did not agree with Mr. Hawdon that propellers for trawlers had always been fully successful in economic terms. Very often the power required for running free had meant that adequate power was available for towing even when operating well below optimum efficiency.

In answer to Mr. Hawdon's questions, the two parts of the paper were linked in that the first part showed that c.p. propeller data did agree with fixed pitch propeller chart predictions. The examples given in the second part thus rested on a reasonable foundation.

The analysis was carried out using thrust identity because this was the method generally used in analysing model experiments. All the data had been given in the paper so that anyone wishing to use torque identity could do so.

As regards the effect of increasing diameter on rev/min the point had not been highlighted in the paper but the numerical values were clearly shown in Fig. 10.

The example given by Mr. Hawdon demonstrated the increase in propeller rev/min which had taken place in recent years. With the introduction of Diesel machinery working at up to 500 rev/min and higher, this situation constantly needed to be reviewed.

The choice of a three-bladed propeller for *Arctic Freebooter* was made by the owners. Most stern trawlers had four-bladed propellers but, because the authors' Unit had equipped *Arctic Freebooter* with a data logging system, she was an obvious choice for trials purposes.

Finally, Mr. Hannan, and later Mr. Hawdon, in his written contribution, made the point that there was nothing new in the fact that larger diameter propellers are more efficient. The authors were at pains to point this out in the paper but would take this opportunity of repeating that the purpose of the study was to quantify the improvement and to provide techno/economic design data on which owners could base their decisions. Where capital costs were high, action would not necessarily follow a recommendation of this nature, unless the economic advantages of investing the additional capital were also presented.

# Marine Engineering and Shipbuilding Abstracts

No. 8, August 1968



### Parallel-gap Welding Techniques Using Automatic Pulse Compensation

The paper describes automatic pulse-width compensation as an important function of parallel-gap welding with emphasis placed on flatpack bonding to printed circuit boards. The basic problem in flatpack welding is outlined and the equipment necessary to do the job is described.

Parallel-gap welding is then defined along with a short discussion on important considerations for welding: electrode force, electrode gap, pulse amplitude and pulse duration.

The effects of integrated circuit lead composition, plating and size variations are clearly illustrated using metallurgical cross section analysis. Automatic compensation is then covered with a discussion of dual sensing as an integral part of parallel-gap welding.

A detailed section on weld scheduling, an area of much concern, features an easy-to-understand method of combining variables graphically.

The relationship of automatic pulse-width control to actual techniques used in parallel-gap bonding is discussed at some length. A comparison is made using Kovar leads welded to copper clad, nickel clad and copper-nickel clad circuit boards in terms of fixed and compensating pulses. Using optimum settings, the effects of both fixed and compensating pulse systems are clearly illustrated with photomicrographs, pull test data and detailed analysis.

Further parallel-gap applications are discussed including methods for flatpack replacement, thin film welding on glass substrates, impulse soldering and solder reflow through insulation.—*Koshinz, E. F., British Welding Jnl, February 1968, Vol. 15, pp. 53–62.*

### Unusual Vessel with Multiple Propulsion Machinery

The cargo motor vessel *Lady Jane,* built by T. van Duijvendijk's Scheepswerf N.V., Lekkerkerk, for B. v.d. Laan Scheepvaart- en Handelmaatschappij, Schiedam, the Netherlands, is an interesting vessel both from the transportation point of view as well as because of her propulsion machinery. The ship has been specially designed for the carriage of heavy and voluminous cargo and she is powered by eight DAF Diesel engines, driving two Schottel rudder propellers via Poly-V belt drives on the Schottel shafts.

The ship was to fulfil a number of requirements which would make her entirely different from other vessels that had hitherto been built. These requirements were:

1) The cargo to be carried is too heavy and too voluminous to be carried by road to the existing seaport. This meant that the vessel would have to have a shallow draught to enable her to load her cargo far inland from the port.
2) Manoeuvrability would have to be high.
3) In order to operate the ship with a small crew she would have to be below the 500 grt limit.
4) Stability would have to be such that the vessel would be capable of carrying approximately 1000 tons of cargo in the shelterdeck.
5) The minimum deadweight capacity had to be at least 1300 tons.
6) A multiple machinery installation was to be used in order not to make the operation dependent on a single engine.
7) The whole of the shelterdeck, from fore to aft, would have to be available for a cargo and the height available on maindeck had to be at least three metres.

The result was a ship of the following principal characteristics:

| | |
|---|---|
| Length, o.a. ... ... ... | 76 m |
| Length, b.p. ... ... ... | 70 m |
| Breadth, moulded ... ... ... | 13 m |
| Depth to maindeck ... ... | 2·90 m |
| Depth to shelterdeck ... ... | 6·20 m |
| Draught ... ... ... ... | 2·85 m |

The superstructure of the vessel consists of two entirely separate parts, arranged on the starboard and port sides of the vessel. The wheelhouse is situated on the starboard side, while an emergency wheelhouse has been built on top of the normal wheelhouse in order to have a clear vision when maneouvring in ports etc. even when the vessel is carrying tall cargo.

The machinery, which was delivered by Frans van Bodegraven N.V., consists of two groups of four six-cylinder DAF Diesel engines, each 165 hp at 1900 rev/min. These engines, which are of the type DP-680 M, drive two Schottel rudder propellers. In normal operation, three of the engines are used for each rudder propeller and are coupled to the Schottel shaft via a Turner Poly-V belt drive. The fourth engine drives at the fore-end a 75 kVA three-phase a.c. alternator via a Wülfel-Elco flexible coupling but can be coupled to the Schottel shaft if one of the three propulsion engines fails. The arrangement chosen for the installation is as follows: six engines with built-on electrically operated friction coupling are flexibly mounted on a bedplate, with after the coupling a Cardan shaft which drives another shaft on which a Poly-V sheave is mounted between two bearing blocks which can be moved to tighten the Poly-V belt. The output of the engines is transmitted from this sheave to the Schottel shaft by means of one Poly-V belt of M profile.

Two engines with built-on electrically operated friction coupling and a 75 kVA three-phase a.c. alternator which is coupled to the engine by means of a Wülfel-Elco flexible coupling, together flexibly mounted on a common bedplate.

Propulsion and steering of *Lady Jane* are effected by two Schottel rudder propellers of the type SRP 300 with a maximum continuous drive-torque of 275 kgm. As each of the propellers receives from the three DAF Diesel engines of 165 hp at 1900 rev/min via the belt drives with 1·1 : 1 reduction and a fluid coupling, a total engine power of 450 hp at 1660 rev/min on the shaft line, loading at 70 per cent is amply below the permissible load. The 4 : 1 reduction in the rudder propeller to the rudder results in a propeller speed of 415 rev/min.—*Holland Shipbuilding, January 1968, Vol. 16, pp. 30; 32; 34.*

### Container Cargo Ship for Rotterdam-Scotland Service

The 1450 dwt cargo motor vessel *Nieuwland* which recently entered the service of the SSM Transport N.V., Rotterdam (Management Scheepvaart en Steenkolen Maatschappij) represents a new development for container service between Rotterdam and Scotland. The ship has been constructed to a design made by the design department of the builders, Firma C. Amels and Zoon, Makkum, and incorporates a number of highly unusual features. Of special interest in this respect is the extremely large main hatchway, which extends over 58 per cent of the ship's length and 81 per cent of her breadth, and is covered by the largest hatchcover yet made by MacGregor. This large hatchway gave rise to the need for special construction, in particular in the transverse structure of the hull, in view of the torsion that might occur due to this arrangment. The solution found for this problem was the strengthening of the hull by a box girder supported on the tweendeck by diaphragm bulkheads and extending over 55 per cent of the length of the hull. This box girder is arranged as a tank.

As *Nieuwland* is called upon frequently to navigate rivers, manoeuvrability must be high. Consultations between the owners and the engine builder led to the conclusion that the ship was to be provided with a twin-engine propulsion installation combined with a controllable pitch propeller. At the same time this arrangement would make it possible to have one of the engines drive a three-phase alternator supplying the electrical energy to drive the bow thruster and, when at sea, to feed the ship's mains.

As a result, the installation is arranged so that in sea navigation the two main engines run at constant speed and in river navigation one engine is used for propulsion with variable speed, while the second engine, with constant speed, drives a generator supplying the electricity for the bow thruster.

The main propulsion machinery consists of two MWM Diesel engines, type TnbD 440-S, each developing 830 hp at 750 rev/min. The mean piston speed of these engines is 6·75 m/s and the mean effective pressure 11·1 kg/cm$^2$. The two engines drive via a Lohmann and Stolterfoht Pneumaflex flexible friction couplings, type KAE180M, and a Lohnmann and Stolterfoht reduction gear-box, type GVA1000W, with 2 : 1 reduction the Lips controllable pitch propeller while the port engine also drives a Still three-phase a.c. alternator which supplies the electricity to the bow thruster. The controllable pitch propeller is of the type 4LH70, with control mechanism 65 V 240. It is of the hydraulic type with pneumatic-electric control. The machinery installation is arranged for a temporarily unattended engine room and reduced manning scale.

The three-gear-designed double reduction gear-unit with single helical spur gears reduces the engine speed to 250 rev/min. The gear-unit is provided with a built-in white metal-lined thrust bearing of the Michell type.—*Holland Shipbuilding, February 1968, Vol. 16, pp. 30–32; 36.*

### Russian Interest in Gas Turbines Afloat

The relatively low weight and size of gas turbines make them suitable for powerful floating electric power stations in regions not connected to an electric power transmission system, where industrial development is in the initial stages or where a new ore deposit is being opened. Gas turbines offer special promise for hydrofoils and hovercraft. In these vessels the controlling factor is the low weight of the prime mover unit and the high power output of the set. Converted aircraft gas turbine plants have been adopted in such vessels as *Burevestnik, Sormovich* and *Tayfoon.* A special place is occupied by the free-piston gas generators. In respect of the latter a number of theoretical, design and experimental investigations has by now been completed which demonstrated engineering and economic feasibility of a joint use of gas turbine plants and free-piston gas producers for shipbuilding purposes. These have a favourable economic performance approaching Diesel units, good tractive indices and long service life while weights and overall dimensions are relatively small. These features of units with free-piston gas producers and the ease they can be mounted in engine rooms aboard ships and finally the possibility of firing heavy grade liquid fuel make them particularly suitable for transport and fishing vessels in the power range up to 5000–6000 hp.—*Gas and Oil Power, February 1968, Vol. 64, p. 61.*

### Variation of the Notch Effect with the Notch Inclination Angle

The angle $\phi$ between the centre-line of V-groove and the axis of a rod specimen was called the notch inclination angle and the variations of notch strengths of mild steels with the variation of $\phi$ were studied at room temperature. In impact tensile experiment, the impact strength varied scarcely with $\phi$ when $\phi$ was near 90° but it varied when $\phi$ was near $\phi_s+25°$, where $\phi_s$ is an angle corresponding to a shoulder fillet. In static tension experiment, although the notch strengthening appeared at breaking load, the relation between actual breaking stress and $\phi$ was similar to that of impact strength qualitatively. Yield strength decreased linearly with the increase of $\phi$ when $\phi$ was not so close to $\phi_s$. The variation of elongation percentage and of the contraction percentage of area corresponded to the variation of the impact strength and of the breaking stress, respectively.—*Maekawa, I., Bulletin Japan Society of Mechanical Engineers, 1967, Vol. 10, pp. 872–880.*

### Fatigue Tests on Butt Welds with Slag Inclusions

The work described in this report forms part of an extensive investigation carried out at B.W.R.A. into the effect of weld defects on the fatigue strength of butt welds in mild steel. It also forms part of a collaborative programme which is being conducted under the auspices of Commission XIII of the International Institute of Welding and other countries are carrying out parallel investigations.

Further tests of $1\frac{1}{2}$ in thick specimens with discrete inclusions and continuous lines of slag are described. These tests were conducted to investigate the effects of defect size, preheat, stress relief and stress ratio and to throw further light on anomalies found in earlier work. One general result which emerged from the work is that small extraneous defects, of a size not normally considered worthy of rejection, frequently acted as fatigue crack initiators in preference to the large intentional defects. The effect of stress relief was not necessarily beneficial. For example, where the defect was a continuous slag line central in the thickness, a considerable reduction in strength occurred.—*Harrison, J. D., British Welding Jnl, February 1968, Vol. 15, pp. 85–94.*

### Influence of Sulphur on Heat Affected Zone Cracking of Carbon Manganese Steel Welds

Sulphur is normally considered a deleterious element in steels which have to be welded and a level of 0·05 per cent maximum is often specified (e.g. BS. 1501 series). The combination of basic electric melting and vacuum degassing in modern steelmaking processes can give a steel which is exceptionally low in sulphur and other non-metallic impurities. Previous experience suggested that such a steel should have excellent weldability but, contrary to expectations, it is shown that steel with abnormally low sulphur can be exceptionally sensitive to hydrogen-induced heat-affected zone cracking.

A mechanism relating sulphur content and void volume to crack susceptibility is briefly introduced and is shown to account for the observed behaviour of material containing various levels of sulphur during the manufacture of components for large steam generators.—*Smith, N. and Bagnall, B. I., British Welding Jnl, February 1968, Vol. 15, pp. 63–69.*

### Automatic Starting for Marine Gas Turbines

The Rolls-Royce Olympus gas turbine engines built into a number of vessels at present under construction for foreign navies are being fitted with automatic starting-up control panels. These panels, which are being installed by Vosper Electric, provide completely automatic control of the engine until it reaches its safe idling speed and enables an operator to start the engine merely by pressing a single button.

Several checks must be made before the starting sequence can commence and while it proceeds. To enable the checks to be made automatically a series of microswitches, pressure transducers and tachogenerators are fitted to the engine and, if the signals from these indicate that the state of the machinery is satisfactory, the 30 s starting sequence will commence once the start button is pressed. The sequence is automatically stopped if any signal occurring before or during the sequence indicates an unsatisfactory condition. If the gas generator fails to light up the system is automatically switched off and the effect is as if the stop button had been pressed.—*Industrial Electronics, February 1968, Vol. 6, p. 90.*

### Propeller Design and Analysis by Lifting Surface Theory

Two problems in propeller theory are considered in this paper: the design of a propeller to meet a specified performance and the calculation of the performance of a given propeller operating under specified conditions.

In the case of the first problem, the requirement is to design a propeller to produce a stipulated thrust under specified operating conditions. It is assumed that an approximate loading distribution has been chosen and that the number of blades, the blade outline and the thickness distribution have been decided. Some technique is then necessary to produce the blade pitch angle and camber line profile at selected radial positions, the torque needed and the distribution of cavitation number over the blade.

The basic equations of lifting surface theory are derived and used to provide methods for the design and evaluation of propellers. For the design calculation, the method requires, as data, the number of blades and the blade outline and thickness distribution, the wake inflow pattern and advance ratio and the required thrust together with the shape of the circulation distribution over the blade surface; the method will then yield a value of the torque required and also the blade pitch angle distribution and the distribution camber, cavitation number and actual circulation values over the blade surface. For the evaluation calculation, the method requires full details of the propeller and operating conditions and will yield thrust and torque values and also surface distributions of circulation value and cavitation number.

The treatment includes the case of a pair of contra-rotating propellers and the similar case of a rotor/stator combination. The method will treat straight or skewed propellers.

The numerical techniques used in the calculations are described. Some information is also given about the KDF9 computer programme which was written to carry out the computation. Numerical examples are presented.—*Murray, M. T., International Shipbuilding Progress, December 1967, Vol. 14, pp. 433–451.*

### Utilization of Propeller Shrouds as Steering Devices

The flow through a propeller shroud or nozzle accelerates or decelerates according to the type of shroud; there is consequently a pressure differential between the inside and outside of the shroud. In the normal axisymmetric shroud, the radial force due to this pressure difference is uniform around the complete circumference and therefore has a resultant value of zero. It is possible to design a shroud to have a circumferential variation in the pressure differential; such a shroud will produce a resultant radial force. This force can be substantial and can be used advantageously to steer the ship or other fluid-borne vehicle.

A simple example illustrating the production of such a radial force is a semicircular shroud of the accelerating (i.e. Kort) type. There is no significant change in the pressure distribution around this half-shroud, except at its tips. The resultant force acts radially inward (there is an internal pressure drop, due to the venturi effect, in way of the half-shroud), and by rotating the half-shroud the force can be directed to either side to steer the ship.

Steering with a shroud instead of with a normal rudder should give these advantages:

1) elimination of rudder drag when going ahead;
2) substantial elimination of rudder drag at all turning rates;
3) elimination of the need to reverse the rudder angle with reversing engines while turning, as direction of side thrust is independent of direction of flow through propeller;
4) shroud can produce upward lift at the stern and so give a virtual decrease in displacement;
5) in the accelerating type of shroud, the trust augmentation should increase propulsor efficiency;
6) shroud loading distribution should minimize the likelihood of propeller cavitation.

Possible disadvantages are:

1) loss of steering capability with engine stopped;
2) structural requirements for shroud support and control;

3) vibration due to propeller/shroud interaction, particularly with partial shrouds.

The shroud need not be incomplete or rotatable; the essential requirement is ability to control the circumferential pressure distribution on the shroud. Among possible alternatives to the partial shroud are:

a) a rotatable complete shroud varying from decelerating sections at the top (i.e. the top during normal straight-course running) to accelerating sections at the bottom;
b) a complete shroud trainable about a vertical axis, usually called a nozzle rudder;
c) a complete shroud consisting of fixed portions and portions which are moved to vary the circumferential pressure distribution.

The paper includes a detailed theoretical consideration of the performance prediction for steering shrouds, with a worked example for the case of a semicircular shroud in conjunction with an 11-ft diameter propeller for 20 knots at 6000 shp. Maximum lift predicted for a spade rudder is only 68 per cent of that predicted for this shroud; also, at maximum lift, the rudder produces a drag of 15 000 lb whereas the shroud gives a thrust of 4000 lb. The method of prediction takes into account the interaction between shroud and propeller.—*Gordon, S. J.* and *Tarpgaard, P. T., January 1967, New England Section of the Society of Naval Architects and Marine Engineers Paper; B.S.R.A. Jnl, December 1967, Vol. 22, Abstract No. 25 964.*

#### Constant Pressure Turbocharging

Experience with large engines has shown that at full load and high b.m.e.p.s, constant pressure turbocharging is superior to the impulse system. In two-stroke engines this superiority depends on the number of cylinders or on the cylinder/blower arrangement used in the impulse system and becomes evident at pressures above 8–9 kp/cm². Constant pressure charging, however, cannot be used at part-loads without some special additional means of supplying air, such as on auxiliary blowers or the injector system used by MAN, which can provide comparable performance and acceleration. MAN's 1020 mm bore three-cylinder experimental engine is at present arranged with the MAN series-parallel system of pressure charging but it is anticipated that a constant pressure system will be fitted for further tests in the future.—*Marine Engineer and Naval Architect, February 1968, Vol. 91, p. 67.*

#### Dynamics of Marine Steam Propulsion

The steam turbine power plant on board ship is, in many ways, analogous to an electronic amplifier. In the electronic circuit, resistances regulate the d.c. current flows, condensers store energy and introduce time-lags. All these parts must work together, or heavy instability will be introduced in the output as the amplifier is moved from steady-state conditions by addition of an input signal.

A few years ago, regulation of all the flows in a steam plant, i.e. the steam, oil and air were carefully watched by a number of operating engineers, but now that one flow controls another in the automated steam plant, the design must carefully be checked on a time scale so that unexpected oscillations do not occur.

In the steam plant, a mild unstable transient is shown by heavy black smoke during manoeuvring, and in the worst cases violent hunting of the forced-draught fans.

The two major problems associated with marine power plants are smoking, caused by turbine transients or burner change, and maneouvrability restrictions, caused by steam pressure deviations or drum-level instability.

The refitting of the ore carrier *Tom M. Girdler*, provided an opportunity to study these problems through an analysis of both the combustion process and the feedwater system. To produce valid data without recording delays, electronic transmission and recording devices were installed on *Girdler*.

The curves show what happens to the steam flow, oil pressure and air flow when the steam turbine throttle is moved. To hold steam pressure within reasonable limits, the fuel and air must decrease rapidly and recover slowly. The fuel, however, must decrease somewhat faster than the air to prevent smoking. The sample system and oxygen analyser total delay was approximately three seconds at all loads.—*Larson, M. N.* and *Martz, L. F., Marine Engineering/Log, February 1968, Vol 73, pp. 61–63.*

#### Ultrasonic Probes Using Shear Wave Crystals

The potential scope of ultrasonic shear wave testing has generally not yet been exploited because of the inherent limitations of bonding, mode conversion or other highly specialized coupling methods used at present. Such shortcomings are substantially avoided by means of pressure or optical contact coupling of SV and SH shear waves at all angles of incidence, or by means of a fluid film for coupling SV shear waves at oblique incidence. Because shear waves interact with materials and stress in a manner different from longitudinal waves, shear waves can be used to determine material properties which are difficult or impossible to obtain with longitudinal waves. Testing possibilities are grouped according to whether the incident shear wave enters the specimen normally or obliquely. Applications of shear waves include measurements of temperature, thickness and flaws at elevated temperature, shear wave inspection and characterization of low velocity materials and determination of stress, strain and material orientations which are sensitive to shear wave polarization. Additional unique advantages of incident waves are obtained by mode converting these waves to Rayleigh, Lamb, longitudinal and Love waves. These waves, in turn, may be applied to Rayleigh, Lamb and Love wave inspection at high frequencies, above 10 MHz, or in low velocity solids such as graphites or plastics. Longitudinal wave tests in fluids may be accomplished in extreme environments by using a shear wave buffer rod to isolate the crystal from the exposed portion of the probe. Longitudinal waves can also be refracted more favourably with respect to certain thermometry and flow measurement problems, by using an obliquely incident shear wave.—*Lynnworth, L. C., Materials Evaluation, December 1967, Vol. 25, pp 265–277.*

#### Excursion into Depth

Diving experiments have provided important new information on long-term dives. Three engineers of the Westinghouse Underseas Division's life support engineering group lived for four days under pressures identical to those found at water depths of 400–600 ft. Manual dexterity and mental alertness tasks were performed and prototypes of new diving equipment were tested.

According to a life support advisory engineer, the experiments proved man's ability to perform work tasks as effectively at 600 ft as he can now perform at the 100-ft depth.

Using the existing Westinghouse prolonged submergence diving system called Cachalot, the engineers, working under a basic pressure equivalent to that of 400 ft of water, made two 600-ft excursion dives of 90 min and 40 min duration.

These excursions proved the accuracy of Westinghouse-developed decompression tables for depths of 400–600 ft.

Among the equipment prototypes tested were a new type of diver breathing apparatus and a helium speech unscrambler. The diving apparatus was a mixed gas, closed circuit piece of equipment weighing only 60 lb and for use at depths of 1500 ft. Both the test chamber and the new diving rig utilize a gas mixture of helium and oxygen. Air is generally not used at depths greater than 150 ft because of the narcotic effect of nitrogen. Closed circuit means that the diver is re-breathing his gas mixture after carbon dioxide is filtered out and oxygen replenished, and none of the gas is allowed to escape into the water. When this closed circuit apparatus comes into use later this year, it will drastically reduce diving costs by saving the expensive helium that is exhaled to the water in conventional deep diving equipment.

The interesting design feature of the diving apparatus is that it reduces the diver's resistance to breathing. In deep water, breathing difficulty greatly contributes to diver fatigue; at the 400-ft depth and beyond, breathing becomes a severe problem.

The helium speech unscrambler also tested in the project is a new device that makes intelligible the high pitched, rapid speech of the helium-breathing diver. The large volume of helium in a breathing mixture makes it less dense than air; the interaction between this lighter density gas and the vocal cords of a diver results in "helium speech".—*Mechanical Engineer, August 1967, Vol. 89, p. 38.*

#### Developments in Navigation Engineering

Steady development of existing navigational devices including the gyro compass, radar, sonar, auto pilots, Decca Navigator and Loran will continue but the most interesting developments for engineers will lie in a number of completely new concepts now in the development stage or already at sea in advanced naval vessels.

Today's gyroscopes are, unfortunately, far from perfect due to such factors as bearing friction in the rotor and gimbal system, windage loss between the rotor and case and the preservation of perfect balance over extended periods of time. A promising development to overcome these drawbacks is being carried out at the present time by using laser or coherent light beams transmitted round an orthogonal or polygonal track by using mirrors to turn the light beam. The rotating mass is almost negligible, but the rotational speed is very high; therefore, gyroscopic inertia is present, but not subject to frictional and balance inaccuracies. Provided optical equipment and associated servo systems and measuring devices can be developed to detect the very small change in the track of the light due to the tilt of the spin axis as the earth rotates and the precision of the mass as the result of applied external forces, then there is no reason why future gyro compasses should not be based on this concept.—*Carr, J., Motor Ship, February 1968, Vol. 48, pp. 533–536.*

#### New Modular Alarm System

Two new surveillance systems, an advanced fully transistorized alarm unit and an electronic temperature monitor with individual alarms are now being marketed by Danish control engineering specialists Søren T. Lyngsø.

The type 214 transistorized alarm system, built up from standard constructional elements to meet precisely the requirements of each individual installation, centres in an advanced 10-channel modular alarm panel. The unit is designed for an alarming sequence which is in widespread use today, a steady green light changing to red, flashing (with accompanying audio warning) in the event of an alarm, to steady red upon acceptance and, finally, to green once again when the parameter returns to its normal state. The method of working has as its basis closed circuit operation of the contacting devices.

Besides meeting all normally occurring alarm conditions the 214 unit is also provided with a cutout circuit to suppress undesired alarms (such as those occurring when auxiliary machinery is started and stopped) and where required transistorized time lag units may be built in to provide adequate delay.

Transistors are of the silicon type and all circuits are printed on glass fibre. The logic circuitry contains no relays or other moving parts; thus a high degree of reliability is ensured. The transistorized circuits, moreover, have been specially developed with a view to suppression of noise pulses that may be induced in the cables for alarm contacts, horn and battery. The source of the induction may often be other cables for light and power and the noise pulses may give rise to false alarms.

The other new instrument, the Thermonitor, is used for load-dependent surveillance of Diesel engines and is designed for monitoring temperatures at up to 12 different measuring points; it can be extended to embrace a further 12 measuring points, bringing the maximum utilization up to a total of 24 points.—*Shipbuilding and Shipping Record, 18th January 1968, Vol. 111, pp. 96–97.*

#### Canadian 200 ft Tuna Seiner

Built by Yarrows Limited, Victoria, B.C., new 200 ft *Atlantic Gennis* has been completed and has proceeded via the Panama to duty on the Atlantic coast. This is the second of two tuna seiners built by Yarrows, for the Canadian Tuna Company of St. Andrew's, N.B.

*Atlantic Gennis* is air-conditioned throughout for service in tropical waters, with spacious accommodation for her crew of about 18. She has a cruising range of 16 000 miles and a normal speed of 13 knots.

Her bridge is equipped with such navigational aids as gyro compass, automatic pilot, radar, loran, electric log and sonar and special gear for locating fish, and for recording water temperatures.

The fish are dropped through hoppers on the upper deck into freezing tanks between decks which can freeze 100 tons of fish a day. A conveyor belt moves the fish to the forward end of the hold for unloading and they are taken to the upper deck by a portable bucket elevator.

The ship is equipped with a bow thrust unit to assist the ship's manoeuvring during fishing, catch loading and berthing.

The main hold, which has a capacity of 900 tons, is kept at a temperature of $-20°C$. Propulsive power is provided by an Atlas-Polar six-cylinder Diesel engine, developing 2400 hp at 250 rev/min.—*Shipbuilding and Shipping Record, 23rd February 1968, Vol. 111, p. 274.*

#### Marine Diesel Exhaust Noise

In order to calculate exhaust noise spectra (e.g. on bridge wings) a simple mathematical model is given of a large turbo-supercharged marine Diesel engine as a source of sound, connected to an exhaust system with or without muffler. The exhaust gas turbines are considered as high internal impedance sound sources. Spectra are computed with the aid of the Fourier analysis of a periodical trapezium gas flow function in combination with the transfer function of the exhaust system. The latter function is determined from scale models. The influences of engine power, number of cylinders, number of revolutions per minute, turbine nozzle area etc. on the sound pressure level are investigated. Continued research is needed especially on some of the tentative conclusions which indicate for example that a factor two increase of the power per cylinder implies an increase of the sound pressure level by 3 dB and, moreover, that this level is reduced by at least 24 dB if the number of rev/min of the crank and propeller shaft is reduced by a factor two.—*Janssen, J. H., Shipping World and Shipbuilder, March 1968, Vol. 161, pp. 529–536.*

#### First of Four Sister Ships for French Owners

The ship described briefly in this article is the first in a series of four similar vessels ordered from the Chantiers Navals de La Ciotat, La Ciotat, France, for the Société Navale

Delmas-Vieljeux. This vessel, named *Lucie Delmas*, 13 281/14 058 dwt, is powered by a Sulzer type Diesel engine and has a service speed of 17·5 knots. She is trading between European and West African ports.

*Lucie Delmas* has been constructed under the special survey of Bureau Veritas, and has been designed to carry general cargo and tropical produce. In addition, she has been strengthened to enable bulk timber to be carried on deck and in the holds. A small amount of refrigerated cargo and vegetable oil may also be carried.

In port all commercial operations are carried out from an office on the after end of the upper deck. From this position the officer on watch is able to communicate by inter-com with the captain, chief officer, and chief engineer: he can also call out the crew by means of an alarm network. The correction of trim, by ballast water or Diesel oil transfer, can also be carried out by remote control from this office.

Principal particulars are:

| | |
|---|---|
| Length, o.a. ... ... ... | 528 ft 6 in |
| Length, b.p. ... ... ... | 482 ft $3\frac{1}{2}$ in |
| Breadth, moulded ... ... | 68 ft $10\frac{3}{4}$ in |
| Depth to upper deck ... ... | 39 ft $4\frac{1}{2}$ in |
| Depth to second deck ... ... | 27 ft $0\frac{3}{4}$ in |
| Draught, summer freeboard ... | 29 ft $9\frac{1}{8}$ in |
| Deadweight, corresponding ... | 13 281 tonnes |
| Displacement, corresponding ... | 19 519 tonnes |
| Draught, maximum freeboard, timber ... ... ... ... | 30 ft 7 in |
| Deadweight, corresponding ... | 14 058 tonnes |
| Displacement, corresponding ... | 20 296 tonnes |
| Cargo hold capacity, bale ... | 608 089 $ft^3$ 17 219 $m^3$ |
| Machinery output, m.c.r. ... | 13 800 bph at 119 rev/min |
| Service speed at 85 per cent normal maximum output ... | 17·5 knots |

*Lucie Delmas* is propelled by a Sulzer type 6 RD 90 two-stroke turbocharged Diesel engine wtih a maximum continuous rating of 13 800 hp (M) at 119 rev/min with a mean effective pressure of 118·5 $lb/in^2$ and maximum output for one hour of 15 180 hp (M). This engine can be controlled remotely both from the wheelhouse or from a soundproof compartment located on the port side of the engine room.—*Shipping World and Shipbuilder, March 1968, Vol. 161, pp. 551–554.*

#### New Long Range Ocean Going Warship

Yarrow (Shipbuilders) Ltd. has recently evolved a new type of warship—Yarrow Frigate. She is 308 ft in length by 34 ft beam, has a load draught of 10 ft 9 in and a load displacement of 1600 tons—dimensions which not only present a craft suited to seagoing requirements but one which, if required, can manoeuvre in inland or restricted waters.

A most revolutionary feature of the whole design is the main propulsion machinery. A twin-screw combined Diesel or gas turbine (CODOG) was selected after detailed investigation into all possible alternatives, and considering such factors as speed, range, weight, size of engine room, reliability, cost, size of propellers, manoeuvrability, military acoustic properties, availability of tried and proven engines and ease of maintenance and overhaul. The saving that has been achieved in length of machinery spaces can easily be seen if the figure of 60 ft is compared with that for any other frigate of comparable speed.

Normal cruising and operational speeds up to about 16 knots are obtained from the Diesel which develops about 3900 hp, whereas speeds of up to 27 knots can be obtained from the gas turbine which is capable of developing almost 20 600 hp. Cruising range employing the Diesel drive is approximately 5200 nautical miles while, alternatively, using the Rolls-Royce Olympus gas turbine, a range of 1000 nautical miles is possible. Each engine is located in a separate compartment, the Diesel being aft of midship with its drive directed in a forward direction into a double reduction gear-box whereas the gas engine in its compartment drives aft to the same train of gears.

The gear-box accepts power from only one prime mover at a time through automatic synchronizing clutches and conveys it to twin propellers each rotating in an opposite direction. As the combined gear-box is already a more complicated gearing design than for a single propeller shaft, it has been considered prudent not to complicate this component by providing for reversing arrangements in the gear but instead to fit controllable pitch propellers.

Twin spade-type rudders and an interlock between the steering and the propellers give the ship a remarkable degree of manoeuvrability.

A final and important refinement of the overall machinery controls guards against excessive stresses which may develop in the inside shaft during a turn. This is achieved by a signal actuated by the rudder movement being fed into the pitch control boxes so that the pitch can be reduced on the appropriate shaft.—*Shipping World and Shipbuilder, March 1968, Vol. 161, Special number "Naval Construction", pp. 17; 19; 21.*

#### Twenty Years of Research under the Ship Structure Committee

The salient results of two decades of research into such broad areas as materials, physical and qualification tests, non-destructive testing of welds, stress distribution, data from ships in service and bending moment determination by models are presented. The brittle fracture phenomenon is attacked from a number of fundamental views: variation of composition and microstructure, prior strain and thermal history, rate of loading, stress intensity and distribution, effect of flaws and variation of test method. All these bits of research are shown to have contributed to the attainment of an engineering solution. Several of these areas are further discussed. Stress distributions at various geometrical discontinuities are reported as are those arising from temperature gradients. Stress intensities measured on ships in service are presented as are the long-term predictions from these measurements. The effect of mill practice on material performance is discussed. The current effort of developing generalizations from results of specific research is described. The trend of the future, the adaptation of computer to exploit generalizations for analysis of whole systems of structure, is set forth. Finally, a listing of nearly 200 reports is given along with information on how to obtain them from Federal repositories.—*Heller, S. R., Lyttle, A. R., Nielsen R. and Vasta, J., November 1967, Society of Naval Architects and Marine Engineers Paper.*

#### Shipboard Electric Surface Heating

Electric surface heating, as opposed to immersion heating, refers to heating units fitted to the surface of a pipe, tank or other object to be heated. The heaters are normally complete units consisting of heating surface, thermal insulation and outer metal enclosure.

Many applications of electric surface heating are conducted as a combined operation. A flexible surface heater is fitted to the equipment to be heated and then separately thermally insulated and protected as necessary. For this purpose electric heating tapes, known as Isotapes, each consisting of an insulated heating element applied or interwoven in the form of a loop with a glass cloth base are used extensively for the heating of pipelines.

With electric heating of pipelines, as compared with steam heating, only the heat losses in the pipe need to be replaced. An accurate determination of load requirements can be made at the planning stage as heat losses to be expected can be determined. Length of pipeline does not affect performance; extreme weather conditions are simply catered for by having a slight amount of power in hand.

Moisture-proof heating tapes Type ITW are most commonly employed, carrying a fluorocarbon insulated heating element providing protection against accidental spillage or wetting.

For frost or anti-freeze protection on board ship Type ITBR/MAR is used, being a PVC extruded heating tape fitted with an external copper braid for protection against accidental mechanical damage.

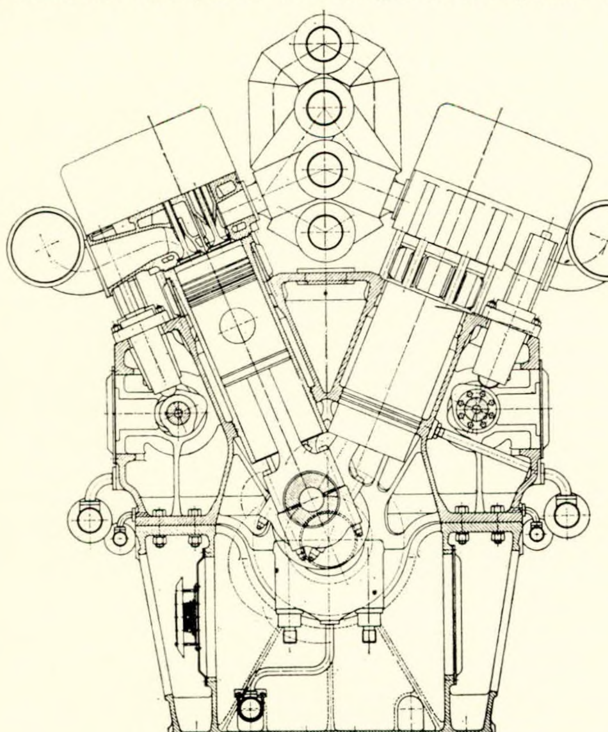
Where especially arduous conditions prevail, Type FTA is applied. The heating element is metal sheathed and magnesia insulated, fitted to a woven glass cloth base and terminated in $\frac{3}{4}$ in glands.

All these tapes have application to a wide range of marine installations; from hull fittings and cargo handling gear to engine room systems and domestic services.—*Young, J. W., Tanker and Bulk Carrier, February 1968, Vol. 14, pp. 553–554.*

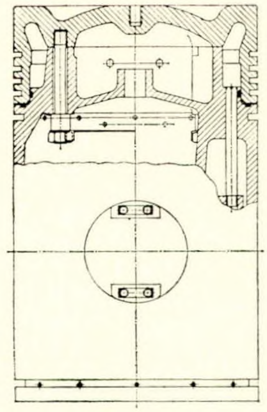
**Twin Geared Engines for Bulk Carriers**

Scott Lithgow Ltd. have received an order from A/S Kristian Jebsen Rederi of Bergen and their associates for six 20 000 dwt bulk carriers for their contract charter business in the chemical fertilizer and mineral sands industries. The ships will be built to norske Veritas Class with the EO notation, denoting an unmanned engine room. The machinery is to consist of two Burmeister and Wain vee-8 8U45HU medium-speed engines with a continuous service output of 4000 bhp at 450 rev/min each and geared to a single shaft through Renk reduction gears. This machinery will be installed by John G. Kincaid and Co. Ltd. of Greenock, B and W licensees since 1924, who celebrate their centenary this year.

The U45H 450 mm × 540 mm four-stroke engine has a cast iron frame carried on an exceptionally deep cast or fabricated bedplate and the crankshaft, a solid forging of 70 kg/cm$^2$ chrome-molybdenum steel with 330 mm diameter, journals and crankpins is underslung from the cylinder block.



*Section through the B and W U45HU engine with deep bedplate and large access doors*



*Section through the two-piece piston. Cooling oil is delivered via the connecting rod and side tube to the gallery around the rim of the crown*

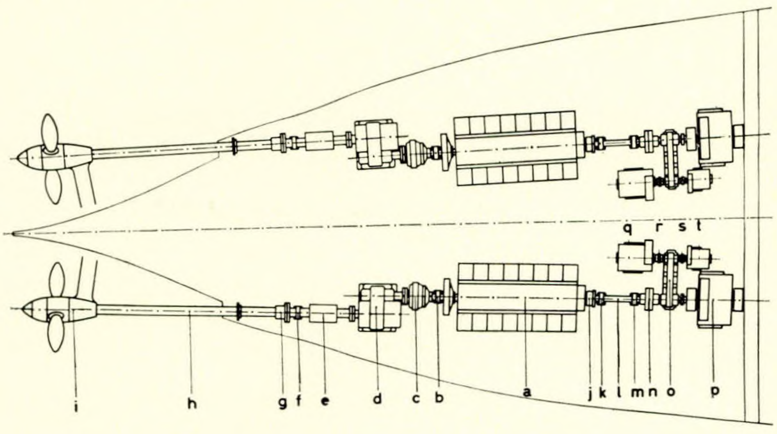
An interesting design feature is that the bedplate/crank chamber diaphragms, which are non-load-bearing, are offset from the main bearings so that the latter can be lowered directly into the crankpit by means of an hydraulic jack. The cylinder block is split transversely and secured by substantial bolts, while the main bearing studs at this point are screwed into cylindrical yokes which extend through both frame parts. Very large crankcase doors give ready access to the main bearings and side-by-side connecting rods which have thin precision-bored bearing shells.

Two inlet and two exhaust valves are operated by short push-rods and forked rockers from the side camshafts. The pistons have lightweight cast iron skirts and bodies supporting a cantilevered steel crown with forced-circulation of cooling oil. The U45HU engines are all vee-form and made with from eight to 18 cylinders covering a power range from 4750 to 9900 bhp. The weight of the eight-cylinder model is 66 tons with cast iron bedplate and 64 tons with a fabricated one.—*Marine Engineer and Naval Architect, February 1968, Vol. 91, p. 47.*

**High Strength Nickel-base Alloy with Improved Oxidation Resistance up to 2200°F**

A high strength nickel-base alloy has been developed which compares favourably in oxidation resistance with known high strength nickel-base alloys. The alloy, although basically a cast material, also possesses workability potential. After 310 h exposure to air at 1900°F, the alloy had a weight gain of 1·8 mg/cm$^2$. The total affected zone, oxidized material plus depletion zone, was 0·4 mil. This compares with a weight gain of 3·0 mg/cm$^2$ and a total affected zone depth of 3·3 mils for René 41 after 100 h exposure at 1900°F. In sheet form after 8 h at 2200°F, its oxidation resistance was approximately the same as that of René 41 at 1900°F. Tensile strengths of the alloy after rolling and heat treatment ranged from an average of 185 000 lb/in$^2$ at 1400°F to 3000 lb/in$^2$ at 2200°F. Maximum elongation was 55 per cent and occurred at the latter temperature. At 1900°F, average tensile strength was 64 500 lb/in$^2$ in the as-cast condition, and 54 000 lb/in$^2$ after rolling and heat treatment. Stress rupture data for low and intermediate stress levels were obtained. In the as-cast condition, temperatures used for 500, 100, and 10-h life at 15 000 lb/in$^2$ were 1815, 1895, and 2010°F, respectively. At 8000 lb/in$^2$ and 2125°F, rupture life was 13 h and compared favourably with some of the strongest known nickel and cobalt-base alloys. The very good high temperature oxidation resistance, good high temperature strength and at least limited workability of this alloy suggest that it may be applicable for use in advanced gas turbine engine components. —*Waters, W. J.* and *Freche, J. C., Trans. A.S.M.E., Jnl Engineering for Power, January 1968, Vol. 90, pp. 1–10.*

**World's Largest Dredger**




*a) M.A.N. main engine*
*b) Journal bearing LRF 330*
*c) Pneumaflex clutch KAE 380*
*d) Reduction gears GUY 7449/7450*
*e) C.r.p. propeller control*
*f) Journal bearing LRL 420*
*g) Connecting flange*
*h) Propeller shaft*
*i) C.r.p. propeller*
*j) Gear coupling DentiLus KDZ 1468*
*k) Journal bearing LRF 330*
*l) Intermediate shaft*
*m) Journal bearing LRL 330*
*n) Elastic shaft coupling*
*o) Transmission gear GAZ 7446*
*p) Generator 2300 kW*
*r) Coupling REWN 450*
*s) Coupling REWN 110*
*t) Generator 240 kW*

*Engine room plan showing arrangement of geared Diesel-propulsion/Diesel-electric dredging power plant*

The largest dredger in the world has been launched at Schiedam. The ship was named *Prins der Nederlanden* and will fly the British flag.

The principal particulars of the dredger, which has been built to Lloyd's Register class, are as follows:

| | |
|---|---|
| Length, o.a. ... ... ... | 469 ft 2 in |
| Length, b.p. ... ... ... | 430 ft 0 in |
| Breadth, moulded ... ... | 47 ft 10 in |
| Depth ... ... ... ... | 39 ft 4 in |
| Draught ... ... ... ... | 33 ft 0 in |
| Hopper capacity ... ... ... | 11 700 yd³ |
| Diameter of suction pipes ... | 48 in |
| Maximum dredging depth ... | 115 ft 0 in |
| Deadweight (approx.) ... ... | 18 000 tons |
| Total installed power ... ... | 20 000 hp |
| Loaded speed (running free) ... | 15 knots |

The main and auxiliary machinery spaces are aft, the pump room is forward and in between are the hoppers. The hopper top is covered by the continuous main deck which adds substantially to the strength of the vessel. A track for a 20-ton crane extends over the full length of the hopper. This crane has an outreach of 36 ft and can reach most of the vessel's deck area. The twin funnels combine the functions of masts, tied by a span at the top which supports a top mast and radar scanner. The open deck aft is used as a landing space for a helicopter, which will be used for relieving the crew and for rapid supply of urgently-needed spares.

The pump room forward of the hoppers, contains two 9 ft 1in diameter dredging pumps, each driven by two 1400 shp 750/940 rev/min motors through flexible couplings and a twin input single output reduction gear giving shaft speeds of 132/166 rev/min.

There is a dredging bridge at each corner of the wheel-house, from which the dredge master can oversee the whole operation, rate of pumping, density of spoil, depth of cut, etc. The hopper has two rows of 13 hydraulically-operated 11 ft 10 in diameter conical doors. These are designed to a successful IHC patent.

A powerful bow thruster, arranged forward, has no need for a separate electric motor drive, since it operates by diverting the discharge from one of the powerful dredge pumps.

The gear-box for each dredging pump also incorporates a turning-gear motor, triple-reduction gear and clutch. The pumps are arranged in a compartment separate from the motors and gear-boxes, to provide watertight integrity in the event of an unexploded bomb or similar object being drawn up the large diameter suction pipes. The suction pipes are suspended on each side by means of three fabricated gantries and electric winches.

Whereas the dredging machinery is Diesel-electrically powered, the ship is propelled by geared Diesel transmission. The main and dredging pump engines consist of two 16-cylinder M.A.N. V8V40/54 engines of the latest highly-rated medium-speed design, each of 8440 bhp at 400 rev/min. These each drive a KaMeWa c.p. propeller of 14 ft 9 in diameter through Pneumaflex pneumatic resilient clutches and 400/125·4 rev/min reduction gears with hardened and ground teeth. The gear-boxes incorporate the propulsion thrust blocks. A Cardan shaft extended from the forward end of each engine, with suitable flexible couplings, drives a 2300 kW generator for the dredging pumps and, by means of a take-off gear, two 240 kW generators for providing power for the dredging winches.—*Marine Engineer and Naval Architect, February 1968, Vol. 91, pp. 62–63.*

**Electric Fishing**

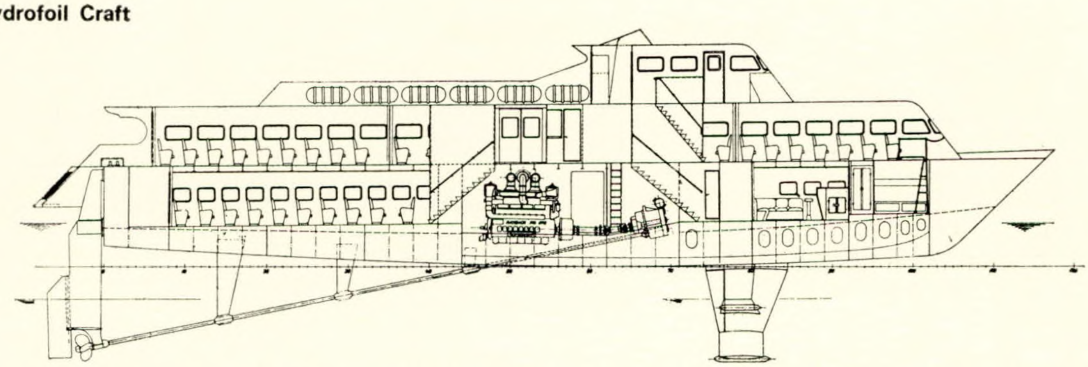
The feasibility of effective, relatively low-cost electronic fishing by which fish are attracted to an electrode placed in the water has been demonstrated at the University of Michigan. The system, which involves lower power supplies than previously considered, will make use of electrode screens operated from boats to sweep or herd fish towards the mouth of a net or the intake of a fish pump.

The system gives promise that power supplies of 5–10 kW can be as effective in fresh water as previous methods which would require 50 kW supplies. A composite pulse with a ten per cent duty cycle has been shown in the laboratory to be just as effective as a solid pulse. The low average power of the composite pulse used is expected to produce less fatigue of the fish, while allowing longer range, lower operating costs and lighter equipment.

Because the high conductivity of salt water requires 100 times the power needed for fresh water, the composite pulse is even more important in holding average power to reasonable levels. Previous electrode arrays often created a strong field which prematurely stunned the fish. Such equipment had an effective range of only ten to twenty feet while the equipment now being studied is effective up to several hundred feet.

Authorities estimate that two or three times as many fish escape as are caught by conventional nets. Proper electronic fishing might increase the catch by 100 to 200 per cent. Further economic savings can be made by combining electronic fishing with the use of fish pumps for emptying trap nets, while a further advantage of electric fishing is that it is selective of fish length.—*World Fishing, March 1968, Vol. 17, p. 39.*

### Seagoing Hydrofoil Craft



*Arrangement of the PT150DC seagoing hydrofoil passenger/car ferry in passenger form with shaft bracket system and 16-cylinder engines but no torque converters. Removal of the 100 seats in the upper saloon aft leaves space for eight cars which could be loaded over the stern*

The largest seagoing passenger hydrofoil craft yet built is now reaching completion by Westermoen Hydrofoil A/S Company at Mandal in Norway. This ship, which has been built to the designs of Supramar AG of Lucerne, is a development of the very successful PT50 and PT20 of this firm.

*Principal particulars:*

| | |
|---|---|
| Length, o.a. ... ... ... | 119 ft 6 in |
| Beam, moulded ... ... ... | 22 ft $11\frac{3}{4}$ in |
| Width over bow foil ... ... | 52 ft $4\frac{1}{2}$ in |
| Draught, maximum ... ... | 18 ft 0 in |
| Draught, foil borne ... ... | 8 ft 10 in |
| Displacement ... ... ... | 130 tons |
| Endurance ... ... ... | 30 miles |
| Speed ... ... ... ... | 38–41 knots |

The PT150DC is intended to commence regular seagoing passenger service about the middle of the year between Göteburg in Sweden and the Danish ports of Aalborg and Frederikshavn.

The main machinery consists of two Maybach MD 1081S 20-cylinder four-stroke engines. These are vee engines of 185 mm bore by 200 mm stroke and follow the familiar Maybach principle, with a welded tunnel crankcase, roller crankshaft main bearings, two overhead camshafts per cylinder bank actuating three inlet and three exhaust valves and combined fuel pump and fuel injectors. Each engine has two vertical-spindle Maybach turbochargers which deliver through intercoolers. The engines are installed amidships with the power train leading forward and aft by a form of vee-drive.

The output is taken through a Geislinger elastic coupling splined to the crankshaft and consisting of a number of flat radial spring packs enclosed in an oil-filled housing and then a torque converter. The case containing the coupling and converter is flanged directly to the drive end of the engine. The driven part of the Geislinger coupling is flanged to the main shaft of the converter gear and between the converter gear output flange and the main shaft is an hydraulically-actuated disc clutch which is engaged during normal service to provide a direct drive from the engine crankshaft.

The hydraulic torque converter is fitted on a shaft parallel to and above the main shaft, with primary and secondary elements coupled to the main shaft by means of separate spur gear trains. The stator of the torque converter is fixed to the gear-case. This converter is filled with fluid and operated during take-off for a rated period of one minute when the hydraulically-actuated disc clutch mentioned above is disengaged. The engine take-off output is then transmitted by the spur gear train from the main shaft to the primary of the torque converter and then transmitted to the secondary through the converter fluid at relatively high efficiency. In this case the output shaft speed no longer equals that of the engine but follows the torque characteristics of the propeller. The operation of the torque converted for take-off and its emptying, after the hydraulically-actuated disc clutch on the main shaft has engaged, is performed automatically and controlled by the difference in speed between the primary and secondary elements.

The shafts are of heat treated stainless steel, flanged to the gear-box output shafts. They comprise five hollow lengths and, except for the one section within the craft, have solid forged flanges at both ends. The flanges of the inboard sections are fitted by the SKF oil injection method and must be removed if the shaft is to be withdrawn through the gland box. There is a short distance piece, immediately after of the gear-box, to which torque and thrust meters may be mounted during trials. The tailshaft is a solid forged stub with integral flange and the usual taper and keyway for securing the propeller. All the shaft flanges are connected by stainless steel fitted bolts and those external to the hull have glass fibre reinforced plastics fairings.—*Marine Engineer and Naval Architect, January 1968, Vol. 91, pp. 8–10.*

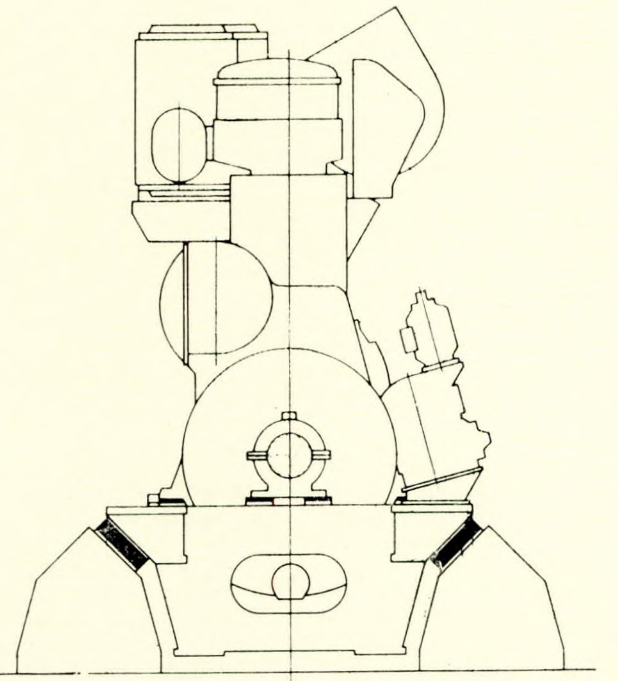
### Dutch North Sea Packet Launched at Birkenhead

An important launch from the Birkenhead shipyard of Cammell Laird (Shipbuilders and Engineers) Ltd. was that of *Koningin Juliana,* a fast multi-engined North Sea passenger and car ferry for the Hook of Holland to Harwich service of the Zeeland Steamship Co.

*Koningin Juliana* has been built to the initial design of Knud E. Hansen of Copenhagen and will carry a total of 1200 passengers in two classes on daylight crossings and 750 passengers on overnight crossings. The car deck and platform deck can accommodate approximately 180 motor cars or a reduced number of private cars with a complement of commercial vehicles or containers. The ship is of straight-through type with large hydraulically-operated bow and stern doors, and the car deck is fitted with portable lifting decks to increase the car stowage arrangements. The principal particulars are:

| | |
|---|---|
| Length, o.a. ... ... ... | 429 ft 6 in |
| Length, b.p. ... ... ... | 377 ft 6 in |
| Breadth, moulded at car deck ... | 65 ft 6 in |
| Depth, moulded at car deck ... | 23 ft 7 in |
| Load draught ... ... ... | 16 ft $4\frac{3}{4}$ in |
| Gross measurement ... ... | 7500 tons |
| Net measurement ... ... | 3700 tons |
| Trial speed ... ... ... | 22·3 knots |
| Service speed ... ... ... | 21 knots |

The two c.p. propellers will each be driven by a pair of Augsburg-built M.A.N. R9V 40/54 engines. These 400 mm ($15\frac{3}{4}$ in) bore 540 mm ($21\frac{1}{4}$ in) stroke trunk-piston engines are rated at 4890 bhp each at 400 rev/min—18 kp/cm$^2$ (256 lb/in$^2$) b.m.e.p. and 7·2 m/s 1417 ft/min mean piston speed



*Sketch showing how engine will be supported resiliently on 45-degree steel and rubber sandwich pieces*

and are coupled through Vulkan EZ couplings and Renk twin input/single output gears.

The engines and propellers are normally controlled from the wheelhouse while the vessel is at sea, engine speed and propeller pitch ranges being automatically selected for day and night service conditions under which two or one engines respectively are coupled to each propeller.—*Marine Engineer and Naval Architect, February 1968, Vol. 91, pp. 44–45.*

#### Fastest Roll-on/Roll-off North Sea Cargo Ship

The 18-knot *Europic Ferry,* the latest addition to the Transport Ferry Service fleet of the Atlantic Steam Navigation Co. Ltd., was built by Swan Hunter and Tyne Shipbuilders Ltd. and is the second ship of her class to be built at the Wallsend shipyard for the Transport Ferry Service.

In most respects the new vessel closely resembles her predecessors, being of the roll-on/roll-off type with a large stern door ramp giving access to a continuous vehicle deck. Maximum cargo space is achieved by locating the machinery completely below the vehicle deck and the upper deck is also kept clear for the carriage of containers, flats and other unitized cargo by siting the accommodation and bridge superstructure well forward.

From abaft the engine room aft bulkhead to the stern door of the vehicle deck can support 70 ton loads on two axles while the forward section of the deck can take 40 tons on two axles.

One of the most important differences between *Europic Ferry* and earlier ships of the series is in the selection of the Pielstick machinery for main propulsion. There are two Lindholmen-built 16 PC2V units in the new ship, each rated at 6780 bhp at 450 rev/min and driving Stone-KaMeWa c.p. propellers through a pair of M.W.D. reduction gear-boxes. Engine/propeller pitch control is normally effected from the bridge via KaMeWa-type Combinator stands but the engines are started and brought up to load from a control compartment in the engine room.

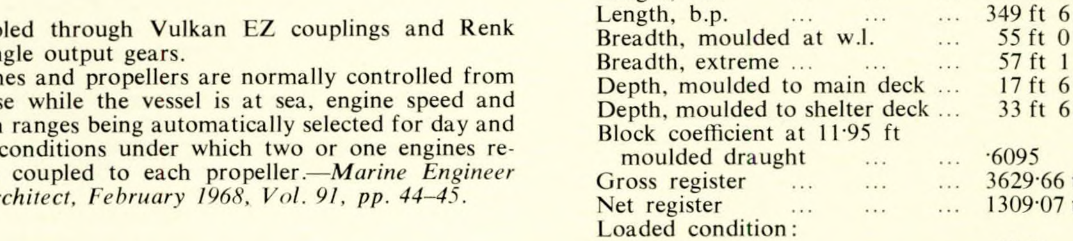
A separate auxiliary compartment, forward of the main engine room, houses four W. H. Allen 470 kW Diesel-engined generators; other auxiliaries include a Sharples Gravitrol 500 fuel purifier and two Sharples Super-Centrifuge machines.—*Motor Ship, March 1968, Vol. 48, pp. 577–580.*

#### First Drive-through Ferry for British Railways

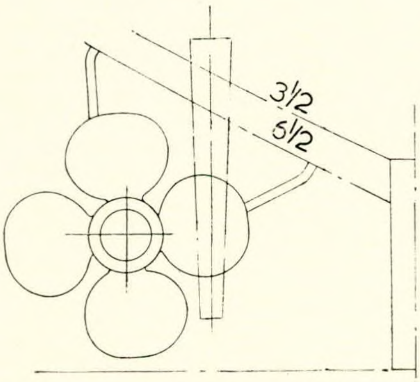
The 3630 gt passenger/vehicle roll-on/roll-off ship, *Antrim Princess,* built by Hawthorn Leslie (Shipbuilders) Ltd., at Hebburn-on-Tyne, has entered service on the 21 mile strait separating the South-West tip of Scotland and North-East Ireland. The vessel is operated by the Caledonian Steam Packet Co. Ltd., a wholly owned subsidiary of British Railways, between Larne and Stranraer.

Main particulars of *Antrim Princess* are:

| | |
|---|---|
| Length, o.a. ... ... ... | 369 ft 0 in |
| Length, b.p. ... ... ... | 349 ft 6 in |
| Breadth, moulded at w.l. ... | 55 ft 0 in |
| Breadth, extreme ... ... ... | 57 ft 1 in |
| Depth, moulded to main deck ... | 17 ft 6 in |
| Depth, moulded to shelter deck ... | 33 ft 6 in |
| Block coefficient at 11·95 ft moulded draught ... ... | ·6095 |
| Gross register ... ... ... | 3629·66 tons |
| Net register ... ... ... | 1309·07 tons |
| Loaded condition: | |
| Draught ... ... ... ... | 12 ft |
| Displacement ... ... ... | 3981 tons |
| Deadweight ... ... ... | 865 tons |
| Freeboard ... ... ... | 5 ft 7 in |
| Light ship: | |
| Draught ... ... ... ... | 9 ft $10\frac{1}{4}$ in |
| Displacement ... ... ... | 3116 tons |
| Capacities: | |
| Fuel ... ... ... ... | 103 tons |
| Water ballast ... ... ... | 683 tons |
| Fresh water ... ... ... | 34 tons |
| Main machinery: | |
| Type ... ... ... ... | S.E.M.T.—Pielstick |
| Output ... ... ... ... | 2 × 7180 bph |
| Speed, engine (maximum) ... | 450 rev/min |
| Speed, propeller (maximum) ... | 310 rev/min |
| Ship speed, service ... ... | $19\frac{1}{2}$ knots |



Europic Ferry




*Half section at frame $3\frac{1}{2}$ and frame $6\frac{1}{2}$ in way of propeller and A-bracket respectively*

The vessel is the first British Rail ferry to be fitted with both bow and stern doors.

The main engines are two S.E.M.T./Pielstick 16 PC 2V units each fitted with four Brown, Boveri VTR 320-type turbochargers. Each engine has a maximum continuous output of 7180 bhp at 450 rev/min. The engines are arranged to burn Class B Diesel fuel and have a fuel consumption of approximately 0·338 lb/bhph at full load. Lubricating oil consumption is 0·0022 lb/bhph.

Each main engine drive through a Geislinger B90 C14-type flexible coupling to a single-reduction gear-box of R. W. Transmission manufacture. The reduction gives a propeller speed of 310 rev/min at an engine speed of 450 rev/min.

Each tailshaft, which is supported by three bearings in a 25 ft long sterntube, turns a Liaaen D80/4 four-bladed stainless steel controllable pitch propeller. Because of heavy loading on the blades during operation the 9 ft 3 in diameter propellers are of the heavy-duty type with a 33·5 in diameter Admiralty hub. As can be seen in the accompanying illustration the designers of *Antrim Princess* have favoured an A-bracket arrangement to support the stern bosses. This lowers the appendage resistance; the propeller tip clearance is about 3 ft. Each propeller, including blades, weighs 5·6 tons.—*Motor Ship, February 1968, Vol. 48, pp. 492–496.*

#### Crosshead with Floating Pin for Large Engines

Service experience has shown that the crosshead bearings of present day large direct-coupled marine Diesel engines are more sensitive and probably the cause of more concern and delay to ships than any other part of this type of main machinery. In close touch with the operation and design features of these bearings is the Research and Technical Advisory Services Department of Lloyd's Register of Shipping which has recently assigned to the National Research Development Corporation, a crosshead design with a fully floating pin, for possible development.

A study of the maximum bearing pressures used by various main engine designers shows a large variation of values, although these design values are not strictly indicative of what actually occurs in practice, as it would appear that no allowance is made for distortion under load. Even if reasonable uniformity of loading is obtained on these bearings, there are sometimes indications from the burnished surfaces of crosshead pins that lubrication is only of a boundary nature, the white metal temperatures apparently being maintained below melting point mainly by the copious flow of cooling lubricant through the many oilways cut in the bearings.

The crosshead design shown in the accompanying illustration, is claimed to offer the following advantages:

1) The upper and lower white metal bearing surfaces, carrying the firing load, extend the full length of the floating cylindrical pin.

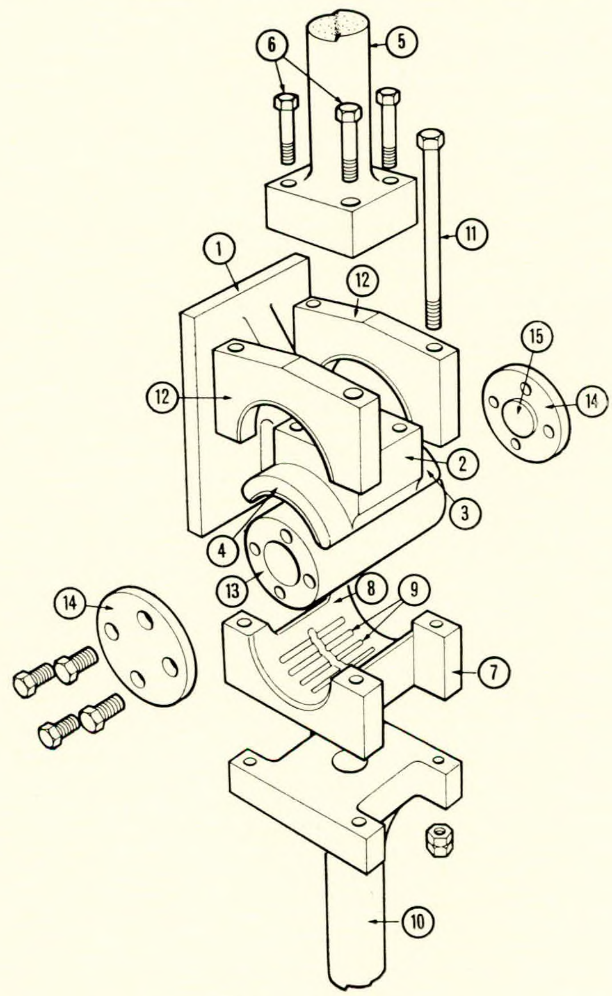
2) The pin is free to turn, thus promoting even wear and assisting lubrication.

3) For the same pin size and specific bearing pressure, twice the bearing surface is always available, compared with conventional crossheads.

4) Any undue friction on one bearing surface will automatically transfer the relative motion to the other bearing surface.

5) The purely cylindrical pin is easily manufactured to very fine limits and surface finish and is easily renewable.

6) Should a hardened pin be required, this can be either



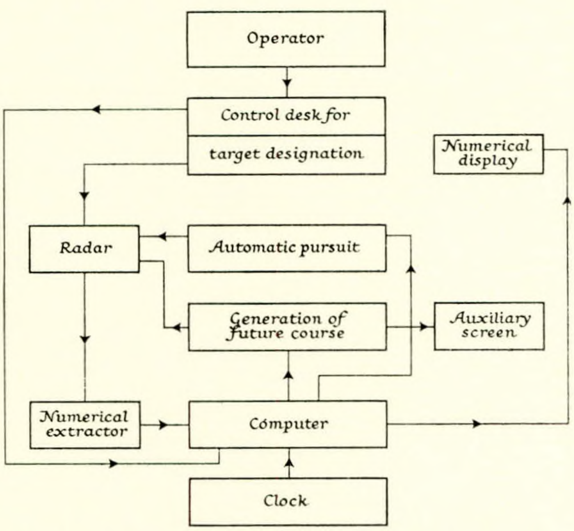
*Key to assembly*

1) *Slipper*
2) *Crosshead block*
3) *Extensions of block (integral)*
4) *White metal lining*
5) *Piston rod*
6) *Piston rod bolts*
7) *Lower half bearing*
8) *White metal lining*
9) *Oilways*
10) *Connecting rod*
11) *Keep bolts*
12) *Keeps*
13) *Hollow floating pin*
14) *Pin locating caps*
15) *Spigot*

*Crosshead with floating pin for large engines* surface-hardened or of through-hardened alloy steel at a relatively lower cost than in conventional designs. In the latter case hollow boring should ensure uniform properties. If case-hardening is adopted, the simple cylindrical form of the pin should also ensure that the case is of uniform depth and hardness, both of which are conducive to uniformity in load carrying capacity.

7) By simply taking the weight of the piston and piston rod and removing one of the pin retaining plates, the pin can be quickly removed from the crosshead assembly for inspection or renewal purposes.—*Motor Ship, February 1968, Vol. 48, p. 523.*

**Automation of Ships' Navigational Systems**




*Block diagram of automatic anti-collision system*

It is now two years since a working group set up under the direction of the French Institute of Navigation at the instance of the Secretariat d'Etat á la Marine Marchande delivered a report which was the fruit of a thirteen-months' study of the various possibilities of bridge-automation for commercial shipping.

The terms of reference of this study embraced a ship all of whose essential functions were under the control of a central computer, the system, briefly including:

1) a dead reckoning loop (ship computer dialogue);
2) a heading speed conversion loop (officer computer dialogue);
3) an observation calculation loop (officer computer dialogue);
4) an anti-collision control loop (radar computer dialogue).

Naturally since this was a formal study, the most complete and complex case was examined, and the approximate cost of the data handling equipment for such a ship was estimated to be of the order of two million francs.

There is, however, one well defined field that offers itself readily to electronic automation and in which there would be an immediate return on investment, once suitable equipment has been developed: this is the problem of safety at sea and the avoidance of collisions.

Automatic plotting necessitates a bi-univocal coupling between the radar equipment and the computer. The essence of the operation is to extract from the radar circuits the bearing range data in real time so that the computer can handle them and feed them back either on the screen or on to an auxiliary screen in the form of future courses.

This automatic plotting necessitates a manual designation of target and the solution that consists in systematically handling all the echoes contained in a corona of predetermined ranges cannot, unfortunately, be profitably retained. The block-diagram of such equipment is shown in the figure. It could function as follows: when a target is designated by placing on the echo a simple marker (made up of radial and circular elements generated as required by a device attached to the radar) a digitalized radar-extractor passes the bearing-range elements of the target to the computer and from that time a frame is placed around the target which is now automatically followed by the computer.

After a minimum time of three minutes the future course of the target is worked out and displayed on the screen by an auxiliary strobe guided by signals sent from the computer.

At the same time a display device makes it possible to attach the values that effect the target selected by the operator. These we may repeat and the bearing and range of the C.P.A. and the time when it will occur.

If the calculated range is less than a tolerated minimum, an alarm immediately warns the officer of the watch, who can then manoeuvre in good time and immediately ascertain the new value produced by an alteration of course of his own ship.

Extraction equipment has already been produced to meet aeronautical requirements but its adaptation to marine radar would call for lengthy research and it will not be easy to perfect it. One must, therefore, envisage a simple device that can be operated without requiring a special screen equipped with a reflection plotter and in normal observation conditions (with vision in position).—*Noetinger, P., Jnl of the Institute of Navigation, 1967, Vol. 20 No. 4, pp. 476–480.*

**Aluminium Alloy Anodes**

The new aluminium alloy Galvalum, developed by the Dow Chemical Co., produces 1280–1300 amp h/lb of metal—that is an efficiency of over 95 per cent, with a potential of $-1{\cdot}1$ V (copper sulphate reference). By comparison, the best quality zinc produces only 355 amp h/lb but with the same efficiency and potential. The reliability of Galvalum has been proved in lengthy field tests under working conditions and typical examples after various exposure times are shown in the illustration. It is estimated that for a cargo ship of 12 000 gross tons, with a wetted hull surface area of 50 000 $ft^2$, a current density of 1·2 m-amps and a hull anode life span of three years, anodes providing 4450 lb of zinc would be required costing £450. For the same protection using these new anodes, 1230 lb of Galvalum would be required and although the anodes are slightly dearer than zinc ones, the total cost would be £319, reducing the hull anode cost to some 30 per cent whilst an additional saving is that there would be nearly 50 less anodes to weld on. For a quick comparison take the weight of zinc required to protect the hull and divide by 3·6. This gives the exact weight of Galvalum needed to provide exactly the same degree of protection for the same length of time.

Anodes should be placed equidistantly around the hull and should be mounted on both sides of the bilge keel. Along the hull they should be mounted approximately 20 ft apart. In the stern area one anode should be mounted just below low water level, another on or adjacent to the tailshaft housing and another on the starboard strake close to the stern frame. In some instances it may be desirable to mount an anode on the rudder but this is not general practice.

When mounting anodes in the bow area consideration must be given to the possibility of anchor chain damage and the anode sited accordingly. If it is known that the vessel uses anchors for manoeuvring, it is a wise practice to fit a cable guard over the leading edge of the anode. It will be appreciated that the use of anodes in the bow area is important because of the damage to paint coating due to anchor chain abrasions.—*Reed's Marine Equipment News, February 1968, p. 10.*

### Inert Gas Welding of 12 per cent Nickel Maraging Steel

This paper summarizes the welding characteristics of 12 per cent nickel maraging steel and provides data on the strength and toughness of gas tungsten arc and gas metal arc welds made with filler metals developed for these processes and this steel. Developments leading to filler metal of the matching type are described. The 12 per cent nickel–5 per cent chromium–3 per cent molybdenum maraging steel is capable of providing 180 000 lb/in$^2$ yield strength on combination with 50 ft lb Charpy V-notch in air-melted plate.—*Lang, F. H., Welding Jnl, January 1968, Vol. 47, pp. 25s–34s.*

### Cold Cracking in Multilayer Welds of Low-alloy High-strength Steel

The NRIM TRC (Tensile Restraint Cracking) test, which was formerly developed in the course of a study of root cracking, was also shown to be as effective in determining a critical restraining stress for cold cracking multilayer welds for each welding condition and hydrogen content and to suggest suitable welding procedures corresponding to actual restraint in welded construction.—*Nakamura, H., Inagaki, M. and Mitani, Y., Welding Jnl, January 1968, Vol. 47, pp. 35s–46s.*

### Hydrodynamic Welding

Hydrodynamic welding is a radical departure from normal tubular joining techniques and has been under investigation for more than four years. The process employs two basic ingredients required by any joining process, namely, heat and pressure. The unique feature of hydrodynamic welding lies in the kinetics of the process, i.e. pressure variables in existing joining methods are pseudo-static in nature, while in this new process, the pressure pulses are dynamic.—*Morin, T. J., Peacock, G. R. and Zotos, J., Welding Jnl, February 1968, Vol. 47, pp. 114–117.*

### Smoke in Gas Turbine Exhaust

Tests were carried out in a small gas turbine (60 bhp at 46 000 rev/min). Reduction of engine speed at constant brake torque resulted in reduced smoke, whereas reduction of load at constant speed led to its increase. A series of tests under air throttling at the compressor inlet showed reduced smoke. A relationship was observed between smoke and exhaust-gas temperature. The tests comprised three types of fuel, gas oil, kerosine and kerosine-lubricating oil mixture.—*Gross-Gronomski, L., 12th–17th November 1967, A.S.M.E. Winter Annual Meeting Paper No. 67-WA/GT-5.*

### Vibration Problems with High Speed Turbomachinery

The paper presents several examples of vibrational problems recently experienced by several turbine manufacturers. These problems include stability, system critical speeds, rotor response to unbalance and balancing of high speed compressors and turbines. The effects of fluid film bearings and seals, bearing pedestals and other parts of support structure on rotor response and critical speeds are discussed. The effects of rotor stiffness, mass and inertia distribution are presented. —*Sternlicht, B. and Lewis, P., Trans. A.S.M.E., Jnl Engineering for Industry, February 1968, Vol. 90, pp. 174–186.*

### Critical Pressure of Flat Acrylic Windows under Short Term Hydrostatic Loading

Flat disc-shaped acrylic windows of different thickness-to-diameter ratios have been tested to destruction under short term hydrostatic loading at room temperatures, where short term loading is defined as pressurizing the window hydrostatically on its high-pressure face at 650 lb/in$^2$-min rate until failure of the window takes place. Critical pressures and displacements of windows with thickness-to-effective-diameter ratios less than one have been recorded and plotted.—*Stachiw, J. D., 12th–17th November 1967, A.S.M.E. Winter Annual Meeting Paper No. 67-WA/UnT-1.*

### Effect of Steam Environment upon Creep Behaviour of Stainless Steel

Some comparative creep-rupture tests on a stainless steel in atmospheres of air and pure steam are described. It is shown that the rupture times in steam are generally greater than those in air despite the formation of many more intergranular cracks in the steam environment. A tentative mechanism of crack restraint due to oxide build up is put forward to explain the results.—*Le May, I., 12th–17th November 1967, A.S.M.E. Winter Annual Meeting Paper No. 67-WA/Met-14.*

### Fuel-cell Power Plant for Deep-diving Submarine

Some of the engineering considerations involved in using a fuel-cell to propel a small deep-diving submarine capable of descending to 20 000 ft are described. A small submarine is considered here to be a vehicle capable of carrying two to four men and having a dry weight in the 50 000–700 000 lb range. At this depth, the submersible would be able to conduct operations over 98 per cent of the ocean floor. It is the author's contention that a fuel-cell power plant based on the hydrogen-oxygen reaction is ready for immediate engineering adaptation to such a vehicle.—*Thurber, W. C., 12th–17th November 1967, A.S.M.E. Winter Annual Meeting Paper No. 67-WA/UnT-2.*

### Underwater Power from Aluminium

A thermochemical power system having the highest theoretical performance (hp-h/ft$^3$ propellants) and particularly suited for underwater applications is described. In this system, aluminium is reacted exothermally with sea water to produce gaseous exhaust products, hydrogen and steam. The exhaust products are expanded through a prime mover to convert the hydrogen to steam for applications that require condensable exhaust products. Open-cycle (shallow depth operation) and closed-cycle (deep depth operation) systems are described.—*Bobb, F. H., 12th–17th November, 1967, A.S.M.E. Winter Annual Meeting Paper No. 67-WA/UnT-3.*

### Design of Pressure Hulls for Small Submarines

Advances in the state of the art for high-strength, lightweight pressure hulls for submersibles during the next decade can be predicted on the basis of research findings. The development of pressure hulls for very deep depths involves the use of lower-ductility hull materials and will require a careful synthesis of design, analysis, material, fabrication and inspection technology.—*Krenzke, M., Jones, R. and Kiernan, T., 12th–17th November 1967, A.S.M.E. Winter Annual Meeting Paper No. 67-WA/UnT-7.*

### Fretting Corrosion Products of Aluminium Alloys

An examination of a black powder produced by the fretting of aluminium alloy surfaces in a ship's hold has been carried out. The fire risk associated with the accumulation of this type of material in confined spaces has been considered.—*Andrew, J. F., Donovan, P. D. and Stringer, J., British Corrosion Jnl, March 1968, Vol. 3, pp. 85–87.*

### Hot Cracking in Austenitic Stainless Steelweld Deposits

Microscopical examination and scanning with an electron probe micro-analyser of austenitic stainless steel weld deposits has revealed that low-melting-point segregates promote hot cracking. These phases were found to contain relatively large amounts of sulphur, phosphorous, manganese, silicon and sometimes niobium. Ferrite tended to dissolve these elements (with the exception of manganese and thus reduced the risk of hot cracking.—*Frediks, H. and van der Toorn, L. J., British Welding Jnl, April 1968, Vol. 15, pp. 178–182.*

### Effect of Fit-up on Heat-Affected-Zone Cold Cracking

The effect of fit-up on the susceptibility of welded joints to heat-affected-zone cold cracking has been studied by ascertaining the minimum pre-heat necessary to make welds without cracks. It appears that with gaps greater than $\frac{1}{64}$ in an increase in pre-heat is needed to avoid cracking and it is recommended that procedural tests (e.g. CTS type tests) should incorporate a gap so as to anticipate poor fit-up in practice.—*Graville, B. A., British Welding Jnl, April 1968, Vol. 15, pp. 183–190.*

### Lamellar Tearing in Silicon-killed Boiler Plate

Types of welded joint typically associated with type of defect are discussed. Tests on plate material taken from a failure at a "J" preparation of a boiler show that lamellar tearing is not due solely to the presence of gross stringer inclusions. It is suggested that the defect is connected with the directional characteristics of the steel which could be due to the presence of silicate films at crystal boundaries.—*Wormington, H., Welding and Metal Fabrication, September 1967, Vol. 35, pp. 370–373; Fuel Abstracts and Current Titles, April 1968, Vol. 9, p. 65.*

### Semi-closed Cycle Gas Turbine Plants

After discussion of the definition of open-cycle, closed-cycle and semi-closed cycle gas turbines, circuits of semi-closed cycle gas turbine plants are explained. Only two circuits have advantages over open-cycle plants, making higher unit ratings and lower capital costs possible. Performance at part load is very good. The machines are started as open-cycle turbines and can be operated as such. Such plants require low sulphur fuels.—*Gasparovic, N., Combustion, November 1967, Vol. 39, pp. 14–25; Fuel Abstracts and Current Titles, April 1968, Vol. 9, p. 71.*

### Wind Tunnel Tests on a Model of a Car Ferry

In 1964, data was published on the wind tunnel tests carried out at the von Karman Institute for fluid dynamics on models of two types of general cargo ship. As sea trials were planned on a car ferry, attention was again given to the wind effects for this type of ship. Wind tunnel tests were therefore carried out on a model of a car ferry at the Karman Institute in a uniform flow and with a wind gradient such as encountered at sea.—*Aertssen, G. and Colin, P. E., International Shipbuilding Progress, March 1968, Vol. 15, pp. 71–77.*

### Investigation of the Scale Effect on Self Propulsion Factors

During the last decade model families have been investigated in several towing tanks to determine the scale effect on the self-propulsion factors. Due to the small number of model families which have been tested, it is at the moment impossible to determine exactly the scale effect on the propulsion data. The purpose of this paper is to give some simple expressions for the thrust deduction and the wake fractions by the investigation of the results of some well known model families.—*Benedek, Z., International Shipbuilding Progress, March 1968, Vol. 15, pp. 78–95.*

### Axial Stiffness of Diesel Engine Crankshafts

A scale model of a crank fixed at one end is loaded by forces and moments at the free end. Measuring of the displacements yields a set of influence numbers. These figures are used for calculation of the axial elasticity of a crankshaft. It is shown that, by means of an extreme simplification, the elasticity of one single crank can also be derived from these influence numbers and that a formula is thus evolved which can be compared with existing formulae.—*Van der Linden, C. A. M., International Shipbuilding Progress, March 1968, Vol. 15, pp. 96–105.*

### Some Special Problems in Surface Effect Ships

This paper is concerned with some problems in the development of large Capture Air Bubble (CAB) craft. The main topics are: configuration; structural materials; design criteria and parameters; side walls (sideboards); bow and stern seals; propulsion; route analysis.—*Waldo, R. D., 1967, A.I.A.A./S.N.A.M.E., Advanced Marine Vehicles Meeting Paper; Jnl B.S.R.A., February 1968, Vol. 23, Abstr. No. 26 160.*

### Isothermal Aerodynamic Investigation of a Suspended Flame Register

The results of tests without firing on a half-scale model of a register are presented, including pressure drop performance and flow patterns. Various swirler designs were tested over a wide range of quarl angle, swirler location and Reynolds number. Significant optimum quarl angles and swirler locations yielding minimum pressure drop are found. Optimum quarl angle varies with swirler diameter. For a fixed swirler diameter a critical matching of optimum quarl angle to blade outlet angle appears.—*Livesey, J. L., Wilcox, P. L. and South, R. D., Jnl Inst. Fuel, April 1968, Vol. 41, pp. 169–186.*

### Large Outboard Motor

An unusual outboard type engine and propeller for use as a thruster for the lead barge of large tows is being tested. The engine, shafting and propeller measures more than 20 ft from top to bottom and swings a $5\frac{1}{2}$-ft diameter propeller. An unusual feature is that a 16 cyl Model 16V-71GM Detroit Diesel engine of 500 hp is arranged vertically to avoid the use of bevel gears, at least at the engine level. The power unit will be remotely controlled from the pilot house of the towboat. The entire 12-ton unit can be raised, lowered, rotated through 180° in either direction, started, regulated and stopped by the pilot.—*Marine Engineering/Log, March 1968, Vol. 73, p. 77.*

### Stern Trawler Designs

This paper describes generally the various design phases required in the development of stern trawlers and covers, in particular, development of the first of a new series of stern trawlers suitable for operations off the Canadian Eastern Atlantic. It is based on an extensive study, initiated by the Department of Fisheries of Canada, and on a comprehensive series of resistance, propulsion and seakeeping tests carried out on three different models by the National Research Council of Canada.—*Kristinsson, G. E. and Doust, D. J., Marine Technology, April 1968, Vol. 5, pp. 105–136.*

**Research Concerning Increasing Fatigue Strength of Press-fitted Shaft Assembly**

The purpose of this investigation was to improve the rotating bending fatigue strength of press-fitted shaft assemblies by varying the geometry of the press-fitted portion. Four groups of 50 mm diameter press-fitted specimens were used. As a result, compared with the fatigue limit of press-fitted plain shaft, maximum increase of the fatigue limit was obtained for the filleted shaft, in which the press-fitted hub had a low protrusion on the face. Taking the reduction of fretting corrosion into consideration, however, the press-fitting of the hub overhanging from the shoulder was most preferable with an increase of fatigue limit about 50 per cent. The fatigue limit of the press-fitted plain shaft could be increased by 50 per cent due to press-fitting of the hub which had the low protrusion.—*Nishioka, K. and Komatsu, H., Bulletin of Japan Society of Mechanical Engineers, 1967, Vol. 10, No. 42, pp. 880–889.*

**Mathematical Model for Steady Operation of Stirling-type Engine**

A mathematical model of Stirling-type engines has been developed. The complexity of the problem has been reduced by analysing the various components of the engine (heat exchangers, regenerator and cylinders) separately for cyclically steady conditions and by selecting pressure, temperature and mass as the independent variables. The required piston displacements are a computed result. Losses due to flow friction, piston blow-by and finite heat transfer rates have been accounted for by applying correction factors.—*Qvale, E. B. and Smith, J. L., 12th–17th November 1967, A.S.M.E. Winter Annual Meeting Paper No. 67-WA/Ener-1.*

**Adaptive Control for Prime Movers**

A novel control system, called Optimizer, enables an engine to adjust its settings continuously for maximum power. An electronic feedback control automatically corrects the engine settings and insures continuous optimum performance. The system can control spark timing, injection timing, air-fuel ratio, blade angle, nozzle area, and bypass adjustments in petrol, Diesel and turbine engines.—*Schweitzer, P. H., 12th–17th November 1967, A.S.M.E. Winter Annual Meeting Paper No. 67-WA/DGP-2.*

**Factors Affecting the Efficiency of Absorption Oil Filtration**

An experimental apparatus and method are described for measuring the effects on dirt removal efficiency of an oil filter of such independent variables as temperature, viscosity and flow rates. Based on these results, an empirical efficiency equation has been developed, which has been tested with success against filter systems obtained from commercial sources.—*Ewbank, W. J., 12th–17th November 1967, A.S.M.E. Winter Annual Meeting Paper No. 67-WA/DGP-3.*

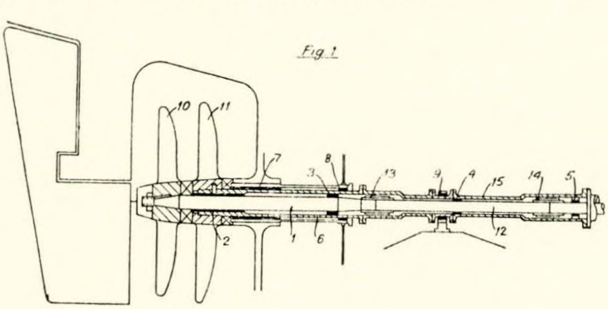
**Heat Treatment of Iron Castings**

Iron-casting heat treatment can be performed on any composition of iron to relieve stresses and improve machinability, ductility, strength and wear resistance. There is a common tendency to forget that heat treatment can improve properties available in a casting beyond its original condition. Iron will exhibit noticeable changes in hardness when it is heat treated but hardness by itself is not an ordinarily useful property.—*Walton, C. F., 12th–17th November 1967, A.S.M.E. Winter Annual Meeting Paper No. 67-WA/Met-13.*

# Patent Specifications

**Propeller Shaft for Two Counter-rotating Propellers**

This invention relates to an arrangement of two co-axially arranged ship propellers of which one propeller shaft is disposed in the other which is tubular in shape.




In Fig. 1 (1) indicates an inner shaft journalled in bearings (2) and (5) in an outer, tubular shaft (6) journalled in bearings (7) to (9). The shafts support contra-rotating propellers (10) and (11). The inner shaft has a portion (12) coupled to the respective ends of the shaft by means of coupling sleeves (13) and (14) which abut the ends of the shaft in such a manner that torque can be transmitted solely by friction. When assembling and dismantling the coupling, pressure oil is introduced between the abutment surfaces, frictional abutment thus ceasing and permitting the sleeves to be displaced axially. This coupling, which is known *per se*, occupies but a small space and is thus well suited for use in this case.

The outer shaft also has a connected portion which may be attached to the respective ends of the shaft by means of flanges and bolts. This shaft portion, which encloses the shaft portion (12), is divided along an axial plane into two halves which are bolted together. For removal of the propeller on shaft (1) the upper half of bearing (9) is removed and one half of portion (15) of the outer shaft is loosened and removed from the other half. The couplings (13) and (14) are loosened and portion (12) and other members of the shaft (1) can be lifted out.—*British Patent No. 1 096 158 issued to Stal-Laval Turbin Aktiebolaget. Complete specification published 20th December 1967.*

**Sealing for Propeller Shaft**

This invention relates to a cathodic system for preventing corrosion of the seal for a propeller shaft. The propeller and the rings (94), (84), (96) and (98), and also the bolts (90) and (92) shown in Figs 3 and 4, are made of non-corrosive metal such as bronze or monel bronze. Sleeve (62) is formed of stainless steel. Had the rings contacted the sleeve (62), the result would be that said rings (94), (96) and (98), and bolts (90) and (92) function as an anode for supplying electrons to

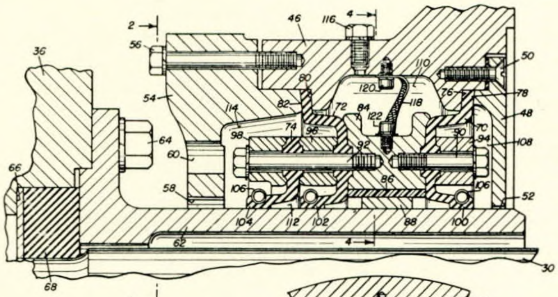


Fig.3

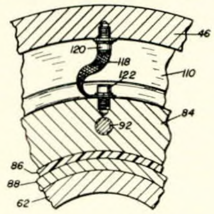


Fig.4

the sleeve (62) and the propeller (36). The reason for this is that the rotation of the sleeve (62) and the propeller causes electrons to be emitted, whereby the sleeve and the propeller function as a sink for electrons and are supplied with electrons by the rings (94), (84), (96) and (98) and the bolts (90) and (92). This causes dissolution of these rings and bolts, resulting in the destruction of the seal, whereby water could leak into the bearings and oil leak out of the tube and about the bearing, causing destruction of the shaft and the bearing. By electrically insulating the ring (84) from the sleeve (62) by the insulating ring (86), the propeller no longer functions as a cathode in that the ring (84) could no longer function as an anode. Nevertheless, all of the seals can follow any radial movement of the rotating element (shaft (30) and sleeve (62)) by reason of the inclusion of the guiding babbitt ring (88).—*British Patent No. 1 090 421 issued to Waukesha Bearings Corporation. Complete specification published 29th December 1967.*

**Anti-pitch System for Ships**

FIG.1.

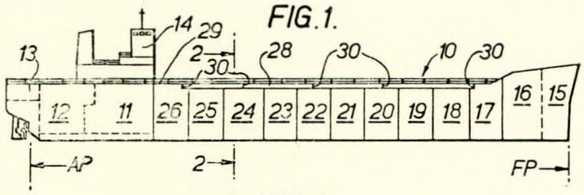


FIG.2.

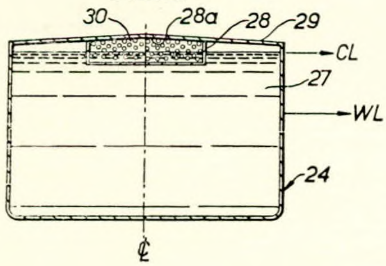



Fig. 1 shows the stabilizing system of this invention embodied in a liquid cargo carrier or tanker (10). Located at the stern of the ship are the engine room (11), boiler room (12), steering gear (13) and superstructure (14), while fore peak tank (15) and forward compartments (16) are located at the bow. Independent cargo carrying tanks (17)–(26) are centrally located and are typically laden with a liquid cargo (27), such as oil.

In accordance with the invention, a longitudinal conduit (28) is installed in the central cargo carrying section and extends for a major portion of the ship's length between the forwardmost cargo tank (17) and the rearmost cargo tank (26). The duct is disposed symmetrically with respect to the centre-line of the ship and is located substantially above the water-line and in intersection with the cargo level (CL), advantageously located immediately below the deck (29), as shown in Fig. 2. Maintenance of the cargo level beneath the upper wall (28a) of the duct (28) provides a free surface. This arrangement accommodates stabilizing wave generation and utilization of free surface effects of generated waves to damp pitching motion.—*British Patent No. 1 097 146 issued to Esso Research and Engineering Company. Complete specification published 29th December 1967.*

**Ship Stabilizer**

This invention provides a ship stabilizer and in particular a stabilizer made mostly of existing ship structures such as the deck and supporting girder plates conventionally found in bulk cargo ships and the like.

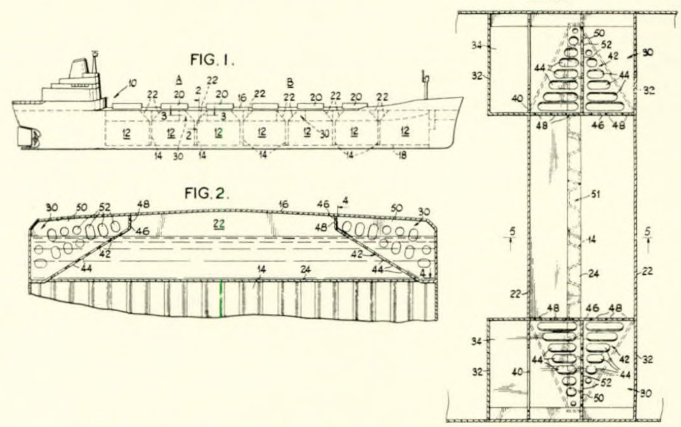



Fig. 1 shows a bulk-carrying cargo ship, generally indicated as (10), having a number of longitudinally spaced holds (12) with bulkheads (14). A suitable number of longitudinal bulkheads (18) extend substantially throughout the length of the ship. Sloping, elongated girder plates (22) are positioned across the hull, and each has one edge in supporting engagement with the underside of the main deck (16) and its other edge rigidly mounted to the edges of a plate (24).

It is readily understood that the sloping deck girders (22) function in their normal manner to give the required structural support and that the space between girders (22) provides a connecting channel (51) between the pair of transversely aligned wing tanks (30) at opposite sides of the hull.

Girders (22) and bulkheads (32), along with bottom plates (34), main deck (16) and the hull of the vessel define an elongated, enclosed liquid stabilizing container or tank which, when filled with a body of liquid, serves as the passive stabilization system for the vessel. As the ship experiences a roll, the liquid within the stabilizer transfers from one side to the other in response to the roll and the effective mass of the liquid shifts to impart a stabilizing moment. Restriction plates (42) provide the appropriate energy dissipation and liquid damping.— *British Patent No. 1 105 715 issued to John J. McMullen Associates. Complete specification published 13th March 1968.*