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SEA TRIALS ON A 9,500-TON DEADWEIGHT MOTOR CARGO LINER

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Read in London at the Spring Meeting of the Institution of Naval Architects on 5 *April* 1955, *Sir Stanley Goodall, K.C.B., O.B.E. (Honorary Vice-President I.N.A.) in the Chair, supported by Mr. Stewart Hogg (Member of Council I.Mar.E.)*

Summary

As second part of the programme sea trials of the Centre Belge de Recherches Navales, M.V. *Lubumbashi*, a newly-built cargo liner of the Compagnie Maritime Belge, was equipped with torsionmeter, thrustmeter, pitometer log, anemometer and windvane and again, in varying conditions of draught, fouling and weather, numerous records were collected of speed through the water, power and fuel consumption, thrust, revolutions, ship motions, wind and waves.

In a similar way as for the Victory ship *Tervaete*(1) the service data are analysed and an attempt is made to ascertain the efficiency and economy of the ship and machinery in different service conditions.

A predominant part of the programme was the ship-model correlation. Two measured-mile trials were carried out, the first as part of the official trials of the ship in ballast condition, the second in loaded condition at the beginning of her maiden trip. Dr. Allan, Superintendent of the Ship Division N.P.L., was good enough to run a model and to make the comparison as part of the investigations (Appendix I).

PART I

Instrumentation for the Trials and Accuracy of Measurements

The position of the principal instruments concerning propulsion is indicated in the general arrangement shown in Fig. 1. The data of the ship and machinery are given in Appendix II.

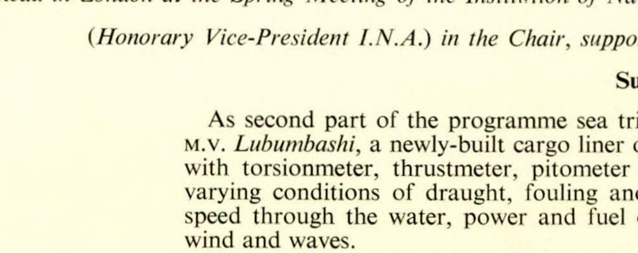

FIG. 1.—INSTRUMENTATION IN M.V. "LUBUMBASHI"

A = Torsionmeter; B = Thrustmeter; C = Pitot log; D = Anemometer; E = Windvane.

The speed through the water is measured with a pitometer log fitted in the ship's bottom. One of the aims of the experiments being the exploration of the boundary layer, the choice fell on a log of the same type as installed in the *Tervaete*.

The log again was manufactured by the British Pitometer Company. The rod, however, was longer and could be transferred to any given position up to 4 ft. from the surface of the hull for measurement. The pitot log was given the most forward position possible, 188 ft. from the forward perpendicular. Calibration was achieved on the measured-mile trials.

The thrust bearing being of the Michell type, a Michell thrustmeter was installed. A thrustmeter of the two-way type, capable of measuring both ahead and astern thrust, would have given an accurate measurement of the thrust, even when the draughts forward and aft are appreciably different. However, the thrustmeter is actually of the one-way type; because of that, corrections have to be made for the weight component when there is a large difference in draught forward and aft. Hence the accuracy of thrust measurement is undoubtedly better for the loaded trial than for the ballast trial with a ship heavily trimming by the stern.

Measurement of power with the Siemens–Ford torsionmeter which came from the *Tervaete* raised the problem of looking for a good location on the shaft in the tunnel, where the torsional vibrations excited by the motor were probably of small amplitude. That is the reason why the meter was installed at the first third of the shaft, there being in the tunnel 4 bearings before and 9 bearings after torsionmeter. The torsionmeter was calibrated and fitted on the shaft, prior to its installation on board, in the engine works.

The propeller revolutions were obtained with the revolution counter and a stop watch.

Wind speed and wind direction, relative to the ship, were measured with a cup-anemometer and a windvane Richard installed on the upper bridge, the records being taken at a distance in the chartroom. The cup-anemometer was checked from time to time with an anemometer of the propeller type held by hand.

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Wave-height and wave-length were obtained by comparing them with known dimensions of the ship, the wave-length being checked by measuring the period of encounter.

During the first voyage pitching and rolling diagrams were taken by means of the gyro-pitch and roll recorder of the National Physical Laboratory.

The course was read from the gyro-compass. During the time a series of propulsion data were collected, the extreme angles to port and starboard, through which the rudder was moved, were observed.

In order to ascertain the efficiency of the power plant, the fuel consumption of both the main engine and the group of auxiliaries had to be measured. It would have been preferable to measure fuel consumption by weighing. Space, however, was not available for the installation of the tanks, balances, etc., which these operations would have required. The day tanks of heavy fuel, to be consumed by the main engine, are of a capacity large enough to allow, by means of simple soundings, a six hours' consumption test. The day tank of diesel oil normally used for the auxiliaries only is small, and although it allowed a correct measurement of auxiliaries consumption, the measurement of the fuel consumption of the main engine was rather rough when operating on diesel oil because of the restricted time available for this last test.

Whenever a consumption test was carried out a sample of fuel was taken for determination of heat value, specific gravity, viscosity and other inspection data. The viscosity was not measured for the diesel oil.

The displacement of the ship when leaving and entering port was calculated from recorded draught fore and aft and density of water. At any time of the voyage the displacement was estimated on a basis of the daily fuel and water consumption.

The accuracy of measurements is within the following limits of error:—Speed through water, the pitometer log being calibrated on the measured mile: smooth water 1 per cent, rough water 2 per cent.

Torque, the shaft being calibrated in the shop: smooth water 2 per cent, rough water 3 per cent.

Thrust: smooth water 4 per cent, rough water 5 per cent.

Revolutions: smooth water 0·5 per cent, rough water 1 per cent.

Heat value of fuel oil: 0·5 per cent.

Indicated horsepower: 4 per cent in smooth water.

Main engine mechanical efficiency: 6 per cent in smooth water.

Fuel consumption per shp main engine: 5 per cent in smooth water.

Fuel consumption per shp auxiliary motors: 5 per cent in smooth water.

Part II

The m.v. "Lubumbashi" Trials

The *Lubumbashi* trials commenced when the newly-built vessel left the yard in ballast condition for her official trial trip at the end of December 1953. The measured-mile trial in ballast condition was carried out on December 22nd at Polperro.

The vessel then left Antwerp in loaded condition for her first Congo-voyage on January 9th, 1954, and ran again progressive trials over the measured mile at Polperro on January 10th. These trials would give a certain basis for others to take place later at sea.

Numerous records were taken during the first voyage from Antwerp to Teneriffe on January 9th to 13th, during the

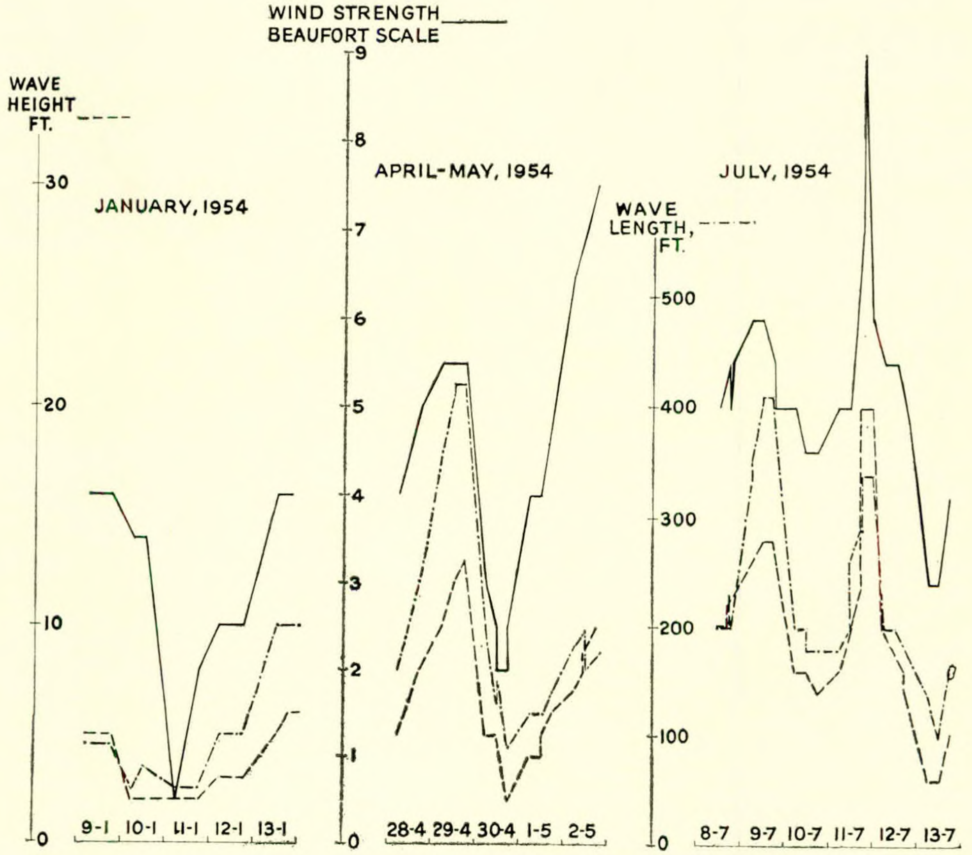


FIG. 2.—WEATHER DIAGRAMS

second voyage from Teneriffe to Antwerp on April 28th to May 2nd, and during the third voyage from Las Palmas to Hamburg on July 8th to 13th.

After this voyage the vessel entered into dry dock and the roughness of the hull was measured in the same way as it had been measured before leaving the yard.

Fig. 2 gives weather experienced during the trials. The figure relates to the tables of Appendix III, where the data given for wind force and waves are mean values. Most of the readings were made by day and it was assumed that weather did not change exceptionally during the night.

Weather conditions were with a wind force varying in the Beaufort scale from 0 to 8–9 and seas varying from calm to very high sea. It seldom happened that the swell was perfectly regular. Furthermore, on many occasions, as the wind had not blown for a long period with a constant force to build up the waves in a regular shape, there was not always a satisfactory correlation between wind force and wave dimensions.

Part III

Analysis of Machinery Data

Records of machinery were taken during each of the three voyages. Records were taken again on July 31st and August 1st when the vessel left Antwerp for her fourth voyage to Congo.

The figures obtained are efficiencies under normal operating conditions, not an ideal performance.

The general features of the machinery are given in Appendix II which contains also the fuel-supply plan of the main engine and the prominent data. The mean indicated pressures (pi), indicated horsepower (ihp), shaft horsepower (shp), and efficiency are given in Table VIII, the results of consumption tests and fuel analysis are given in Tables IX and X.

The 6,000 bhp main engine was built for heavy fuel and operated regularly on heavy fuel, except for entering and leaving ports and rivers. However, during the whole first Congo voyage, which commenced January 10th with the Polperro measured-mile trials in loaded condition, diesel oil was used, except for the two days following the Polperro trials when the motor for the first time operated on heavy fuel.

Consumption tests, of a duration varying from two to eight hours, were carried out during the three voyages. Weather was fine during each of these tests, so the fuel level in the day-tanks could be taken with accuracy and the engine was running on constant revolutions. The fuel consumption and the fuel rate given in Table IX have been corrected for a standard high heat value of 18,500 B.Th.U. A mean value of the fuel rate is 0·422 lb. per shp per hour.

It was of some interest to ascertain the fuel rate when operating on diesel oil. It has been mentioned that this measurement could not be very accurate for two reasons: the day tank for diesel oil is small and therefore the duration of a consumption test is only about half an hour; furthermore, the consumption of the auxiliaries has to be subtracted. An attempt however was made, and the day before the Polperro loaded trial, on January 9th, three consecutive tests of half

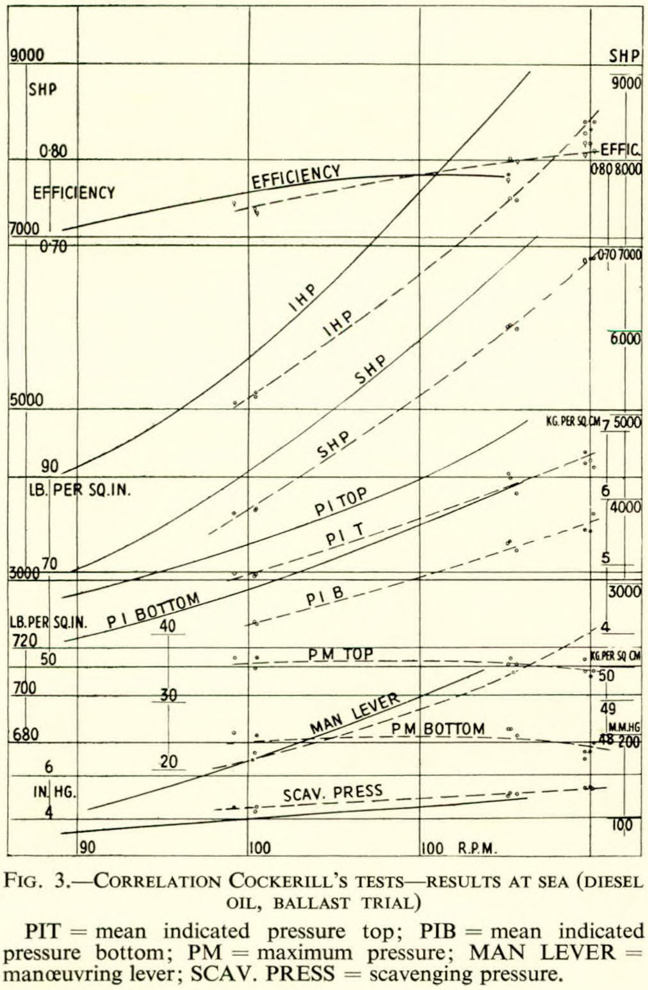


Fig. 3.—Correlation Cockerill's tests—results at sea (diesel oil, ballast trial)

PIT = mean indicated pressure top; PIB = mean indicated pressure bottom; PM = maximum pressure; MAN LEVER = manœuvring lever; SCAV. PRESS = scavenging pressure.

Read Kg per Sq. Cm., not GR per Sq. Cm,

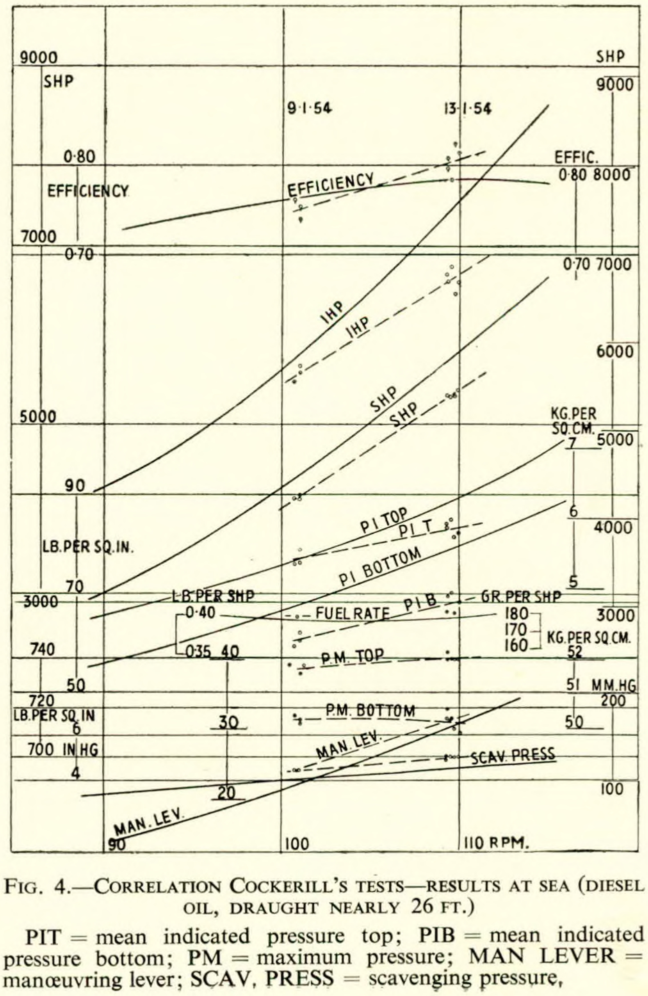


Fig. 4.—Correlation Cockerill's tests—results at sea (diesel oil, draught nearly 26 ft.)

PIT = mean indicated pressure top; PIB = mean indicated pressure bottom; PM = maximum pressure; MAN LEVER = manœuvring lever; SCAV. PRESS = scavenging pressure.

an hour each were carried out, the main engine operating on diesel oil. The consumption of the auxiliary motors was measured later on, at sea, the main engine operating on heavy fuel, and found to be 0·397 lb. per shp per hour. So the subtraction could be made and the consumption of the main engine was found to be 0·398 lb. per shp per hour with a diesel oil of 19,397 B.Th.U.

Tests have been carried out at Cockerill's, the engine builder, the main engine operating on diesel fuel, and the data, given in full lines, are the basis of Figs. 3, 4, 5, 6, 7.

Fig. 3 gives the comparison between the data obtained at Cockerill's and the data of the measured-mile trial in ballast condition. The data at sea are given in dotted lines. Maximum pressures pm at sea are also shown,

As was expected, ihp, pi, and shp plotted on *rpm* are at sea well beneath the lines of the engine builder, drawn for a normal draught condition. The engine shp is obtained from measured power at torsionmeter by adding to it *rpm*/2 (cf. Part IV).

Fig. 4 gives the comparison between Cockerill's data and the data obtained in fine weather at sea with a draught nearly 26 ft., the engine operating on diesel fuel.

The fuel rate of 0·398 lb. per shp per hour at 100·9 *rpm* obtained at sea with a diesel fuel of 19,397 B.Th.U. is to be compared with the consumption line of Cockerill obtained with a diesel oil of 19,278 B.Th.U.

Fig. 5 shows the comparison between Cockerill's data and the data obtained at sea in fine weather with a draught varying from 24 to 26 ft., the engine operating on heavy fuel.

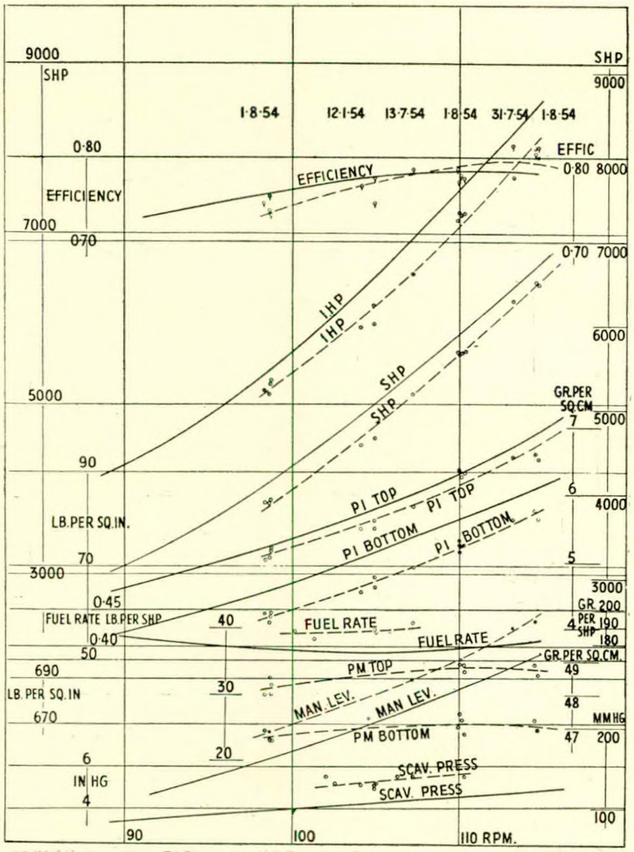


FIG. 5.—CORRELATION COCKERILL'S TESTS—RESULTS AT SEA (HEAVY FUEL, DRAUGHT 24 TO 26 FT.)

PIT = mean indicated pressure top; PIB = mean indicated pressure bottom; PM = maximum pressure; MAN LEVER = manœuvring lever; SCAV. PRESS = scavenging pressure.

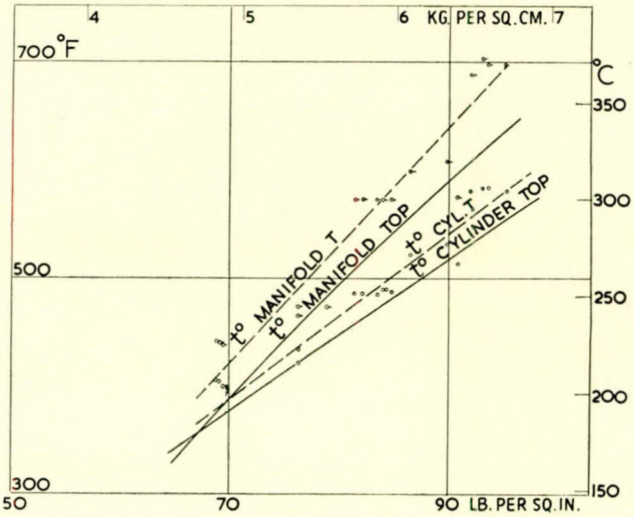
For the five consumption tests the fuel rate was corrected and reduced to a standard high heat value of 18,500 B.Th.U., the consumption line of Cockerill being obtained with a diesel fuel of 19,278 B.Th.U. Operating on heavy fuel gives lower maximum pressures than operating on diesel fuel.

It should be noted that the scavenging pressure which is higher when the engine is operating on heavy fuel as compared with diesel fuel increases during a fortnight's voyage from Antwerp to Congo, the rate of increase at constant *rpm* being roughly 1 in. Hg.

Fig. 6 and 7 show, plotted on pi, the exhaust gas temperatures, Fig. 6 for the engine operating on diesel fuel, Fig. 7 for the engine operating on heavy fuel, compared in both cases with the Cockerill lines obtained with diesel fuel. Since the efficiency is the same the engine operating with diesel fuel as the engine operating with heavy fuel, the comparison can be made on a base of pi.

The temperatures at sea are higher than at Cockerill's, even for diesel fuel, and are slightly higher for heavy fuel than for diesel fuel. It should be noted that for a given pi, the *rpm* are higher at sea than at Cockerill's.

During the trials in fine weather, the motor ran usually 107 revolutions with a power near 5,500 shp, a mean indicated



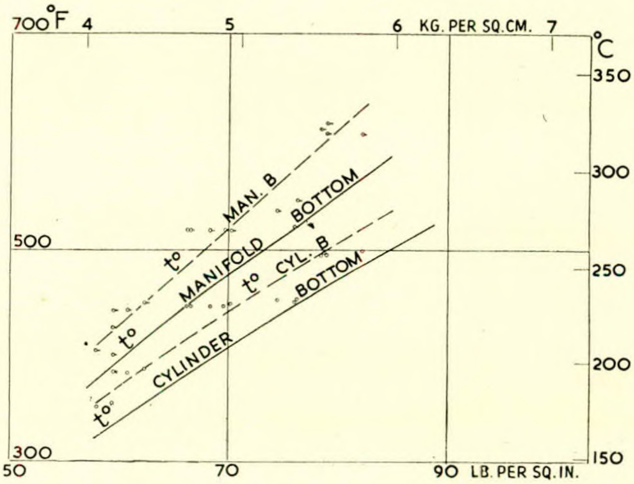


FIG. 6.—CORRELATION COCKERILL'S TESTS—RESULTS AT SEA

Relation exhaust gas temperature—mean indicated pressure (diesel oil).

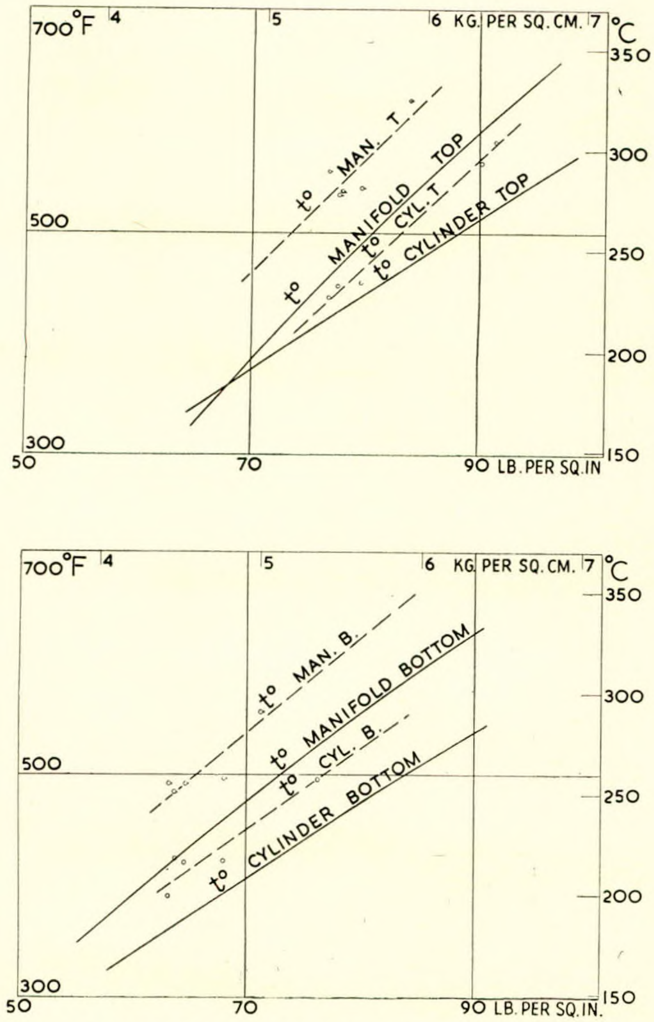


FIG. 7.—CORRELATION COCKERILL'S TESTS—RESULTS AT SEA

Relation exhaust gas temperature—mean indicated pressure (heavy fuel).

pressure nearly 86 lb. per sq. in. top, 71 lb. per sq. in. bottom. The mean temperatures of exhaust gas were:

(i) for diesel oil: cylinder top 500° F., cylinder bottom 450° F.; manifold top 590° F., manifold bottom 520° F.
(ii) for heavy fuel: cylinder top 520° F., cylinder bottom 465° F.; manifold top 610° F., manifold bottom 545° F.

The maximum pressures *pm* were with diesel oil 740 lb. per sq. in. top, 710 lb. per sq. in. bottom, with heavy fuel 700 lb. per sq. in. top, 670 lb. per sq. in. bottom.

In the exhaust general manifold the pressure was varying from 8 to 12 in. water. The steam pressure of the exhaust gas boiler was varying from 70 to 100, with a mean value of 90 lb. per sq. in., the inlet temperature of water being nearly 125° F.

The fuel inlet temperature in the main engine was: for diesel oil 100° F., for heavy fuel 255° F. to 265° F.

The temperatures of cooling water were:—

(i) for diesel oil: inlet 122° F., outlet piston valves top 130° F., bottom 140° F., cylinders 127° F.;
(ii) for heavy fuel: inlet 122° F., outlet piston valves top 133° F., bottom 145° F., cylinders 132° F.

The temperatures of piston-cooling oil were:—
(i) for diesel oil: inlet 104° F., outlet 131° F.;
(ii) for heavy fuel: inlet 104° F., outlet 133° F.

The fuel valves were water cooled, the engine operating on heavy fuel: the temperatures were 96° F. at inlet, 100° F. at outlet.

The temperature of seawater was varying between 50° F. and 70° F.

The purifiers were operating on temperatures varying from 175° F. to 185° F., the clarifier on temperatures from 185° F. to 195° F.

The mean load of the auxiliaries was for the first voyage 254 kW, for the second voyage 215 kW, and for the third voyage 211 kW.

At the moment this paper was written the *Lubumbashi* was operating on heavy fuel for almost a year and no special incidents occurred which were connected with the use of heavy fuel. Except for the first voyage, heavy fuel was used continuously at sea, from pilot to pilot, even in a high sea, the revolutions being reduced to 80, as happened during the third voyage. Use was never made of the bi-fuel system, heavy fuel on top, diesel fuel on bottom of cylinder.

The consumption of lubricating and cylinder oil was practically of the same amount as for the vessels operating on diesel fuel, cylinders 18 gall., crankcase 20 gall. per day. The wear of the cylinders was in excess of not more than 25 per cent of the wear of the cylinders of similar engines operating on diesel fuel. After 3,000 hours the mean wear per 1,000 hours was 0·016 in. It must be emphasized that the motor was usually running at a power not higher than 5,500 shp.

The purifiers and clarifiers were cleaned once a day. The engine-room had a very clean appearance during the three voyages, the temperature near the manœuvring lever varying from 82° F to 97° F.

The figures of fuel analysis (Table X) give some idea of the type of fuel burnt in the main engine. The fuels used were as heavy as the fuels burnt in the boilers of the Victory ship *Tervaete* during her trials of the years 1951 and 1952.[1] The specific gravity was about 0·97 at 70° F. The viscosity Redwood No. 1 at 100° F. was around 2,000 during the second and was more than 3,000, with a maximum of 3,472, during the third voyage, although there was no exhaust smoking. The sulphur content varied from 2 to 3 per cent. The pour point was rather low, the asphaltene content and the Conradson carbon rather high. During the third voyage the asphaltene content was about 8, the Conradson carbon about 12 per cent, the ash percentage however being rather low, about 0·06 per cent.

PART IV

Analysis of Propulsion Data

A. The Measurement of Hull Roughness.

The vessel left the yard in very good condition. A first coat of linolin was given before launching. A few days before the ballast trial the ship was docked. The hull was sandblasted and paints of following types were used: two coats of linolin, one coat of Hempels' anticorrosive and one coat of Hempels' antifouling. Between 16 ft. and 24 ft. and 25 ft. on fore and after ends the hull was covered with a coat of Hempels' boot-topping instead of a coat of Hempels' antifouling.

At the same time six steel plates were distributed around the vessel, two at the fore end, two amidship, and two at the after end. Inclined against the walls of the dry dock, the plates were treated, sandblasted, and painted, in the same way and simultaneously with the hull of the vessel.

During two days preceding the undocking of the ship the roughness of the hull was measured by means of the pneumatic instrument currently used in the General Hydraulic Laboratory of the University of Liége (Belgium) for the roughness measurements of pipes.(2)

In his paper to the Grenoble meeting Jorissen shows that the correlation between the industrial roughness of steel pipes, measured with the pneumatic feeler, and Nikuradse's artificial roughness is satisfactory. The measurement of the industrial roughness with a pneumatic feeler is not very accurate because only the average height of the irregularities is measured. The differential pressure which relates to the average height of the irregularities is measured by a water U-tube manometer. The feeler is calibrated by measuring grooves of known depth and breadth in a plate.

The instrument is adequate for measurement of roughness of a hull in dry dock. It is portable and suits the usual compressed air supply of say 100 lb. per sq. in. The instrument has a quick response. The accuracy, however, is not better than some 20 per cent.

Most of the measurements were taken the day of undocking, the hull being suitably dry. Since the feeler has a tendency to penetrate into the rather mellow coat of paint, a large number of measurements had to be eliminated.

The wind blowing starboard, most of the measurements were taken that side of the hull which was quicker drying than the port side.

The vessel was practically smooth over the whole hull. Out of the 855 measurements taken starboard, 10 gave a roughness higher than twice the mean value. The keel plate was rougher, 7 measurements out of 97 gave a roughness higher than twice the mean value.

The total mean roughness was:—

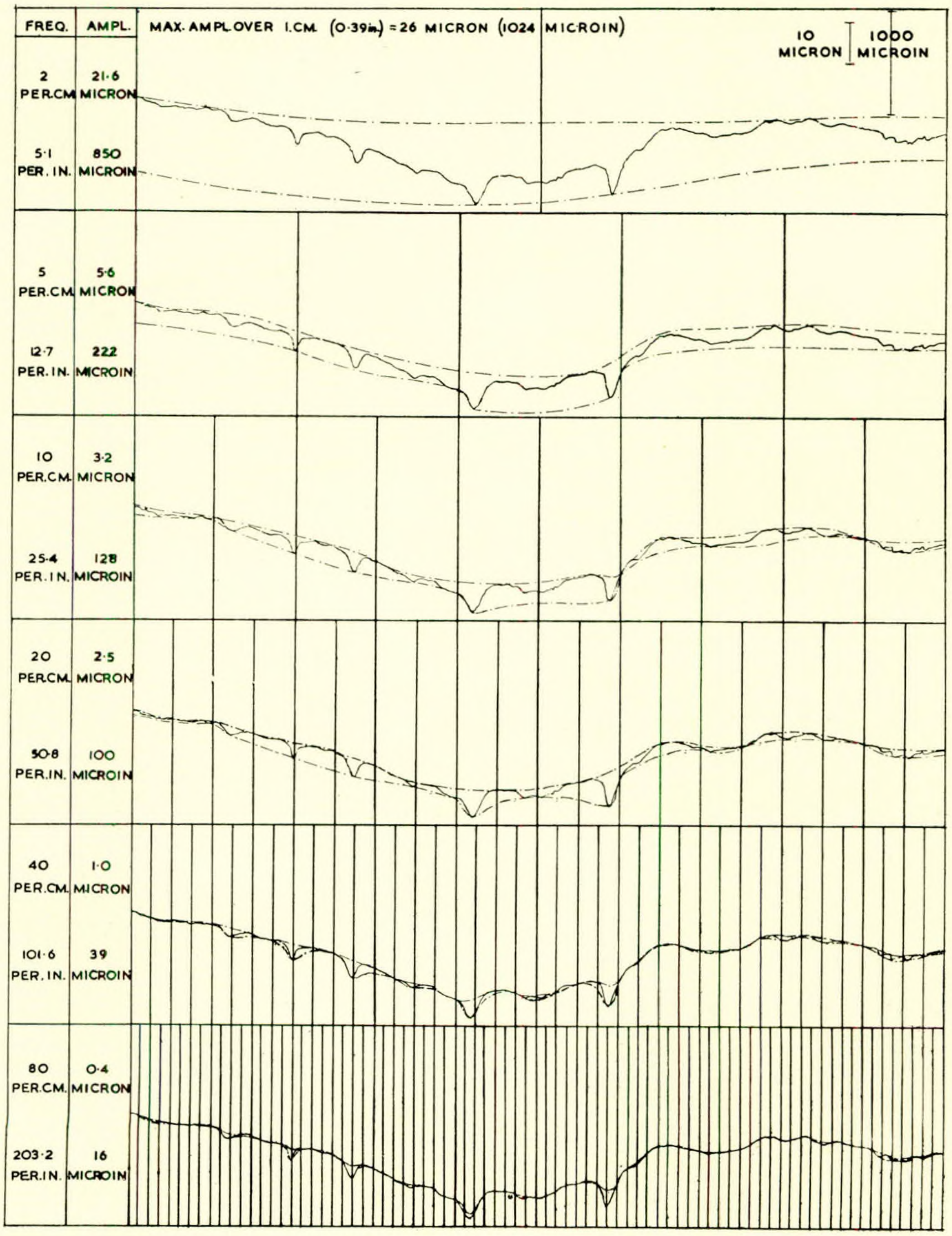


FIG. 8.—ANALYSIS OF TALYSURF RECORD B.S.R.A. METHOD

(i) for the part of the hull painted with antifouling 2,170 microinches.
(ii) for the part of the hull painted with boot-topping 2,050 microinches.

The keel plate had a roughness of 2,520 microinches. The highest value of roughness recorded was 10,000 microinches, the lowest value 1,650 microinches. The mean roughness of the hull was 2,130 microinches. No appreciable difference of roughness was stated between fore and aft.

The 6 sample plates, 4 of which were painted with anti-fouling, the other 2 being painted with boot-topping were sent to the University of Ghent for examination in the Talysurf machine installed there.

A large number of records were taken, both with the sample plates covered with anti-fouling paint and with the samples covered with boot-topping. 95 records were analysed, of which 57 with anti-fouling and 38 with boot-topping. The same method has been followed here as was used by the B.S.R.A. for the *Lucy Ashton* experiments.(3) The magnification was 1,000 for the amplitudes, 20 for the horizontal scale.

An example of roughness analysis has been given in Fig. 8. This is the analysis of a Talysurf record of a sample plate painted with anti-fouling. Not only is the maximum amplitude very low but for a frequency of 100 per in. the mean apparent amplitude is not higher than 40 microinches.

At given frequencies the means of the amplitudes of roughness have been calculated, first for the 57 samples with anti-fouling, then for the 38 samples with boot-topping; the ratio of mean amplitude-wavelength was also calculated and the results given in Fig. 9.

The sample plate of which the Talysurf record was given (Fig. 8) has been investigated with the pneumatic feeler: the roughness had a good uniformity over the whole plate and was about 1,060 microinches.

This figure, to be compared with the hull figure 2,130, calls for an explanation. There is no doubt about the ship being rougher than the samples. However, at the time this hull was explored by means of the pneumatic feeler, the instrument had not been in use for measurement of hull roughness. It was only used for the measurement of the roughness of industrial pipes varying from 2,000 to 20,000 microinches. The accuracy of measurement in the lower range of roughness was rather poor and the instrument was, after that first ship, modified in order to give more reliable results for newly-built fresh-painted ships. It is exactly with this somewhat modified Solex equipment of the instrument that the roughness of the sample plate was measured and found to be 1,060 microinches. It is apparent that the figure 2,130 is too high and that the accuracy of roughness measurements is not better than 20 per cent. Moreover, it requires consideration that a great number of measurements had to be made in a very short time in hard conditions, the instrument with the two operators, suspended in the hook of a crane, being led along the shell plating.

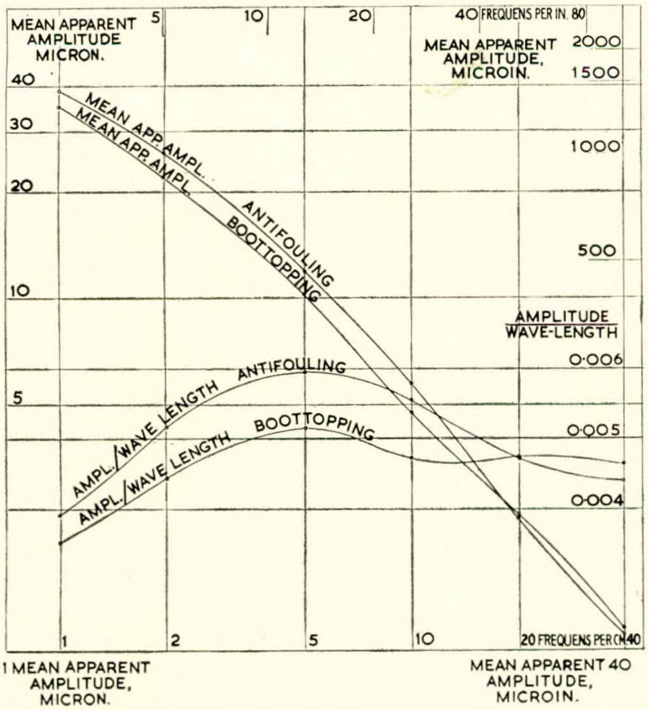


FIG. 9.—RESULTS OF ROUGHNESS ANALYSIS B.S.R.A. METHOD

The roughness of the hull was measured again in dry dock after six months' service, in order to make it possible to correlate this renewed measurement with the propulsion data of the third voyage. The estimation of the roughness of the dirty vessel is based solely upon the measurements made with the pneumatic feeler. The mean roughness over the whole hull was 3,620 microinches, the bottom being smoother than the sides and the after end being 10 per cent rougher than the fore end. The roughness was much more irregular than for the clean ship, some values being as high as 18,500 microinches.

However, the pneumatic feeler was not capable of measuring a roughness higher than say 20,000 microinches. The hull was clean, though paint had generally disappeared. A large number of rust stains covered the whole of the hull. Rust blisters of a height 0·05 to 0·1 in. are spread over the hull. They are numerous at the foreshoulder of the vessel, at the fore end of and beneath the bilge keels where they reach a height of 0·1 to 0·2 in. and a surface of 0·4 to 0·6 in. length and 0·2 to 0·4 in. breadth. These blisters are more numerous on the 4th, 5th, and 6th plate, counted from the stem, of strake D and the 8th plate of strake E. They appear on starboard as well as on port, but are more numerous on port where they are seen especially on 2 plates of strake C amidships. The density of these blisters was nearly 1 per sq. in. Beyond doubt these blisters have a strong influence, so that the roughness of the hull must be largely in excess of the measured value 3,620 microinches.

B. Velocity Distribution in the Boundary Layer.

The roughness of the hull was furthermore described by the shape of the velocity curve in the friction belt. The speed through the water being measured by means of a pitot log with extensible rodmeter, the velocity distribution could be investigated in the boundary layer. Fig. 10 shows two velocity curves: the first taken during the first voyage, immediately after the Polperro loaded trials, the second at the end of the third voyage, the vessel being six months out of dry dock.

In a similar way as for the *Wrangel*(5) the curves have been analysed by the method of Scholz.(4) There is certainly an influence of potential flow. No attempt, however, has been made to correct for this potential flow. The analysis is based upon the assumption that the flow from the stem to the pitot log located at a distance 147 ft. from the base of the stem is quite equivalent with the frictional longitudinal flow along a flat surface.

The curve of January 13th, for a ship's speed, U, of 16·0 knots, yields a momentum thickness of 1·58 in., 20·9 in. being the thickness of the boundary layer. Hence the relative roughness l/k_s is 7×10^5 and for $l = 147$ ft., $k_s =$ 2,530 microinches.

The curve of July 11th, for a ship's speed of 14·95 knots, yields a momentum thickness of 1·84 in., 22·8 in. being the thickness of the boundary layer. The relative roughness is now $2{\cdot}9 \times 10^5$ and $k_s = 6{,}100$ microinches.

Fig. 11 shows the same velocity distribution curves, $\log y/\delta$ being plotted on a base of $\log u/U$. The law of velocity distribution appears to be, according to Allan:(6)—

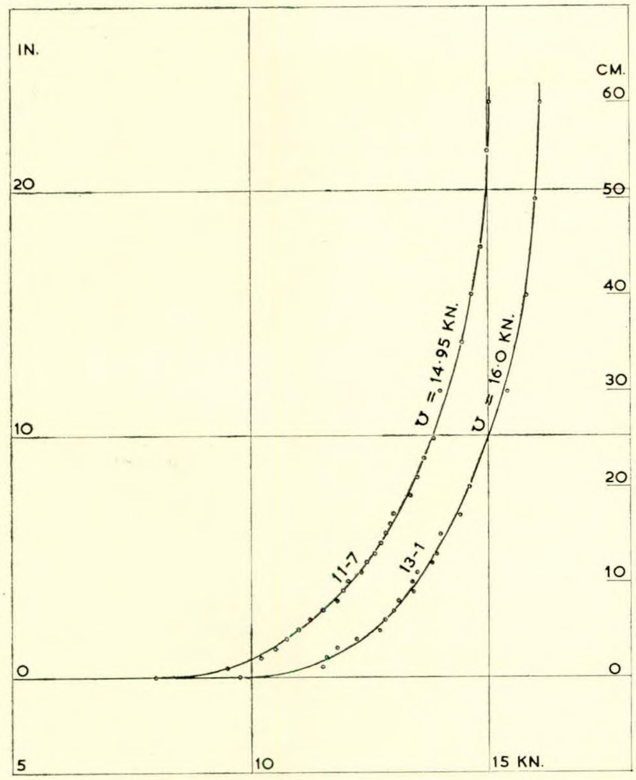


FIG. 10.—VELOCITY DISTRIBUTION IN BOUNDARY LAYER

$$\text{on January 13th } \frac{u}{U} = 1\cdot025\left(\frac{y}{\delta}\right)^{0\cdot109}$$

$$\text{on July 11th } \frac{u}{U} = 1\cdot028\left(\frac{y}{\delta}\right)^{0\cdot133}$$

The agreement between the sand roughness obtained on a base of momentum thickness and the roughness on a base of exponent n is not very satisfactory, although the increase of n from 0·109 to 0·133 for six months' service appears to be quite possible.

From momentum thickness the increase of ΔC_f from January 13th to July 11th is established at 0·0003, which relates to a reasonable increase of frictional resistance for a cargo vessel after six months' service.

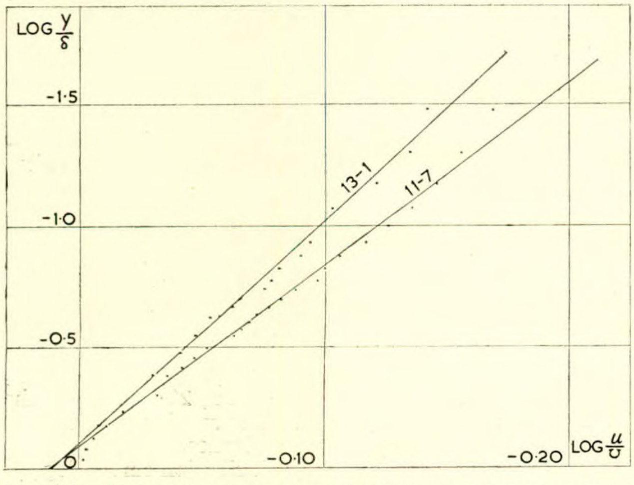


FIG. 11.—VELOCITY DISTRIBUTION IN BOUNDARY LAYER, LOG–LOG PLOTTING

It is therefore presumed that the sand roughness numbers 2,530 microinches, compared with 6,100 microinches established on a base of momentum thickness, are not far from the truth. They correlate very well with the roughness measurements of the hull established about 2,000 compared with 3,620, if account is taken of:—

(i) The rust stains which must have given the ship a substantial roughness increase in service conditions;

(ii) The structural roughness of this newly-built vessel which, with her butts welded and her seams and frames riveted, must not be very important.

C. Effect of fouling and weather.

Readings of speed, torque, thrust, revolutions, wind force, ship's course, pitch and roll, were made on every occasion that a change in weather conditions or revolutions occurred. Circumstances did not permit of all these readings being taken simultaneously. The time taken in collecting all the data during each observation was about half an hour. It was unlikely that weather and state of sea would change before that time elapsed.

Tables XI, XII and XIII give weather data for each observation (Appendix III). The relative wind force and wind direction were measured by anemometer and windvane. The true wind speed was calculated and the obtained strength in the scale Beaufort is given in the tables for use of further calculations. A comparison with the wind force as given by the deck log shows that the data are close, the tendency of the ship's officers being rather to over-estimate the wind force. Height and length of the waves are often not in agreement with the recorded wind force, as shown by Fig. 2. Nevertheless, power increase and loss of speed are given in terms of wind force.

It was of great importance to have an accurate value of the speed. Frequent measurements of the instantaneous speed were made with the calibrated pitot log for each observation number, especially in rough weather. The revolutions are taken by means of counter and stop-watch. The effective horsepower is derived from recorded thrust and speed with introduction of a thrust-deduction coefficient taken from the model-tests (Appendix I). The torsionmeter gives, after due correction for the zero, the measured power, mhp. An attempt has been made to have correct values for the delivered horsepower. Especially during the ballast trial the speed of the last group of runs was taken very low in order to make it possible to establish the shaft losses from torsionmeter to propeller. Plotting for both measured-mile trials (Fig. 12) mhp/N against N^2 yields for $N = 0$ a mhp/N $= 1$, hence a shaft loss of 1 hp per revolution. This value is in complete agreement with a loss of 2 per cent at full power. Thus dhp is calculated by subtracting from mhp the *rpm*.

Furthermore, due to the actual location of the torsionmeter on the shaft in the tunnel, the engine shp is obtained by adding *rpm*/2 to mhp.

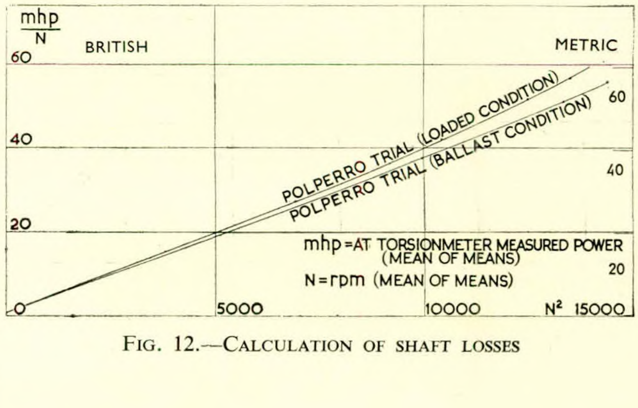


FIG. 12.—CALCULATION OF SHAFT LOSSES

Since no measurements were made south of the Canary Islands, the data are not corrected for water temperature.

It was of some interest to have the correlation between the propulsion data and the ship motions. During the first voyage pitching and rolling were recorded with the gyro-instrument kindly lent by the N.P.L. Unfortunately, the instrument was not on board during the subsequent voyages and the angles had to be recorded with a pendulum. The pitch angles out to out are in the weather tables, as mere information on the response of the vessel to wave motion. The rudder angles out to out were not higher than 10 deg.

Tables XIV, XV, XVI, XVII (Appendix III) give the propulsion analysis. A diagram of dhp plotted on speed shows how to obtain the effect of weather on ship's speed (Figs. 13, 14, 15). The diagram is basic for all further calculations. For each observation number dhp is corrected for draught unto the basic draught of the Polperro load trials. The dhp-line in a smooth sea and still air is well known from these trials and due reference is made to this line for each observation in order to obtain the increase of power and loss of speed. However, for the second and third voyage, the effect of fouling has changed the smooth water curve. The primitive

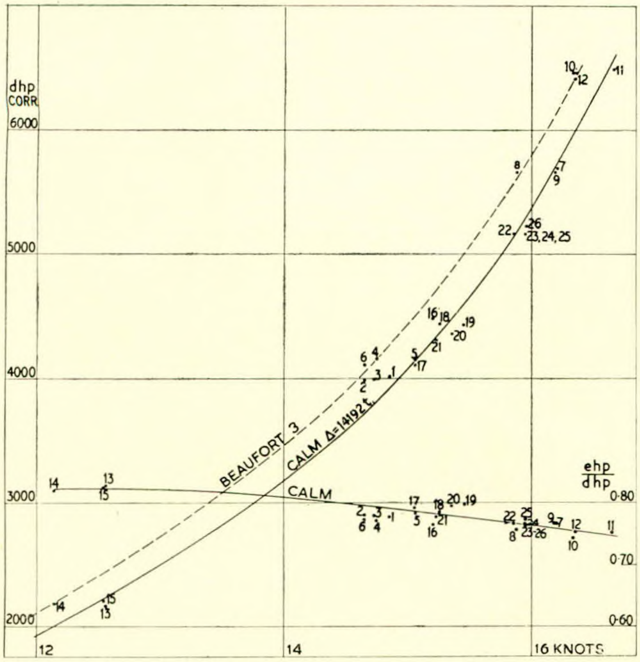


FIG. 13.—RELATION DHP-SPEED. VOYAGE ANTWERP–TENERIFFE

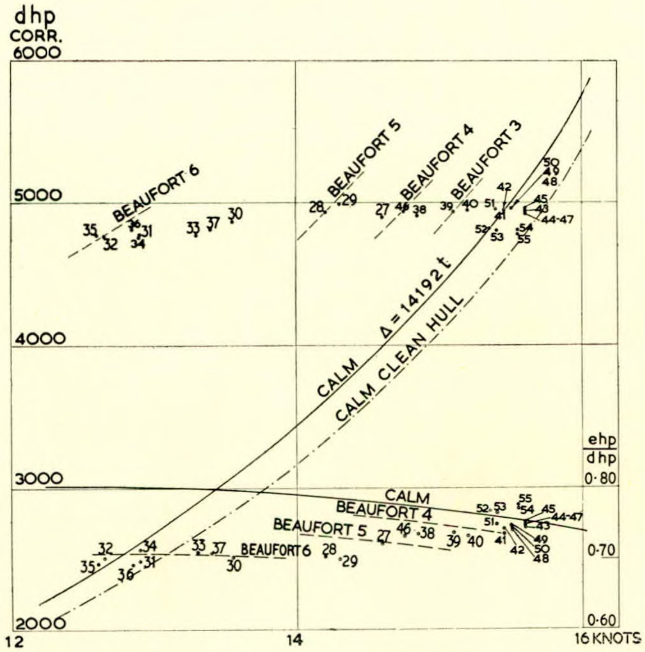


FIG. 14.—RELATION DHP-SPEED. VOYAGE TENERIFFE–ANTWERP

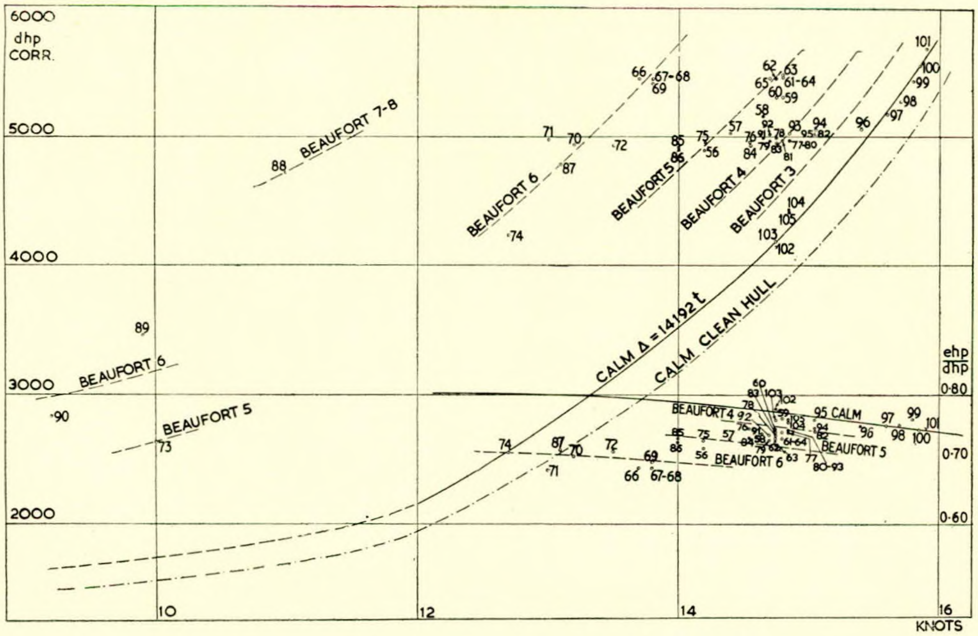


FIG. 15.—RELATION DHP-SPEED. VOYAGE LAS PALMAS–HAMBURG

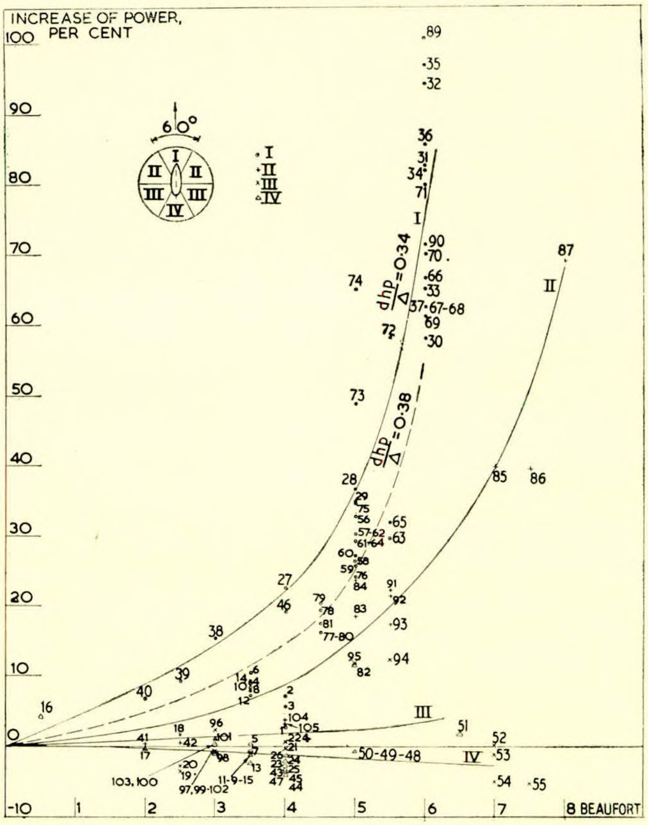


FIG. 16.—RELATION INCREASE OF POWER—WEATHER BEAUFORT

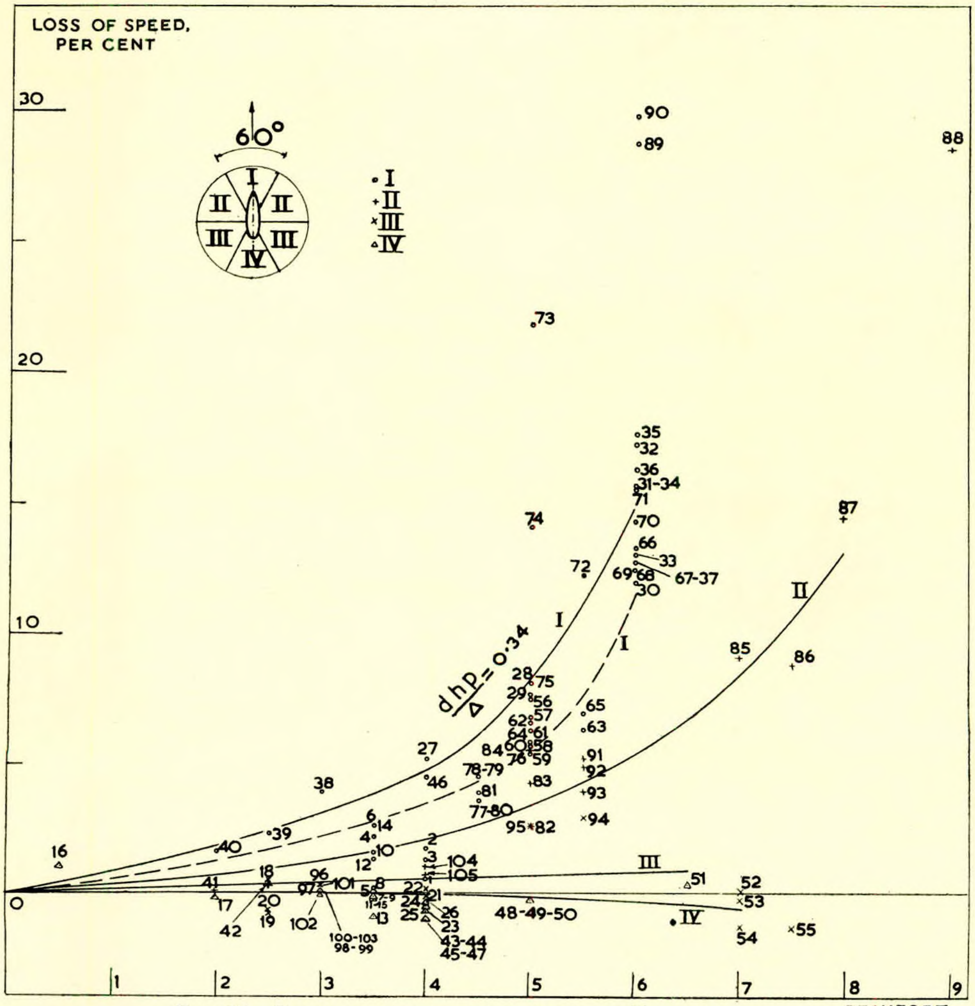


FIG. 17.—RELATION LOSS OF SPEED—WEATHER BEAUFORT

curve is drawn in dotted line in Fig. 14 and Fig. 15. The air resistance was established for the first voyage at 2·7 per cent of the total resistance.

The diagrams give also the propulsive efficiency in smooth and rough water.

Isoweather curves are drawn in the diagrams. They relate to different numbers in the scale of Beaufort and seem to confirm in the known parts—unfortunately they are not numerous—Telfer's opinion that all isoweather power-speed curves are naturally parallel to the basic zero weather relation,[8] although there is a small increase of the power loss with speed. The left part of Fig. 15 is somewhat questionable, due to the extrapolation of the smooth water curves on a basis of the data of the Polperro ballast trial. The description of sea disturbance and wind strength are in accordance with Baker.[9]

In Fig. 16 and Fig. 17 the increase of power and the loss of speed are plotted on a basis of weather in the Beaufort scale. The isoweather curves of Figs. 13, 14, 15 assume the vessel facing a sea well built-up by a wind of known strength. Following Bonebakker's method[7], the author has now grouped his observations I, II, III, and IV, although this appears somewhat difficult in the transition belt between successive groups. Furthermore, the power-displacement ratio dhp/Δ is important. All the lines of power increase and loss of speed are drawn for dhp/Δ = 0·34. It was observed that for a dhp/Δ = 0·38 the power increase and loss of speed are substantially lower. The curves for this higher power-displacement ratio are drawn in dotted lines. It should be noticed that, when grouping the observations for the calculation of increase of power and loss of speed, account is taken of the direction of waves rather than of the direction of wind.

In a following sea this vessel gains speed.

Accurate measurements of the thrust allowed to establish the relation between loss of propulsive efficiency and wind strength or sea disturbance (Fig. 18). The loss is slight for a

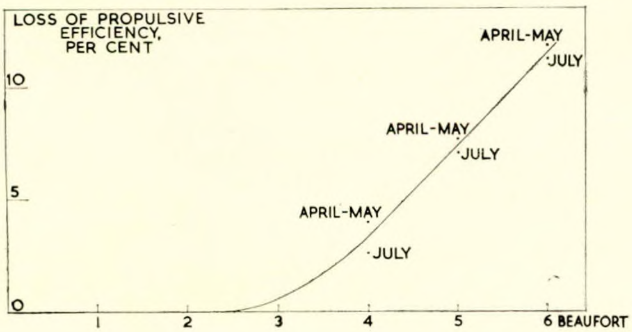


FIG. 18.—RELATION LOSS OF PROPULSIVE EFFICIENCY—WEATHER BEAUFORT

strength 3, but is more than 10 per cent for a strength 6 in the Beaufort scale. There is an uncertainty in that the propulsive efficiency is calculated on a base of thrust deduction coefficient obtained with the model tests.

Observation number 46, with a ship's course 180 deg. more than number 45, allows the separation of air and sea-waves resistance. The moment before she turned round, the vessel, in a following sea, had practically the same speed as the wind blowing aft. Hence the relative wind strength, wind ahead, was 30 knots. Assuming that the wind resistance follows a quadratic law with speed[9] this resistance is 11 per cent of the total resistance; and if finally account is taken of a loss of propulsive efficiency of 5 per cent (Fig. 14), the resistance of the waves amounts to 6 per cent in this moderate sea.

Finally, an attempt is made to calculate what is commonly called the margin of power.

The mean value of the increase of power due to weather conditions was:—

2·9 per cent for the voyage Antwerp–Teneriffe;
21·8 per cent for the voyage Teneriffe–Antwerp;
32·5 per cent for the voyage Las Palmas–Hamburg.

Taking 2·9 as a mean value for the outward voyage, 27·1 for the homeward voyage, the total mean appears to be 15 per cent.

On the other hand, the increase of power due to fouling was, at 15 knots, for the second voyage 7·9 per cent, and for the third voyage 9·2 per cent. Hence the effect of fouling, which amounts to 9·2 per cent after six months' service, is estimated 12 per cent after a year's service. The mean value over the year is 9 per cent.

For this newly-built vessel, weather and fouling allowances on dhp in still air and calm sea amount together to 25 per cent.

The author is well aware of the fact that ascertaining an allowance with the results of only three voyages is somewhat hazardous. He thinks, however, that the figures obtained on this basis are practically correct for the route Antwerp–Canary Islands.

Acknowledgements

This investigation was carried out under the auspices of the Centre Belge de Recherches Navales (Ceberena), with the financial assistance of the Institut pour l'Encouragement de la Recherche scientifique dans l'Industrie et l'Agriculture (Irsia).

The author wishes to acknowledge with thanks the valuable co-operation of Dr. Allan, Superintendent of the Ship Division of the National Physical Laboratory, that of the shipowners, the Compagnie Maritime Belge (Lloyd Royal), and that of both engine works, John Cockerill and Mercantile Marine Engineering and Graving Docks Co. The author also enjoyed the assistance of Mr. van Maanen, technical director of Ceberena, Professor Coppens, Messrs. Pluys, Vancraeynest, Ferdinande, and Mainil.

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Appendix I

NATIONAL PHYSICAL LABORATORY REPORT

May 10, 1954

Report on Experiments Resistance and Propulsion

Model Hull No. 3492
Model Screw No. U8

Ship Division

Experiments ordered by: Centre Belge de Recherches Navales, Rue Des Drapiers 21, Brussels—Belgium.

Ship Designation: Motor Vessel—*Lubumbashi.*

Model Scale and Material: 1/24th Wax Model Hull. 1/24th White Bronze Propeller.

Tables with this Report:—

- Table I. Particulars of ship as tested.
- Table II. Resistance experiment results.
- Table III. Particulars of screw as tested.
- Table IV. Propulsion experiment results and ship estimates.
- Table VA/B. Ship Trial results—details of runs.
- Table VIA/B. Ship Trial results corrected to calm air conditions.
- Table VIIA/B. Comparison of Ship and Model results.

Diagrams with this Report:—

- Fig. 1A. Body sections and endings as tested.
- Fig. 2A. Resistance experiment results.
- Fig. 3A. Stern arrangement as tested.
- Fig. 4A. Particulars of single screw U8.
- Fig. 5A/B. Comparison of Ship and Model results.

Design Particulars:—

See Ship Trial results.

Remarks on Model Experiments:—

1. The model was made to the moulded lines plan supplied by Centre Belge de Recherches Navales, and run for resistance in two conditions of displacement, corresponding to the loaded and ballast ship trials.
2. During these experiments a trip wire 0·036 inch diameter was fitted to girth the model at a section 5 per cent *LBP* abaft *FP*. The effect of the trip wire is estimated to have increased the ehp predictions by some 3½ per cent at 16·5 knots, and 2¼ per cent at 18·0 knots in the loaded and ballast conditions respectively.
3. Self propulsion experiments were conducted in both conditions of displacement, in conjunction with propeller U8 designed at the Wageningen Tank.
4. Open water propeller tests were conducted, the immersion to the axis corresponding to the screw diameter.

Remarks on Ship Trials:—

(*a*) Ballast condition. M.V. *Lubumbashi* conducted at Polperro, December 22, 1953.
Ship Particulars: 446·2 ft. *LBP* × 61·355 ft. Breadth mld. × 17·083 ft. mean mld. draft. Trim/stern = 6·67 ft.
Extreme displacement = 8,945 tons (35 ft³/ton).
Length of measured course:—
6,080 ft.—bearing E.W. 266°.
W.E. 86°.
Air Temperature: 13° C. Depth of water 22–25 fathoms under keel.
Water Temperature: 12° C.

Propeller.
Single screw (R.H.) 17·65 ft. diameter.
Mean designed face pitch = 14·70 ft. (measured deviation = 0·267 per cent).
Developed blade surface area = 112·8 ft².

Hull surface.
Riveted seams, welded butts, sandblasted hull.
Sea. Moderate. *Wind:* Beaufort scale 3 to 4 increasing to nearly 5 for group IV. Direction S.W.
Wave heights—2½–4·0 ft.
Wave lengths—82–98 ft.

(*b*) Loaded condition. M.V. *Lubumbashi* conducted at Polperro, January 10, 1954.
Ship particulars:—446·2 ft. *LBP* × 61·355 ft. Breadth mld. × 25·833 mean mld. draft. Trim/stern = 0·25 ft.
Length of measured course:—
6,080 ft.—bearing E.W. 266°.
W.E. 86°.
Air temperature: 10° C. Depth of water 22–26 fathoms under keel.
Water temperature: 11·5° C.

Propeller.
As for ballast trials.

Hull surface.
As for ballast trials.
Sea: Moderate. *Wind:* Beaufort scale 3 to 4. Direction N.W. going to N. at end of trials.
Wave heights—1·5–2¼ ft.
Wave lengths—49–66 ft.

Both ballast and loaded trials have been corrected to still water and calm air conditions as shown in Tables 6A/B.

Thrust and torsionmeters were fitted for both trials, the shp and thrust (corrected for trim) values being supplied by Professor G. Aertssen.

(*a*) *Comparison of Model Predictions, based on N.P.L. Method, with Ship Trial Results:—*

1. For this type of ship, having flush welded butts and riveted seams, a ship correlation factor of 1·0 is used to predict the dhp at the propeller.
viz. dhp at propeller
$$= \frac{\text{ship correlation factor} \times \text{ehp}_{\text{FROUDE}}}{\text{QPC}}$$
2. This analysis is shown in Table IV, for both loaded and ballast conditions of displacement.
3. Comparison of the model predictions based on a ship correlation factor of 1·0 with the ship trial results corrected to calm air conditions, is shown in Figs. 5A and 5B.
4. The comparison shows that a ship correlation factor of 1·0 is substantially correct for the ballast trials. In the loaded condition a ship correlation factor of 0·95 would be more appropriate.

(*b*) *Comparison of Model Predictions with Ship Trial Reports:*

1. From the ship trial reports of the M.V. *Lubumbashi*, ehp values have been derived from the measured shp on the assumption that the quasi-propulsive coefficients obtained from the model experiments are applicable to the ship.
This ehp = ship trial dhp for calm air conditions × model QPC
2. These ehp values have been compared with those according to Froude and Schoenherr, the latter

SHIP DIVISION — N.P.L.

REPORT ON EXPERIMENTS WITH MODEL No 3492

FIG. 1.A BODY SECTIONS & ENDINGS

SCALE:- 1/48 (SHIP)

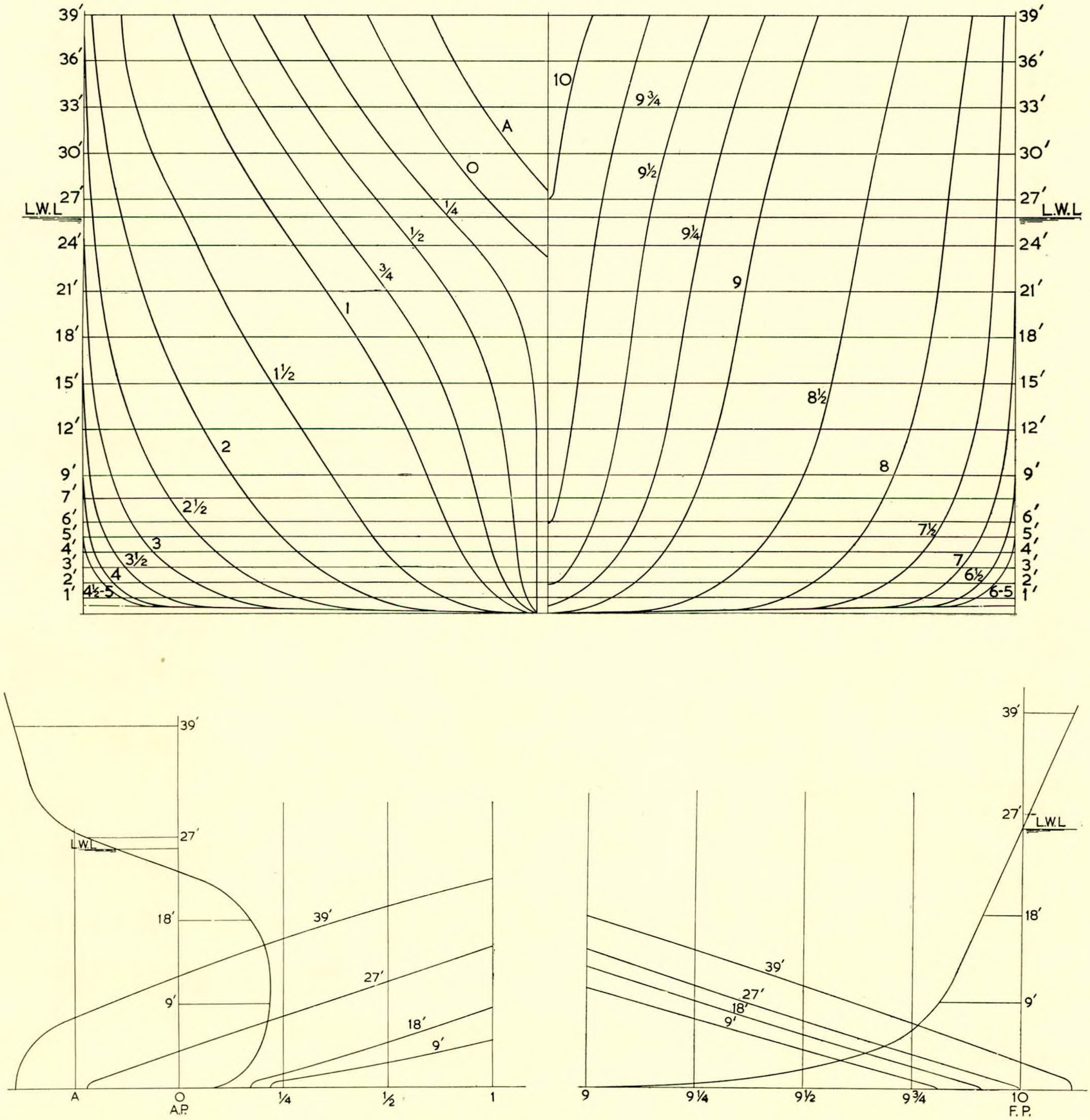


Fig. 1a

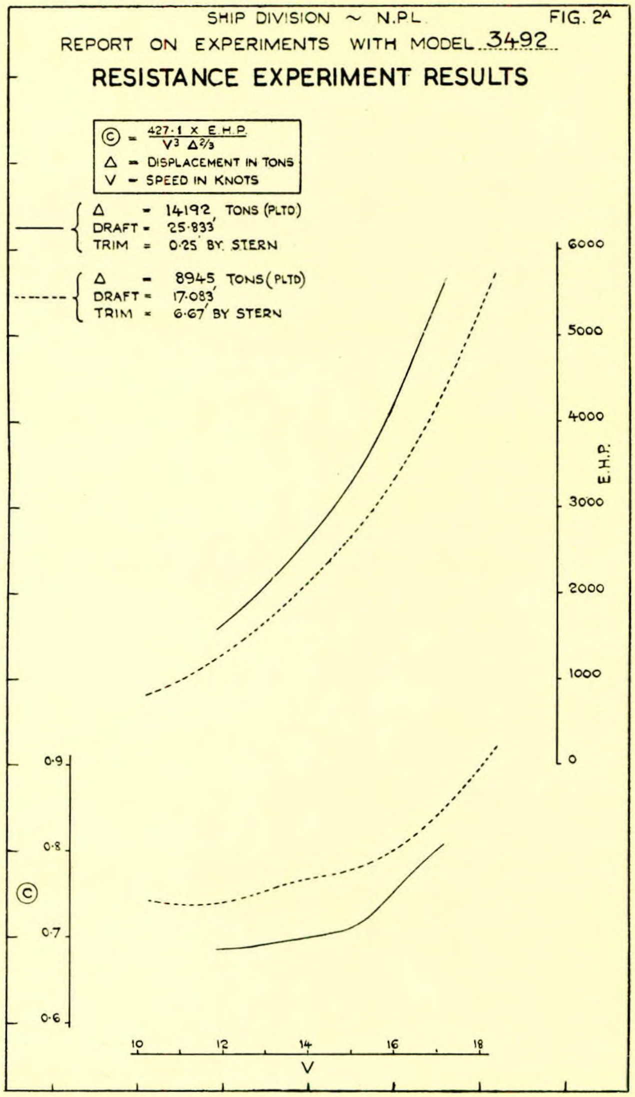


FIG. 2A

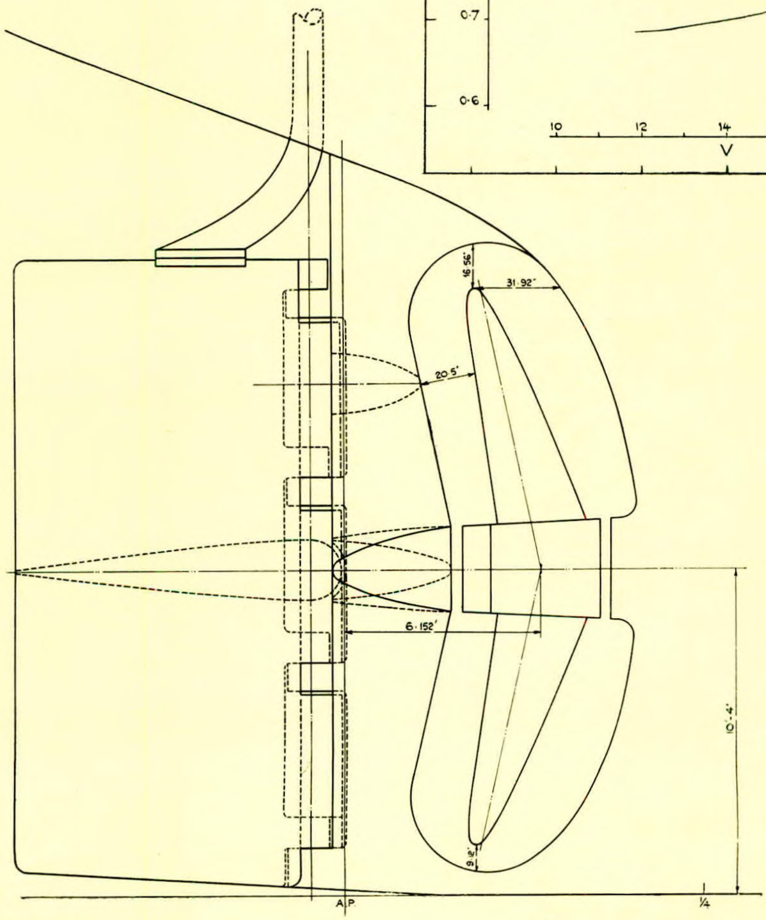


FIG. 3A.—STERN ARRANGEMENTS AS TESTED

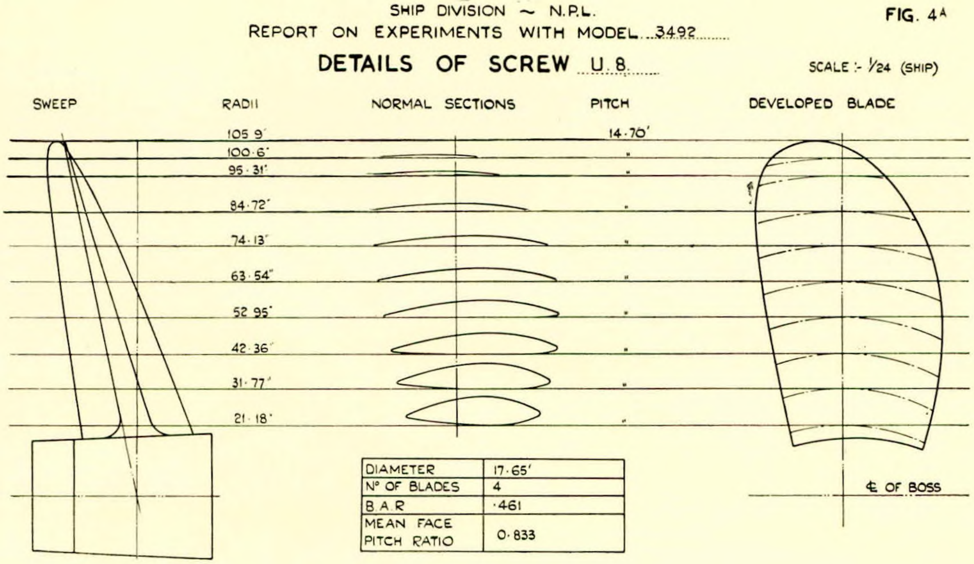


FIG. 4A

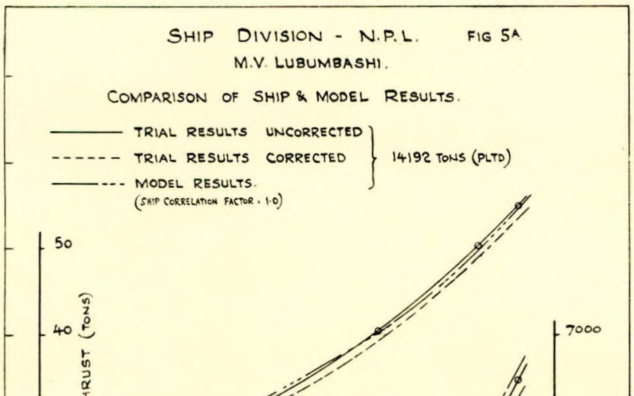


FIG. 5A

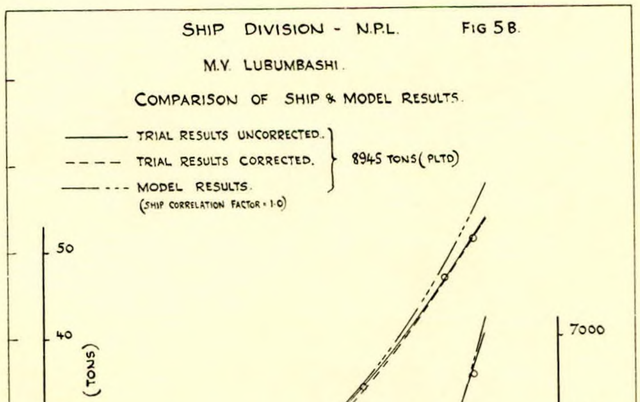


FIG. 5B

including no allowance for roughness. (See Tables VII A/B.)

3. The comparison shows that the Froude ehp values have to be multiplied by a factor of between 0·92 and 0·95 for correspondence with the ship ehp's, in the loaded condition, and between 0·975 and 1·027 in the ballast condition. The Schoenherr ehp values have to be multiplied by a factor varying between 1·04 and 1·07 in the loaded condition, and 1·08 to 1·18 in the ballast condition, for correspondence with the ship ehp's derived from the shp.

4. From the ship trial results of the M.V. *Lubumbashi*, ehp values have been derived from the measured thrusts, on the assumption that the thrust deduction fractions obtained from the model experiments are applicable to the ship.

   This ehp $= 6{\cdot}88 \times$ ship trial thrust (tons) $\times (1 - t)$ $\times$ ship speed in knots, for calm air conditions.

5. These ehp's have been compared with those according to Froude and Schoenherr. The comparison shows that the Froude ehp values have to be multiplied by a factor varying, between 0·96 and 1·04 in the loaded condition, and 0·93 to 1·02 in the ballast condition, for correspondence with the ehp values for the ship. The Schoenherr ehp values have to be multiplied by a factor varying between 1·09 and 1·19 in the loaded condition, and 1·03 to 1·18 in the ballast condition for correspondence with the ship ehp's derived from thrust measurements.

6. Ship *rpm* values have been compared with the corresponding model *rpm* at the same power absorption. In the loaded condition the ship *rpm* are between 0 and 2 per cent less than predicted. In the ballast condition the ship *rpm* are 0 to 2½ per cent higher than predicted.

7. The ship propeller was measured for accuracy of pitch and blade thickness by the manufacturer, and the model propeller as measured corresponded very closely with these records.

TABLE I

PRINCIPAL SHIP PARTICULARS CORRESPONDING TO CONDITIONS FOR WHICH MODEL TESTS WERE MADE

Model Hull No.		3 492	
Scale of Model		1/24	
Length *BP* (ft.)	*LBP*	446·2	
Breadth moulded (ft.)	B_{mld}	61·355	
Mean draft moulded (ft.)	d_{mld}	25·833	17·083
Trim at rest, in *LBP* (ft.)		0·25 by stern	6·67 by stern
Equivalent mean draft moulded at level trim (ft.)		—	17·12
Designed rake of keel, in *LBP* (ft.)		—	—
Displacement moulded (tons)	Δ_{mld}	14,117	8,887
Displacement with shell (tons)	Δ_{ext}	14,192	8,945
Block coefficient	Cb	0·699	0·664
Midship-area coefficient	C_m	0·985	0·977
Prismatic coefficient	C_p	0·709	0·679
Longitudinal centre of buoyancy from amidships *BP* (ft.)	*LCB*	1·24 forward midships	—
LCB in trimmed condition (ft.)		—	6·69 aft midships
½ angle of entrance of waterline	$\frac{1}{2}\alpha_e$	16·5	15·5
Length of entrance (ft.)	L_e	178·5	
Length of parallel (ft.)	L_p	66·9	
Length of run (ft.)	L_r	200·8	
Bilge radius (ft.)		4·90	
Rise of floor (ins.)		6·90	

Coefficients and *LCB* are for moulded displacement, including cruiser stern for moulded dimensions and level trim. Equivalent level draft used for coefficients where different from mean draft.

TABLE II

(C) AND EFFECTIVE HORSEPOWER VALUES FOR SHIP

Model Hull No.		3,492			
Mean draft moulded (ft.) Trim at rest, in *LBP* (ft.) Displacement with shell (tons)	d_{mld} Δ_{ext}	25·833 0·25 by stern 14,192		17·083 6·67 by stern 8,945	
Speed (knots)	V	(C)	ehp	(C)	ehp
	10½	—	—	0·740	865
	11	—	—	0·737	990
	11½	—	—	0·737	1,130
	12	0·685	1,625	0·739	1,290
	12½	0·687	1,840	0·745	1,465
	13	0·691	2,085	0·752	1,665
	13½	0·695	2,345	0·761	1,890
	14	0·699	2,630	0·767	2,125
	14½	0·703	2,940	0·771	2,370
	15	0·709	3,285	0·777	2,645
	15½	0·725	3,705	0·787	2,955
	16	0·751	4,220	0·801	3,310
	16½	0·778	4,795	0·818	3,705
	17	0·800	5,395	0·841	4,170
	17½	—	—	0·866	4,680
	18	—	—	0·895	5,265
	18¼	—	—	0·910	5,580

1. The model was made to the moulded lines and tested naked, i.e. without appendages, at the moulded displacement. A trip-wire 0·036 in. diameter, was fitted to girth the model at a section 5 per cent *LBP* abaft FP.

2. The estimated ehp values are for ship at displacement with shell, and are assumed to apply to a clean painted riveted ship in smooth deep salt water. No allowance is included for appendage or air resistance.

3. Skin friction correction from model to ship is according to R. E. Froude, and the results are corrected to a temperature of 59° Fahrenheit (15° Centigrade).

TABLE III

PRINCIPAL PARTICULARS OF THE SHIP SCREW

Model Hull No.	—	—	3,492
Model Screw No.	—	—	U8
No. of Screws	—	—	1
Scale of Model	—	—	1/24th
Designed by	—	—	Wageningen
Design No.	—	—	757–95–900A
Type of boss	—	—	Solid
Material	—	—	Bronze
Screw Details			
No. of blades	—	—	4
Diameter	D	ft.	17·65
Boss diameter (max.)	—	ins.	38·0
Boss diameter at rake line	D_b	ins.	35·4
Designed face pitch (max.)	P_f	ft.	14·70
Designed face pitch (mean)	P_m	ft.	14·70
Developed area outside boss	A_d	sq. ft.	112·8
Cylindrical thickness at root	t_r	ins.	9·15 at 21·18 ins. radius
Thickness at shaft axis	t	ins.	12·34
Rake aft	—	degs.	12
Boss diameter ratio	$Db/12D$	—	0·1671
Mean face pitch ratio	P_m/D	—	0·833
Blade area ratio	$4A_d/\pi D^2$	—	0·461
Thickness ratio	$t/12D$	—	0·058
Screw Position			
Centre of propeller—forward of A.P.	—	ft.	6·152
—above mld. base	—	ft.	10·33
Clearances			
Single screw:			
Trailing edge and fin	—	ins.	20·5
Top of aperture—above tips	—	ins.	16·56
—forward of tips	—	ins.	31·92
Bottom of aperture—below tips	—	ins.	9·12

Notes: 1. Mean face pitch is obtained by taking moments of pitch at equally spaced radii about the shaft axis.

$$P_m = \frac{\Sigma P_r r}{\Sigma r} \text{ where } P_r = \text{Face pitch at radius } r.$$

2. Centre of propeller is taken at intersection of rake line and shaft axis.

TABLE IV

EXPERIMENT RESULTS AND ESTIMATES FOR SHIP

Model hull No.		3492							
Mean draft moulded (ft.)	d_{mld}	25·833				17·083			
Trim at rest, in *LBP* (ft.)		0·25 by stern				6·67 by stern			
Displacement, with shell and appendages (tons)	ext	14,192				8,945			
Stern arrangements		Streamlined double plate rudder and fin				Streamlined double plate rudder and fin			
Model screw No.		U8				U8			
Direction of turning		R.H.				R.H.			
Experiment Results									
Speed (knots)	V	12·11	13·39	14·94	16·77	10·53	13·06	15·39	18·25
Wake fraction (Froude)	w_f	0·520	0·515	0·484	0·470	0·616	0·590	0·545	0·504
Wake fraction (Taylor)	w_t	0·342	0·340	0·326	0·320	0·381	0·371	0·353	0·335
Thrust deduction fraction	t	0·198	0·219	0·220	0·223	0·212	0·206	0·222	0·237
Hull efficiency	η_h	1·219	1·183	1·157	1·142	1·274	1·263	1·202	1·147
Screw efficiency in open water	η_o	0·584	0·587	0·594	0·591	0·583	0·593	0·605	0·607
Screw efficiency behind hull	η_b	0·627	0·635	0·644	0·634	0·632	0·640	0·652	0·650
Relative rotative efficiency	η_r	1·074	1·082	1·084	1·073	1·084	1·079	1·078	1·071
Quasi-propulsive coefficient	*QPC*	0·764	0·751	0·745	0·724	0·805	0·808	0·784	0·745
Revolutions per minute	N	82·6	92·3	104·3	121·2	66·2	82·7	99·9	125·0
Estimates for Ship									
Speed (knots)	V	12·0	13·5	15·0	16·5	10·5	13·0	15·5	18·25
Effective horsepower (naked)	ehp	1,625	2,345	3,285	4,795	865	1,665	2,955	5,580
Ship correlation factor		1·0	1·0	1·0	1·0	1·0	1·0	1·0	1·0
Quasi-propulsive coefficient	*QPC*	0·766	0·750	0·744	0·728	0·805	0·808	0·783	0·745
Horsepower delivered to screws	dhp	2,120	3,125	4,415	6,585	1,075	2,060	3,775	7,490
Revolutions per minute	N	81·8	93·0	104·7	118·5	66·0	82·4	100·7	125·0

1. The above predictions are for the ship with a smooth clean painted surface and measured mile conditions corresponding to Beaufort scale zero.
2. The ship correlation factor of 1·0 is based on recent experience of ship-model correlation and applies to ocean-going vessels with flush butts and riveted seams.
3. $\text{dhp} = \text{ehp} \times \dfrac{\text{Ship Correlation factor}}{QPC}$.
4. Transmission losses should be added to dhp to give bhp at engine.
5. The above revolutions per minute correspond to the dhp and are obtained from the model torque/revolution curve.

TABLE VA

SHIP TRIAL RESULTS—DETAILS OF RUNS. SHIP: M.V. "LUBUMBASHI" (LOADED CONDITION) MEASURED MILE: POLPERRO. DATE: JANUARY 10, 1954

Group and run		Direction	Time at start G.M.T. hrs.	min.	Ground speed knots	Tide knots	Water speed V knots	N *rpm*	mhp at torsion-meter	dhp	Thrust T tons	Range of rudder angles degrees	Weather: Sea	Relative wind: Speed knots	Relative wind: Direction	N/V	$\frac{dhp}{N^3\times10^3}$	$\frac{T}{N^2\times10^3}$
I	1	W.	10	26	14·41	+0·42	14·83	101·67	4,256	4,156	40·49	2° P.–4° S.	Moderate	27	20° S.	6·856	3·955	3·917
	2	E.	10	58	15·29	−0·34	14·95	102·57	4,254	4,153	40·21	5° P.–5° S.		14	60° P.	6·861	3·849	3·822
	3	W.	11	23	14·55	+0·24	14·79	101·22	4,204	4,104	40·31	4° P.–4° S.		26	20° S.	6·844	3·957	3·934
II	4	E.	11	50	16·29	−0·13	16·16	112·36	5,805	5,694	50·15	2° P.–2° S.		14	60° P.	6·953	4·014	3·972
	5	W.	12	23	16·00	0	16·00	111·31	5,776	5,666	50·38	1° P.–3° S.		31	10° S.	6·957	4·108	4·066
	6	E.	12	53	15·89	+0·15	16·04	111·94	5,773	5,663	50·09	2° P.–3° S.		19	60° P.	6·979	4·037	3·997
III	7	W.	13	27	16·74	−0·33	16·41	115·60	6,578	6,464	55·01	2° P.–4° S.		33	0°	7·044	4·184	4·116
	8	E.	13	56	16·15	+0·45	16·60	116·74	6,607	6,492	54·84	2° P.–4° S.		17	60° P.	7·033	4·081	4·024
	9	W.	14	22	16·93	−0·51	16·42	115·49	6,532	6,418	55·04	3° P.–4° S.		27	20° S.	7·033	4·166	4·126
IV	10	E.	14	55	11·94	+0·50	12·44	83·56	2,254	2,172	26·33	2° P.–2° S.		15	75° P.	6·717	3·723	3·771
	11	W.	15	28	12·71	−0·45	12·26	82·58	2,262	2,181	26·76	1° P.–3° S.		22	10° S.	6·736	3·873	3·924
	12	E.	16	00	12·03	+0·40	12·43	83·96	2,295	2,212	26·79	0 –3° S.		15	80° P.	6·755	3·737	3·800

1. Transmission losses between torsionmeter and screw taken from information supplied by Professor G. Aertssen.
2. True wind 10–18 knots north-westerly.
3. Thrust not corrected for hydrostatic head. This correction is −0·9 ton.

TABLE VB

SHIP TRIAL RESULTS—DETAILS OF RUNS. SHIP: M.V. "LUBUMBASHI" (BALLAST CONDITION). MEASURED MILE: POLPERRO. DATE: DECEMBER 22, 1953

Group and run		Direction	Time at start G.M.T. hrs.	min.	Ground speed knots	Tide knots	Water speed V knots	N *rpm*	mhp at torsion-meter	dhp	Thrust T tons	Range of rudder angles degrees	Weather: Sea	Relative wind: Speed knots	Relative wind: Direction	N/V	$\frac{dhp}{N^3\times10^3}$	$\frac{T}{N^2\times10^3}$
I	1	E.	9	27	15·957	−0·57	15·387	100·4	3,764	3,665	34·0	4° P.–3° S.	Moderate	12	45° S.	6·525	3·621	3·373
	2	W.	9	50	14·754	+0·44	15·194	99·2	3,743	3,645	34·3	4° P.–3° S.		22	25° P.	6·529	3·734	3·486
	3	E.	10	12	15·517	−0·30	15·217	100·5	3,783	3,684	34·2	4° P.–4° S.		12	45° S.	6·604	3·629	3·386
II	4	W.	10	33	16·949	+0·18	17·129	115·2	5,911	5,798	46·9	3° P.–4° S.		23	25° P.	6·725	3·792	3·534
	5	E.	10	47	17·308	−0·11	17·198	115·7	5,890	5,776	46·8	4° P.–5° S.		12	35° S.	6·728	3·729	3·496
	6	W.	11	05	17·142	+0·01	17·152	115·3	5,916	5,802	47·1	3° P.–3° S.		23	20° P.	6·722	3·785	3·543
III	7	E.	11	34	17·734	+0·14	17·874	120·2	6,695	6,576	51·3	3° P.–3° S.		14	45° S.	6·725	3·787	3·551
	8	W.	11	55	18·036	−0·23	17·806	119·7	6,667	6,549	51·1	2° P.–4° S.		27	20° P.	6·722	3·818	3·566
	9	E.	12	16	17·578	+0·26	17·838	120·0	6,693	6,575	51·3	2° P.–3° S.		12	45° S.	6·727	3·805	3·563
	10	W.	12	37	18·072	−0·28	17·792	119·7	6,686	6,568	51·3	3° P.–3° S.		27	20° P.	6·728	3·830	3·580
IV	11	E.	13	00	10·830	+0·25	11·080	71·27	1,316	1,246	16·3	4° P.–5° S.		8	80° S.	6·432	3·442	3·209
	12	W.	13	20	10·588	−0·21	10·378	68·20	1,248	1,181	–	4° P.–4° S.		22	35° P.	6·572	3·723	—
	13	E.	13	42	10·539	+0·15	10·689	70·60	1,292	1,222	16·5	5° P.–5° S.		10	75° S.	6·605	3·473	3·310

1. Transmission losses between torsionmeter and screw taken from information supplied by Professor G. Aertssen.
2. True wind 9–15 knots south-westerly.
3. Thrust not corrected for hydrostatic head. This correction is −0·6 ton.

TABLE VIA

SHIP TRIAL RESULTS—MEANS ANALYSIS. SHIP: M.V. "LUBUMBASHI" (LOADED CONDITION)

Group	Water speed V knots	N *rmp*	dhp	T tons	Estimate for calm air condition					
					N		dhp		T	
					Correction per cent	Corrected	Correction per cent	Corrected	Correction per cent	Corrected
I	14·88	102·01	4,142	40·30	−0·70	101·3	−3·33	4,004	−2·98	39·10
II	16·05	111·74	5,673	50·25	−0·66	111·0	−2·93	5,507	−2·53	48·98
III	16·51	116·15	6,467	54·96	−0·70	115·4	−2·91	6,279	−2·60	53·53
IV	12·35	83·17	2,187	26·66	−0·44	82·8	−2·10	2,141	−1·87	26·16

Weather conditions are given in Table VA. Thrust without static head.

TABLE VIB

SHIP TRIAL RESULTS—MEANS ANALYSIS. SHIP: M.V. "LUBUMBASHI" (BALLAST CONDITION)

Group	Water speed V knots	N *rmp*	dhp	T tons	Estimate for calm air condition					
					N		dhp		T	
					Correction per cent	Corrected	Correction per cent	Corrected	Correction per cent	Corrected
I	15·25	99·83	3,660	34·20	−0·53	99·3	−2·29	3,576	−2·05	33·5
II	17·17	115·47	5,788	46·90	−0·06	115·4	−0·311	5,770	−0·298	46·76
III	17·83	119·88	6,565	51·23	−0·15	119·7	−0·59	6,526	−0·55	50·95
IV	10·57	69·57	1,208	16·40	−1·64	68·43	−7·37	1,119	−6·70	15·30

Weather conditions are given in Table VB. Thrust without static head.

TABLE VIIA

LOADED CONDITION

V	Froude ehp	Schoenherr ehp	ehp derived from ship trial dhp	ehp derived from ship trial thrust	Ship trial *rpm*	Model predicted *rpm* at same power absorption	ehp derived from dhp / ehp Froude	ehp derived from dhp / ehp Schoenherr	ehp derived from thrust / ehp Froude	ehp derived from thrust / ehp Schoenherr	Ship *rpm* / Model *rpm* at same power absorption
12·0	1,625	1,425	1,490	1,690	80·3	80·3	0·917	1·046	1·040	1·186	1·000
12·5	1,840	1,615	1,695	1,835	83·9	84·1	0·921	1·050	0·997	1·136	0·998
13·0	2,085	1,830	1,915	2,030	87·4	87·9	0·918	1·046	0·974	1·109	0·994
13·5	2,345	2,065	2,155	2,260	91·0	91·7	0·919	1·044	0·964	1·094	0·992
14·0	2,630	2,325	2,420	2,535	94·7	95·5	0·920	1·041	0·964	1·090	0·992
14·5	2,940	2,600	2,735	2,850	98·5	99·5	0·930	1·052	0·969	1·096	0·990
15·0	3,285	2,910	3,085	3,220	102·3	103·6	0·939	1·060	0·980	1·107	0·987
15·5	3,705	3,295	3,510	3,650	106·3	107·9	0·947	1·065	0·985	1·108	0·985
16·0	4,220	3,765	3,990	4,135	110·5	112·7	0·945	1·060	0·980	1·098	0·980
16·5	4,795	4,295	4,570	4,715	115·2	117·7	0·953	1·064	0·983	1·098	0·979

Thrust without static head.

TABLE VIIb

Ballast Condition

V	Froude ehp	Schoenherr ehp	ehp derived from ship trial dhp	ehp derived from ship trial thrust	Ship trial *rpm*	Model predicted *rpm* at same power absorption	ehp derived from dhp / ehp Froude	ehp derived from dhp / ehp Schoenherr	ehp derived from thrust / ehp Froude	ehp derived from thrust / ehp Schoenherr	Ship *rpm* / Model *rpm* at same power absorption
11·0	990	860	1,015	1,010	71·2	69·5	1·025	1·180	1·020	1·174	1·024
12·0	1,290	1,130	1,325	1,305	77·5	75·8	1·027	1·173	1·012	1·155	1·022
13·0	1,665	1,465	1,690	1,655	83·9	82·4	1·015	1·154	0·994	1·130	1·018
14·0	2,125	1,875	2,125	2,090	90·4	89·5	1·000	1·133	0·984	1·115	1·010
15·0	2,645	2,340	2,670	2,610	97·5	97·0	1·009	1·141	0·987	1·115	1·005
16·0	3,310	2,940	3,340	3,220	105·1	104·8	1·009	1·136	0·973	1·095	1·003
17·0	4,170	3,730	4,165	3,970	113·2	113·1	0·999	1·117	0·952	1·064	1·001
18·0	5,265	4,765	5,140	4,900	121·8	122·0	0·976	1·079	0·931	1·028	0·998

Thrust without static head.

Appendix II

General Data of the Ship and Particulars of the Machinery

Ship's name: M.V. *Lubumbashi.*
Ship's owners: Compagnie Maritime Belge, S.A., Antwerp.
Shipbuilders: S.A. John Cockerill, Chantier Naval, Hoboken (Belgium).
Condition of hull: butts welded, seams riveted.
Engine builders: S.A. John Cockerill, Seraing (Belgium).
Lines of the vessel and screw particulars are given in Appendix I.

Dimensions:

Length overall	146·60 m. (480·99 ft.)
Length between perpendiculars ..	136·00 m. (446·20 ft.)
Breadth moulded	18·70 m. (61·36 ft.)
Depth to maindeck (D deck) ..	9·45 m. (31·01 ft.)
Depth to shelterdeck (C deck) ..	12·00 m. (39·37 ft.)
Loaded moulded draught in sea-water	abt. 8·23 m. (27·00 ft.)
Deadweight	abt. 9,500 tons
Service speed, loaded	abt. 15 knots

Height of superstructures:

From C to B deck	2·60 m. (8·53 ft.)
Forecastle	2·60 m. (8·53 ft.)
From B to A deck	2·60 m. (8·53 ft.)
From A deck to navigating bridge	2·50 m. (8·20 ft.)
Roof on navigating bridge ..	2·50 m. (8·20 ft.)

The main engine is a Burmeister & Wain double acting, two-stroke diesel engine 6·59 W.F. 125/45, developing in normal service conditions 6,000 bhp at 112 *rpm*, and especially built to operate on heavy fuel.

Dimensions:

Bore	590 mm. (23 3/16 in.)
Stroke of piston	1,250 mm. (49 3/16 in.)
Diameter of piston-valve top ..	592 mm. (23 4/16 in.)
Diameter of piston-valve bottom	588 mm. (23 2/16 in.)
Stroke of piston-valves	450 mm. (17 3/4 in.)
Diameter of sheath of piston-rod	236 mm. (9 5/16 in.)

From the double-bottom tanks where it is heated to about 100° F., the heavy fuel is drawn by the transfer pump to the settling tanks (120° F.). The fuel is then transferred to two purifiers in parallel of a capacity of 1 ton per hour each (from 175° F. to 185° F.), further to one clarifier of a capacity of 2 tons per hour (from 185° F. to 195° F.). Purifiers and clarifier are of Westfalia construction and are cleaned by hand.

Fig. 19 shows the way the fuel is drawn to the motor from the day tanks where the temperature is about 165° F. The piping is of a system adequate to operate:

(i) on diesel oil only;
(ii) on heavy fuel only;
(iii) on heavy fuel on top and diesel oil on bottom.

The last contingency has been made clear in the figure where full lines show the way to the top of the cylinders (heavy fuel), while dotted lines show the way to the bottom of the cylinders (diesel oil). There is one single tank of diesel oil for main engine and auxiliary motors.

The boiler installation comprises an exhaust gas boiler and an auxiliary donkey boiler of 7 kg. per sq. cm. (99·5 lb. per sq. in.).

For the auxiliaries are provided three sets of 250 kW with four-stroke diesel engine 428 *rpm*, 5 cylinders, bore 310 mm. (12 3/16 in.), stroke 390 mm. (15 5/16 in.), operating on diesel oil.

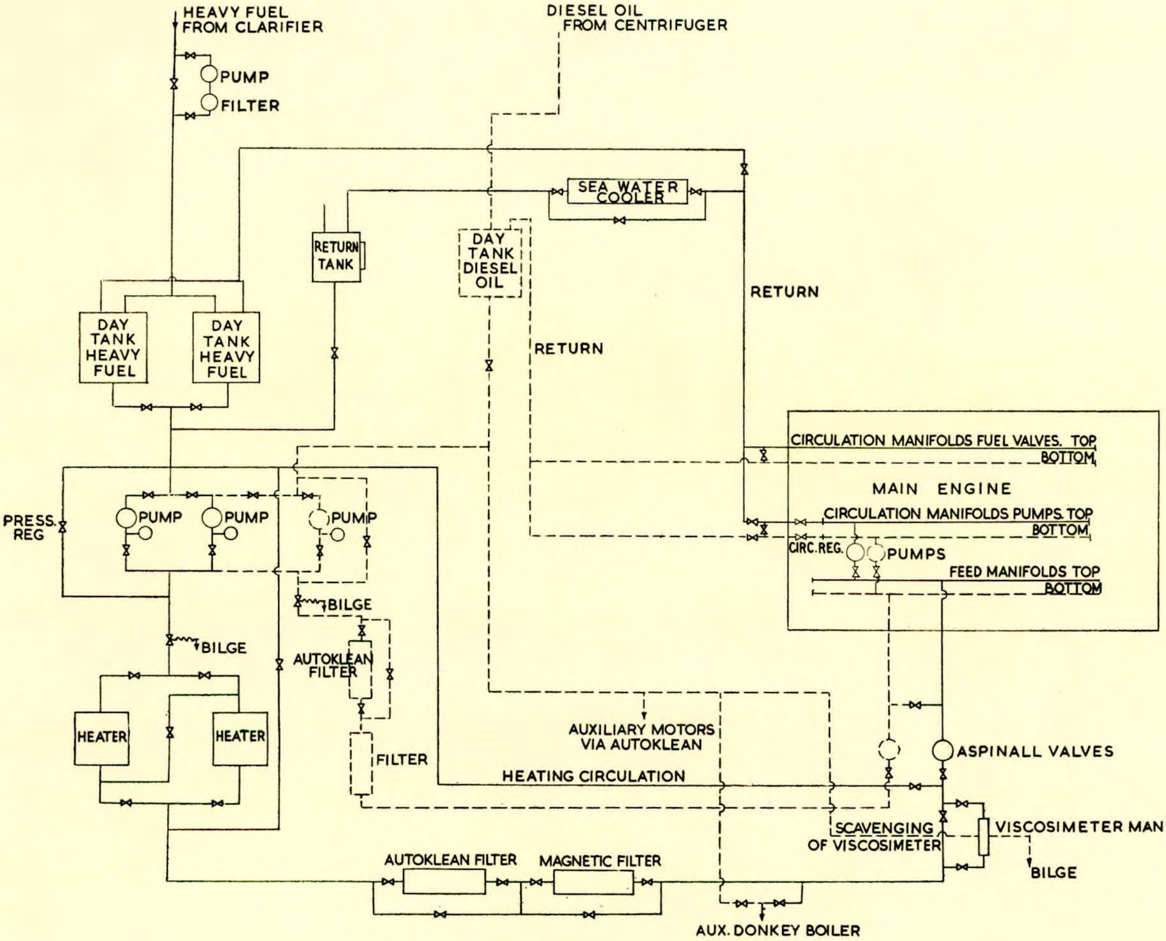


FIG. 19.—FUEL SUPPLY MAIN ENGINE

TABLE VIII

Engine Data

Date	Hour		rpm	pi lb. per sq. in.		ihp	shp	Efficiency	Mean draught, ft.	Manœuvr. lever in notch
				Top	Bottom					
	hrs.	min.								
22–12–53	9	30	100·4	68·7	59·5	5,128	3,810	74·3	17·3	21·5
	9	50	99·8	69·3	57·9	5,061	3,791	75·0	17·3	21·5
	10	10	100·5	68·8	59·4	5,198	3,831	73·8	17·3	21·5
	10	30	115·3	90·9	76·1	7,680	5,966	77·7	17·3	34·3
	10	50	115·8	86·6	74·6	7,440	5,945	79·9	17·3	34·3
	11	0	115·4	89·9	76·2	7,445	5,972	80·2	17·3	34·3
	11	40	120·2	92·0	82·2	8,335	6,758	81·1	17·3	40·5
	11	50	119·7	95·3	79·0	8,332	6,722	80·7	17·3	40·5
	12	20	120·1	93·8	78·6	8,241	6,756	82·0	17·3	40·5
	12	40	119·7	93·2	78·8	8,212	6,740	82·1	17·3	40·5
9–1–54	15	0	101·0	76·2	62·2	5,574	4,201	75·4	26·0	24·3
	15	30	100·7	76·2	59·5	5,461	4,161	76·2	26·0	24·3
	16	0	101·0	79·0	60·8	5,640	4,170	74·0	26·0	24·3
12–1–54	10	30	104·9	77·8	63·8	5,921	4,590	77·5	25·8	26·4
	15	0	104·9	79·7	67·7	6,160	4,590	74·5	25·8	26·4
	17	0	104·1	77·8	64·4	5,899	4,511	76·5	25·8	26·4
13–1–54	14	0	109·3	84·1	66·4	6,580	5,321	80·9	25·7	31·3
	15	0	109·5	84·8	70·3	6,766	5,319	78·6	25·7	31·3
	16	0	109·3	83·6	69·7	6,680	5,320	79·7	25·7	31·3
	17	0	109·7	81·5	66·2	6,460	5,321	82·4	25·7	31·3
	18	0	109·9	82·2	68·3	6,598	5,379	81·6	25·7	31·3
28–4–54	14	30	105·0	89·9	74·0	6,868	5,336	77·7	27·7	32·0
30–4–54	14	30	107·2	89·5	73·3	6,965	5,361	77·0	27·5	32·0
31–5–54	—		106·5	84·8	66·3	6,440	5,140	79·8	27·3	32·0
13–7–54	10	0	100·1	76·8	63·3	5,597	4,404	78·8	26·5	26·0
31–7–54	11	0	107·2	82·6	69·8	6,520	5,120	78·6	24·1	32·3
	16	0	113·2	91·6	78·0	7,655	6,220	81·3	24·1	40·0
1–8–54	5	40	98·4	71·3	59·9	5,147	3,840	74·6	24·0	24·3
	6	20	98·7	74·0	60·0	5,279	3,864	73·2	24·0	24·3
	7	0	98·6	73·2	59·6	5,228	3,860	73·8	24·0	24·3
	7	40	98·6	71·8	57·9	5,100	3,860	75·7	24·0	24·3
	8	30	110·3	90·1	74·4	7,230	5,601	77·5	24·0	35·2
	9	0	110·0	90·6	74·4	7,248	5,599	77·2	24·0	35·2
	9	40	110·1	88·9	75·4	7,210	5,615	77·8	24·0	35·2
	10	0	109·9	89·9	72·8	7,150	5,614	78·5	24·0	35·2
	10	50	114·7	92·6	80·1	7,890	6,402	81·2	24·0	41·0
	11	30	114·5	93·8	81·8	8,000	6,422	80·3	24·0	41·0

TABLE IX

Consumption Tests of Main Engine

Date	Duration of test in hours		Nature of fuel	rpm	shp	Fuel cons. lb. per hr.	Fuel rate lb. per shp per hr.*
	hrs.	min.					
9–1–54	1	30	Diesel oil	100·9	4,177	1,662	0·398
12–1–54	7	39	Heavy fuel	104·5	4,550	1,929	0·424
28–4–54	2	0	Heavy fuel	105·0	5,336	2,240	0·420
30–4–54	2	0	Heavy fuel	107·2	5,361	2,316	0·432
13–7–54	2	0	Heavy fuel	100·1	4,404	1,854	0·421
13–7–54	4	40	Heavy fuel	101·3	4,640	1,911	0·412

* Fuel rate is corrected for heat value in case of heavy fuel (standard = 18,500 B.Th.U.).

TABLE X

FUEL ANALYSIS A

Date	Nature of fuel	Specific gravity at		Viscosity seconds Redwood 1 at 100° F.	Heat value high B.Th.U. per lb.
		68° F. = 20° C.	167° F. = 75° C.		
9–1–54	Diesel oil	0·864	0·831	—	19,397
12–1–54	Heavy fuel	0·966	0·935	1,239	18,209
28–4–54	Heavy fuel	0·973	0·939	2,219	18,394
30–4–54	Heavy fuel	0·966	0·927	1,356	18,803
13–7–54	Heavy fuel	0·966	0·937	3,472	18,360
13–7–54	Heavy fuel	0·972	0·940	3,458	18,000

FUEL ANALYSIS B

Date	Ash per cent	Sulphur per cent	Asphaltenes per cent	Conradson carb. per cent	Flash point PM closed, deg. F.	Pour point deg. F.
12–1–54	0·051	2·17	6·49	11·62	196	21
28–4–54	—	3·01	—	—	183	—
30–4–54	0·071	2·98	3·20	8·94	186	48
13–7–54	0·056	2·23	8·03	12·10	183	23
13–7–54	—	2·30	—	—	180	—

Appendix III

TABLE XI

WEATHER DATA, VOYAGE ANTWERP–TENERIFFE

No.	Date	Hour		Course deg.	Description of sea	True wind		Rel. wind		Waves			Pitch angle deg.
						Beaufort scale	Direction	Strength knots	Direction deg.	Height feet	Length feet	Direction deg.	
		hrs.	min.										
1	9–1–54	15	0	239	Rath. rough	4	N.N.W.	23	20 S.B.	5	90	20 S.B.	—
2		15	30	239	Rath. rough	4	N.N.W.	23	20 S.B.	5	90	10 S.B.	—
3		16	0	239	Rath. rough	4	N.N.W.	23	20 S.B.	5	90	10 S.B.	—
4	10–1–54	10	30	266	Moderate	3–4	N.W.	27	20 S.B.	2	50	20 P.	—
5		11	0	86	Moderate	3–4	N.W.	14	60 P.	2	50	160 S.B.	—
6		11	30	266	Moderate	3–4	N.W.	26	20 S.B.	2	50	20 P.	—
7		12	0	86	Moderate	3–4	N.W.	14	60 P.	2	50	160 S.B.	—
8		12	30	266	Moderate	3–4	N.W.	31	10 S.B.	2	70	20 P.	—
9		13	0	86	Moderate	3–4	N.W.	19	60 P.	2	70	160 S.B.	—
10		13	30	266	Moderate	3–4	N.W.	33	0	2	70	20 P.	—
11		14	0	86	Moderate	3–4	N.W.	17	60 P.	2	70	160 S.B.	—
12		14	30	266	Moderate	3–4	N.W.	27	20 S.B.	2	70	20 P.	—
13		15	0	86	Moderate	3–4	N.W.	15	75 P.	2	70	160 S.B.	—
14		15	30	266	Moderate	3–4	N.W.	22	10 S.B.	2	70	20 P.	—
15		16	0	86	Moderate	3–4	N.W.	15	80 P.	2	70	160 S.B.	—
16	11–1–54	15	0	207	Smooth ripple	0–1	N.N.E.	16	0	2	50	160 S.B.	—
17		18	0	207	Smooth ripple	2	N.N.E.	10	0	2	50	160 S.B.	—
18	12–1–54	10	30	199	Moderate	2–3	N.N.E.	8	20 P.	3	100	135 S.B.	0·4
19		15	0	199	Moderate	2–3	N.N.E.	10	20 S.B.	3	100	135 S.B.	0·4
20		17	0	199	Moderate	2–3	N.N.E.	8	20 S.B.	3	100	135 S.B.	0·4
21	13–1–54	11	0	198	Rath. rough	4	N.N.E.	2	0	5	200	140 S.B.	0·6
22		14	0	197	Rath. rough	4	N.N.E.	0	0	6	200	140 S.B.	0·6
23		15	0	197	Rath. rough	4	N.N.E.	0	0	6	200	150 S.B.	0·6
24		16	0	197	Rath. rough	4	N.N.E.	0	0	6	200	155 S.B.	0·6
25		17	0	197	Rath. rough	4	N.N.E.	0	0	6	200	160 S.B.	0·6
26		18	0	197	Rath. rough	4	N.N.E.	4	0	6	200	160 S.B.	0·6

TABLE XII

WEATHER DATA, VOYAGE TENERIFFE–ANTWERP

No.	Date	Hour		Course deg.	Description of sea	True wind		Rel. wind		Waves			Pitch angle deg.
						Beaufort scale	Direction	Strength knots	Direction deg.	Height feet	Length feet	Direction deg.	
		hrs.	min.										
27	28–4–54	8	30	19	Rath. rough	4	N.W.	23	30 P.	5	160	25 P.	2·5
28		15	30	19	Rath. rough	5	N.W.	27	30 P.	8	240	30 P.	2·5
29		17	30	19	Rath. rough	5	N.W.	25	50 P.	8	240	30 P.	2·5
30	29–4–54	5	30	19	Rough	6	N.W.	27	45 P.	10	360	30 P.	3·0
31		9	30	19	Rough	6	N.W.	31	45 P.	12	410	30 P.	3·5
32		11	30	340	Rough	6	N.N.W.	35	0	13	410	0	6·5
33		12	0	19	Rough	6	N.N.W.	27	35 P.	13	410	30 P.	4·5
34		14	30	19	Rough	6	N.N.W.	31	35 P.	13	420	30 P.	5·5
35		15	0	19	Rough	6	N.N.W.	29	35 P.	13	420	30 P.	6·5
36		17	0	19	Rough	6	N.N.W.	31	35 P.	13	420	10 P.	5·0
37		18	0	19	Rough	6	N.N.W.	31	35 P.	13	420	10 P.	5·0
38	30–4–54	7	0	19	Moderate	3	N.N.W.	21	15 P	5	200	15 P.	1·0
39		9	0	19	Smooth	2–3	N.W.	19	15 P.	5	130	15 P.	1·0
40		11	30	19	Smooth	2	W.W.N.	14	20 P	5	150	20 P.	1·0
41		16	0	28	Smooth	2	S.W.	10	0	2	90	30 P.	0·5
42		17	30	28	Smooth	2–3	S.W.	8	0	2	90	30 P.	0·5
43	1–5–54	9	0	28	Moderate	4	S.S.W.	2	0	4	120	150 P.	0·5
44		10	0	28	Moderate	4	S.S.W.	2	0	4	120	150 P.	0·5
45		11	49	28	Moderate	4	S.S.W.	2	0	4	120	150 P.	0·5
46		12	0	208	Moderate	4	S.S.W.	30	0	5	120	30 S.B.	—
47		14	0	28	Moderate	4	S.S.W.	2	0	5	120	150 P.	0·5
48		15	0	25	Rough	5	W.S.W.	15	90 P.	6	140	150 P.	—
49		16	0	25	Rough	5	W.S.W.	16	90 P.	6	140	150 P.	0·5
50		17	30	25	Rough	5	W.	16	90 P.	6	140	150 P.	—
51	2–5–54	6	0	69	Very rough	6–7	S.W.	14	160 S.B.	7	180	160 S.B.	0·5
52		9	0	69	Very rough	7	S.W.	21	140 S.B.	8	200	140 S.B.	0·5
53		10	30	69	Very rough	7	S.W.	19	140 S.B.	8	200	140 S.B.	0·5
54		14	30	69	Very rough	7	S.W.	19	140 S.B.	9	160	140 S.B.	1·0
55		15	45	77	Very rough	7–8	S.W.	25	120 S.B.	10	170	120 S.B.	1·0

TABLE XIII

WEATHER DATA, VOYAGE LAS PALMAS–HAMBURG

No.	Date	Hour		Course deg.	Description of sea	True wind		Rel. wind		Waves			Pitch angle deg.
						Beaufort scale	Direction	Strength knots	Direction deg.	Height feet	Length feet	Direction deg.	
		hrs.	min.										
56	8–7–54	10	30	60	Rough	5	N.	31	30 P.	10	200	30 P.	2·0
57		11	0	60	Rough	5	N.	31	30 P.	10	200	30 P.	2·0
58		11	30	60	Rough	5	N.	31	30 P.	10	200	30 P.	1·5
59		11	45	60	Rough	5	N.	31	30 P.	10	200	30 P.	1·5
60		12	0	60	Rough	5	N.	31	30 P.	10	200	30 P.	1·5
61		13	0	60	Rough	5	N.	27	30 P.	10	200	30 P.	1·5
62		14	0	32	Rough	5	N.	33	15 P.	10	200	15 P.	1·5
63		15	0	32	Rough	5–6	N.	35	25 P.	11	230	30 P.	2·5
64		16	0	32	Rough	5	N.	31	25 P.	11	230	25 P.	2·5
65		17	30	32	Rough	5–6	N.N.E.	35	0	10	200	0	2·5
66	9–7–54	6	0	13	Rough	6	N.N.E.	37	0	13	330	0	4·5
67		7	0	13	Rough	6	N.N.E.	39	0	13	340	0	4·5
68		8	0	13	Rough	6	N.N.E.	39	0	13	350	0	3·0
69		9	0	13	Rough	6	N.N.E.	37	0	13	360	0	3·0
70		12	0	13	Rough	6	N.N.E.	35	0	14	400	0	4·0
71		14	30	13	Rough	6	N.N.E.	35	0	14	410	0	5·0
72		15	30	13	Rough	5–6	N.N.E.	31	0	14	410	0	4·0
73		17	0	13	Rough	5	N.N.E.	29	0	14	410	0	4·0
74		17	30	13	Rough	5	N.N.E.	29	0	14	410	0	4·0
75	10–7–54	5	30	13	Rath. rough	5	N.N.W.	31	20 P.	8	200	20 P.	2·0
76		7	0	13	Rath. rough	5	N.N.W.	31	20 P.	8	200	30 P.	2·0
77		10	0	13	Rath. rough	4–5	N.N.W.	27	20 P.	8	200	30 P.	2·0
78		12	0	13	Rath. rough	4–5	N.N.W.	27	20 P.	8	200	30 P.	1·5
79		13	30	13	Rath. rough	4–5	N.N.W.	27	20 P.	8	180	30 P.	1·5
80		15	0	13	Rath. rough	4–5	N.N.W.	27	20 P.	7	180	30 P.	1·5
81		16	0	13	Rath. rough	4–5	N.N.W.	27	20 P.	7	180	30 P.	1·5
82	11–7–54	5	30	27	Rough	5	N.W.	25	50 P.	8	180	60 P.	2·0
83		10	0	27	Rough	5	N.W.	25	50 P.	10	200	60 P.	2·0
84		11	30	27	Rough	5	N.W.	21	60 P.	10	200	60 P.	2·0
85		15	0	27	Rough	7	W.N.W.	33	70 P.	12	260	70 P.	3·5
86		16	15	27	Rough	7–8	W.N.W.	37	70 P.	16	300	60 P.	3·5
87		17	45	27	High sea	8	N.W.	43	60 P.	18	330	60 P.	6·0
88		19	30	27	High sea	9	N.W.	49	60 P.	20	340	60 P.	6·0
89		20	0	350	High sea	6	N.W.	29	0	20	340	0	6·0
90		22	0	350	High sea	6	N.W.	29	0	20	340	0	6·0
91	12–7–54	7	0	47	Rath. rough	5–6	N.W.	29	50 P.	10	200	60 P.	2·0
92		10	30	47	Rath. rough	5–6	N.W.	27	60 P.	9	200	70 P.	1·5
93		11	45	47	Rath. rough	5–6	N.W.	27	60 P.	9	200	60 P.	2·0
94		15	0	58	Rath. rough	5–6	N.W.	27	60 P.	8	180	120 P.	1·0
95		17	0	58	Moderate	5	N.W.	21	60 P.	7	180	90 P.	1·0
96	13–7–54	3	45	68	Smooth	3	N.W.	14	45 P.	3	140	120 P.	0·5
97		4	0	68	Smooth	3	N.W.	14	45 P.	3	140	120 P.	0·5
98		4	30	68	Smooth	3	W.N.W.	12	55 P.	3	140	150 P.	0·5
99		5	0	68	Smooth	3	W.N.W.	12	55 P.	3	140	150 P.	0·5
100		6	30	68	Smooth	3	W.N.W.	12	55 P.	3	100	150 P.	0·5
101		8	0	68	Smooth	3	W.N.W.	10	45 P.	3	100	150 P.	0·5
102		9	30	55	Smooth	3	W.	8	20 P.	3	100	150 P.	0·5
103		10	0	55	Smooth	3	W.	8	20 P.	3	100	150 P.	0·5
104		19	0	39	Moderate	4	S.S.W.	0	0	5	160	45 P.	0·5
105		20	0	39	Moderate	4	S.S.W.	0	0	5	160	45 P.	0·5

TABLE XIV

PROPULSION DATA, BALLAST TRIAL

Run	Speed knots	*rpm*	dhp	Thrust in tons	ehp	$\frac{\text{ehp}}{\text{dhp}}$
1	15·62	100·4	3,665	34·0	2,839	0·774
2	14·92	99·2	3,645	34·3	2,751	0·755
3	15·56	100·5	3,684	34·2	2,844	0·772
4	16·87	115·2	5,798	46·9	4,185	0·722
5	17·52	115·7	5,776	46·8	4,320	0·748
6	16·88	115·3	5,802	47·1	4,204	0·724
7	18·07	120·2	6,576	51·3	4,871	0·741
8	17·63	119·7	6,549	51·1	4,740	0·724
9	18·02	120·0	6,575	51·3	4,860	0·739
10	17·67	119·7	6,568	51·3	4,770	0·726
11	11·53	71·3	1,246	16·3	1,022	0·821
12	9·92	68·2	1,181	—	—	—
13	11·18	70·6	1,222	16·5	1,004	0·821

Displacement 8,945 tons. Trim by stern 6·67 ft.

Thrust not corrected for hydrostatic head. This correction is −0·6 ton.

TABLE XV

PROPULSION DATA, VOYAGE ANTWERP–TENERIFFE

No.	Speed knots	*rpm*	dhp	Thrust in tons	ehp	$\frac{\text{ehp}}{\text{dhp}}$	Δ	dhp corr. for Δ	Increase of power per cent	Loss of speed per cent
1	14·86	101·0	4,050	39·5	3,150	0·778	14,260	4,021	2·7	0·6
2	14·66	100·7	4,011	39·8	3,131	0·781	14,260	3,985	7·1	1·7
3	14·73	101·0	4,021	39·7	3,138	0·780	14,260	3,995	5·7	1·3
4	14·75	101·7	4,156	40·5	3,201	0·771	14,192	4,156	9·1	2·2
5	15·07	102·6	4,153	40·2	3,251	0·783	14,192	4,153	0·4	0·1
6	14·66	101·2	4,104	40·3	3,171	0·773	14,192	4,104	10·5	2·5
7	16·21	112·4	5,694	50·1	4,357	0·766	14,192	5,694	−0·5	−0·1
8	15·88	111·3	5,666	50·4	4,285	0·756	14,192	5,666	9·5	1·7
9	16·19	111·9	5,663	50·1	4,342	0·767	14,192	5,663	−0·6	−0·1
10	16·34	115·6	6,464	55·0	4,810	0·744	14,192	6,464	8·3	1·6
11	16·65	116·7	6,492	54·8	4,881	0·752	14,192	6,492	−1·3	−0·2
12	16·36	115·5	6,418	55·0	4,820	0·752	14,192	6,418	7·1	1·3
13	12·56	83·6	2,172	26·3	1,800	0·828	14,192	2,172	−2·6	−0·9
14	12·14	82·6	2,181	26·8	1,790	0·820	14,192	2,181	9·0	2·6
15	12·54	84·0	2,212	26·8	1,826	0·825	14,192	2,212	−0·6	−0·2
16	15·21	104·6	4,481	42·0	3,428	0·765	14,160	4,483	4·2	1·0
17	15·06	102·4	4,110	40·2	3,250	0·791	14,160	4,112	−0·4	−0·2
18	15·26	104·9	4,431	41·8	3,424	0·773	14,130	4,435	1·7	0·5
19	15·46	104·9	4,431	41·8	3,468	0·782	14,130	4,435	−3·6	−0·8
20	15·36	104·1	4,357	42·0	3,461	0·795	14,130	4,359	−2·7	−0·7
21	15·23	103·8	4,304	40·9	3,341	0·777	14,090	4,312	−0·3	−0·1
22	15·86	109·3	5,160	46·5	3,952	0·766	14,090	5,168	0·6	0·1
23	15·96	109·5	5,157	46·1	3,943	0·765	14,090	5,161	−2·5	−0·6
24	15·95	109·3	5,159	46·4	3,965	0·769	14,090	5,165	−2·1	−0·5
25	15·96	109·7	5,160	46·6	3,987	0·773	14,090	5,168	−2·4	−0·6
26	15·96	109·9	5,219	46·6	3,987	0·764	14,090	5,223	−1·4	−0·3

Trim by stern: varying from 0·1 ft. on 9–1–54 to 0·8 ft. on 13–1–54.

Thrust not corrected for hydrostatic head. This correction is −0·9 ton.

TABLE XVI

PROPULSION DATA, VOYAGE TENERIFFE–ANTWERP

No.	Speed knots	*rpm*	dhp	Thrust in tons	ehp	$\frac{ehp}{dhp}$	Δ	dhp corr. for Δ	Increase of power per cent	Loss of speed per cent
27	14·60	105·9	5,160	47·5	3,720	0·721	15,300	4,900	22·5	5·2
28	14·20	104·9	5,195	48·0	3,656	0·704	15,300	4,931	36·6	8·1
29	14·30	105·5	5,260	48·0	3,680	0·700	15,300	4,996	34·8	7·7
30	13·55	103·5	5,118	49·3	3,589	0·702	15,230	4,868	58·3	12·0
31	12·90	102·5	5,020	50·2	3,499	0·697	15,230	4,774	82·6	15·7
32	12·65	102·1	5,008	51·2	3,510	0·701	15,230	4,766	94·2	17·3
33	13·30	102·3	5,019	49·8	3,560	0·710	15,230	4,773	64·9	13·1
34	12·90	102·0	5,000	51·2	3,569	0·714	15,230	4,757	82·0	15·7
35	12·60	102·5	5,020	50·8	3,479	0·693	15,230	4,774	96·8	17·6
36	12·85	102·2	5,049	50·3	3,492	0·692	15,230	4,803	85·8	16·3
37	13·40	103·5	5,066	49·8	3,585	0·708	15,230	4,821	62·6	12·7
38	14·85	106·3	5,165	47·7	3,799	0·735	15,230	4,917	15·6	3·9
39	15·10	106·5	5,199	47·2	3,820	0·736	15,230	4,943	9·2	2·3
40	15·20	107·0	5,200	46·7	3,808	0·732	15,230	4,947	6·7	1·6
41	15·45	107·2	5,204	46·2	3,828	0·735	15,230	4,952	0·3	0·0
42	15·45	107·4	5,218	46·7	3,870	0·742	15,230	4,961	0·5	0·3
43	15·60	107·2	5,185	46·1	3,858	0·744	15,180	4,950	−3·5	−1·0
44	15·60	107·4	5,161	46·1	3,858	0·747	15,180	4,928	−3·9	−1·0
45	15·60	107·4	5,198	46·6	3,900	0·751	15,180	4,960	−3·3	−1·0
46	14·75	106·0	5,180	47·9	3,790	0·732	15,180	4,943	19·1	4·5
47	15·60	107·3	5,160	46·1	3,858	0·748	15,180	4,924	−4·0	−1·0
48	15·50	107·3	5,200	46·3	3,850	0·740	15,180	4,962	−0·8	−0·3
49	15·50	107·3	5,200	46·6	3,873	0·745	15,180	4,962	−0·8	−0·3
50	15·50	107·3	5,200	46·6	3,873	0·745	15,180	4,962	−0·8	−0·3
51	15·40	106·8	5,180	47·0	3,880	0·749	15,150	4,960	1·8	0·3
52	15·35	106·3	5,038	47·0	3,869	0·768	15,150	4,821	0·1	0·0
53	15·40	105·8	5,020	46·5	3,840	0·765	15,150	4,807	−1·3	−0·3
54	15·55	106·3	5,020	46·5	3,877	0·772	15,150	4,807	−5·0	−1·3
55	15·55	105·9	5,000	46·5	3,877	0·776	15,150	4,788	−5·4	−1·3

Trim by stern: varying from 3·6 ft. on 28–4–54 to 2·3 ft. on 2–5–54.

Thrust not corrected for hydrostatic head. This correction is —1·0 ton.

TABLE XVII

PROPULSION DATA, VOYAGE LAS PALMAS–HAMBURG

No.	Speed, knots	*rpm*	dhp	Thrust in tons	ehp	$\frac{\text{ehp}}{\text{dhp}}$	Δ	dhp corr. for Δ	Increase of power per cent	Loss of speed per cent
56	14·20	104·0	5,020	47·4	3,610	0·719	14,705	4,895	32·8	7·5
57	14·40	104·7	5,160	48·9	3,778	0·732	14,705	5,032	30·4	6·8
58	14·65	105·8	5,300	49·9	3,920	0·740	14,705	5,167	26·5	5·8
59	14·80	107·0	5,450	52·4	4,160	0·764	14,705	5,318	25·6	5·4
60	14·75	107·0	5,450	52·4	4,143	0·760	14,705	5,318	27·2	5·7
61	14·80	107·8	5,610	51·8	4,110	0·733	14,705	5,470	29·4	6·3
62	14·75	107·3	5,596	51·8	4,100	0·733	14,705	5,456	30·4	6·6
63	14·80	108·0	5,628	50·8	4,031	0·716	14,705	5,488	29·6	6·3
64	14·80	107·7	5,613	51·8	4,112	0·733	14,705	5,472	29·4	6·3
65	14·70	107·3	5,597	51·8	4,082	0·730	14,705	5,458	32·0	7·0
66	13·70	105·8	5,595	52·2	3,835	0·686	14,705	5,456	66·6	13·3
67	13·80	105·8	5,595	51·8	3,834	0·686	14,705	5,456	62·7	12·7
68	13·80	105·8	5,595	51·8	3,834	0·686	14,705	5,456	62·7	12·7
69	13·80	105·3	5,548	52·2	3,865	0·696	14,705	5,414	61·4	12·4
70	13·20	101·7	5,066	50·3	3,571	0·705	14,705	4,942	70·2	14·3
71	13·00	102·3	5,101	49·8	3,490	0·684	14,705	4,976	80·0	15·6
72	13·50	103·0	5,058	49·8	3,610	0·713	14,705	4,930	58·0	12·3
73	10·00	82·7	2,698	—	—	—	14,705	2,630	49·0	21·9
74	12·70	96·4	4,342	44·9	3,090	0·712	14,705	4,236	65·2	14·2
75	14·20	104·0	5,080	48·7	3,709	0·730	14,660	4,960	34·6	7·8
76	14·55	104·0	5,080	48·7	3,800	0·748	14,660	4,960	24·1	5·5
77	14·85	104·8	5,101	47·7	3,799	0·744	14,660	4,980	16·2	3·6
78	14·75	105·3	5,120	47·7	3,772	0·737	14,660	4,998	19·4	4·5
79	14·70	105·2	5,100	47·2	3,720	0·730	14,660	4,978	20·4	4·5
80	14·85	105·2	5,100	47·7	3,800	0·745	14,660	4,978	16·2	3·6
81	14·80	105·2	5,100	47·7	3,785	0·742	14,660	4,978	17·5	3·9
82	15·05	105·9	5,145	47·6	3,840	0·746	14,635	5,023	11·4	2·6
83	14·75	104·5	5,080	47·6	3,762	0·742	14,635	4,960	18·5	4·2
84	14·55	104·5	5,060	47·6	3,711	0·734	14,635	4,940	23·6	5·5
85	14·00	103·0	5,041	49·1	3,688	0·732	14,635	4,925	39·9	9·1
86	14·00	103·5	5,030	48·6	3,650	0·726	14,635	4,911	39·6	8·8
87	13·10	100·8	4,898	49·6	3,500	0·715	14,635	4,782	69·0	14·4
88	11·00	98·0	4,829	—	—	—	14,635	4,716	150·3	27·6
89	9·90	87·8	3,541	—	—	—	14,635	3,460	100·5	28·8
90	9·20	81·8	2,898	—	—	—	14,635	2,830	71·5	29·8
91	14·65	104·4	5,100	48·0	3,770	0·740	14,600	4,998	22·3	5·2
92	14·70	105·1	5,126	48·0	3,783	0·738	14,600	5,022	21·4	4·9
93	14·85	105·3	5,136	48·0	3,822	0·745	14,600	5,030	17·4	3·9
94	15·05	105·9	5,175	48·0	3,872	0·749	14,600	5,068	12·4	2·9
95	15·05	106·0	5,139	48·5	3,913	0·762	14,600	5,037	11·7	2·6
96	15·40	106·6	5,158	46·9	3,875	0·752	14,550	5,068	2·4	0·6
97	15·60	107·4	5,278	47·4	3,966	0·752	14,550	5,185	−0·6	0·0
98	15·70	108·1	5,376	48·1	4,050	0·754	14,550	5,282	−1·1	−0·3
99	15·80	109·1	5,534	49·7	4,209	0·761	14,550	5,440	−0·8	−0·3
100	15·85	109·5	5,660	49·7	4,220	0·746	14,550	5,562	0·0	0·0
101	15·90	110·3	5,793	50·9	4,337	0·749	14,550	5,695	1·2	0·3
102	14·75	100·3	4,220	41·5	3,284	0·778	14,550	4,150	−0·8	−0·3
103	14·75	100·1	4,256	41·0	3,243	0·762	14,550	4,183	0·0	0·0
104	14·85	101·6	4,517	43·0	3,426	0·758	14,550	4,440	3·6	1·0
105	14·85	101·3	4,498	43·0	3,426	0·762	14,550	4,420	3·1	0·7

Trim by stern: varying from 2·1 ft. on 8–7–54 to 0·5 ft. on 13–7–54.

Thrust not corrected for hydrostatic head. This correction is −0·9 ton.

DISCUSSION

Dr. J. F. Allan, D.Sc. (*Member of Council I.N.A.*): I am sure you will all agree that this is a very interesting paper and it has a lot of points of contact with the work which has been going on in this country, by the British Shipbuilding Research Association and the National Physical Laboratory.

Professor Aertssen has referred, particularly in reference to myself, to the ship-model comparison picture. But I should like to make a few general remarks.

First of all on the question of accuracy, referred to in Part I of the paper, the measurement of the speed through the water is dependent on the pitometer log and that is calibrated on the measured mile trials. In so far as it is calibrated on the measured mile in calm water it is probably very reliable in calm water, but did the author form any impression during the voyage trials as to the loss of accuracy in the pitometer log in pitching conditions?

The other accuracies I do not propose to discuss in detail, but it seems to me that the accuracies stated at the end of Part I tend to be on the optimistic side, even considering the fact that the torsionmeter was calibrated on the shaft.

Regarding the roughness measurements—I made some reference to them this morning—this is a matter of some difficulty because the data are not entirely complete as between the *Lucy Ashton* and the *Lubumbashi*. However, the comparable increases in grain roughness on the two vessels after six months are similar and yet the increase in resistance on the *Lucy Ashton* is much greater than than on the *Lubumbashi*. I have previously suggested that this is a point of some considerable importance. There is the growth of barnacles on the *Lucy Ashton*, which various speakers this morning have indicated may be responsible for the large increase in resistance; against that there are the rust blisters on the *Lubumbashi*, which are mentioned by Professor Aertssen. These, of course, would not be so rough and pointed as barnacles, but they are a factor to be taken into consideration.

One might also remark on the effect of slimy scum on the resistance of a ship. When a model has been in the tank for some weeks it acquires a scum which is smooth and slimy to the touch, and there is a marked increase of resistance; frequently ships which are docked after a period at sea have a similar slimy fouling and although they look comparatively clean they have a marked increase in resistance. It appears therefore that other factors than the physical roughness measured by existing gauges can have a material effect in increasing the resistance.

With regard to the calculation of shaft losses (Fig. 12), the method of extrapolation to zero *rpm* has been used and this has been discussed on previous occasions. Unless data are available for very low revolutions, one should not place too much reliance on this method of determining the friction of the tailshaft of the transmission line. Obviously a very slight variation in the extrapolation of the curves in Fig 12 could make a material difference to the answer which is taken off at the zero ordinate.

The analysis of the weather results I find extremely interesting, and although one could criticize the details in many ways, the broad picture cannot be denied. The most interesting point is the very rapid increase in the power required to maintain a given speed when the weather factor exceeds a certain amount. I think that in general there is a tendency to assume rather low weather factors. My own policy based on experience is to allow a factor for service conditions rather larger than is generally acceptable. You will note that in Fig 16, up to about a Beaufort number of 4 or 5 there is a fairly moderate increase of power, and after that a very spectacular increase, which suggests that it is not a practical policy to apply additions beyond those indicated for Beaufort 5. If one takes an increase of the order of 25 per cent at a Beaufort number 5 and there is fouling in addition, an overall figure of 35 per cent is obtained, which I submit is a safe figure to work with for most sea routes.

Referring now to the ship-model comparison, and in particular Figs 5*a* amd 5*b*, it will be agreed that the general picture is fairly satisfactory. Looked at more closely, however, there are some discrepancies. The measured results based on shaft horse power require a factor from 0·92 to 0·95 in the loaded condition and from 0·975 to 1·027 in the ballast condition applied to Froude ehp. These factors compare with N.P.L. standard practice of a unity factor for this type of shell construction, so that in the load condition the performance is some 5 per cent better than prediction. This is not outside the margin of variation indicated in the results given in this Institution a year ago by Mr. Canham and myself.

Professor Aertssen says he does not think there is very much effect on the performance due to the shell landings with the flush welded butts. With completely flush welded hulls we have obtained factors of the order of 0·8–0·9 to Froude ehp, as is given in the paper already referred to. The accurate determination of the effect of various structural roughnesses is a long term research and the approach from the analysis even of accurate ship trials is difficult because the influence of structural roughness is masked by the influence of plating and paint roughness which are now emerging as very significant factors.

I am sure you will all join me in expressing appreciation of this very useful work which the Belgian Institution has carried out on the practical ship trials and the ship-model correlation.

Mr. A. W. Davis, B.Sc. (*M.I.N.A. and M.I.Mar.E.*): It is particularly interesting to hear the author's views as to the relation between the increase of power and the idiosyncrasies of the weather. It is more or less the practice for vessels on the North Atlantic to allow an increase of power of about 30 per cent for maintaining a particular reduced speed under average bad weather conditions and with an average dirty hull as compared with the trial condition at that speed. From Fig. 16 it would seem that this is a fair estimate of the requirements.

The author refers to the relative effects of running with diesel oil and heavy fuel. This is a subject as pronounced in the variety of experience as in the variation of the quality of so-called heavy fuel. As a point of minor amendment, it is thought that in the third last paragraph of Part III on page J5 the expression "excess of" should read "excess by." This would remove ambiguity whereby it is possible to interpret that cylinder liner wear on heavy oil was less than on diesel oil. This might seem facetious were it not that such a claim has already been reported from another quarter. It is in fact possibly more the general experience that cylinder liner wear with average heavy fuel and normal cylinder lubricating oil is double that with diesel oil and it would be interesting to know from the author whether an additive was being employed in the cylinder lubricant in this instance.

The remarks by the author on hull roughness are most interesting. It has been suggested that skin resistance increases 2 per cent for each day out of dry dock, but this must be a function of many variable factors.

Professor A. M. Robb, D.Sc. (*Vice-President I.N.A.*): May I first endorse Dr. Allan's expression of our thanks to Professor Aertssen, and to the Compagnie Maritime Belge, for their generosity in presenting a mass of very valuable information. My only regret is that I have not had opportunity to give the paper the consideration it deserves. I cannot pretend to discuss it in detail. But I should like to raise two points. One arises from a remark made by Mr. Davis. Twenty to thirty years ago a very well known and respected Lloyd's surveyor in Liverpool, John Dykes, never tired of dilating on the folly of shipowners in taking ships out of dry dock before the fresh bottom paint had had a chance of hardening; he was completely satisfied that the cost of an extra day in dock would be more than recovered by the resulting improvement of the bottom coating. No doubt the concern was primarily with depreciation of the shell plating. It is, however, possible that the improved condition of the bottom might be beneficial in other ways. My other point concerned Dr. Allan rather than Professor Aertssen. I suggest that proper ship-model correlation can never be attained until all the available data are examined. The point here is that the relation between the thrust and the torque for the ship propeller is known, and Dr. Allan must know the relation between the thrust and the torque for the model propeller in open water. It is a fortunate fact that, at least over the working range, the plotting of thrust constant on a base of torque constant gives, for model propeller results, a line which is straight, or very nearly so. Unfortunately, such limited data as are available show the results for the ship propeller lying well above the line for the model propeller; in effect, for given torque constant the thrust constant of the ship propeller is appreciably greater than that of the model propeller. It is not good enough blandly to dismiss the discrepancy as being merely scale effect. It is necessary to determine, if possible, whether there is actually always such a discrepancy, and if so why does it occur and what is the limit. If there is indeed always such a discrepancy as has been indicated there is an interesting implication; namely that under-assessment of the thrust delivered by a ship propeller as derived from a propulsion experiment conceals an under-assessment of the ship resistance.

Mr. H. J. S. Canham (*A.M.I.N.A.*): Since this is a very full and interesting paper, it is not possible to examine it closely within a short time and I shall therefore confine my remarks to certain aspects of the instrumentation, the roughness measurements, and the tank tests.

I am a little surprised that Professor Aertssen considers there is greater accuracy of thrust measurement in the loaded trial than in the ballast trial because only the ahead type of Michell thrustmeter was fitted. You can make a correction to take into account the weight of the shafting and you can apply that to the thrust readings. I cannot see, therefore, why there should be less accuracy in the ballast trial.

I was pleased to see that Professor Aertssen had the torsionmeter calibrated on the shaft on this occasion, and this prompted me to look up the figures for accuracy of torque measurement given in his paper of two years ago. In the present paper it is stated, for instance, that the torque (the shaft being calibrated in the shop), was measured to within 2 per cent in smooth water. Am I right in assuming that is $\pm$ 1 per cent, or is it in fact $\pm$ 2 per cent?

Professor Aertssen: It is $\pm$ 2 per cent.

Mr. Canham: The point here is that in the *Tervaerte* paper Professor Aertssen quoted 3 per cent for an uncalibrated shaft. At the B.S.R.A. we think that an allowance of $\pm$ 3 per cent must be made to take account of a possible error in the assumed value of the modulus of rigidity of the shaft. Therefore I think the figures given in his last paper were a little optimistic, but in the present case he should get the torque measurements within $\pm$2 per cent.

B.S.R.A. have carried out tests on the Michell thrustmeter, as the result of which it was concluded that an accuracy within $\pm$ 2 per cent can be achieved. Professor Aertssen gives 4 per cent in smooth water and 5 per cent in rough water. I do not question the rough water figure, but I would like to know why he considers that in smooth water the errors are within $\pm$ 4 per cent.

I was a little surprised to read that the main engine mechanical efficiency was measured to within 6 per cent in smooth water. I was once assailed by a distinguished marine engineer on the question of torsionmeter accuracy, his point being that there were often indications that torsionmeter readings were wrong. I think he based his opinion on bhp and ihp readings taken in the shop. I wonder whether he achieved the same degree of accuracy for mechanical efficiency as is quoted in this paper.

In connection with the measurement of hull roughness, I would take the liberty of expressing a point of view. It seems to me that what is important in this matter is to measure the roughness of the hull at the time of the measured mile trial. In almost every case ships are coated with an anti-fouling paint before the trials, and the period between the painting and the trials varies. A notable exception was the *Lucy Ashton*, which had no anti-fouling paint. There is a marked difference between an anti-fouling and an anti-corrosive paint; an anti-fouling paint is essentially one which dissolves or leaches into water, and I know from my own experience that these paints do get very soft indeed. Therefore, it seems to me that there is some action between the paint and the water as soon as the ship is undocked, and that our hope of knowing what the surface is actually like at the time of trial is rather remote, even if we cut down the interval between the undocking and the trials to a very short period of time.

The point I am getting at is that there have been attempts in the past to explore the nature of the roughness of paint surfaces; we did it in connection with the work on the *Lucy Ashton*, and Professor Aertssen has done the same thing here with test panels. I would question whether the results obtained from these panels really give us much indication of the condition on the ship hull during the measured mile trials. From my own point of view I think it would be better to consider the grosser forms of roughness that are usually encountered on the painted hull.

A feature of the *Lubumbashi* is the remarkably smooth hull. I do not think that B.S.R.A. has yet taken roughness records on a sandblasted hull, but our records show that generally speaking, new hulls are very much rougher than that of the *Lubumbashi*. Our experience has prompted us at the B.S.R.A. to develop rather a different kind of instrumentation from that used by Professor Aertssen. We have developed a mechanical gauge which gives a record of the profile of the shell surface. It has the advantage that we can take a very great number of records quickly, although I am not suggesting that we get an absolutely accurate picture of the surface. A disadvantage of the pneumatic feeler is that it produces a reading, but you have no record at any time of what the surface looks like. The accuracy I interpret here to be $\pm$ 20 per cent.

So far as the investigation on the *Lubumbashi* is concerned, I feel that we cannot place very much reliance on the pneumatic feeler gauge readings, except to note that they do register an expected increase of roughness over six months. I think it would be much better to rely on the figures interpreted through the boundary layer traverses, and from which are deduced the equivalent sand roughness of the hull. It is interesting to note the marked increase that took place over six months and to dwell on the fact that that increase accounts for about 9 per cent in the total power after six months in service. I cannot agree with Professor Aertssen that the figures for the sand roughness correlate very well with the measurements of the roughness of the hull as established by the pneumatic method. I agree that account must be taken of rust blisters and other surface defects and I think that a mechanical gauge is more suitable for that purpose.

I look upon the method of estimating the shaft loss as simply proving that this loss lies within limits of $\pm$ 5 per cent.

With a thrust measurement limit of error of $\pm$ 4 per cent in smooth water, as given early in the paper, I was rather surprised to find that the thrust correction figures are given in Table VI to three decimal places in one case and to two decimal places in other cases. I had always been taught never to quote more figures than are necessary.

Turning to the weather data in Table XI, I notice that on January 10th, for observations Nos. 4, 5, 6, and 7 the sea is described as moderate, with waves of 2 ft. in height, 50 ft. length, and 160° starboard in direction. On January 16th and 17th the sea is described as a smooth ripple, again with waves of 2 ft. in height, 50 ft. in length, and 160° starboard in direction. It seems to me odd that the same waves should warrant quite different descriptions. If I were told that a trial took place in moderate weather conditions, it would suggest that the data would not be of very great accuracy. But it there was a smooth ripple, I should expect the weather effect to be very small and the results to be correspondingly more reliable.

Referring to the tank test report, I was rather surprised that the comparison of the ship *rpm* and model *rpm* at the same power absorption differed between the loaded and the ballast conditions. In the loaded trials, the comparison gives a factor, generally speaking of 0·99, whereas in the ballast trials the factor was greater than unity, except at top speed. I am rather puzzled as to why there should be that difference between the results for the two conditions. In a paper* given before this Institution last year Dr. Allan and I gave ship-model comparison results for thirty-seven ships. In almost every case the ship *rpm* was greater than the *rpm* predicted from the model, and only in five cases did we find the reverse; and I think Dr. Allan will agree that in two cases out of those five we shall probably revise our opinion shortly. This makes me wonder whether the *rpm* comparison for the *Lubumbashi* in the loaded condition is correct.

Professor E. V. Telfer, D.Sc., Ph.D. (*Vice-President, I.N.A., M.I.Mar.E.*): This paper is a valuable continuation of Professor Aertssen's previous work; and like this previous work will not only well repay detailed study but because of its generous supply of basic data will admit of such detailed study being easily and pleasantly undertaken.

It is admittedly somewhat unfortunate that in the general problem of ship-model correlation the engine has to intervene and complicate the correlation. Long experience has shown, however, that this simply means that two correlations become necessary; the engine must first be considered in relation to the known performance of sister engines. For this purpose it is probably most convenient to adopt, for diesel engines, the following graphical presentation of the basic data. Using a base of (mep) mean effective (or brake) mean pressure, the corresponding mean indicated pressure (mip), available from test-bed trials can be plotted as ordinates. From each mip ordinate the corresponding mep value can be set down; and the residual ordinate represents the combined running and static friction losses. A statistical linear plot through these residual ordinates, obtained from the data of many similar engines becomes very valuable in formation. It should be used in preference to (if necessary) the much fewer data usually available from a single engine trial; and if Professor Aertssen could produce such a line for the Burmeister & Wain–Cockerill engine it would add usefully to the value of his paper.

With these data available the bolder step of ship-model correlation is facilitated. The first step here is again the engine performance; and as the engine is now propeller controlled it is now preferable to present the mip values, and the shp values converted to mep to a base of revolutions squared. If the mip values are first modified by the statistically appropriate friction residuals, the resulting meps should be slightly in excess of those derived from the torsion meter readings and should serve as an excellent control on these latter, particularly when it is remembered that they avoid all question of shaft modulus, temperature effect on modulus, and torsion meter zero error. The control on this latter is greatly increased by carrying the revolutions down to their lowest reliable values. It is not necessary that this be done at any convenient time but preferably in the mile direction which corresponds to the with-weather runs. If runs are made for example at 35 *rpm* when full power *rpm* are say 110 this gives a nine to one extrapolation when the revolutions squared plotting is adopted. It is evident from Dr. Allan's and Mr. Canham's remarks that they have not had much opportunity to explore this end of the scale, but if they really hope to improve the general accuracy of their correlation work I urge them to insist that such opportunity be accorded them.

I hope on another occasion to show how the *Lubumbashi* data analysed by generalized power diagram methods compare with Table VA and VIA of Dr. Allan's appendix to the author's paper. It is evident that Dr. Allan has not used such methods, nor does his determination of tide appear to allow for the distinctly different wind resistance values on the opposite runs on the mile. This is evident from his final N/V values in Table VA failing to correlate with the dhp/N^3 . 10^3 values. This criticism is not a quibble. If Dr. Allan wishes to get more accurate ship knowledge of smooth and structurally rough surface frictional resistance he must analyse his ship data better; and in any case he should show how the data are analysed, since in this particular case, thanks undoubtedly to Professor Aertssen, all the data are published.

The final point I wish to make is in connection with the particular presentation of speed loss used by Professor Aertssen. In this he follows a suggestion recently made by Professor Bonebakker and plots percentage speed loss to a base of Beaufort number. From Professor Bonebakker's last N.E.C. paper where he made this suggestion one gets the impression that the Beaufort number is considered to be something physical, thus lacking the disadvantages of some merely descriptive systems more recently introduced. Without attempting to defend, for example, my own descriptive weather intensity, no better condemnation of the use of the Beaufort number could be forthcoming than contained in Figs. 16 and 17 of the present paper. The rapidly increasing loss towards the higher Beauforts shows that this number certainly does not linearize the loss. Clearly, a far more fundamental presentation, seeing that relative wind speeds and directions are given by the author, would be simply to plot percentage loss against relative wind velocity squared. This simple device applied to Professor Aertssen's data undoubtedly does linearize them; and a development of this basic idea would appear to have some interesting possibilities. For example, Professor Aertssen tells us that the *Lubumbashi* can evidently derive some assistance from following weather. This appears to be so and is a very useful observation since not all vessels by any means can derive such assistance. What are the features of a vessel's design which make such assistance possible? If Professor Aertssen persists in the good work he is doing and returns from time to time to our joint Institutions to report his findings, I feel certain that this question will not much longer remain unanswered.

Mr. J. Foster Petree (*M.I.N.A.*): The torsionmeter is stated to have been calibrated on the shaft, but I do not see in the paper the diameter of the shaft, and there is no description of how the torsionmeter is calibrated. I have had some experience of calibrating torsionmeters on shafts, though only of optical types—the Bevis-Gibson and the Hopkinson-Thring—and only with shafts of less than 12 in. in diameter. That was sufficiently difficult, and as the shaft in this case is obviously larger, I should like to know how it was done; because, unless there has been a great advance in the technique, it seems possible that there might be a 1 or 2 per cent zero error in the calibration itself.

Mr. H. Lackenby, M.Sc. (*M.I.N.A.*).) A previous speaker referred to the effect on resistance of a coating of scum on the hull surface as distinct from barnacles and grasses. In this connection I should like to mention that some information on this was forthcoming from the tests carried out on the *Lucy Ashton* and is given in Dr. Smith's paper.* Briefly, this was as follows: after the ship had been laid up in the Gareloch for a period of forty days from December 21, 1950, some runs were made and it was found that the total resistance had increased on the average about 3½ per cent

* ALLAN and CANHAM: "Ship Trial Performance and the Model Prediction," TRANS. I.N.A., 1954, p. 287.

* "B.S.R.A. Resistance Experiments on the *Lucy Ashton*," Part IV, see p. 525.

compared with that measured immediately before she was laid up, the state of the hull then being: clean with sharp seams and aluminium paint. Subsequent docking after the forty day period at the buoy showed that there was no visible evidence of fouling apart from the fact that the hull surface was covered with a light scum which was slimy to the touch. There was also some exfoliation of the top coat of aluminium paint, so it would appear that the effect of the scum was to increase the resistance by not more than the amount stated viz. $3\frac{1}{2}$ per cent, which corresponded to about 5 per cent on the skin friction.

I was very interested to read the account of the very careful and comprehensive records taken of hull roughness and the analysis of these made on the lines adopted in the *Lucy Ashton* tests. I was specially interested, too, to see that the roughness gauge used had a "pneumatic feeler." On the face of it this would appear to be a very desirable development, because a mechanical probe might be expected to indent the paint to some extent depending, of course, on the condition of the paint and the pressure applied to the probe. In spite of this I note that reference is made to the "pneumatic feeler" having a tendency to penetrate into the paint coating. Perhaps the author would comment on this.

Another aspect of the paper in which I was particularly interested was the measurement of velocity distribution in the boundary layer. I should like to ask Professor Aertssen whether the method he uses to calculate the skin frictional resistance from the velocity distribution is equivalent to calculating the loss of momentum in the usual way from the "velocity defect" over the width of the boundary layer. I should also like to ask Professor Aertssen what evidence there is for the influence of potential flow to which he refers. Further, as it is stated that the log was calibrated on the measured mile trials, why it is that the maximum of "free-stream" speeds indicated by the curves in Fig. 10 are both slightly greater than the speeds marked on the curves? If the log speeds were calibrated from speeds taken in the usual manner on measured mile trials, the velocity distributions in the boundary layer are presumably not absolute measurements of water speed, but are reduced to give the maximum speed at the outer extremity of the boundary layer equal to the "free-stream" speed well clear of the ship (i.e. considering the water moving relative to the ship). In the circumstances it might be expected that the velocities given in Fig. 10 would not be influenced to any great extent by potential flow. Perhaps the author would comment on this.

I would suggest that it might be of interest to plot both curves in Fig. 10 with the speeds expressed as fractions of the maximum speed. This would reduce both curves to the same maximum speed ordinate of unity and one would see at a glance the change in contour due to the change in frictional resistance for the two ship conditions. It would also be of interest to see the mean frictional resistance coefficients calculated from the velocity distributions plotted at their respective Reynold numbers against the background of the Nikuradse scale of sand roughness as in Fig. 4 of the *Lucy Ashton* paper—Part IV. These ship frictional resistance coefficients would of course refer to an elemental strip of hull surface extending from the stem to the log position. It is presumed that this in fact is how the equivalent sand roughness values (l/k_s) quoted in the paper have been determined and I should be glad if the author would confirm this as I did not find the derivation of these values very clear in Part IV B of the paper.

Mr. John Brown, B.Sc. (*Member of Council, I.N.A.*): Would Professor Aertssen agree that in the analysis of the effect of weather on performance, the size of the ship would also be a factor? A previous speaker has referred to a possible allowance of 30 per cent increase of power to counteract the average weather effect on the North Atlantic. I suggest that if there is a size effect, on the bigger ships the loss would be less than that; even in the case of ships of moderate area I believe the owners accept a lower percentage than 30 as the allowance for weather. Dr. Allan is inclined to think that we underestimate its value. Can he indicate the probable effect of size?

Following what Mr. Canham has said about tolerances in measurements, I have recently examined the records made on one of our own ships and I admit to some surprise at the variation in thrust measurements, even in a measured mile run. May I ask the author if his figure of 5 per cent is "plus or minus"?

Commander L. A. Rupp, B.S., M.S., U.S.N. (*M.I.N.A.*): In connection with corrections to thrustmeter readings, I also question the author's statement on the first page of the paper as to the relative accuracy of one-way and two-way thrustmeters. Regardless of which type is used, the weight component of the rotating masses in the axis of the shafting must be calculated and proper corrections applied to the thrust reading. This correction cannot be determined accurately by measuring the astern thrust when the shafting system is at rest, since the static friction of the shafting in the bearings and stuffing boxes would introduce a considerable error.

I would also like to ask the author whether the thrust correction for static head of the water acting on the shaft cross-section in way of the stern tube was considered. If this correction were not applied, some of the inconsistencies of the ship-model correlation data in Table VII might be explained. In Table VIIA the ehp derived from dhp over ehp Froude varies from 0·917 at 12 knots to 0·953 at 16·5 knots, while the ehp derived from thrust over ehp Froude varies from 1·186 to 1·098 for the same speeds. The thrust correction due to static pressure on the shaft is nearly constant and is subtractive from the measured thrust. At the lower speed it may amount to perhaps 6 to 8 per cent of the measured thrust, and at the higher speed condition probably the correction is of the order of 2 per cent. Consequently, if this correction has not been applied in analysing the trial data, consideration of it would make the ship-model correlation based upon thrust data more nearly constant over the speed range and also more consistent with that derived from horsepower measurements.

Another item which several speakers have mentioned is the relationship between the increase in power required and the Beaufort scale of the wind. We know that wind velocity produces various effects on ship resistance. Wind acting on the part of the ship above the water effects the resistance depending on the area and above water shape of the ship and the square of the wind velocity. Wind also affects the condition of the sea. The ship resistance increase due to surface waves does not follow any simple law. This increment is very directly affected by the relation between the length of the ship and the length of wave it encounters. The resistance becomes very much greater as the length of the ship approximates the length of the waves. Consequently, I question the utility and accuracy of attempting to relate power increase and Beaufort wind scale *per se*.

The Chairman (Sir Stanley Goodall) (*Hon. Vice-President, I.N.A.*): We are extremely grateful to Professor Aertssen for this paper, which again gives us a mass of information to digest. In asking you to show your appreciation of the work he has done, I should also like to say that we hope we shall have "Volume III" at some time or other; and I would ask him to convey our thanks to the Centre Belge des Recherches Navales and the Institut pour l'Encouragement de la Recherche scientifique dans l'Industrie et l'Agriculture.

Written Contributions to the Discussion

Professor Dr.Ir. W. P. A. van Lammeren (*M.I.N.A.*): Since the Netherlands Ship Model Basin (N.S.M.B.) carried out the model tests in the stage of design of the M.V. *Lubumbashi* the author invited our establishment to compare the results of these tests with those obtained on the trials as described in the paper. It is a pleasure to have the author's agreement to add the results of this comparison to those already given by Dr. Allan in Appendix I.

Model and propeller were made to scale $22\frac{1}{2}$. Resistance and self-propulsion tests with the model and propeller in question were carried out on three different draughts, viz.:

(*a*) Ballast; draught = 17·2 ft.; trim = 6·67 ft. by stern.
(*b*) Loaded; draught = 26 ft.
(*c*) Loaded; draught = 27 ft.

The tests were carried out according to the Continental Method, using a trip-wire (ϕ = 1 mm. = 0·04 in.) and applying the Froude skin friction values.

The results for the corresponding trial conditions, A (draught = 25·833 ft.; trim = 0·25 ft. by stern) and B draught = 17·083 ft.; trim = 6·67 ft. by stern) have been derived from the results of the above tests by cross fairing over the whole speed range, for each half knot. They are given in the tables below, together with the trial results of the actual ship and the allowances derived therefrom.

From the tables it appears that the allowance on dhp for trial condition B are some per cents higher than those for condition A. This is due to the fact that the general condition as to sea and wind in case A was somewhat more favourable than in case B. For the rest, the allowances on dhp are considered to be satisfactory. They agree quite well with allowances of a number of ships of the same type. As to the number of revolutions of the propeller, for both conditions it appears to be somewhat too low or, in other words the pitch of the propeller appears to be too high. The allowances on thrust do not give a clear picture. They are in general, however, smaller than those on bhp which is in agreement with our experience.

Comparing the above results with those given by Dr. Allan in Table IV of Appendix I it appears that there is a fair agreement between the dhp values for the ballast condition. The values for dhp for the loaded condition, however, are about 4 per cent higher than our values. For both conditions the number of revolutions estimated for the ship by N.P.L. is about 2 per cent higher. I should like to ask the author whether a correction for wake scale effect is included in the figures given in Table IV. If not, the discrepancies with the values estimated according to the Continental Method will still be greater.

In Appendix I Dr. Allan mentions the length of the measured course at Polperro mile to be 6,080 feet. The Admiralty charts in our possession mention 6,079 feet. Although the effect on speed calculations is negligible it would be interesting to know which figure is right.

Finally I should like to express my admiration for the careful way in which the authors have conducted the full scale trials and model tests. I am sure that this is the only way to bring the correlation problem nearer to its solution.

Mr. R. E. Clements, B.Sc. (*A.M.I.N.A.*): Having carried out a somewhat similar series of trials for the B.S.R.A. on a passenger-cargo vessel in the North Atlantic, I find this

TABLE XVIII

RESULTS OF MODEL TESTS CARRIED OUT BY N.S.M.B.
(trip-wire; standard temp. 59° F.)

A. Draught = 25·833 ft.
Trim = 0.25 ft. by stern.

B. Draught = 17·083 ft.
Trim = 6·67 ft. by stern.

V	A: Metric Values		A: *rpm*	B: Metric Values		B: *rpm*
	dhp	Thrust (tons)		dhp	Thrust (tons)	
12	2058	23·92	80·0	—	—	—
$12\frac{1}{2}$	2338	26·20	83·7	—	—	—
13	2658	28·84	87·4	2042	23·03	80·7
$13\frac{1}{2}$	3000	31·40	91·0	2322	25·20	84·3
14	3365	33·96	94·7	2620	27·37	87·7
$14\frac{1}{2}$	3750	36·46	98·2	2938	29·62	91·2
15	4200	39·30	101·7	3300	32·22	95·0
$15\frac{1}{2}$	4755	43·03	105·8	3700	34·81	98·7
16	5497	47·98	110·4	4170	37·98	102·5
$16\frac{1}{2}$	6360	53·60	115·3	4683	41·20	106·3
17	7270	58·90	120·4	5270	44·86	110·7
$17\frac{1}{2}$	—	—	—	5955	48·67	114·9
18	—	—	—	6772	53·60	119·6

TABLE XIX

ALLOWANCES ON POWER AND RPM FOR TRIAL CONDITIONS A AND B AT FULL POWER (6200 BHP-METRIC)

Condition	TRIAL			ALLOWANCES			*rpm** predicted
	V ship	Thrust (tons)	*rpm*	dhp	bhp	Thrust	
				%	%	%	
A	16·22	52·90	113·7	2·8	5·8	5·6	115·6
B	17·33	48·50	116·2	5·2	8·2	2·2	117·4

* The figures include an allowance of 2·2 per cent for wake scale effect.

second paper by Professor Aertssen of considerable interest. Appreciating the enormous amount of analysis work involved, Professor Aertssen is to be congratulated upon the concise way in which the data are presented.

Turning first to the collection of the various data, I notice that most observations were made visually. This method of observation has several failings; firstly, no permanent record is obtained, secondly, it is difficult under heavy weather conditions to assess mean values accurately and thirdly, if all records are to be obtained simultaneously a large number of observers is necessary. I should be grateful therefore if the author would give us some idea of the synchronization of the various records, particularly of power, thrust, and *rpm*, and the corresponding weather data. Had autographic records been taken of the speed I think the author would revise his ideas regarding the accuracy of this measurement in rough water. It was our experience that on a vessel only a little shorter than *Lubumbashi* the fluctuation of speed when running into heavy head seas was as much as $\pm$ 12 per cent which, even assuming the recorded speed to be accurate, did not allow the mean to be assessed to within $\pm$ 2 per cent. The autographic recording of data also enables a reduction to be made in the time taken for each observation thus reducing the possibility of changes in power or of sea conditions.

Regarding the estimation of sea conditions, this was probably the most difficult observation to make. Nevertheless, I was a little surprised that the seas were sufficiently regular for wave height and length to be quoted in every observation.

The section dealing with boundary layer traverses is very interesting. The fact that the increase in C_f for the six months in service was of the right order indicates that much useful information might possibly be obtained from this approach. The information becomes even more valuable if records can be obtained of the corresponding hull roughness by taking measurements in dry dock either immediately before or immediately after the boundary layer traverses are made. It would be unwise, however, to draw conclusions from a logarithmic plotting of y/δ and u/U extending over the full width of the boundary layer. Recent developments indicate that the velocity distribution curve of a turbulent boundary layer can be divided into three parts, each of which can be represented by different functions. As these are the first data available for a vessel having a surface which has been previously sand-blasted and also as Fig. 10 is too small for analysis purposes, I should be grateful if the author could supply full details of the spots given in this figure.

Data of the kind presented in this paper are of course invaluable for establishing and comparing various methods of analysing the service performance data normally supplied in the form of ships' logs. A certain amount of work has been done on this subject by the B.S.R.A. where, following Professor Bonebakker's first paper on the subject,* methods have been developed using multiple regression analysis to determine the separate effects of weather and of fouling on performance.

Applying these methods to the *Lubumbashi* data, the trial still-air dhp of 15 knots estimated from the analysis was 1½ per cent higher than the actual still-air dhp, while the estimated *rpm* agreed exactly with the trial value. The average effect of weather for this route was found to be 17½ per cent. The effect of fouling was to increase the dhp by 6½ per cent for the voyage Teneriffe–Antwerp and by 8½ per cent for the voyage Las Palmas–Hamburg, figures which are in good agreement with the author's.

These are the early results of our researches. The author can be assured that every effort will be made to extract the maximum of information from the valuable data he has placed at our disposal.

* BONEBAKKER, J. W.: "The Application of Statistical Methods to the Analysis of Service Performance Data." *Trans N.E.C.I.E.S.*, Vol. 67, 1950–1, p. 277.

Professor J. W. Bonebakker (*M.I.N.A.*), and **Mr. J. Gerritsma:** Fig. 16 (relation between increase of power, and wind force and direction) is based on a restricted number of observations (105), but apparently the curves are easily faired through the spots. No doubt this is due to the painstaking care with which the observations were taken. The restricted number of observations accounts for the difficulty, stated by the author, in assessing observations in the transition zone between successive sectors to their proper group. With, say, a thousand observations, this difficulty would be greatly diminished.

The statement that wave direction is of more importance than wind direction is borne out by the writers' experience. Out of a large number of observations, 75–85 per cent will show that wave direction and wind direction come in the same sector.

The mean value of the increase of power due to weather conditions on the voyage Antwerp–Teneriffe—2·9 per cent—is particularly interesting. From Table XI it would seem that this is due to the prevailing following sea, overriding the influence of the opposing wind. This is a confirmation of the statement mentioned above.

It would have been interesting if the regression equations for the three voyages had been computed; the position of the corresponding regression lines would have shown at a glance the influence of fouling.

The method, followed by Professor Aertssen in computing his Figs 16 and 17, is also advocated by Lewis and Morrison of Stevens Institute, Hoboken, N.J., in *International Shipbuilding Progress*, Vol. 2, No. 7 (1955).

Author's Reply

To *Dr. Allan.* I feel that the accuracies referred to in Part I are more on the pessimistic than on the optimistic side, as stated by Mr. Canham in the discussion. The accuracy of the pitot log being 1 per cent in smooth and 2 per cent in rough water can be established from Tables XV, XVI and XVII. For observations 66 to 69, extending over 3 hours, made in rough weather Beaufort 6, with pitching angle 4 deg., *rpm* 105·3 to 105·8, the speed varied from 13·7 to 13·8; for observations 34 to 36, extending over 2½ hours, made in rough weather Beaufort 6, with pitching angle 5·5 deg., *rpm* 102 to 102·5, the speed varied from 12·6 to 12·9 knots. In a high sea, however, for observation number 90, the ship surging heavily, speed variations were observed from 9·0 to 9·5; this does not mean, however, that speed cannot be estimated with an accuracy better than 6 per cent, as it is possible to have many readings taken over a small lapse of time.

Regarding the increase in resistance after six months' service, there is, I think, a satisfactory correlation between the increase of power due to fouling, 9 per cent, and the increase of $\Delta C_f = 0{\cdot}0003$ as measured by boundary traverses, and the increase of roughness.

Both measured-mile trials were conducted on low revolutions, 70 for the ballast condition, 80 for the loaded condition, when full power *rpm* are 120, not only to have a good correlation, but also to allow the calculation of shaft losses. With a revolutions squared plotting this gives a three to one extrapolation. It is not evident that data at very low revolutions would have given better results: they are surely nearer to zero, but one should bear in mind that, especially with a diesel engine, it is very difficult to run at constant speed on low revolutions. That is why no attempt was made for having runs carried out at revolutions lower than 70.

I agree with Dr. Allan that it is difficult indeed to establish the effect of structural roughness of a painted hull. He states that this is an important factor in ship's resistance. That is why it would be so very useful if for many new-built ships boundary layer traverses could be taken immediately after roughness measurements were made. The traverses give the total roughness, measurements allow the establishment of paint roughness and by difference the structural roughness can be calculated.

To *Mr. Davis.* Cylinder liner wear on heavy oil is indeed 25 per cent more than on diesel oil. This is the experience on the ships of the Compagnie Maritime Belge equipped with this type of engine, Burmeister & Wain–Cockerill. It should be remarked that no additive was being employed in the cylinder lubricant in this instance. I agree entirely that it is the general experience that cylinder liner wear with average heavy fuel and normal cylinder lubricating oil is double that with diesel oil.

An increase of 2 per cent for each day out of dry dock is a very high figure indeed, applying perhaps only to ships lying in tropical harbours. The increase of ΔC_f for six months' service was established at 0·0003, relating to an increase of frictional resistance of 18 per cent, or 0·1 per cent per day out of dock. This is a very low figure as compared with the figure of 2 per cent, but, as Mr. Davis said, there are many implications in this increase of skin resistance due to fouling.

To *Professor Robb.* I am glad the question arises of the opportunity of taking ships out of dry dock before hardening of the resh bottom paint. *Lubumbashi* had a hull suitably dry when she was undocked on the evening of the day the roughness measurements were taken. The shell plating was remarkably smooth and had the appearance of a mirror. However, it is believed in many quarters that exfoliating of the antifouling coat that prevents fixing of the barnacles proceeds better when the paint is not hardened. This argument is inconsistent with the behaviour of *Lubumbashi*: the shell plating had a hardened paint coat; nevertheless, she was not dirty after six months' service, being covered only by rust blisters. Moreover, at sea, a non-hardened paint coat would probably be rougher than a dry coat. The appearance, however, of numerous rust blisters depreciating the shell plating remains an open question.

Regarding the efficiency of the propeller in the ship-model correlation, it can be said that, when account is taken of thrust correction due to the hydrostatic head on the end of the shaft, which amounts to 2 per cent at 16 knots in loaded condition, there is, for the loaded condition, no appreciable scale effect in the quasi propulsive coefficient of this ship.

To *Mr. Canham.* With a one-way thrustmeter a correction can be made indeed to take account of the weight component of the shaft, but there remains a certain doubt as to whether the weight of some attached parts of the shaft should be added, entirely or partly, to the weight of the shaft.

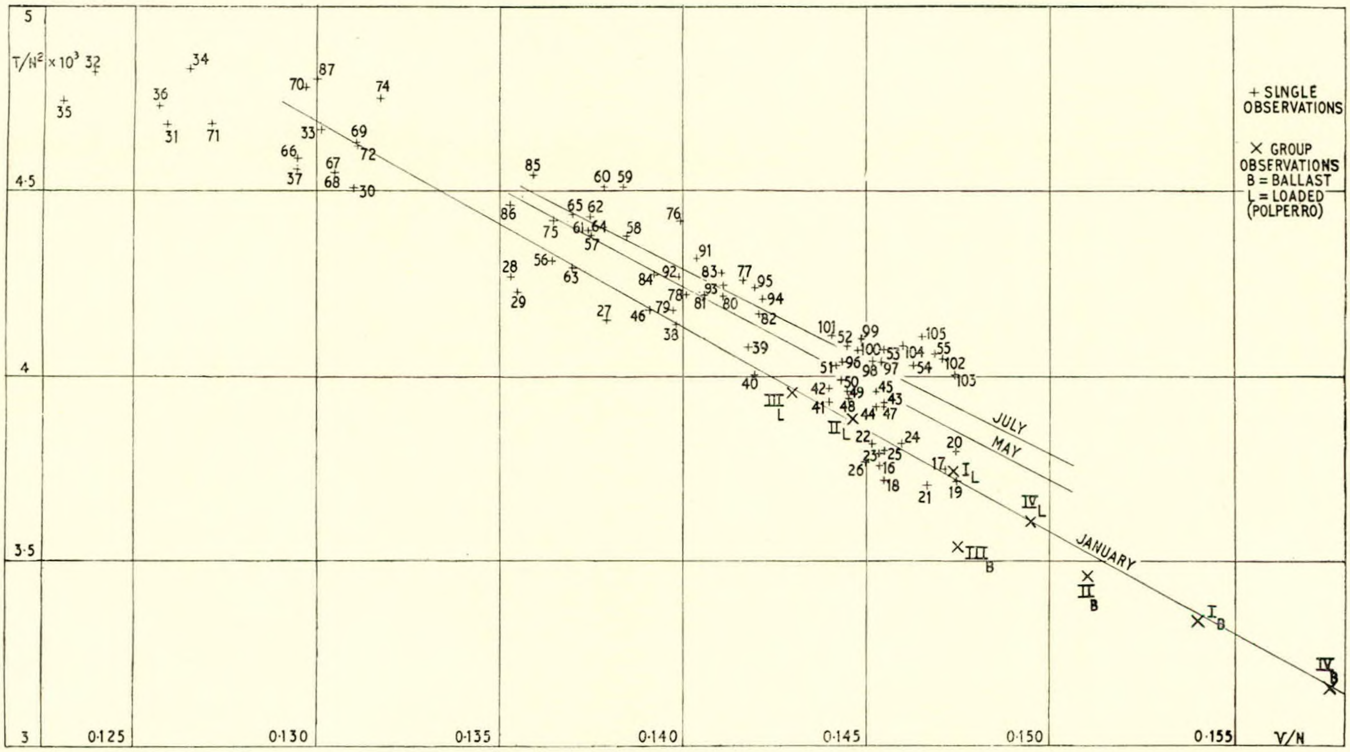


FIG. 20

That is the reason why there is less accuracy on the trial data in ballast condition than on the trial data in loaded condition. With a two-way meter it is possible to establish by experiment the correction weight component plus hydrostatic head on the shaft end.

The accuracy of 2 and 4 per cent quoted in the paper for torque and thrust measurements is established from the scattering in a diagram dhp/N^3 . 10^3 and T/N^2 . 10^3 plotted on V/N. This last relation is given in Fig. 20. The values of thrust here are corrected for the hydrostatic head (−0·6 ton in ballast, −1 ton and −0·9 ton in loaded condition). Obviously the scattering of thrust readings is no worse than the scattering of torque readings and on the face of it, it would look fairly plausible to conclude, as B.S.R.A. did, that an accuracy of 2 per cent was achieved. However, though the manometer was calibrated in the University of Ghent, there is certainly a zero error on this one-way thrustmeter: there is the friction on the pistons, a possible error on the diameter of the pistons, the unknown pressure at the astern part of the thrust collar, which though very low, might have an effect. The error, brought on by these factors, may be important, 1 or 2 per cent, and is to be added to the error shown by the scattering in the diagram. Having revised my opinion, this error is now quoted 1 per cent instead of 2 per cent, which brings the total limit of error to 3 per cent instead of the 4 per cent of the paper. The remark of Mr. Canham on this item is therefore fully appreciated.

Considering Fig. 20 again, it is possible now to analyse the deviations of *rpm*. Mr. Canham wonders whether the *rpm* comparison for *Lubumbashi* in loaded condition is correct. This remark urges on checking *rpm* with the diagram. It is of interest to note that all the data of the ballast trial and the first voyage, when the hull was clean, are remarkably in line. The 2×4 groups of runs on the measured mile, marked B for the trial in ballast condition, L for the trial in loaded condition, must give very accurate values of $T/N^2 . 10^3$ on V/N, and these groups determine the line for the first voyage as well. The fact that group III of the ballast trial, and in some way group II, too, are out of line, might be an indication of cavitation. The value $T/N^2 . 10^3$ is most feeling to an error of N and evidently there is a close agreement between the *rpm* of the ballast and of the loaded condition.

The mechanical efficiency is the ratio shp/ihp. The error on this ratio is the sum of the errors on shp and ihp taken separately. The error on ihp is 4 per cent indeed with indicator cards taken on board and calculated at home. Table XX, in my reply to Professor Telfer, gives the difference mip–mep, representing the combined running and static friction losses. The deviations from the mean values, 31·6 lb. per sq. in, for the ballast trial, 33·5 lb. per sq. in. for the loaded trial, are characteristic for the accuracy in appreciating the mechanical efficiency.

It is certainly difficult to know what the surface of the hull is like at the time of the measured mile trial, even if the interval between undocking and trial is no more than a few days. If account is taken, however, of the frictional resistance being a large part of the total resistance and of the effect of roughness on frictional resistance, it is certainly worth while measuring this roughness in dry dock. It is useful to have the record of the profile of the surface taken by means of the mechanical gauge developed by the B.S.R.A. But how to interpret this record in hydraulic terms? There is indeed a doubt whether the results obtained with the exploration of test panels gave a good indication of the surface of the ship, and that is why the surface of the shell was explored by means of the pneumatic feeler. I estimated the accuracy of this measurement not better than some 20 per cent. Mr. Canham's feeling is that the accuracy is even worse than that. I give here the opinion of Professor Schlag, Director of the General Hydraulic Laboratory of the University of Liége, where, as I mentioned in the paper, this pneumatic feeler is currently used for the roughness measurement of pipes:

"The instrument proved successful for pipe roughness measurement, and I cannot agree with Professor Aertssen's statement of the accuracy not being better than 20 per cent. When the Solex equipment suits the roughness to be measured and when the explored surface has a well-defined roughness—for instance, a calibration marble—the experimental data show a scattering from a mean curve of not more than 2 per cent. I feel that, if indeed on *Lubumbashi* larger deviations were recorded, they existed actually on the hull's surface, which certainly has an irregular roughness, and that the deviations are not an effect of the lack of accuracy of the feeler. The instrument is portable and easy to be manipulated. Provided a great number of measurements are taken, it yields a reliable mean roughness for the hull. The mechanical gauge of the B.S.R.A. gives a profile of the surface, but I feel that actually the equivalent roughness number of Nikuradse, although it gives the roughness by merely one length number, is the best way to describe the roughness. Mr. Canham prefers to rely on the figures obtained from the boundary layer traverses, from which are deduced the equivalent sand roughnesses. I cannot wholly agree, since the velocity distribution is merely an effect of the roughness and the relation between physical roughness and its effect on ship's hydrodynamics remains worth investigating."

Although I cannot entirely endorse the arguing of my colleague of Liége, I am glad to see that my assertion of an accuracy of not better than 20 per cent turns out to be half-way between the opinions of Prof. Schlag and Mr. Canham. It is my conviction that the pneumatic feeler readings give more than an increase of roughness from one moment to the other. The correlation between the feeler readings and the figures obtained from the boundary layer traverses is satisfactory, if account is taken of the numerous blisters on the hull after six months' service. It should be emphasized that the feeler readings are the roughness of the surface without blisters, which probably give an increase from the 3,620 microin. measured by the feeler to the 6,100 microin. established from the traverses.

I agree that it remains difficult to account for the roughness given by the blisters, but I am not convinced that the mechanical gauge will give a figure which can be interpreted in hydraulic terms.

Regarding the establishing of the shaft losses, I feel that the drawing, which was made on a scale 1 ft. $\times$ ½ ft., shows clearly that the upper limit of the losses is lower than 5 per cent and these these losses are very close to 3 per cent. The thrust data are given in Tables XIV to XVII with one decimal. These data are the basis of all my calculations.

There is no real inconsistency between the weather data of January 10th and January 11th. It should be mentioned that on January 10th the ship was in a rather sheltered place, on the measured mile, a wind 3–4 in the Beaufort scale building up waves no higher than 2 ft., while on January 11th on the Atlantic there was practically no wind and the sea was smooth, the waves of 2 ft. high making no more than ripples. The apparent inconsistency, mentioned by Mr. Canham, is very often found in the deck-logs.

To *Professor Telfer*. The statistical line mip–mep based on the data obtained from many engines and giving running and static friction losses for a known type of diesel engine, i.e. the Burmeister & Wain–Cockerill double acting two-stroke engine, would have been very useful indeed for the control of the zero of the torsionmeter. Unfortunately, this line could not be produced for a series of similar engines. It has been determined for the main engine of *Lubumbashi*: Table XX gives for wide varying loads at sea the difference mip–mep.

tp being the mean top pressure, bp the mean bottom pressure given by Table VIII, the mean indicated pressure for this type of engine is obtained from mip $= 1 \cdot 035$ tp $+ 0 \cdot 868$ bp.

A mean value of mip–mep is, for the ballast condition 31·6 lb. per sq. in., and for the loaded condition 33·5 lb. per sq. in.

The difference mip–mep affords a good control for the torsionmeter readings, so far as it gives an indication when the zero has shifted and must be established again. It can, however, not be a substitute for the calibration of the torsionmeter in the shop. I agree that the temperature effect on modulus may not be neglected and at regular intervals the temperature was taken in the tunnel. During the trials the torsionmeter readings were checked against mechanical efficiencies obtained previously and well established by taking means. Whenever this was possible, in port, the zero was checked. It should be mentioned, too, that the relationship of the torsionmeter is important and can show some deviations.

The shaft losses were established from runs at 70 *rpm*, full power *rpm* being 120. It would have been difficult with a diesel engine to obtain reliable results at lower *rpm*.

I fully appreciate the suggestion of Professor Telfer to plot

TABLE XX

Date	Hour		*rpm*	lb. per sq. in.				
				1·035 tp	0·868 bp	mip	mep	mip–mep
22–12–53	9	30	100·4	71·1	51·6	122·7	91·2	31·5
Ballast . .	9	50	99·8	71·7	50·2	121·9	91·4	30·5
	10	10	100·5	71·2	51·6	122·8	90·6	32·2
	10	30	115·3	94·1	66·0	160·1	124·3	35·8
	10	50	115·8	89·6	64·8	154·4	123·3	31·1
	11	0	115·4	93·0	66·2	159·2	127·6	31·6
	11	40	120·2	95·2	71·4	166·6	135·0	31·6
	11	50	119·7	98·6	68·5	167·1	135·0	32·1
	12	20	120·1	97·0	68·2	165·2	135·5	29·7
	12	40	119·7	96·5	68·4	164·9	135·4	29·5
1–8–54	5	40	98·4	73·8	52·0	125·8	93·8	32·0
Loaded . .	6	20	98·7	76·6	52·1	128·7	94·2	34·5
	7	0	98·6	75·7	51·7	127·4	94·0	33·4
	7	40	98·6	74·3	50·3	124·6	94·4	30·2
	8	30	110·3	93·2	64·6	157·8	122·3	35·5
	9	0	110·0	93·7	64·6	158·3	122·2	36·1
	9	40	110·1	92·0	65·4	157·4	122·4	35·0
	10	0	109·9	93·0	63·2	156·2	122·5	33·7
	10	50	114·7	95·8	69·5	165·3	134·1	31·2
	11	30	114·5	97·1	71·0	168·1	135·0	33·1

increase of power and loss of speed against relative wind velocity squared and am much interested in the development of this presentation.

To *Mr. Foster Petree*. The shaft had a diameter of 404 mm. (15·9 in.). With the torsionmeter fitted on it, this shaft was bolted at one end on a bracket, the other end being supported by means of steel rollers which were very close to the second end of the shaft. The calibrating beam is bolted on this second end. From normal full power and *rpm* the normal torque is calculated and, with an allowance of more than 10 per cent for overload, the maximum load which will be suspended at the end of the lever is established at 9 tons. The calibration is carried out successively with decreasing and with increasing loads.

To *Mr. Lackenby*. A mechanical roughness gauge might be expected to indent the paint to some extent indeed. The pneumatic feeler, because of the pressure applied by hand, also gives rise to deterioration of the surface when the paint is wet. The feeler then penetrates into the mellow coat and, as no air can get out, the reading then relates to a surface which is infinitely smooth; such reading, although erroneous, has an advantage: the error is evident and erroneous measurements are very easily eliminated. That dependence of the feeler on the surface being more or less mellow, exists only for a fresh painted hull. No readings must be eliminated when after six months' service the roughness of the hull was measured again by means of the pneumatic feeler.

I am glad that Mr. Lackenby has raised the question of the assessment of frictional resistance. C_f, deduced from boundary layer traverses, is calculated indeed from the loss of momentum and refers to an elemental strip of hull surface extending from stem to log. The velocities given by the curves, Fig. 10, are actual velocities observed on the log: they are the velocity of the ship relative to the water at any distance of the shell plating. The velocity beyond the boundary layer, as remarked by Mr. Lackenby, is in excess by 0·1 knot of the ship's speed, which is indicated on the curves. This 0·1 knot represents the interference of potential flow, which has been made clear by the calibration of the log on the measured mile. For a Pitot log which has been correctly adjusted, the ultimate calibration on the mile is no more than the assessment of the influence of potential flow. This influence is very low indeed, and the loss of momentum is calculated from the curves in the usual way for a flat surface. However, in order to get C_f, this loss of momentum is divided by square ship's speed and not by square potential speed. There is a certain error doing that, but I feel it is the best way to obtain C_f. On January 13th, C_f is 0·00178 with a Reynolds number $3 \cdot 4 \times 10^8$; on July 11th, C_f is 0·00208 with a Reynolds number $3 \cdot 3 \times 10^8$.

The equivalent sand roughness is established according to Scholz's paper in the German Jahrbuch. In his tables Scholz gives a correlation between relative roughness, ratio momentum thickness to physical thickness and exponent n. For a ratio momentum-thickness to physical thickness 0·076 on January 13th, the relative roughness is 7×10^5; the exponent n should be for this roughness 0·101. For a ratio momentum-thickness to physical thickness 0·084 on July 11th, the relative roughness is $2 \cdot 9 \times 10^5$; for this roughness n should be 0·108. Obviously roughness is better described by momentum thickness, which relates to C_f, than by exponent n. The logarithmic law characterized by this exponent n relates only to the medium zone of the boundary layer.

To *Mr. Brown*. I certainly agree that the size of the ship is an important factor in the analysis of the effect of weather. The allowance of 30 per cent for the Atlantic referred to in the discussion is, I agree, too high for bigger ships, which have a length much greater than normal Atlantic waves, 300 ft. long. But there is another factor to be considered than the ratio wavelength–shiplength. Figs. 16 and 17 show that the effect of waves becomes much greater when, for a given displacement, the engine power lessens: the ratio dhp/Δ is important. That is why big high-speed passenger-ships do not suffer much from bad weather.

The allowance depends also upon the route. For the Teneriffe route the *Lubumbashi* required, as a mean of three voyages, 15 per cent. A Victory ship, the *Tervaete*, required for the North Atlantic, as a mean of two voyages, 28 per cent; for the route Congo–U.S.A., with a single voyage, an allowance of 19 per cent was obtained. The limits of error given in Part I of the paper are possible errors in plus or minus.

To *Commander Rupp*. The thrust correction for the effect of the hydrostatic head acting on the shaft cross-section in way of the stern tube was not considered, neither for *Lubumbashi* nor for *Tervaete*. This correction theoretically should be applied for the ship-model correlation—provided the no-load correction was applied too in the model basin—but has been questioned sometimes. This correction is recommended by the Code on Instruments and Apparatus for Ship Trials, 1952, of S.N.A.M.E. Different authors, however, do not mention this correction in their thrust calculations. Bauer gives a detailed calculation of thrust on different ships and corrects only for the weight component.* Saunders writes an important paper on thrust evaluation and mentions in his zero correction only the weight component.† In the discussion of the same paper a detailed calculation of the thrust on a T.S. turbine steamer, *Empress of Australia*, is given by Hamilton Gibson, and again it is corrected only for the weight component.

It seems better, however, for propulsion analysis to apply the no-load correction to model tests—as tanks as a rule do—and to correct ship's thrust for the hydrostatic head as well as for the weight component, and I thank Commander Rupp for having focused attention on it.

The thrust correction for hydrostatic head is −0·6 ton for the ballast trial, −0·9 ton for the loaded trial, the first and the third voyage, and −1·0 ton for the second voyage. Taking into account this correction, the quasi-propulsive coefficient in loaded condition at 16·5 knots is now 0·74, against 0·76 at 14·5 knots, as to compare with the model results of 0·73 at 16·5 knots, against 0·75 at 14·5 knots. There is now a better ship-model correlation indeed for the loaded condition. In loaded condition the ehp derived from dhp over ehp Froude varies from 0·917 at 12 knots to 0·953 at 16·5 knots, while the ehp derived from thrust over ehp Froude varies from 1·004 to 0·967 for the same speeds.

Regarding the increase of power in a rough sea, it is quite certain that the relation shiplength to wavelength is of the utmost importance. In Figs. 13 to 18 the effect of weather is calculated only for the Atlantic where very often waves are observed of 300–400 ft. in length, 13 ft. in height for a wind 6 in the Beaufort scale, and where a known relation exists between wind force and wave dimensions. It is clear also from the diagrams that even in a very rough sea, 6 or 7 Beaufort, the effect of the waves on ships' motions and resistance is small when these have a reduced length as was experienced on May 2nd in the Channel and North Sea (observ. 51 to 55).

* G. Bauer: "Messungen und Untersuchungen an Schiffsschrauben," *Jahrb. S.T.G.*, 1923.

† H. E. Saunders: "Measurement of Propeller Thrust on Shipboard," *Trans. S.N.A.M.E.*, 1934.

To *Professor van Lammeren*. Although the data of the N.S.M.B. for the measured mile trial in loaded condition are derived from results of model tests carried out at draughts somewhat different from the draught of the trial, the comparison with the model tests of N.P.L. is very interesting. Both predictions are based on Froude with factor 1. The N.P.L. relation ehp derived from dhp to Froude ehp is 0·94, the N.S.M.B. allowance on power is +2·8 per cent. It should be remarked that the tanks refer to a power curve still air condition calculated from the *Polperro* trials. The difference between both curves is small, but here is a rise more of discrepancy between both tanks. Tables XIV to XVII give Pitot log speeds, and the reference curves calm air clean hull (Figs. 13, 14, 15) are established according to Taylor's *Speed and Power of Ships* with the speeds taken by Pitot log on the measured mile. According to this method, still air conditions are following for the loaded trial, thrust corrections being made for the static head of the shaft. They are compared with the results of N.P.L. and N.S.M.B.

TABLE XXI

Speed in knots	12	13	13·5	14	14·5	15	15·5	16	16·5
dhp trial	1,930	2,500	2,810	3,160	3,575	4,075	4,650	5,370	6,260
Thrust trial in tons	24·7	27·5	30·2	32·3	35·1	38·7	42·8	47·5	52·6
dhp N.P.L.	2,120		3,125			4,415			6,585
dhp N.S.M.B.	2,030	2,620	2,960	3,320	3,700	4,140	4,690	5,420	6,275

A correction for wake scale effect is not included in the figures given in Table IV. According to the Admiralty charts the length of the measured course at Polperro is 6,079 ft.

To *Mr. Clements*. It is indeed important to synchronize the various records, and it would have been useful to have the records taken by more observers. Every observation was taken visually and, as I mentioned in the paper, it took half an hour collecting all the data of one observation number. Even then, ship's officers had to be helpful by taking weather records. Unfortunately, better arrangements could not be made.

It is difficult, indeed, to draw conclusions from a logarithmic plotting of y/δ and u/U, and that is why in the paper the roughness is established from momentum thickness only, not from exponent n. Mr. Clements would divide the boundary layer in three parts. I agree and I feel that I give form to his idea if I say that only the medium zone of the friction belt is represented by the logarithmic law of Fig. 11. It is clear from this diagram that very near to the shell plating and in the transition zone to the potential flow there is another law.

I fully appreciate the multiple regression analysis applied to the voyage data in order to determine the separate effects of weather and of fouling on performance. No doubt more observations would have given even better fouling and weather factors.

To *Professor Bonebakker*. Computing the regression equations for the three voyages is indeed interesting. This has been done by Mr. Clements who has determined by this method the separate effects of weather and fouling. I agree, more voyages on different routes would have given more complete results regarding the performance of this vessel.

INSTITUTE ACTIVITIES

Sections

Kingston upon Hull and East Midlands

The first meeting of the Kingston upon Hull and East Midlands Section was held at the Royal Station Hotel, Kingston upon Hull, on Thursday, 27th October 1955, at 7.30 p.m. Two films, kindly lent by Ruston and Hornsby, Ltd., of Lincoln, were shown and a short account of the production of Diesel engines by that firm was given by their representative, Mr. G. K. Baguley.

An interesting discussion, opened by Mr. Bryan Taylor, B.Sc.(Eng.), was continued by Messrs. C. J. Potter, T. Sherburn, A. E. Walker and A. W. B. Edwards; visitors to the meeting also entered into the discussion.

A vote of thanks to Ruston and Hornsby, Ltd., and to Mr. Baguley, was proposed by Mr. G. Hill and seconded by Mr. C. J. Potter. Seventy-seven members and friends attended.

Scottish

Scottish Section members should note that all correspondence addresssed to the Honorary Secretary, Mr. J. D. B. Mundie, M.I.Mar.E., should be sent to:

Stow College of Engineering Annexe,
167, West Graham Street,
Glasgow, C.4.

Sydney

A meeting of the Sydney Section was held at Science House, Gloucester Street, Sydney, on Friday, 30th September 1955. Mr. W. G. C. Butcher (Member) was in the Chair and there were fifty-two members and guests present.

Mr. R. T. B. McKenzie delivered a lecture entitled "A Brief Survey of Modern Developments in Marine Refrigeration" which dealt with the application of refrigeration to domestic units, ships' cold rooms, refrigerated holds and air conditioning of passengers' quarters; the author illustrated his lecture by showing lantern slides. Messrs. Buls, Redford, Thornton, Long and Searby contributed to the discussion which followed.

A vote of thanks to the lecturer was proposed by Mr. D. S. Carment and carried by acclamation.

West Midlands

At a General Meeting of the West Midlands Section held at the Birmingham Exchange and Engineering Centre at 7.0 p.m. on Thursday, 13th October 1955, Mr. H. E. Upton, O.B.E. (Chairman of the Section) was in the Chair and there was an attendance of fifty-five members and guests.

Mr. G. A. Plummer (Member) presented a paper entitled "Steam Boilers—Trends and Tendencies". He showed a short introductory film illustrating each phase of modern watertube boiler manufacture and then outlined the basic principles of watertube boiler design, going on to describe the progress made towards higher efficiencies, greater outputs and smaller, more compact, installations. The author concluded by discussing the latest techniques of design for units operating at advanced steam conditions and compared the relative merits of natural and forced circulation boilers with respect to efficiency, rating and reliability.

Ten members took part in the discussion which followed, and the Chairman expressed his appreciation to Mr. Plummer for his excellent and highly informative paper. The meeting closed at 9.15 p.m.

Student Meeting

A meeting of the Student Section was held at 85 Minories, London, E.C.3, on Monday, 25th April 1955, at 6.30 p.m., when films entitled "The Sea Shall Test Her" and "Handling Ships" were shown. Messrs. K. Abel (Associate Member) and A. T. Webb (Member) answered questions regarding the former and Commander W. R. Symon, R.D., R.N.R., regarding the latter film. Mr. F. D. Clark (Associate Member) was in the Chair. Seventy-one members and visitors were present and eight speakers took part in the discussion.

A vote of thanks proposed by the Chairman was accorded by acclamation. The meeting ended at 8.50 p.m.

Membership Elections

Elected 2nd November 1955

MEMBERS

John Alfred Blockley
Henry Burton Brett
Charles Tregarthen Brinkman
Peter Brodie
William Brown
(formerly Member 9777)
Arthur Norman Davies, Lt. Cdr., R.N.
Alfred Ernest Day, Lt. Cdr., R.N.
Ronald A. Fenwick
Alfred Harrison
Thomas Alfred Hoyland
Alexander Benjamin Ives
James Haddon Kemp
Harry Knight
Klaas Kruimink, Lt. Cdr. (E), R.N.N.
William MacDonald
Duncan MacGregor
David McLelland
Keith McCallum Murray
James Sinclair Porteous
Alan James Richmond, B.Sc. (Eng.), Ph.D
Donald Adrian Rose
Theodore Calvert Scovell
Frederick Edward Smith
George Henry Wheeliker
Bartel Wilton

ASSOCIATE MEMBERS

Robert Barber
Allan Stuart Bridgwater
Robert Findlay Campbell
Michael Chilton, B.Sc.
Edwin Ernest Clayton
Wilfred Norman Copeland
Ronald Dent
Anthony Edgar Derbyshire
Frederick Henry Evans
Raymond Foster

John William Grant
William Simpson Harper
Elias Hatzitheodorou
Dennis Victor Hyde
Ivor Reginald Jordan
Gerard Paul Kiernan
Douglas George Cooper Koster
Joseph Lee
John McCabe Mair
Reginald Stanley Mills
Nellari Poongankandi Mukundan, Lieut. (E), I.N
Stanley Lynham Pickles
William James Clarke Robertson
Harry Short
Jack Tinneveld, Lt. Cdr. (E), R.N.N.
Donald Edmund Walters
Gordon Ivor Watkins
Charles Herbert Bradwell Watson
George Campbell Watt

COMPANIONS

John Peter Ford
Reginald Stewart MacTier, C.B.E.
(Elected 3rd October 1955)

ASSOCIATES

Geoffrey George Cope
Benjamin William Edwards
John William Goldsmith
Franz Uri Levy
Robert Wilfred O'Gorman
Harry Frederick Ford Prosser
Patrick Francis Cleary Smyth
Harold Douglas Waghorn
Hubert Guy Webley
Mansell William Wilde

GRADUATES

Satya Prakash Agarwala, Lieut. (E), I.N.
James Duncan Atkinson
William Clifford Bambrough
Jeremy Dickson Bates, B.Sc. (Eng.)
George Buchanan
Rajeswari Prasadarao Chitra
David Ceiriog Hughes
Dennis Frederick McLaren
Stanley Oliver
Charles Brown Peacock
Clifford Granville Solloway
Andrew Thompson
Henry Topping
James Brown Trail

STUDENTS

Richard Bruce Bellchambers
Peter Albert Dust
Pudupakam Rabindra Ganesh
John Stanage Porter, B.Sc. (Eng.)

PROBATIONER STUDENTS

Malcolm Harris Allsop
Graham Leslie Boram
Michael Bretherton
Jeffrey Fielding Brown
Geoffrey Christian
Brian Collinson
Peter Dunderdale
Ian Dunn
Rodney Edgar Foreman
Robert Arthur Gittings
Roger Edward Goddard
Leslie Greenberg
John Christopher George Halliday
John Caldwell Harrison
John Henry Heffernan
Brian Clifford Jackman
Bernard Porter Jeffery
Stanley William Jones
Anthony Mansfield Kidd
Kevin Arthur Lockett
Neil Kellett McGarr
Duncan George Matthews
Spencer Arthur Morrison
Anthony John Morton
Philip Edward Norris
Peter Charles Ormerod
James Richard Petrie
David John Probert
William Robert Rawlings
Malcolm Reid
Ronald Robins
Robert Martin Shapley
William Humphrey Shepherd
David James Lichfield Smalley
John Taylor
Barry David Thomson
Alan Alfred Turner
Richard John Williams

TRANSFER FROM ASSOCIATE TO MEMBER

George Watson

TRANSFER FROM ASSOCIATE TO ASSOCIATE MEMBER

Oku Ekpe Asuquo
Neville Donald John Bhardwaj
Ronald Francis Coghill
John Alexander Coull
Noel Joseph D'Sylva
Donald Gill English
Leslie Joseph Spencer
William Peter Waddell

TRANSFER FROM GRADUATE TO ASSOCIATE MEMBER

Robert Reuben Hochstadter

TRANSFER FROM STUDENT TO GRADUATE

William Edward Sedgwick

TRANSFER FROM PROBATIONER STUDENT TO STUDENT

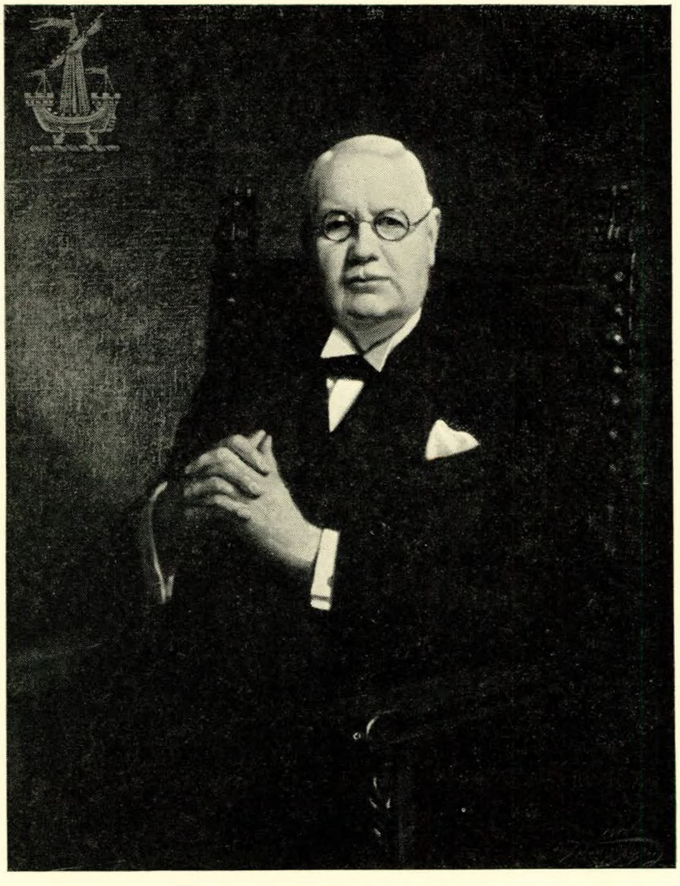
Richard Vincent Clarke
Keith Ronald Corless
Peter Robert Davies
Geoffrey Joseph Dixon
Douglas Hague

THE RT. HON. LORD INVERFORTH, P.C.

An appreciation by J. C. Lowrie (Vice-President)

Many will regret the death of Lord Inverforth, which occurred at his home, "The Hill", Hampstead, London, on Saturday, the 17th September 1955.

Lord Inverforth became President of the Institute of Marine Engineers in the year 1925. During his period of office he was highly esteemed and continued at all times to take a keen interest in the welfare of the Institute. For instance, in 1928 Sir Alan G. Anderson, G.B.E., was President of the Institute but he was in Australia on 7th March of that year and was unable therefore to attend the Annual Dinner held at the Guildhall at which the Duke of Windsor, then Prince of Wales, was the chief guest. Lord Inverforth deputized as chairman for him on that important occasion.



He was born at Kirkcaldy on the 24th April 1865, and was educated at the High School under the Rectorship of Dr. Scott. At all times, whilst attending school, he was very interested in geography and history. In these early years his thoughts were then of foreign lands, and even ships, for he liked to visit the small schooners that called at the port in search of knowledge from those wonderful men of the sea who so successfully navigated the craft to wherever produce was offered.

After leaving school he served his apprenticeship in a local bank and it was there he learned the value of money and how to use it to the best advantage. On leaving the bank he served a Glasgow firm of shipowners; after acquiring sufficient knowledge of how ships should be traded, he established in Glasgow the firm which is known today throughout the world, Andrew Weir and Company.

His first sailing vessel was a barque of 862 tons registered named the *Willowbank*; the *Olivebank,* 2,792 tons gross, and built in 1895, was another—she was one of the last to be built. In all, the Company had some fifty-two sailing ships which sailed under the British Flag.

One cannot but appreciate the farsightedness, courage and business ability of Lord Inverforth, for he was ready enough to acknowledge and make use of the advantages of steam. It was in 1896 that Messrs. Andrew Weir and Company acquired their first steamer, the s.s. *Duneric,* which had a deadweight carrying capacity of 3,050 tons. She marked the beginning of a new era; she represented the change from sail to steam. Lord Inverforth was also amongst the first to appreciate the good points inherent in the marine internal combustion engine. The passenger vessels, trading from India to Africa, the cargo liners, and oil carrying vessels are all propelled by various types of Diesel machinery, a total of seventy-five. Thus it will be seen that in the course of Lord Inverforth's lifetime he had made the fullest possible use of all three three methods of ship propulsion.

On 27th November 1918 he was appointed to the Cabinet and in 1919, in recognition of his services to the country, he was created a Baron.

Perhaps one of the outstanding traits in the dynamic personality of Lord Inverforth was his gift of securing the willing co-operation of his employees, many of whom have spent a lifetime in his service. Shortly after his ninetieth birthday he celebrated the Seventieth Anniversary of the founding of the world-wide shipping organization which he had created.

He served in many capacities in the commercial world; he held many directorships, of which the principal were Cable and Wireless, Ltd., and The United Baltic Corporation, Ltd. His knowlwedge of cable-laying vessels and of their remote duties was unique.

Those who had the pleasure of his acquaintance fully realized his greatness, and so passes one of Britain's shipping magnates, if not the greatest of our century.

He was a home-loving man, a devoted friend, honest, god-fearing and true of purpose.

He is succeeded by his only son, now Lord Inverforth, who has been closely associated for many years with the Company's interests and who without doubt will prove a worthy successor.

HAMISH FERGUSON (Associate 11090) was born in Rugby in 1904. He attended Tonbridge School from 1919-21 and then studied medicine at Birmingham University and University College, London, for the next four years but gave up this medical course to undertake an apprenticeship with the English Electric Company at Rugby from 1927-29, continuing with the Company as a draughtsman in their Diesel section until 1932 and as an outside erector until 1935. For the next nine years he was engaged in the inspection of oil engines for P. H. Smith and Company, Diesel engine consultants in London. From 1944 he was part-time secretary of the Diesel Engine Users' Association and a consulting engineer on his own account, giving up the secretaryship in 1953. Mr. Ferguson, after having been very ill for some time, appeared to be fully recovered when he attended the International Internal Combustion Engine Congress at The Hague in May 1955; however, he was suddenly taken ill again and died there on 28th May. He was elected to membership of the Institute in 1946.

ERIC GORDON HARBOTTLE (Member 13768), who was born in 1892, was apprenticed to Walker Brothers of Pagefield Ironworks, Wigan, from 1908-13, and attended the Wigan Mining and Technical School during the same period. He then joined the Royal Mail Steam Packet Company (now Royal Mail Lines, Ltd.) and was in their service until his death, after several months' illness, on 12th September 1955. He obtained a First Class Steam Board of Trade Certificate in 1923 and a First Class Motor Endorsement in 1931. From 1945 he sailed as chief engineer and retired from active sea service in 1953 as commodore chief engineer of the company.

Mr. Harbottle was elected a Member of the Institute in 1952.

ALEXANDER ANDERSON JAMIESON (Member 4346) died on 8th September 1955, in his eighty-third year. He served an apprenticeship with Scott and Company, Greenock, from 1888-94 and then sailed with the Clyde Shipping Company until he obtained a First Class Board of Trade Certificate. He then joined the China Navigation Company in 1902 and continued in their service until he retired in 1927. Mr. Jamieson was a native of Rhu in Dunbartonshire and returned to live there throughout his retirement. He was elected a Member of the Institute in 1920.

ALEXANDER HAROLD MCBURNEY (Member 12465). Our seagoing members visiting the Australian coast will be sad to learn of the death on the 29th September of Mr. Alex. McBurney.

Alexander Harold McBurney was born at Sydney, N.S.W., on the 8th September 1909, and was educated at North Sydney High School and Sydney Technical College. He served an engineering apprenticeship at Mort's Dock and Engineering Co., Ltd., at Sydney, and then served as an engineer at sea in steam and motorships of the Australian and Oriental Line and Burns Philp and Co., Ltd., gaining his First Class Marine Engineer's Certificates.

He left the sea to become works manager of A. E. Goodwin, Ltd., general and structural engineers of Sydney, where he was engaged in the design and construction of barges and small seagoing craft and oil storage tanks, but his great interest in ships attracted him back to Mort's Dock and Engineering Company where, during the latter stages of World War II he worked as an engine designer in connexion with the building of naval and merchant ships.

He joined Lloyd's Register of Shipping as an engineer surveyor in March 1945, and was stationed in Sydney until February 1954, when he was transferred to Fremantle, Western Australia, to open an Exclusive Surveyors' office at that port. He was promoted to the rank of senior ship and engineer surveyor as from the 1st July 1955. He was elected to Membership of the Institute in 1949.

Alex. McBurney died at Perth on the 29th September 1955, after a short illness. He leaves a widow and one young daughter. W.J.F.

WILLIAM JARDINE MARTIN (Associate Member 3247) died on 15th August 1954, aged sixty-five. He served an apprenticeship with Hawthorns, Ltd., of Leith, and then spent some years at sea, obtaining a Second Class Board of Trade Certificate. In 1930 or so he was appointed assistant foreman in the fitting-out department of Scott's Shipbuilding and Engineering Co., Ltd., Greenock, employment in which he continued until his death.

Mr. Martin was elected an Associate Member of the Institute in 1917.

HAROLD EDWARD PINCHES (Member 9363) was born in 1914. He served an apprenticeship from 1931-35 with Grayson, Rollo and Clover Docks, Ltd., and was then a seagoing engineer until 1946, with the Harrison Line of Liverpool until 1941 and then with the Lyle Shipping Co., Ltd., sailing eventually in their ships as chief engineer. In July 1946 he was appointed engineer surveyor with the British Engine Boiler and Electrical Insurance Co., Ltd., Manchester, the position he held at the time of his death, of coronary thrombosis, on 30th August 1955.

Mr. Pinches was elected an Associate of the Institute in 1942 and was transferred to full Membership in 1945.

FREDERICK GEORGE SCARLETT (Member 9369) was born in 1895. From 1910-15 he was apprenticed with Hotchkiss and Sons, Ltd., Eastbourne, and then joined the Royal Navy, serving as an engine room artificer throughout the remainder of the First World War. From 1919-26 he was assistant works manager with Burnard and Company, London, leaving them to join the Dairy Outfit Company, London, as chief engineer. He was general works manager with Guest and Chrimes, Rotherham, from 1932-34, and works manager of Minimax, Ltd., Feltham, for the next six years. Then, for a year, he was general works manager with Commercial Structures, Ltd. From 1941 until 1947 he was senior production officer at the Directorate of Engine and Airscrew Production, London, and his final appointment, which continued until his death on 20th August 1955, was with the Distillers Co., Ltd., as contracts engineer.

Mr. Scarlett was elected to Membership of the Institute in 1942.

WILLIAM TAYLOR TURNER (Member 3672) served an apprenticeship with Scott's Shipbuilding and Engineering Co., Ltd., of Greenock from 1905-11, which included a year in their drawing office; at the same time he undertook a four years' course in naval architecture at the Royal West of Scotland Technical College, Glasgow. For the next four years he was assistant marine superintendent to the Anglo-American Oil Co., Ltd., but joined the army as a sapper in the Royal Engineers in 1915, being demobilized in March 1919 with the rank of major. During the latter part of the war he was deputy assistant director of the Controller's Department of the Admiralty. From 1919-22 he was a consulting surveyor in New York, acting for various British and American shipowners, and was agent for Wilton's Engineering and Slipway Company of Rotterdam. During 1922 and 1923 he opened a shipbroker's office in London which had to close on account of the shipping depression; he also gave technical evidence in various shipping law cases. From 1924-29 he was superintendent of the New York Oil Storage and Transfer Company's plant at Bayway, New Jersey, for the next two years he was chief of construction and maintenance of Roosevelt Field, Inc., Mineola, N.Y., and for a further two years supervizing engineer of M.A. Hanna Company, Cleveland, Ohio. For a period Mr. Turner was surveyor to the American Bureau of Shipping in Cleveland and in 1937 he took the appointment in which he remained until his death, as plant engineer at Hanson-Van Winkle-Munning Company, Matawan, N.J.; he died on 18th February 1954.

Mr. Turner had been a Member of the Institute since 1919.