STATISTICAL INVESTIGATION INTO DIESEL ENGINE NOISE

conducted by the CIMAC Working Group "Noise"

Chairman: Dr.-Ing. W. Hempel

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A world-wide statistical investigation into the noise emitted by Diesel engines has been conducted, to which the members of the CIMAC working group "Noise" submitted the data on 191 four-stroke Diesels. The evaluation, which initially consisted of calculating the free-field-corrected and A-weighted 1m distance sound pressure level as well as the noise power and efficiency levels, finally established a relationship between the radiated sound power and the output and speed of an engine.

BASIS OF INVESTIGATION

At the first meeting of the "Noise" committee of the Congrès International des Machines à Combustion (CIMAC) in Brussels in February 1967 it was agreed to carry out a statistical investigation into the noise of Diesel engines. A total of 267 completed questionnaires was received from the various CIMAC member countries.

Of these questionnaires 209 were sufficiently complete to permit useful evaluation. The 18 large two-stroke engines were purposely omitted from the analysis and the following evaluation deals only with the 191 four-stroke Diesels covering a range of 10–8000 bhp and 200–3000 rev/min. Most of the subjected engines were water-cooled and turbo-charged, although some normally aspirated and air-cooled engines, primarily below 100bhp in output, were also included.

DEFINITIONS AND FORMULAE

For the purpose of the evaluation it was a prerequisite to convert the acoustical readings obtained under differing measuring conditions to comparable values. The standard distance for measurements from the idealized engine surface was assumed to be $a = 1$m. The various effects of the testroom were eliminated by correcting the levels observed to those of an undisturbed "free field"[1].

Correction to Free-field Conditions

Assuming a semi-diffuse sound field with the engine installed on a reflecting surface and radiating noise only in the upper hemisphere, correction to conditions of a free field is made as follows:

$$L_{pfr} = L_p - 10 \log \left(\frac{1}{2\pi r^2} + \frac{4}{A}\right) + 10 \log \frac{1}{2\pi r^2} \qquad (1)$$

where L_{pfr} [dB] is the corrected level

L_p [dB] is the observed level

The distance r [m] is obtained from the equation:

$$r = a + R$$

where

a [m] is the measuring distance from the engine surface

R [m] is the mean engine radius: $R = \frac{L + B}{\pi}$

L [m] is the length of the engine

B [m] is the width of the engine

For an accurate determination of the equivalent absorption surface A as a function of the frequency it would have been necessary to perform extensive measurements of the reverberation period. To simplify this, the questionnaires merely required information on the type of the room (reverberant room, normal industrial room, dampened room) and its volume.

This information is sufficient for obtaining a rough estimate of the mean absorption surface in each case.[2]

The relationship between the absorbent wall surfaces of rooms and the volume is as follows:

$$F = k \sqrt[3]{V^2} \qquad (2)$$

If a parallelepiped is assumed for the room, the factor k will be equal to 6.

The equivalent absorption surface is proportional to the wall surface:

$$A = \alpha \cdot F \qquad (3)$$

The mean absorption coefficient being estimated at $\alpha = 0{\cdot}08/0{\cdot}16/0{\cdot}25$, equations (2) and (3) above will result in the following absorption surfaces:

a) very reverberant room $A = 0{\cdot}5 \sqrt[3]{V^2}$

b) normal industrial room $A = 1{\cdot}0 \sqrt[3]{V^2}$

c) partially dampened room $A = 1{\cdot}5 \sqrt[3]{V^2}$

Standardizing the Measuring Distance

The levels corrected to free-field conditions are converted to a reference distance of $a = 1$m from the engine surface. Based on a free-field level $L_{p,a}{}^*$ as measured at a distance of a^* [m], the 1m level L_A according to (1) is as follows:

$$L_A = L_{p,a}{}^* + 20 \log \frac{(R + a^*)}{r_o} - 20 \log \frac{(R + 1)}{r_o} \qquad (4)$$

where R [m] is the mean engine radius and $r_o = 1$m is the reference distance.

Noise Power

The noise power level L_{PA} is calculated from the free-field-corrected and A-weighted 1m sound pressure level as follows:

$$L_{PA} = L_A + 10 \log \frac{2\pi (R+1)^2}{So} \quad (5)$$

where the reference surface is $So = 1m^2$ and the reference power is $Po = 10^{-12}W$

Acoustic Efficiency

Basic differences in the acoustic characteristics of various engines are especially easily recognized by relating the radiated noise power $P(A)$ to the brake horsepower. This results in a comparison factor in the form of the acoustic efficiency:

$$\eta = \frac{P(A)}{N_e} \quad (6)$$

However, as this factor is very small, and therefore not sufficiently elucidative, it is advisable to introduce a so-called efficiency level in dB:

$$\eta^* = 10 \log \frac{P(A)}{N_e} \quad (7)$$

$$\eta^* = L_{PA} - L_{N_e}$$

EVALUATION

The questionnaire was so formulated as to call for a minimum of measuring work. It asked, in fact, for the A-weighted sound pressure level in dB(A) averaged around the engine together with details of the measuring distance as well as the type and volume of the room. The engine data required were: speed, output, swept volume, cycle, design (in-line or vee-form), cooling system (water or air), turbo charging, engine weight and dimensions, mean piston speed c_m and mean effective pressure p_e.

Using a computer and the formulae shown above, the following quantities were determined:

1) Free-field-corrected, A-weighted 1m sound pressure level L_A
2) Noise power level L_{PA}
3) Acoustic efficiency level

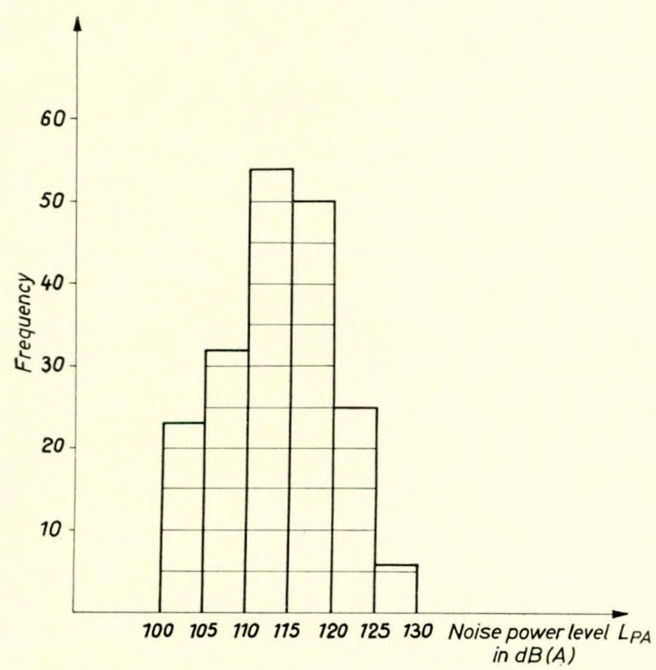

FIG. 1—*Noise power level of Diesel engines*

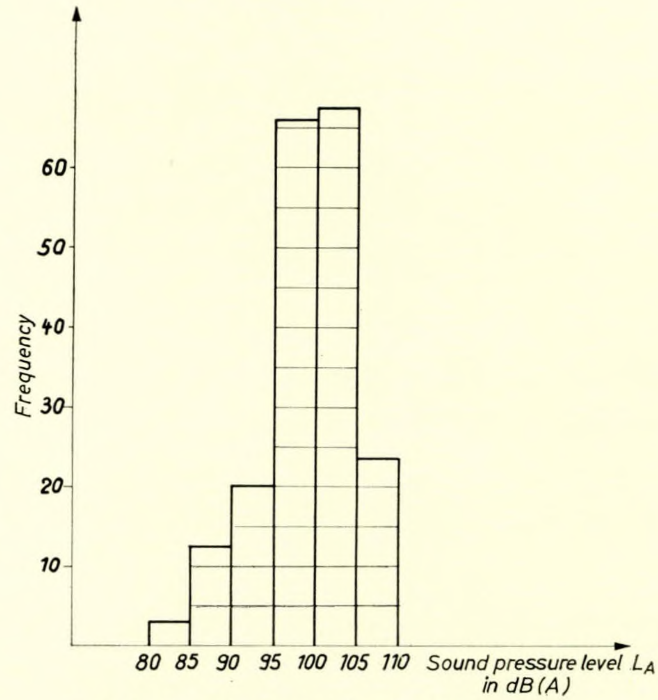

FIG. 2—*Sound pressure level of Diesel engines*

Deriving a Formula for Calculating the Sound Pressure Level

A histogram of the noise power levels L_{PA} and 1m sound pressure levels L_A covering the 191 engines is given in Figs 1 and 2. Most of the engines fall within a sound pressure level range of 95–105dB(A) and a noise power level range of 110–120dB, the difference between the overall minimum and maximum levels being about 30dB in either case. As this is not sufficiently elucidative, a number of regressions were made, similar to previous investigations into the noise of heavy-duty gear-boxes.[2] The input variables were those that have already been indicated as noise factors in a number of existing papers[1,3,4,5] on Diesel engine noise. They are essentially:

i) engine speed, n — determining frequencies and partly amplitudes in pulsations;

ii) mean piston speed, c_m — main factor determining piston slap and mass forces;

iii) number of cylinders, z — determining number of sound sources and engine surface;

iv) stroke, s — determining engine surface, piston slap, mass forces.

These quantities are expressed in a number of formulae for calculating Diesel sound pressure and noise power levels respectively. Two basic types may be considered here[3]:

References 3, 4: $L_A = c_1 \log n$ (*or* c_m) $+ c_2 \log z + c_3 \log s + c_4$

References 1, 5: L_A or $L_{PA} = c_1 \log n + c_2 \log N_e + c_3$

The approach according to Cordier and Reyl[3] and Zincenko[4] is undoubtedly theoretically better founded, but at the same time it is more complicated than those of Pflaum and Hempel[1] and Slavin[5]; this investigation therefore, makes use of both approaches.

The evaluation showed that, due to the errors inherent in corrections to 1m distance and free-field conditions, a regression according to Cordier and Reyl and Zinčenko would not be worthwhile. Typical of this are Figs 3 and 4 using constant stroke s and plotting sound pressure level L_A with the number of cylinders z as a parameter against speed n and mean piston speed c_m respectively. In either case, no useful trend became apparent. The regression formula of Pflaum and Hempel, and Slavin, on the other hand, showed practical results, as shown later.

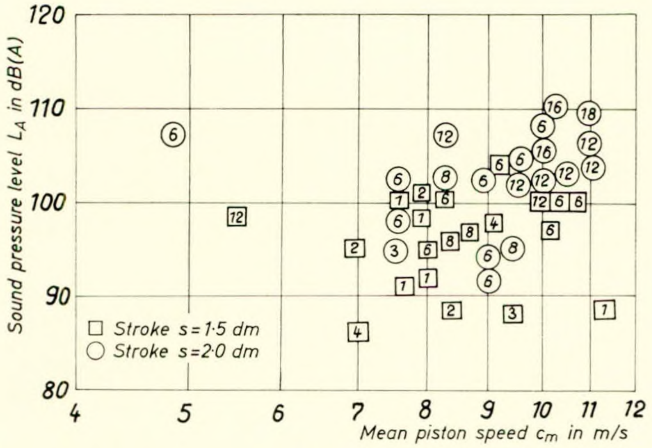

FIG. 3—*Sound pressure level/piston speed—Figures in symbols refer to number of cylinders z*

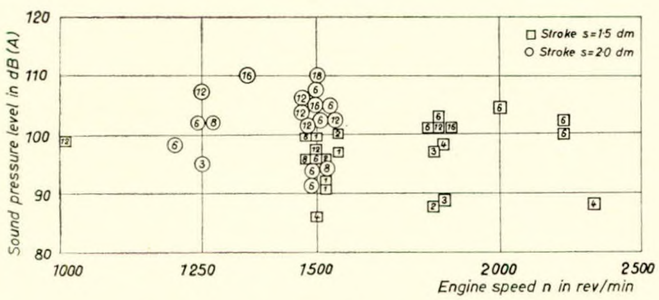

FIG. 4—*Sound pressure level/engine speed—Figures in symbols refer to number of cylinders z*

The equation

$$L_A = c_1 \log n + c_2 \log N_e + c_3 \tag{8}$$

is a linear double regression, the constant of which can be determined graphically. To this end, the sound pressure levels of all engines having the same rated speed were plotted against the logarithm of the rated output. Fig. 5, for instance, uses the speed $n = 1500$, 1000, 750 rev/min and shows a difference between maximum and minimum values of 16dB which is significantly smaller than that shown in Figs 1 and 2, the increase when doubling the rated output being 3dB.

The increase of 3dB for each doubling of output makes constant $c_2 = 10$. When introducing for $L_{PA} - 10 \log N_e$ the acoustic efficiency level η^*, formula $L_A = c_1 \log n + c_2 \log N_e + c_3$ is reduced to:

$$\eta^* = c_1 \log n + c_3 \tag{9}$$

In Fig. 6 the efficiency level η^* of all 191 engines tested is plotted against the logarithm of the rated speed n_N. We find a relatively narrow spread that increases, as should be expected, with the rated speed. While of the engines rated at over 100bhp, 68·3 per cent account for a spread of ±4dB, the engines with lower horsepower—mostly single or two-cylinder units—show heavier deviations to higher efficiency levels. For the purposes of exactly determining constants c_1 and c_3 of formula (9), the distribution of the points in the case of engines above 100bhp provided for determining an approximate straight line $n^* = f(n)$ such as to achieve the minimum sum of the squares of the distances from the ordinates. This calculated straight line, which shall be designated here as the standard straight line of the acoustic efficiency level (see Fig. 6) can be reproduced by means of the formula:

$$\eta^* = 10 \log \left(\frac{n_N}{n_o}\right) - 91\text{dB} \tag{10}$$

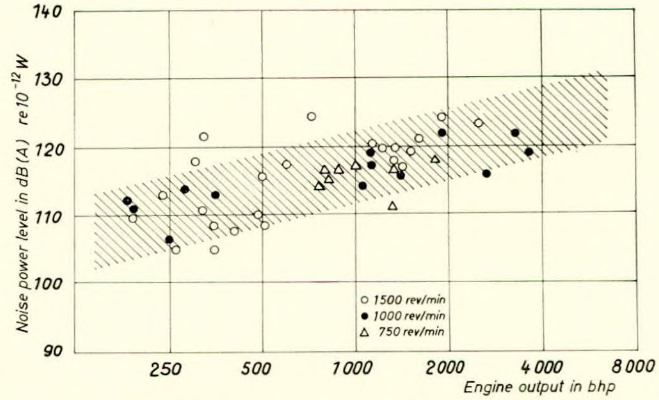

FIG. 5—*Noise power level/horsepower*

where the reference revolution is $n_0 = 1$ rev/min.

This formula results in an increase of the efficiency level amounting to 3dB for each doubling of the speed, which seems to be at variance with the results[1,6] obtained from the individual engines, where speed doubling at constant m.e.p. caused the mean noise power level to rise through 9dB and thus the efficiency level through 6dB.

This difference should be due to the fact that an increase in the rated speed is accompanied by a decrease in the swept volume and thus in the radiating surface, and in particular to the fact that when considering different engines, doubling of the engine speed will not cause the mean piston speed c_m to rise to the same extent as with the individual engine. The latter fact is stressed quite particularly by plotting the efficiency level against the logarithm of the mean piston c_m (see Fig. 7), which results as with the individual engine—if with a large spread—in a rise of about 6dB when doubling the c_m.

In Fig. 6, no connexion can be established between the efficiency level and the number of cylinders z, nor between the efficiency level and the engine design (in-line or vee-form). Nor does the efficiency level appear to depend on turbo-charging. In fact, turbocharged engines do not show more unfavourable efficiency levels than normally aspirated engines or, in other words, engines with a high m.e.p. are not louder than those with a low m.e.p. (unless no provision has been made for adequate silencing of the intake noise, particularly in the case of high-charging slow-speed engines.[7]

Fig. 6 finally shows that the type of cooling (air or water) has no bearing on engine noise. Although indeed, small single and two-cylinder engines present, more unfavourable efficiency levels, this is not due to the type of cooling as such but to the unfavourable engine surface/horsepower ratio.

A conversion of equation (10) for calculating the noise power level of the average engine results as a function of the rated speed $H^n{}_N$ and the rated output N_{eN} is given in the following formula:

$$L_{PA} = 10 \log \left(\frac{n_N}{n_o}\right) + 10 \log \left(\frac{Ne_N}{Po}\right) - 91\text{dB} \tag{11}$$

where the reference speed is $n_0 = 1$ rev/min and the reference power is $Po = 10^{-12}$ W.

Almost the same relationship was established in a study of 17 engines by Pflaum and Hempel[1], except that they found a constant $c_3 = -93$. The reason for this deviation should be that they disregarded the intake noise, while it has been included here in the majority of cases.

To calculate the noise power level at different rating conditions it is necessary only to introduce a speed correction factor of $30 \log \left(\frac{n_N}{n}\right)$ into equation (11), making allowance for the speed function of the radiated sound power as determined for the individual engine.[1,6] If the influence of the mean effective pressure is negligible, a variation of horsepower can likewise be

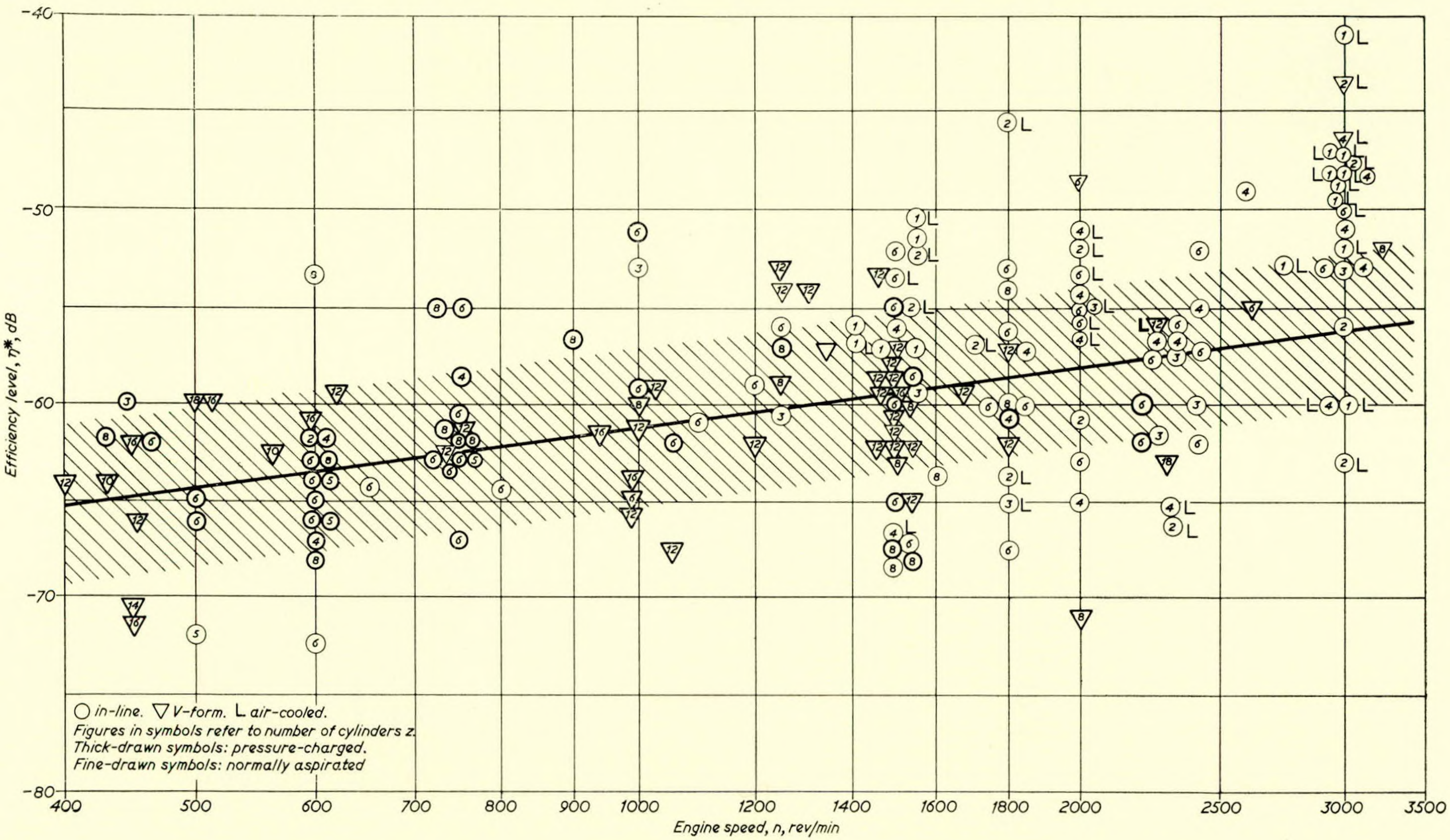

FIG. 6—*Efficiency level/engine speed*

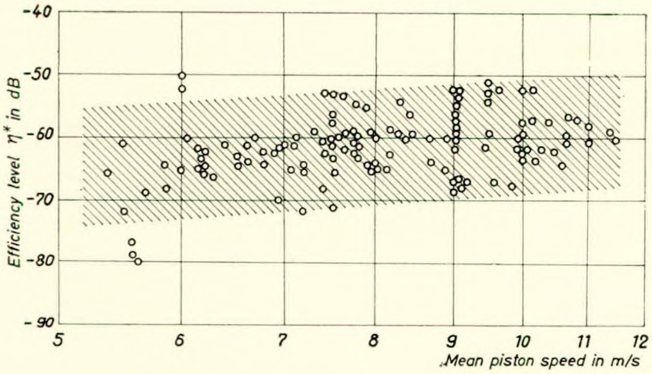

FIG. 7—*Efficiency level/mean piston speed*

neglected. At any operating condition, therefore, the equation for the noise power level will be as follows:

$$L_{PA} = 10\log\left(\frac{n_N}{n_o}\right) + 10\log\left(\frac{Ne_N}{No}\right) - 30\log\left(\frac{n_N}{n}\right) + 58 \quad (12)$$

where n [rev/min] is the operating speed
n_N [rev/min] is the rated speed
Ne_N [bhp] is the rated output
No = 1bhp is the reference power

The relationship between the sound power level and the sound pressure level (5).

$$L_A = L_{PA} - 10\log S/So \quad (13)$$

where the reference area is $So = 1\text{m}^2$, results in the corresponding equation for the sound pressure level as follows:

$$L_A = 10\log\left(\frac{n_N}{no}\right) + 10\log\left(\frac{Ne_N}{No}\right) - 30\log\left(\frac{n_N}{n}\right) - 10\log\left(\frac{S}{So}\right) + 58 \quad (14)$$

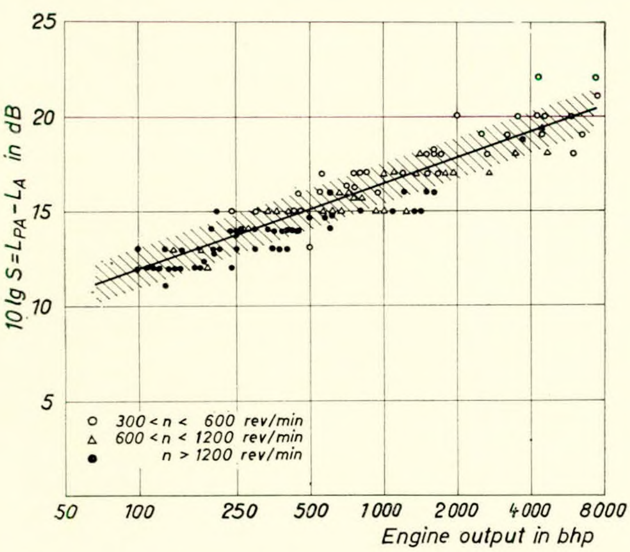

FIG. 8—*Sound power/sound pressure level deviation*

As shown in Fig. 8 plotting the difference between the 1m sound pressure levels L_A and the noise power levels L_{PA}, the term $10\log S/So$ can be expressed with an error of only ± 1dB as a function of the rated horsepower Ne_N as follows:

$$10\log\left(\frac{S}{So}\right) = 4{\cdot}5\log\frac{Ne_N}{No} + 3\text{dB}$$

where S [m²] is the prescribed surface surrounding the idealized engine surface at 1m distance. Accordingly, for calculating the 1m pressure level L_A there results an equation that now depends on output and speed only:

$$L_A = 10\log\left(\frac{n_N}{no}\right) + 5{\cdot}5\log\left(\frac{Ne_N}{No}\right) - 30\log\left(\frac{n_N}{n}\right) + 55 \quad (16$$

which for the rated speed has a simplified form:

$$L_A = 10\log\left(\frac{n_N}{no}\right) + 5{\cdot}5\log\left(\frac{Ne_N}{No}\right) + 55$$

Comparison with other Formulae

As mentioned, several formulae already exist for determining sound levels, but the corresponding results differ widely. The most important of these formulae are the following:

Zinčenko[4]:

$$L_A = 15{\cdot}8\log\left(n_N\frac{1}{k}\ \sqrt[3]{D\cdot s}\ \sqrt[2]{\frac{z}{6}}\right) + 75{\cdot}5$$

where s [m] is the stroke
D [m] is the bore
z [1] is the number of cylinders
n_N [rev/min] is the rated speed
k = 1 is the factor for four-stroke and k = 1·3 for two-stroke.

The study considers 60 marine engines of 10–3500bhp and 100–2000 rev/min, an unweighted sound pressure level measured at 0·5m distance, making no allowance for effects of the room.

Slavin[5]:

$$L_A = 30\log n_N + 12\log Ne - 9\text{dB}$$

where n_N [rev/min] is the rated speed
Ne [bhp] is the rated output

This investigation was primarily concerned with marine Diesel engines, an unweighted sound pressure level measured at 1m distance, making no correction to free-field conditions.

Cordier and Reyl[3]:

$$L_A = 30\log cm + 5\log s + 5\log z + 69$$

where c_m [m/s] is the mean piston speed
s [dm] is the stroke
z [1] is the number of cylinders

This formula is based on a study of 49 engines having a stroke of 1–6·6dm and a rated speed of 250–3000 rev/min. Measurements were taken at 1m distance from the loudest point of the engine.

Taking, for instance, an eight-cylinder four-stroke Diesel engine rated at $Ne = 1000$bhp and $n_N = 1000$ rev/min, with a bore $D = 2{\cdot}3$ dm, mean effective pressure $p_e = 12$ kg/cm² and mean piston speed $c_m = 7{\cdot}6$ m/s, formulae (3, 4, 5) plus the formula developed here will result in the following sound pressure levels:

Slavin[5]:	117 dB (LIN)	at 1·0m distance
Zinčenko[4]:	114 dB (LIN)	at 0·5m distance
Cordier and Reyl[3]:	102 dB (A)	at 1·0m distance
CIMAC:	101 dB (A)	at 1·0m distance

Even if formulae of Slavin and Zinčenko use an unweighted level, disregarding the effects of the room, and measurements according to Zinčenko were taken at 0·5m distance, these cannot be the only factors accounting for the heavy deviations of their formulae[4,5] from those of Cordier and Reyl and CIMAC. Since the formulae of Slavin and Zinčenko attribute excessive importance to the speed, they should have become inadequate for describing the noise development of modern engines.

SUMMARY

Based on the acoustic data of nearly 200 Diesel engines a formula has been developed form statistics enabling the sound pressure level and sound power level of an average engine to be determined at a given output and speed. This formula can be used to estimate the expected noise during planning operations or to judge the results of noise measurements taken on an engine. In view of the errors inherent in measuring and converting operations, only deviations of more than ±4dB from the comparative value calculated for the "standard engine" will be a safe measure for determining the quality of an engine.

As a rule, the noise levels determined with the fairly old formulæ, such as those of Zincenko[4] and Slavin[5], are too high and may no longer be representative for noise emitted by modern engines.

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Discussion

MR. H. D. ADAM (Member) could not understand why the authors had not mentioned noise made by bearings. He felt that the main noise was due to bearing knock, and depended on design clearance, which was increased due to wear after a given number of running hours or excessive wear due to lack of lubrication; and many major breakdowns had been due to bearing failures. On the other hand he was pleased that the authors mentioned piston slap, which could become excessive without causing breakdown and fracture of parts.

MR. G. T. WILLSHARE, B.Sc., wrote that in publishing the results of the work done by CIMAC, the author had presented those faced with the problems resulting from Diesel engine noise with a valuable design aid. The simple and well substantiated formula for the calculation of airborne noise provided a quantitative starting point on which an economic treatment for noise could be based at an early stage in the design of plant layout.

Apart from the fact that it was based on a larger quantity of data, the formula had the advantage over those attributed to other authors in that the dB(A) sound level calculated was for free field conditions to which suitable corrections could be made for particular rooms. Furthermore the calculation took into account both rated speed and operating speed.

There was perhaps one important factor not accounted for in the expression; that was the noise produced by the transmission of engine vibration to the structure of a ship. B.S.R.A. in conjunction with the National Ports Council had recently carried out a survey of noise levels in the engine rooms of small ships (tugs, hoppers, dredgers etc.) and it was found that the majority of dB(A) sound levels were in the 100–110dB region, i.e. 5dB higher than that shown for the majority of Diesel engines in Fig 2. Noise measurements taken during the survey suggested that for a particular engine the dB(A) level generated in a large test shop was 6dB lower than that produced by the same engine fitted in a ship. This difference was only partly accounted for by the difference in the acoustic characteristics of the two enclosures (it was hoped that the results of the survey would be published in the near future).

In an attempt to estimate the dB(A) sound level in an engine room a modified version of Slavin's formula was used and reasonable agreement between calculated and observed levels obtained for a sample of 22 ships, covering engine powers of 44–3000 bhp and speeds of 500–1800 rev/min.

This was a relatively small sample and the expression probably gave too much weight to engine speed. Nevertheless, the results were sufficiently successful to suggest that given more data and using as a basis, the more rigorously established formula presented by Dr. Hempel, an expression might be derived which would afford a reasonably accurate estimation of the dB(A) sound level produced by a given Diesel engine in a ship's engine room.

MR. T. E. NOBLE (Associate Member) in a written contribution commented that the different formulæ used to calculate sound pressure and noise power had as input variables: engine speed, mean piston speed, number of cylinders and stroke, which, with the exception of stroke, were related to definite frequencies radiating from the engine. The effect of "white" noise emanating from the turbocharger and its associated ducting and acoustic resonances were not considered in the formulæ. Were allowances made for these in determining the various constants?

The readings taken in response to the questionnaire were A-weighted sound pressure levels averaged around the engine. The A-weighting implied attenuation at lower frequencies, particularly at the fundamental pitched frequencies of most Diesel engines. With this in mind, why was the A-weighted network selected in preference to linear measurements? Also were ISO noise rating values obtained for the engines considered; as a result of the survey, could the noise rating or even the frequency spectra be predicted for a given load and speed?

MR. K. RATCLIFFE wrote that the four formulæ for the prediction of Diesel engine noise levels put forward by the CIMAC Working Group on Noise had been used by his company to compare with actual noise levels measured on eight different engine types. The Slavin and Zincenko formulæ were rejected, being based on dB'Lin' readings, but the Cordier and Reyl and the CIMAC formulæ were found to give a reasonable prediction. The best agreement between measured and predicted noise levels was found to be given by a formula recently developed by Professor T. Priede which used engine crankshaft speed and cylinder bore as the parameters.

TABLE I—RESULTS

Measured	Priede prediction	Error	Cordier and Reyl predicted	Error	Measured CIMAC corrected	CIMAC prediction	Error
104·4	102·3	−2·1	102	−2·4	101·4	100·3	−1·1
102·2	103·4	+1·2	102·6	+0·4	99·2	100·7	+1·5
96·5	97·9	+1·4	99·2	+2·7	93·5	97·5	+4·0
97·3	96·3	−1·0	98·4	+1·1	94·3	97·6	+3·3
103·1	102·3	−0·8	102·8	−0·3	100·1	100	−0·1
102·0	102·3	+0·3	103·7	+1·7	98·5	100·9	+2·4
100·6	101·2	+0·6	98·4	−2·2	96·4	98·6	+2·2
104·3	104·3	0	102·7	−1·6	100·3	101·8	+1·5
Range of error		−2·1 to +1·4		−1·6 to +2·7			−1·1 to +4·0
Average of error		−0·4		−0·1			+1·7
Standard deviation		1·19		1·88			1·68

The distributions of the errors about zero for the three formulae are shown in Fig. 9.

23 different engines were tested comprising eight different types.

Range of horsepowers	42·5 to 170 bhp
Range of swept volumes	108 in³ to 510 in³
Range of engine speeds	2000 to 4000 rev/min
Range of cylinder bore	3·125 in to 4·5 in
No. of cylinders	3 to 8

Sound pressure levels dB(A) were recorded one metre from both sides, front and top of engine, to give an average level at full load rated speed. The test environment was a specially constructed semi-anechoic noise cell of volume 27 m³ except in the case of the largest engine which was tested in an open test shop with other engines shut down. A reduction of 2dB(A) was made from these levels to account for environmental differences, this being an allowance found by experiment. The measured levels were corrected according to the method given in the CIMAC paper. The noise cell was defined as a "partially dampened room" for which the equivalent absorption surface $A = 1{\cdot}5\sqrt[3]{V^2}$ where V was the room volume (m³). The other formulæ were used with the measured data directly, but with the constant terms in the formulæ reduced. In Priede's method the constant term was reduced by 1dB(A) to give the best fit. In the Cordier and Reyl formula the reduction was 2dB(A). These differences could be accounted for by differences in room acoustics.

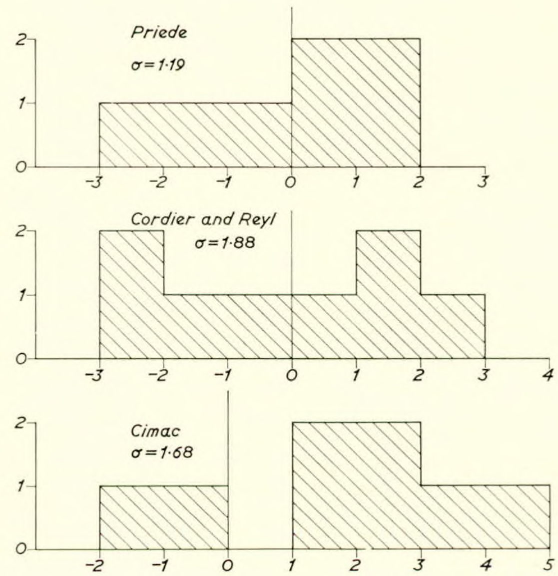


FIG. 9

Priede's Formula

Based on crankshaft speed and cylinder bore:

$$L_A = 50 \log_{10} B + 30 \log_{10} \eta - 31{\cdot}5$$

where L_A = SPL measured from 1 metre [dB(A)]
B = cylinder bore (in)
η = crankshaft rev/min

It was found that the measured data fitted the formula best when the constant was changed i.e.

$$L_A = 50 \log_{10} B + 30 \log_{10} \eta - 30{\cdot}5$$

Priede's formula was developed from an earlier one in which swept volume was used as a parameter instead of of cylinder bore*. The use of the parameters engine speed and bore implied of course that the same noise would be produced by engines of the same bore running at the same speed irrespective of the number of cylinders. This would plainly not be the case in the extreme cases of a single-cylinder engine compared with say an eight-cylinder engine. However, the rule held good for comparisons of engines having between three and eight cylinders.

* PRIEDE, T., GROVER, E. C. and LALOR, N. "Relation between Noise and Basic Structure of Vibration of Diesel Engines." S.A.E. mid-year meeting Chicago May 19–23, 1969. Paper No. 690450.

Reply by Dr. Hemple

In reply to Mr. Willshare, the author wrote that he was himself convinced that, especially in the case of marine application, besides the directly-reflected air noise, the body sound transmission into the vessel's hull was important and that therefore the kind of elastical suspension, the impedance of the foundation, and the structure of the vessel played a special role. This accounted especially for the sound development outside of the engine room, and within the engine room virtually only in the case of insufficient suspension and foundation of the engine. Dr. Hempel did not think it possible to cover the influence of sound conducted through solids by world-wide consultation. Only body sound spectra, measured at engine suspension points, were comparable with each other if the investigated engines were either very softly suspended or if the impedance of the foundation was also measured in each case; the latter would, according to the standing of the measuring technique with the individual engine manufacturers, lead to too great an expense. For this reason it would also be difficult to develop, for cases with dominating body sound transmission, a modified formula for the sound pressure and the acoustic power level, besides the engine data (as such data of the structure of the vessel and the foundation had also to be determined).

In reply to Mr. Noble, the author maintained that in the majority of international circulars and measuring specifications the A-valued sound pressure level had been acknowledged as the authoritative value for machine noises. For a standardization, preference had therefore been given to the dB (A)-level contrary to the linear level or NR-values. Optionally, mean engine spectra could also have been stated on the questionnaires. From the material obtained for 60 engines it had been attempted to work out mean standardized engine spectra for quick and slow running engines as well as super and non-supercharged engines. For this purpose the difference between the octave levels and the dB (A)-level had been plotted in each case. The great divergences which resulted showed that such a standardization in the case of Diesel engines with their immense number of always different and important individual sound sources would not be possible. For the same reason it was also not possible to take into consideration the noise radiated by the turbocharger and its piping system in the formulae worked out with its own constants.

In reply to Mr. Adam, the author believed it was correct to say that the crankshaft bearing was an occasional source of damage. However, on the basis of the present bearing research this source of damage was continuously declining. As a major noise source the bearing could not be made responsible in any investigation carried out up to the present time except where bearing damage had already occurred.

In reply to Mr. Ratcliffe's contribution the author wrote that the formula of ISVR gave good results for truck engines. According to his opinion the influence of the speed seemed to be overestimated. The dependency

$$LA = 40 \log N + 50 \log B - 66{\cdot}5$$

for supercharged engines, did not take into account the fact that with supercharged engines with dominant exhaust gas turbocharger noise, the noise depends primarily upon the airflow i.e., from the engine output, and not the speed; for highly supercharged engines with speeds around 400 rev/min the ISVR-formula gave values which were accordingly also lower by 10 db (A). As the ISVR-formula had been especially developed from data of truck engines of a very limited speed and power range it was understandable that it offered partially better approximations for this narrow parameter range than did the CIMAC-formula which had been determined from engine data from all power ranges and for which it was valid.

Marine Engineering and Shipbuilding Abstracts

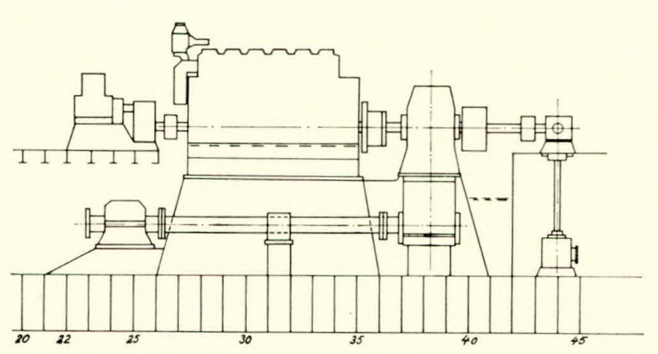
No. 12, December 1970

Gas Turbines for Methane Tankers

The application of marine gas turbines to lng tankers offers cost reductions in initial ship design, construction, and operation. These conclusions were reached after comparing a gas turbine propulsion system for an lng ship requiring 20 000 to 30 000 shp to a steam plant of equal power. Comparison shows the gas turbine plant to have 11 major parts (many of which are self-contained within the turbine package) rather than 24 for the steam system. The Pratt & Whitney FT4 marine gas turbine, for example, includes a self-contained lubricating oil system, fuel and control system, starter, instrumentation sensors, and is pre-wired and pre-piped for installation as a unit. Ship construction advantages include reduction in structural foundations inter-connecting piping and support, increased construction scheduling flexibility, and reduced machinery plant size, resulting in reduction in material cost and related shipyard labour. Turbines are also noted for requiring minimum maintenance and maximum installed engine life. Other advantages include increased ship availability, additional cargo space, reduced manning, and simplified ship design and construction. Optimum gas turbine system depends upon ship size and speed requirements. For the tankers built for Phillips and Marathon, the power selected is 20 000 shp maximum continuous and 18 000 shp normal for a service speed of 17 knots. A gas turbine system capable of meeting present propulsion plant power requirements with the capability of providing up to 28 000 bhp would include a single Pratt & Whitney Aircraft FT4 dual-fuel marine gas turbine, a reduction gear capable of reducing a maximum free turbine speed of 3600 rev/min at maximum power rating to the maximum propeller speed of 100 rev/min and a controllable reversible-pitch propeller. A gas compressor receiving the lng boil-off at atmospheric pressure must be installed capable of boosting the pressure up to 250 to 350 lb/in^2 to ensure proper injection of the fuel into the combustion section of the engine.—*ASME Paper 69-GT-47, presented by Smitt, H. F., in Cleveland, Ohio. Marine Engineer and Naval Architect, June 1970, Vol. 93, p. 290.*

Proposed Engine Room Arrangements with UDAB Geared Diesel Machinery

Simultaneous with the development of the UDAB engine various machinery arrangements for different types of ship have been studied and drawn up in detail. Transmission systems for main and auxiliary power as well as piping systems, overhaul and maintenance procedures have also been developed for the best utilization of this new type of Diesel engine. The twin-engine installation is the most advantageous but single-engine and four-engine plants are also of interest. The twin-engine plant having a maximum



Arrangement for ships with short engine rooms, featuring two elevated engines and double-reduction gear-boxes forward —UDAB engine arrangements.

power of almost 40 000 hp will, of course, find its applications in large or fast ships.

The arrangement shown in the diagram has the gear-box forward of the main engines and the two engines in an elevated position. For ships where a large cubic capacity is essential this arrangement will make it possible to design an extremely short engine-room. The lines in the afterbody are usually of some vee-form and consequently the engines can be moved aft if they are lifted above the double-bottom.

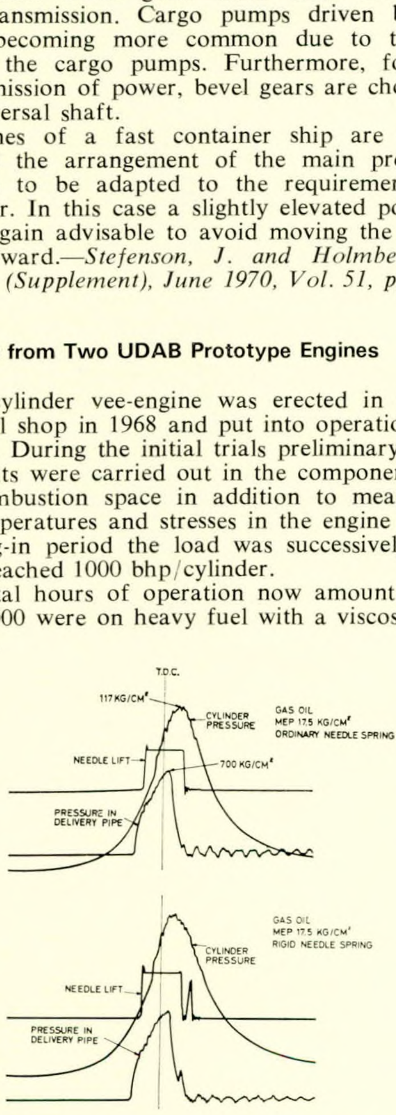
The design of this gear-box having about 4·0 m vertical distance between engine shaft centres and propeller shaft centre has been studied. To avoid intermediate wheels with tooth contacts on both sides of the teeth, a double-reduction gear-box is chosen. This special gear-box will be more expensive but there will be a gain in cubic capacity of the ship and cargo pumps can be driven over the quill shafts from the Diesel engines. Bevel gears have been chosen for the auxiliary transmission. Cargo pumps driven by the bevel gears are becoming more common due to the very low suction on the cargo pumps. Furthermore, for the transverse transmission of power, bevel gears are chosen—operating a transversal shaft.

The lines of a fast container ship are usually very slender and the arrangement of the main propulsion machinery has to be adapted to the requirements from the hull designer. In this case a slightly elevated position of the engines is again advisable to avoid moving the engine room further forward.—*Stefenson, J. and Holmberg, O., The Motor Ship (Supplement), June 1970, Vol. 51, pp. 35–37.*

Test Results from Two UDAB Prototype Engines

A six-cylinder vee-engine was erected in Götaverken's experimental shop in 1968 and put into operation at the end of the year. During the initial trials preliminary temperature measurements were carried out in the components surrounding the combustion space in addition to measurements of bearing temperatures and stresses in the engine frame. After this running-in period the load was successively raised and very soon reached 1000 bhp/cylinder.

The total hours of operation now amount to 1700 out of which 1000 were on heavy fuel with a viscosity of 1500 s Redwood 1.



Injection characteristics and cylinder pressure variation—load 1000 bhp/cyl, 425 rev/min—UDAB engine tests

The second experimental engine, a three-cylinder in-line unit, was built in the Diesel laboratory of NOHAB, in Trollhättan. Naturally this engine has a great number of details identical with those of the vee-engine, the cylinder unit including the connecting rod, and the exhaust valve components being notable examples.

With a great amount of experience already available from the vee-six trials it was possible to press forward with the development of the in-line engine from the beginning. At the time of writing the total trial hours amount to 1300 out of which the last 800 concern heavy fuel operation—of 1500 s Redwood 1 at 100°F. A 400 h trial at a cylinder output of 1200 bhp corresponding to an m.e.p. of 21 kg/cm² was recently completed giving quite satisfactory temperature levels in the combustion space components as confirmed by measurements on practically the same scale as for the vee-six prototype. The unit is now running on 3500 s fuel at an output of 1200 bhp/cylinder.

Soon after commencing the vee-six tests it was found that the fuel injection pressure did not, in the first place, depend on the atomizer hole area but rather on throttling in the internal ducts of the injection valve. Therefore, these ducts were bored up whereafter the engine could be operated at maximum speed and at a cylinder output of 1000 bhp/cylinder, giving a specific fuel consumption of 157 g/bhp h.

Meanwhile redesign of the injection valve was carried out. The diagram shows the injection pressure and needle lift variations for gas oil operation with these new injection valves. The corresponding fuel consumption was 153 g/bhp h at 1000 bhp/cylinder. The gas oil concerned had a viscosity of 32 s Redwood 1 at 100°F.

However, heavy fuel with a viscosity of 1500 s Redwood 1 at 100°F and heated to 98°C gave a considerably higher viscosity than the gas oil referred to above and, as a consequence, 100 kg/cm² higher injection pressure. This pressure increase evidently improved the combustion process in the cylinder giving, as a consequence, a lower fuel consumption. The heavy fuel consumption was measured to 150 g/bhp h as converted to the heat value of Diesel fuel.—*Halldin, N. and Johansson, J., The Motor Ship (Supplement), June 1970, Vol. 51, pp. 11–15.*

Crankcase Explosion Relief Valves

A recent survey by the Marine Division of the Board of Trade, London, has indicated an increase in average yearly ship losses due to fire and explosion of approximately 217 per cent over the period reviewed from 1958 to 1967.

The full significance of this figure is only realized when it is considered that this rate of increase is far in excess of any other cause of such losses and that during this period the number of ships being considered (that is over 500 grt) has grown by only 18 per cent.

Since crankcase explosions fall within this rapidly growing category it is desirable that further consideration be given to this hazard. The provision of crankcase explosion relief valves has for many years now been a standard requirement on heavy oil engines by all marine registration societies.

In view of the above trend, it is therefore not surprising that the principal registration societies now qualify the above requirements by recommending that the crankcase relief valves be designed in such a way as to prevent the emission of flame.

Two methods are currently employed. One is to provide a relief valve fitted with an external cover designed so as to deflect the flame emitted to a safe direction. The other more recent development is the provision of a relief valve incorporating a flametrap capable of preventing the passage of flame through the valve.

Since it is essential that such relief valves be capable of rapid pressure release the degree of directive control effected by a deflecting cover must of necessity be very limited. In practice these covers are invariably designed to deflect the valve discharge through 90° and so direct the flame issuing along the side of the engine.

The aim of this measure is to provide protection for engine room personnel, which is, of course, of paramount importance. Whether this is adequately successful may however, be in doubt since issuing flame lengths from 8 to 12 ft have been witnessed, which become quite uncontrollable due to other deflecting influences in the flame path and the blast conditions prevailing.

A further factor created by more recent developments, that of unattended engine rooms, is the danger of fire gaining hold before attention can be given because of the absence of engine room personnel.

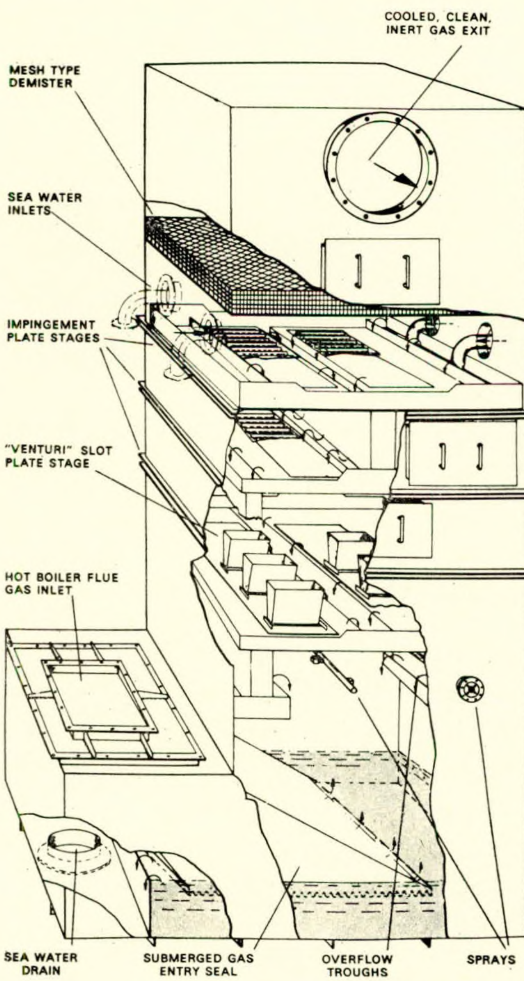
This must inevitably pose the question as to whether any direction can be considered a "safe direction" for the passage of naked flame in an engine room.

The alternative valve is marketed as a patented flame-proof valve and the manufacturers claim that the efficiency of this valve to effect rapid pressure release and to extinguish flame completely has been proved by extensive test results under explosion conditions.—*Shipping World and Shipbuilder, July 1970, Vol. 163, pp. 939; 942.*

Full Control of Explosive Hydro-carbon Vapours with Inert Gas System

Peabody Ltd. has completed development work on a new inert gas system for tankers. The system is designed to give 100 per cent effective control of explosive hydrocarbon oxygen vapours in cargo tanks.

The basis of Peabody's inert gas system is the use of a well-tried design of scrubber-cooler which is said to give exceptionally high efficiencies in the removal of solid matter, sulphur compounds and in cooling the gas to close temperature limits. Utilising boiler flue gases, the system passes these gases through the scrubber, cleaning them and cooling them and the cargo tanks while cargo or ballast is being discharged. Under proper boiler operation the flue gas contains no more than 4–5 per cent oxygen by volume. When purified, this low oxygen gas is incapable of supporting combustion and initiating an explosion as danger does not arise until oxygen concentrations approach 12 per cent.



Peabody direct contact gas scrubber

After discharge of cargo or ballast, the cargo tanks are filled with inert gas from the system mixed with residual hydrocarbon vapours from the cargo. An explosion-proof tank atmosphere is thus maintained. The complete system consists of four principal components:

1) The marine type flue gas scrubber-cooler which uses sea water supplied from existing water service systems or by a separate pump.
2) The blower, fully spark proof to comply with all classification society requirements overcomes total resistance in flue gas system and delivers the requisite gas volume at a positive pressure. The blower may be driven by steam turbine or electric motor.
3) An O_2 or CO_2 analyzer measures and records the oxygen level in flue gases during the inerting procedure. Automatic shut down occurs if O_2 level should exceed or CO_2 fall below a predetermined limit.
4) A control panel with a start-stop button is electrically interlocked with limit switches. The system cannot start until all the limits are satisfied and failure to comply with any limit causes an automatic shut down of the system.

   An additional benefit claimed is the considerable inhibition of corrosion by the removal of the sulphur dioxide in the tank atmosphere.

—*Shipbuilding and Shipping Record, 19 June, 1970, Vol. 115, p. 48.*

Low-temperature Process for Depositing Wear-resistant Coatings

The Caubet process (named after its original developer) is a patented electrochemical deposition technique which reduces friction and increases the resistance of ferrous metals to seizure and wear by providing their surfaces with a thin layer of iron sulphide. This process has a number of inherent advantages, one of the most outstanding of which is that it involves low-temperature treatment of the parts to be coated. As a result, it can be applied to a wide variety of case-hardened, induction-hardened, or through-hardened steel parts without any risk of causing them to lose their hardness.

In the Caubet process, treatment is effected in a special molten-salt bath at a temperature between 360 and 390°F. Sulphur diffuses into the steel and forms a layer of iron sulphide, about 0·0002 to 0·0003 in depth, with a root-like penetration into the steel matrix. The treatment cycle is short and a period of less than 30 minutes is sufficient to produce optimum surface properties, which include marked anti-welding characteristics and excellent absorption of films of lubricating oil. In addition, it is stated that dimensional changes can be considered insignificant.—*Metal Progress, U.S.A., March 1970, p. 9; Engineers' Digest, May 1970, Vol. 31, p. 37.*

Tanker Explosions

After five months of intensive research work, it now seems that a speedy and simple solution to the problem of what caused the explosions on *Marpessa, Mactra* and *Kong Haakon VII* is unlikely to manifest itself.

Co-ordinated by the International Chamber of Shipping, the investigation in which tanker owners and research establishments have participated has now polarized around the twin problems of identifying potential sources of ignition and on the measurement of gas quantities inside the cargo tanks. Electrostatic charging, frictional sparks, auto-ignition and compression ignition have all been the subject of considerable research and this is continuing.

Of these sources of ignition the electrostatic charging solution has continued to be that most widely researched. Suspecting the high pressure tank washing equipment as a source of ignition, the use of this gear was stopped on a world-wide basis, causing in itself further problems where repairs or drydocking were required. However, the manufacturers of this equipment stated that provided the wash water was cold and clean, tank washing with their machines could be resumed.

It has been found that both the temperature and the cleanliness of wash water contribute towards the amount of charging present during washing operations. The possibility of auto-ignition through heat in the tank or compression ignition by means of a trapped gas bubble also forms part of the continuing inquiry.

The quality of gas found inside tanks during various stages of operations has formed the other principal field for research. A considerable mystery has been attached to this as *Kong Haakon VII* was operating the popular "too rich" method of cleaning (where the atmosphere inside the tanks is kept too rich for explosion) while the other vessel, in common with Shell's policy, a "too lean" system. Here again, the information available has been found inadequate and a programme of intensive research into the atmospheres inside the tanks is under way.—*Shipbuilding and Shipping Record, 22 May 1970, Vol. 115, p. 11.*

Developments in Sea Transport

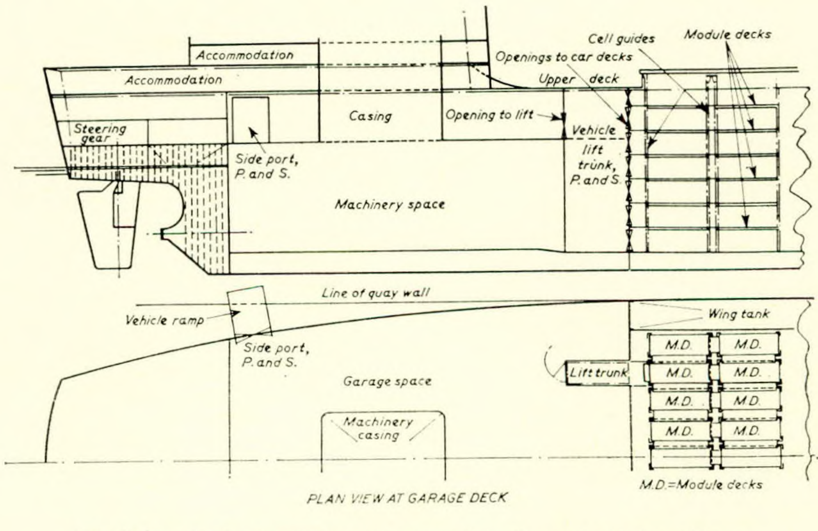
The true cargo liner will survive—this is clearly shown by present trends—but in a very much more specialized form than before. Heavy lift capabilities are increasing. Hatches and holds for long cargo units are lengthening but, even so, cargo ships today must be designed in the context of pallets and containers and hence, to some extent are hybrids. Moreover it is likely that the current trend towards higher speeds will continue but mainly as a result of changes in size or function. Thus, for example, the very large tankers of today have the same or greater speeds than their predecessors although speed/length ratios have decreased and block coefficients have increased.

Container bulkers by virtue of large size and light cargo are ideally adapted to high and growing speeds. They are not, however, significantly faster in terms of speed/length ratio than the break bulk ships they are replacing. Larger container bulkers may be expected to reach speeds of around 30 knots before long and this increase over the 20+ knots of the cargo liner is mainly by virtual change of function and hence size.

Quite a number of "peripheral" bulk carriers exist. In the lumber trade two or three distinct types of bulker are becoming increasingly common. Chipwood carriers deal with approximately 30 per cent of the original lumber and 50 per cent is transported in finished wood form. Thus a very large tonnage of wood pulp chips must be transported. This is a bulky cargo stowing at approximately $200ft^3$/ton or compressed at approximately three-quarters that value. Ships carrying pulp chips are, therefore, very high cubic capacity bulkers, normally self-unloading and yearly increasing in size. Where cargo is not so homogeneous and continuously available, the container bulk ship may be expected to become a hybrid, resulting in ships where deck area is the prime consideration. This will particularly apply to the carriage of finished vehicles which require a large number of decks of low height.

The car carrier which can be used as a containership is not inherently very efficient in the latter mode, while there are means available whereby the containership may be made quite efficient at carrying vehicles while retaining its extremely high cargo handling capability as a purely cellular container bulk ship.

The large, fast cellular containership is with us in increasing numbers and will develop in size and speed along lines easily foreseen, but so will the hybrid ship with both container and vehicle carrying ability so that more harbour facilities will probably have to be provided for such vessels. The diagram shows a proprietary arrangement whereby vehicles may be carried in containerships by the use of modular car decks. Side or stern entry allows vehicles to load at upper 'tweendeck level and lifts can deliver the vehicles to any required level in a cargo hold. The vertical container cells can be spaced off into cargo decks by loading modular platforms instead of containers. By this means a large number of vehicles may be loaded, unloaded at the other end, the modular platforms removed, stowed perhaps



*Modular decks to allow car-carrying in a containership—
Developments in sea transport*

forward, and the ship loaded with cellular container cargo, all with a minimum of delay.

The efficient mechanization of pallets has produced a distinct type of ship. Generally design avoids storing pallets more than one-high and this limits clear deck height beneath girders. At the same time pallet lifts have been introduced which require large openings in the side shell and the cutting of sheer and stringer strakes.

The Ro-Ro commercial vehicle ferry does not seem to have an inherently sound future. It obviously has a place though, where distances are small and where traffic density does not justify, economically, the introduction of mechanized container block systems. To some extent it will survive and develop in this context, but it is not improbable that it will become absorbed in the general unit load system by the introduction of subsidiary Ro-Ro capacity into containerships.—*Paper by Corlett, E. C. B., presented at the ICHCA conference, June 1970, Shipbuilding and Shipping Record, 12 June, 1970, Vol. 115, pp. 44–45.*

3000 bhp Berthing Tug with Triple-Screw Propulsion

Representing one solution to the problem of minimum bollard pull on large berthing tugs is the novel installation on the new 299 gross ton *Keston* of three fixed-pitch propellers working in Kort nozzle rudders. Each screw is powered by an independently controlled 1000 bhp main engine and no clutches are fitted, each engine driving through a coupling and reverse/reduction gear-box.

The *Keston* was built at Lowestoft by Richards (Shipbuilders) Ltd., for J. P. Knight Ltd.

Principal particulars are:

Length, o.a.	114 ft 0 in
Length, b.p.	100 ft 0 in
Breadth, moulded	30 ft 0 in
Depth, moulded	14 ft 6 in
Draught, fore	11 ft 0 in
Draught, aft	11 ft 8 in
Gross register	299·6 tons
Speed, free-running	10·5 knots
Bollard pull, maximum (approx.)	40 tons
Bollard pull, minimum (approx.)	2½ tons
Crew	5
Total engine output, continuous service rating	3 × 1000 bhp at 900 rev/min

Propulsive power is supplied by three Lister Blackstone EWSL8MGR uni-directional units, fitted with Napier turbochargers and Woodward governors. Each engine is coupled to a Lohmann and Stolterfoht GUW 450 3:1 reverse/reduction gear-box—of 1500 hp for added reliability and reduced maintenance—through a Lister Blackstone patent nodal damper coupling, which gives a smooth drive to the gear-box over the complete speed range. A Twiflex 100 per cent torque disc brake is fitted on each shaft aft of the gear-box.

Each shaft drives a fixed-pitch four-bladed manganese bronze propeller of 6 ft 6 in diameter and 5 ft 8 in pitch, operating in a Kort nozzle rudder. The *Keston* is believed to be the first vessel in the world to be fitted with three Kort nozzles.

The steering gear for the three nozzle rudders is of the Donkin electro-hydraulic two-cylinder type with telemotor control and is situated in a deckhouse, as is often the custom on low-freeboard tugs. Two Donkin swashplate steering pumps are fitted, the second to enable rapid manoeuvring to be carried out.

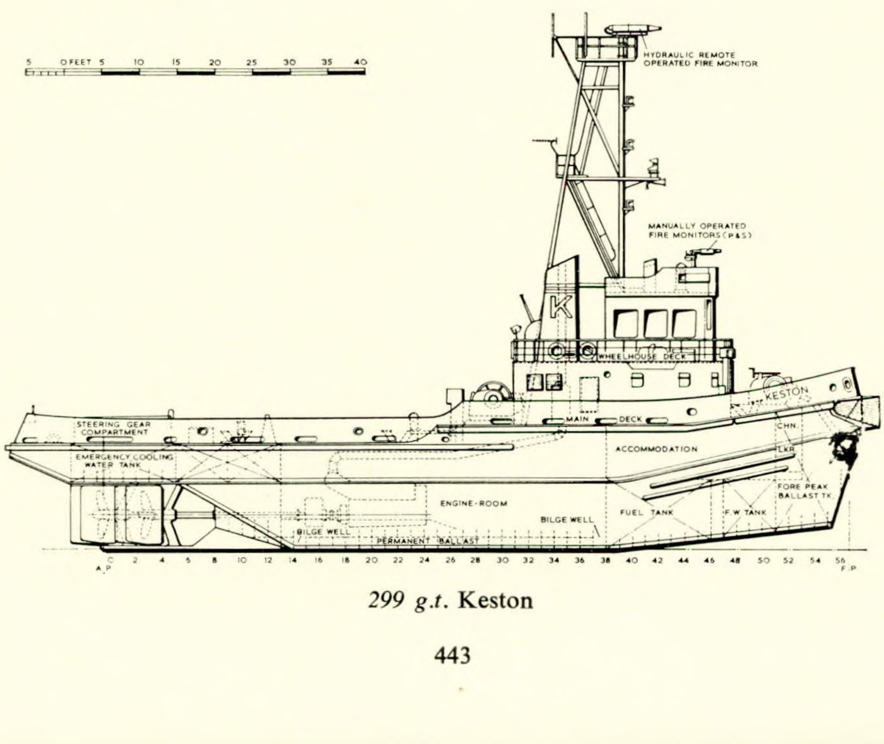
The hull form incorporates a practically flat bottom with a chine just below the waterline and no sheer over most of its length. While this reduces hull resistance, it offers good manoeuvrability and stability on a draught (aft) of 11 ft 8 in; this figure is considerably less than those for tugs of similar size which may draw up to 16 ft. In addition, the flat bottom considerably reduces construction time and enables the vessel to be easily beached for careening on the shingle banks of the Medway. To accommodate the three propellers the vessel has a broad after section, as can be seen from the accompanying plans.—*The Motor Ship, June 1970, Vol. 51, pp. 137–139.*

Polish Ship Lengthening Gives Increased Deadweight at Original Draught and Speed

A standard Polish B459 type ship, the 2250 dwt *Geisha* has recently been lengthened to provide increased deadweight without affecting the original draught and service speed.

Geisha is owned by Gerner Mathisen A/S Rederi, of Oslo, Norway, and was built in 1967 by Stocznia Gdansk in Poland whose Ship Repair division has also been responsible for modifying the vessel.

Prime requirements in the conversion were that speed



299 g.t. Keston

and draught would remain the same as in the original specification while the gross register would not exceed 1600 tons. Designers from the Polish Ship Design Research Centre have, apparently, met the owners' requirements in these respects.

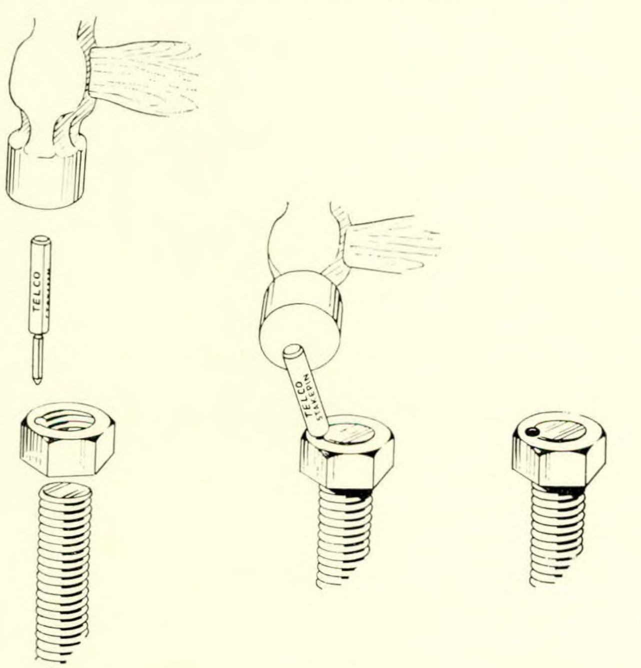
Principal particulars are:	*before*	*after*
Length, o.a.	74·36 m	87·56 m
Length, b.p.	66·92 m	80·02 m
Breadth	13·00 m	13·00 m
Depth	5·90 m	5·90 m
Deadweight	2250 t	2790 t
Speed	12·1 kn	12·1 kn
Gross register	1198 t	1498 t

When the ship arrived at the yard, a 132 m section had already been built and hull reinforcement and fitting elements were prepared for the reconstruction job. Cutting the hull was a relatively simple operation and the sections were easily slid apart. The new midbody section was then floated into the dock and work commenced on making good the two joints.—*Shipbuilding and Shipping Record, 22 May 1970, Vol. 115, p. 33.*

On-site Bolt Making

The extensive piping networks on board ship mean many sizes of flanges each requiring different diameters and lengths of bolts. Nuts and bolts are also used elsewhere on board ship and large stocks have be carried. However, there are instances where a job requires a bolt size that is not carried and a length of threaded rod and two nuts is the obvious alternative. If the fitting has to be made in an awkward corner then the normally simple job of bolting-up can become difficult as quite often one nut will go on the threaded bar easily while the other will perhaps go on for two or three turns and then become tight, so that the load on the threads is uneven.

A simple but effective answer to both these problems is the Stakenut which allows a bolt to be made to the exact dimension required. The Stakenut is a normal nut which has a small fixing slot in one side of the thread. This nut is threaded onto a length of screwed rod until the end of the rod is flush with the end of the nut. A stakepin is fitted into this slot by driving it home with a light hammer and this stakepin locks the nut and screwed rod together. A sideways hammer blow knocks off the head of the stakepin leaving the surface of the nut completely flush. Thus, by cutting the required length of screwed rod, a bolt can be made to any size in a very short period of time.—*Shipbuilding International, June 1970, Vol. 13, P. 36.*



On-site bolt making

Second Containership for Irish Sea Service

The second of two £1 million containerships for British Rail's Irish Sea services was launched in Cork from the same slipway vacated by her sistership a few weeks earlier. This vessel, named *Rhodri Mawr*, and *Brian Boroime*, each with a capacity for 184 standard 20 ft containers, and now completing at the Verolme Cork Dockyard, will operate a new container service linking Holyhead with Belfast and Dublin.

Principal particulars are:	
Length, o.a.	351 ft 6 in
Length, b.p.	328 ft 0 in
Breadth, overall	57 ft 0 in
Breadth, moulded	55 ft 0 in
Depth, to main deck	26 ft 0 in
Load draught	13 ft 6 in
Deadweight	2645 tons
Cargo deadweight	2460 tons
Gross register	3800 tons
Service speed	14 knots
Complement	18
Containers, on deck	48
Containers, in holds	136

To restrict draught, the ratio of breadth to depth has been increased and a moulded beam of 55 ft chosen, which also allows side clearance between containers to be improved from 6 in in the earlier ships to 10 in, while still stowing containers five abreast, to give the crane greater latitude in positioning. Excessive stiffness is avoided by installing wing ballast tanks and leaving the main body of the double-bottom dry. This arrangement gives a GM which varies between three and 10 ft for different conditions and for this reason it was not considered practical to install a stabilizing system.

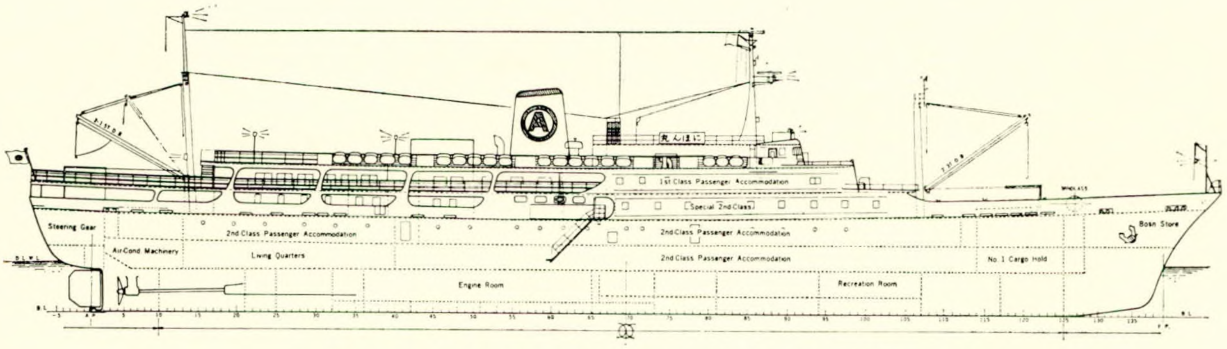
Although a bulbous bow might have added a $\frac{1}{2}$ knot to the performance, a bow rudder was considered more necessary, particularly for manoeuvring past other shipping at Holyhead. Manoeuvring is further assisted by using twin rudders aft and a 350 hp KaMeWa bow thruster.

The mooring system includes eight Roto-bollards, manufactured by Pusnes Mek. Verksted, and four Stothert and Pitt capstans, the forward pair of which handle the ground tackle. A line may be heaved tight by capstan after turning up on a pair of rotating bollards, which are then set to hold the vessel with an hydraulic automatic tensioning control. This obviates any need for stoppers and makes quick adjustments very simple.

Main propulsion machinery is the same as that for the Sea Freightliners, namely two Mirrlees National KLSSGMR six-cylinder engines with a maximum rating of 2100 bhp driving fixed-pitch screws through MWD 2:1 reverse reduction gear-boxes giving an output speed of 225 rev/min. However, whereas a Twiflex clutch/flexible coupling was fitted between the gear-box and main engine in the earlier ships, this coupling arrangement is now abandoned and compensation sought by increasing the diameter of the intermediate tailshaft.—*The Motor Ship, June 1970, Vol. 51, pp. 127–128.*

Japanese Coastal Passenger/Cargo Vessel

A recent addition to the coastal fleet has been the 2998 grt passenger/cargo vessel *Nihon Maru*, built by Mitsubishi's Shimonosuki Yard for Mitsubishi Shoji Kaisha and to be operated by Oshima Unzu K.K. Her employment will



Japanese coastal passenger cargo vessel, Nihon Maru

be in the passenger service around Japan, mainly from her base at Kajoshima, near the southern tip of Kiushu, to Tokyo and the southern waters down to Okinawa.

Nihon Maru is a complete superstructure vessel, with machinery slightly aft of midships. In determining the shape of the hull, which has a raked stem and cruiser stern, particular consideration was paid to seaworthiness. Sufficient flare was given to the stem above the water line, and special attention was also paid to the height of the bulwark. The hull form was finalized after model tests, and a twin-screw single-rudder system was chosen.

The hull structure, as well as the form of the base of the bow, has been designed to enable the vessel to cruise at high speed even in rough seas. To reduce vibration twin engines and twin-screw propulsion were used and careful consideration was given to the distribution of large areas devoted to public rooms. Highly satisfactory results were in consequence achieved.

The main propulsion plant consists of twin Mitsubishi type 7UE 45/75C medium speed Diesel engines driving twin propellers. Starting, stopping, reversing and speed can be remotely controlled from a central console in the wheelhouse to facilitate manoeuvring.

Principal particulars are:

Length, o.a.	106·34 m
Length, b.p.	95·50 m
Breadth, moulded	13·90 m
Depth, moulded	6·20 m
Draught, full load	4·50 m
Gross register	2998 tons
Net register	1716 tons
Deadweight	1111 tons
Propulsion output ...	2 × 4400 bhp
Service speed	20·5 knots

—Shipbuilding and Shipping Record, 3 July 1970, Vol. 116, pp. 30–32.

New Prince Line Ship for Mediterranean Service

Malvern Prince has been built as a single-screw, shelter-decker having two continuous decks, with engine room and accommodation aft. She has been classified by Lloyd's Register as ✠ 100 A1, UMS.

The propelling machinery consists of a Ruston and Hornsby type 9 ATCM four-stroke turbocharged Diesel engine having a continuous rating at 600 rev/min of 2350 shp on the gear-box output coupling. This engine drives through a Metalastik coupling to a Modern Wheel Drive reverse/reduction gear with a ratio of 2·18:1 giving a propeller speed of 275 rev/min. The machinery is designated "UMS" and the machinery spaces will be unmanned for 16 hours per day. An electric standby lubricating oil pump having a capacity of 3900 gal/h at 85 lb/in² has been installed. *—Shipping World and Shipbuilder, July 1970, Vol. 163, pp. 1009–1010; 1013–1015.*

OBO Carrier Built by A. G. Weser

A large Ore-Bulk-Oil (OBO) carrier has been delivered to her owners Wilh. Wilhelmsen of Oslo, by the West German builders A.G. Weser, Bremen. This vessel, *Tarim*, 149 900 dwt and powered by a Diesel engine which at an output of 28 000 bhp (metric) gave the ship a trial speed of 16·75 knots, exceeding the contract speed of 16·2 knots by over half a knot, is the largest vessel of this type to enter service.

Tarim's design has been developed in close co-operation with the classification society and the naval architects Arnesen, Christensen & Co., Oslo. The hull is constructed entirely of high tensile steel, material NV 36 (yield point 36 kg/mm²) being used for the top scantlings of hull girder amidships, whereas the remaining hull is built in NV 27.

Principal particulars are:

Length, o.a.	292·89 m	960 ft 11 in
Length, b.p.	281·00 m	921 ft 11¼in
Breadth, moulded ...	42·50 m	139 ft 5¼ in
Depth, moulded ...	24·70 m	81 ft 0½ in
Draught, summer free-board	17·46 m	57 ft 3¼ in
Corresponding dwt ...	152 300	149 900
Dry cargo capacity ...	169 500 m³	5 985 750 ft³
Oil cargo capacity (100% full)	171 250 m³	6 047 750 ft³
Machinery output, m.c.r.	30 400 bhp (M) at 103 rev/min	

The plating of both the inner and outer bottom was carried out with steel of different thicknesses in the full and empty holds which occur when carrying ore. A thickness reduction from CL to the sides was also carried out. The lateral lower hopper tank bulkheads and the lower trapezoidal stools of the transverse bulkheads were used for fixing and supporting the double bottom grillage.

Speed trials, carried out using the H.S.V.A. speed log, indicated that the contract speed had been exceeded.

Turning circle trials were run at full rudder and maximum engine output, the path of the vessel's stern being computed from successive radar readings. A significant point in connexion with trials of this type is that the exact position of the radar scanner should be taken into account as different results can be obtained depending on its location on the ship.

Det norske Veritas do not require full "crash stop" manoeuvres to be carried out, but only stops from 2/3 of the maximum speed. This manoeuvre was carried out with these results:

Starting speed	13 knots
Engine speed	80 rev/min
Engine reversed and starting astern after:	1 min 22 s
Vessel stopped at 75 rev/min astern after:	10 m 30 s
Distance travelled (developed) ...	1·33 nm

The main engine is a single-acting slow-running two-cycle turbocharged marine Diesel engine type 8K98 FF, built under Burmeister & Wain licence by Fried. Krupp GmbH, Essen. Two B & W exhaust gas turbochargers are fitted.

Principal particulars are:

Normal output ...	30 400 shp (metric) at 103 rev/min
Max. cont. output	28 000 shp (metric) at 100 rev/min
Number of cylinders	8
Cylinder bore	980 mm
Piston stroke	2000 mm

Being a multiple use OBO carrier the vessel has a relatively extensive steam system. Two double pressure boilers generate the steam for the two cargo oil turbines, the two stripping pumps and the tank cleaning pump, as well as the gas freeing fan. They further supply steam for preheating and heating cargo oil, for bunker heating, heavy oil preheating and for heating living spaces.

During normal operation of the vessel at sea an exhaust gas boiler supplies 5 t/h of superheated steam to the turbogenerator, thereby ensuring the entire supply of electric energy, and 2 t/h of saturated steam for heating and preheating purposes.

The boilers operate according to the double pressure system, i.e. the directly oil-heated primary unit delivers the heat contained in the steam to the heat exchanger of the secondary system.—*Shipping World and Shipbuilder, July 1970, Vol. 163, pp. 1041–1045.*

Fast Vehicle Ship for Mediterranean Freight Traffic

As in other parts of the world, the Ro-Ro ferry has brought prosperity in its wake, with greatly improved services to Naples, Bari, Brindisi and Merrina from the North.

Latest of the interesting ships that have entered this growing service is *Freccia Rossa,* meaning Red Arrow, first of two vehicle ferries to enter service this year for the Soc. Grandi Traghetti. Built by Cantiere Naval Breda SPA in Venice, the 21 knot vessel will be capable of a passage time of only 20 hours berth to berth between Genoa and Palermo.

Principal particulars are:

Length, o.a.	134·00 m
Breadth, moulded	21·20 m
Depth, moulded	15·50 m
Deadweight	4000 tons
Service speed	21 knots
Capacity	90 trucks or trailers and 130 private cars
Complement ...	30 crew and 12 passengers, 40 drivers

A load of 90 trucks and trailers (at 12 m each) may be carried, together with 130 private cars. Machinery is capable of coping with 30 freezer trailers, there being a brisk northbound trade in foodstuffs.

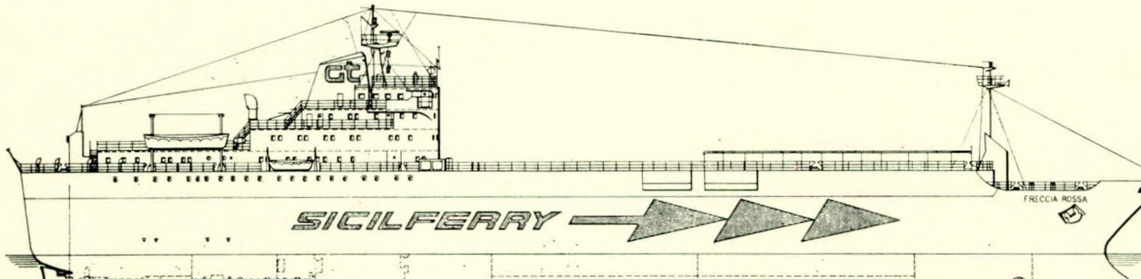
Built to the special survey and to the highest class of Registro Italiano Navale, *Freccia Rossa* has three continuous vehicle decks, below which are four car garage spaces. Trucks and trailers are carried on the main and intermediate decks and on the weather deck, a particularly useful feature in view of the growing hazardous cargo trade. Sole access to the ship for vehicles is through three large stern doors, across the full width of the transom stern, doubling as loading ramps and measuring 6 × 5 m each. These doors, of MacGregor design, have been designed to support a load of 45 tons on three axles and enable simultaneous cargo working on the four "streets" giving access to the vehicle decks.

Propelling machinery consists of two B608S two-stroke Fiat engines each developing 7360 hp at 220 rev/min. Each engine directly drives an Escher Wyss stainless steel c.p. propeller. The system includes automatic adjustment of load between the two engines and a single control for engine output and propeller pitch fitted in both bridge and machinery control room. Power is provided by three 420 kW Diesel alternator sets driven by 4-stroke Fiat engines at 600 rev/min flexibly coupled to brushless, statically excited Marelli alternators.

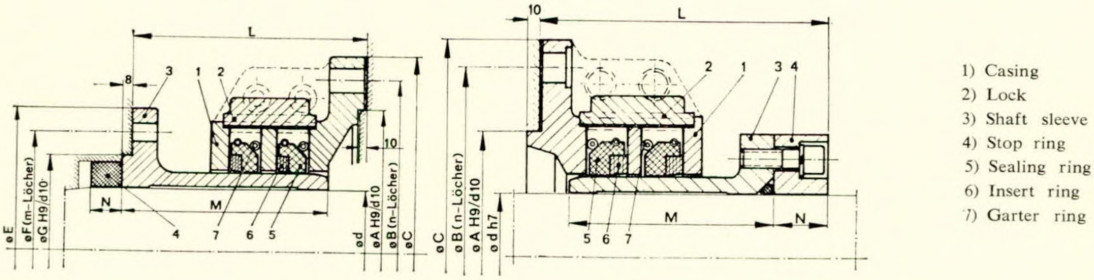
The engine room is of particularly compact proportions, this being enabled by the low height of the B608S engine. The machinery space only occupies the full width of the ship on one deck, there being a vehicle ramp onto the main deck at the level of the tops.—*Shipbuilding and Shipping Record, 26 June 1970, Vol. 115, pp. 24–26.*

Circumflex Seals

The Shell 18 000 dwt general purpose tankers *Hinnites* and *Kenia* have been fitted with a new design of split ring seal for their oil lubricated stern tubes, which enable repairs to be carried out without withdrawing the propeller or docking the ship. The stationary parts of both inboard and outboard seals are interchangeable. The normal radial clearance of 3 to 4mm between the stationary and rotating parts can be increased up to 12mm to compensate for wear in the bearing. The rotating mass of the seal is negligible, being about 10lb for a 700mm diameter shaft, thus allowing the seal to follow movements and vibrations of the shaft. All metal parts, including the shaft sleeve, can be made of bronze to avoid corrosion and all the rubber parts are



Fast vehicle ship for Mediterranean freight traffic



Outboard and inboard elements of Huhn Circumflex tailshaft gland seals

of simple split design. They can be supplied in Viton for increased service life in tropical waters. No bellows are used. The overall length of the seal can be very small, permitting a reduction in the distance between propeller boss and bearing on new construction and facilitating the conversion of existing tonnage to oil lubricated stern tubes.—*Marine Engineer and Naval Architect, July 1970, Vol. 93, p. 339.*

Polish Universal Tramp Ship

The vessel described in this article is the first in a series of B441 ships designed at the Ship Design and Research Office, Szczecin Shipyard, Poland, as a replacement for the fast disappearing Liberty ship. Named *Bailundo* this new ship of 16 618 dwt is owned by Companhia Colonial de Navegacao, Portugal. The ship is intended for the carriage of general cargoes, bulk cargoes such as grain, coal and ore, as well as containers. Vegetable oils can be loaded into the deep tank in the forehold, and timber may also be carried.

The propelling machinery in *Bailundo* consists of a six-cylinder type 6RD68 Cegielski-Sulzer Diesel engine having a maximum continuous output of 7200 bhp at 135 rev/min, and designed to run on heavy fuel of 3500 Redwood No. 1 viscosity at 100°F. Steam is generated in a watertube type donkey boiler rated at 2300 kg/h steam output at 6 kg/cm^2 (85 lb/in^2), and in a main engine exhaust waste heat boiler of 2000 kg/h output at 6 kg/cm^2.

Electricity is supplied by three Diesel-driven alternator sets, each of 400 kVA output. These sets comprise a Cegielski-Sulzer type 8BAH22 Diesel running at 500 rev/min direct coupled to a 380/220 volt 50 cycle alternator. In addition there is a 120 kVA emergency power generator which can also be used as a harbour duty set. This unit has been installed in an insulated compartment on the upper deck.

Principal particulars are:

Length, o.a.		519 ft $6\frac{5}{8}$ in
Length, b.p.		469 ft $0\frac{1}{2}$ in
Breadth		68 ft $10\frac{5}{8}$ in
Depth to upper deck		44 ft $5\frac{3}{8}$ in
Depth to tonnage deck		31 ft $4\frac{7}{8}$ in
Draught, summer		32 ft $9\frac{3}{4}$ in
Draught, b.l. tonnage mk. ...		27 ft $5\frac{3}{4}$ in
Deadweight, summer		16 618 tons (M)
Deadweight, b.l. tonnage mk. ...		12 508 tons (M)
Gross tonnage		11 430·13
Cargo capacity:		
grain	844 074 ft^3	23 902 m^3
bulk	765 467 ft^3	21 676 m^3
Deep tank, grain ...	18 504 ft^3	524 m^3
Speed on trials		15·2 knots
Cruising radius		1300 n. miles

—*Shipping World and Shipbuilder, June 1970, Vol. 163, pp. 803–805.*

Japanese 30 000 shp Planetary Gear

The demand for powerful steam turbine sets, to propel the giant tankers now being ordered in quantity has given rise in some countries to gear manufacture problems. Notable among these are the need to provide very large gear-cutting machines for main wheels. These are exceedingly costly if they are to achieve the high accuracy necessary and, since they are not readily applicable to many other products, tend to be under utilized. The Ship Bureau of the Japanese Ministry of Transport has therefore sponsored a development project for planetary reduction gears, which have considerably smaller components. This has been placed with Toyo Seimitsu Zohki who have considerable experience in epicyclic gear transmissions for industrial drives and has supplied many sets of Diesel turning reduction gear for deck machinery and so on.

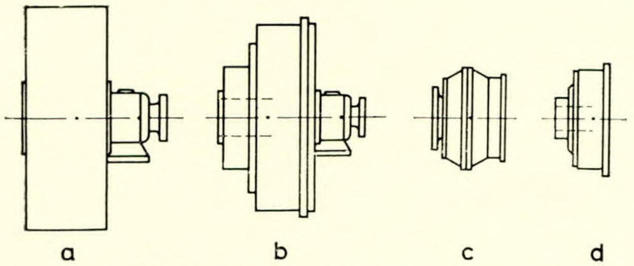
Manufacture has been carried out at the Kobe works of Kawasaki Heavy Industries of two sets, each capable of transmitting 30 000 shp. Tests, which are continuing, indicate that for this transmitted power a 50 per cent reduction in weight and a substantial reduction in size from that of conventional configuration can be made.

A fundamental problem of planetary gearing is the need to share the load equally among each planetary gear and various solutions have been found to achieve this. Toyo Seimitsu, in their IMT system, carry the planet wheels on floating intermediate wheels, using the "spring effect" of a pressurized oil film between them. The arbors which support the bushes are secured in the planet wheel carrier by spherical bearings.

Notable is the use of double spur gears throughout, instead of conventional helical gears. The two separate internal-tooth spur wheels which form the annulus are supported by "spring rods" located by a spherical bearing in the centre. This is made possible in this application in the IMT planetary system because the load is distributed evenly between the planet gears and vibration is absorbed by the relatively thick oil film. The result is greatly simplified machining and assembly operations and reduced production costs.—*Marine Engineer and Naval Architect, July 1970, Vol. 93 pp. 322–323.*

Compact Reduction Gears for Medium-speed Diesels

The design values for marine gears laid down by the classification societies are based on experience but naturally include a generous safety margin for extreme operating conditions. Nevertheless, some fundamental conditions must be accepted for highly-loaded gear-boxes. The most important assumptions are a sufficient contact picture and the avoidance of additional external forces such as can be excited by the Diesel engine and uneven propeller thrust. To avoid the imposition of inadmissible forces due to torsional vibrations, suitable clutches and couplings are necessary and robust thrust bearings, firmly secured to the ship's seatings, are needed to absorb the thrust pulsation. Whereas hydraulic and electro-magnetic slip couplings dominated this field for many years, highly-elastic clutches and couplings of reduced size and weight are used today, see diagram. The use of these clutches implies exact calculation of the torsional vibration situation which it is designed to overcome, no great problem with today's computer techniques. The ability to vary the torsional stiffness and relative damping of such clutches and couplings makes it possible to solve almost any application problem where potentially dangerous loads resulting from torsional vibration must be avoided.



a) Hydrodynamic 2·2m dia—6·12 tons
b) Electromagnetic, 1·95m dia—5·92 tons
c) Highly-elastic (with clutch), 0·99m—1·59 tons
d) Highly-elastic (without clutch), 0·99m—0·97 tons

Various shaft couplings for 2250 hp at 375 rev/min

The thrust block is normally installed on the after side of the gear-case but in situations where the gear-box is located particularly far aft it may be necessary to arrange this at the forward end. For very high outputs, however, it is usual to provide a separate thrust block with its own seatings, as is standard practice in high power turbine gears today.

A high proportion of geared Diesel installations provide for driving the main ship's service generator from the main machinery. Sometimes power is transmitted through a quill shaft extending through the hollow pinion. Another method is to drive the generator by means of step-up gears, the driving wheel of this train running on its own bearings freely outside the main input shaft drive and itself connected to the primary side of the elastic clutch through a second elastic clutch arranged in tandem. By this means the generator can be driven by that particular engine, even with a disengaged propeller shaft. This is particularly suitable for high output generators since the elasticity of the generator coupling can be designed to suit the particular torsional vibration situation. *—Hiersig, H. M. and Burkhardt, W., Marine Engineer and Naval Architect, June 1970, Vol. 93, pp. 254; 256.*

Stern Trawler *Protea*

Hall Russell and Co. Ltd. of Aberdeen, has completed the first of six freezer trawlers for Irvin and Johnson Ltd. of Cape Town.

One of the most advanced designs of trawlers ever built, the *Protea* incorporates a number of features not yet adopted by British freezer trawlers. These include a fish receiving bunker cooled by refrigerated salt water, and an extensive system of twelve conveyors which carry the fish from the receiving bunkers to a sorting point where unacceptable fish are discarded. From here the fish are carried through a descaling machine to two Baader heading and gutting machines after which the fish are conveyed into further chilled salt water holding and bleeding tanks to bring the temperature of the fish down to a lower level and to remove all traces of blood.

On completion of this process, the fish are conveyed to packing stations where they are placed into standardized aluminium trays and weighed. These trays are then loaded into a bank of four Jackstone horizontal plate freezers where the fish are cooled down to −20°F and frozen into blocks. After removal from the freezers, the frozen blocks are removed from the aluminium trays and packed in owner's standard wrappers before being conveyed to the refrigerated fish hold by means of a pneumatic lowering platform.

A conveyor extends over the length of the fish hold to provide for rapid handling of the blocks within the hold. This conveyor discharges into a banana-type unloader which will be placed on board at the unloading berth.

Propulsion machinery comprises an English Electric 8-cylinder Diesel engine developing 1800 bhp and driving a Lips controllable pitch propeller via a Modern Wheel Drive gearbox to give a cruising speed of about 14½ knots. Electrical power is provided by three 420kW alternators. The refrigeration plant, comprising five compressors, is located in the engine room and was supplied, together with associated evaporators, chillers and cooling grids.

Principal particulars are:

Length, o.a.	200 ft 0 in	(60·96 m)
Breadth, moulded ...	38 ft 6 in	(11·73 m)
Depth, moulded	24 ft 3 in	(7·39 m)
Fishroom capacity ...	18 000 ft^3	(509 m^3)
Freezer plant	35 tons/day capacity	

—Shipbuilding International, June 1970, Vol. 13, pp. 30–31.

Swedish-built OBO Carrier for Norwegian Owners

An OBO carrier of 101 600 dwt has been built by Oresundsvarvet AB, Landskrona, Sweden, for Halfdan Ditlev-Simenson & Company of Oslo. Named *Varenna*, this vessel is the first in a series of six similar ore-bulk-oil carriers ordered from the Landskrona shipyard, and the largest ship yet built at this yard. The engine powering *Varenna* is of Götaverken's latest type with an increased mean indicated pressure due to modifications to the cylinder head.

Principal particulars are:

Length, o.a.	256·523 m
Length, b.p.	244·018 m
Breadth, moulded	38·938 m
Depth, moulded	20·600 m
Draught, summer	15·112 m
Block coefficient	
Deadweight	101 600 tons
Displacement	
Lightweight	19 620 tons
Block coefficient	0·8354
Gross tonnage	55 031·88
Cargo hold capacity	
grain	115 003·0 m^3
bale	107 709·5 m^3
oil	119 206·0

The propelling machinery in *Varenna* consists of a nine-cylinder turbocharged Götaverken two-stroke Diesel engine type 850/1700 VSG-9U rated at 19 800 bhp at 115 rev/min. This engine if of the latest type with redesigned cylinder heads and exhaust valves; it is capable of producing a normal continuous rating of 21 600 bhp at 115 rev/min with a m.i.p. of 11·1 kg/cm^2, and a maximum continuous rating of 23 850 bhp at 119 rev/min, m.i.p. 11·9 kg/cm^2. *Varenna* is the third vessel to be powered by the new type G.V. 850/mm bore engine.

The cylinder head was modified to improve the action of the cooling water on the surface facing the combustion chamber, and the piston crown surface facing the combustion space has been modified to achieve a reduction in thickness in the warmest part.

The exhaust valve has a reduced diameter and an increased lift, resulting in a lower fuel consumption and exhaust gas temperature.

On sea trials a mean average speed of 15·47 knots was attained. The engine output at this speed was 20 850 bhp at

118·95 rev/min. The KaMeWa four bladed controllable pitch propeller is of stainless steel and has a diameter of 6800 mm.—*Shipping World and Shipbuilder, July 1970, Vol. 163, pp. 1017–1021.*

New 55 000 dwt Capacity Floating Dock at Japanese Shipyard

A floating dock with a capacity for handling vessels of up to 55 000 dwt is now in operation at Mitsubishi Heavy Industries' Kobe shipyard. Intended for repair work, the new dock is well equipped, and on the dock's port side a large floating fender and mooring facilities, incorporating auto-tension winches, have ben provided so that large vessels can be moored safely.

The dock is of the sectional pontoon type, seven sections making up the complete dock floor. The pontoons are built on the transverse system with floors and girders and are divided longitudinally by bulkheads into four watertight compartments. The walls are of longitudinal construction with transverse webs. Dimensions of the dock are:

Overall length, including overhanging platform and gangway	233·20 m
Length of pontoon	209·20 m
Overall breadth,	
pontoon	44·00 m
wall	45·00 m
inside	37·00 m
between fenders	33·42 m
Wall height	14·50 m
Keel block height	1·40 m
Buoyancy	27 000 ton

The dock has been designed to take a vessel having the following maximum dimensions:

Length, b.p.	220·0 m
Breadth, moulded	33·0 m
Draught	6·5 m
Gross tonnage	32 000
Deadweight	55 000 ton

When empty, the time taken to flood the dock, changing the condition from 0·5 m freeboard to 8·15 m depth of water over the pontoon, is one hour. A similar time is needed to raise the dock, when empty.—*Shipping World and Shipbuilder, July 1970, Vol. 163, p. 1039.*

Unusual Catamaran

Westermoen Hydrofoil specialize in the production of fast surface vessels for civil and military use. Their yard has developed a completely new type of vessel under the name Westamaran. It is mainly in the hull design that the vessel is new. Captain Henriksen, project leader of the yard, took his point of departure, in the theory that from a well known and well tested single hull, a vessel with the same hull, but asymmetrically divided could be built to achieve a better ratio between length and breadth than can be achieved with a single hull. The aim was to gain greater capacity, improved stability; reduced resistance and thereby improved safety and comfort.

To divide a hull asymmetrically means to divide it down the centre and to connect the two parts in a certain relationship to each other with a deck built over the two. Thus a Westamaran is markedly different from a traditional catamaran. The catamaran usually has two identical hulls with a deck arrangement over them giving the vessel greater capacity and stability. However, this form is not suitable for high speed because turbulence occurs in the water streaming between the hulls. With the Westamaran an even stream of water is achieved between the two hulls and the sum of resistance of the parts separated at a certain distance is less than the sum of resistance for the same hull as a single unit.

Main particulars are length overall 86 ft, draught about 4 ft, displacement about 65 tons and it will carry 142 passengers. The speed will be 28 knots and range 220 nautical miles.

The propulsion machinery will consist of twin Diesel engines, each developing 1100 bhp at 1400 rev/min.

The vessel is to be built after comprehensive trials in co-operation with the model tank at the Norwegian Institute of Technology. Resistance curves have been prepared which show that by means of the new hull construction 50 per cent greater capacity is achieved with only seven per cent increase in resistance if comparison is made between a single hull and a Westamaran of the same length.—*Ship and Boat International, June 1970, Vol. 23, pp. 27–28.*

Engine Inching Control System

An electro-pneumatic inching control system has been developed by GX Engineers Ltd., which enables one engineer to carry out adjustments to the ship's engines or the propeller shafts without the aid of a second person to control the engine-turning motor. Hitherto, engine tuning adjustments usually required two men. One man carried out the actual adjustment, the second controlled the electric engine-turning motor to shouted commands. With the new system the man making the adjustments controls the engine-turning motor himself by pressing an "ahead" or an "astern" push-button, housed in a compact portable box connected to the control gear of the engine-turning motor. The push buttons energize two solenoid-operated spool valves which, in turn, control the two sections of a duplex cylinder of $1\frac{1}{2}$ in (38·1 mm) bore and 3 in (76 mm) stroke each end, to give forward, off and reverse positions of the turning-motor controller.—*Shipping World and Shipbuilder, July 1970, Vol. 163, p. 1059.*

Norway's New Research Ship

The Norwegian Institute of Marine Research has acquired a replacement for its 20-year-old research vessel *G. O. Sars*. Retaining the same name, the new *G. O. Sars* is 70 m o.a. and probably has the most advanced fish detection complex of any comparable vessel.

G. O. Sars was designed by Bergens Mekaniske Verksteder in collaboration with the institute's planning group, and the builders were Mjellem and Karlsen, of Bergen. The vessel is built as a stern trawler, with full trawling equipment including a pelagic trawl drum, netzsonde winch etc. and it is also equipped for purse seining, having a Triplex net hauler to starboard, with KaMeWa bow and stern thrusters. A trawl bridge is fitted on the stern gantry right aft.

This fishing equipment is not intended for use in gear research, but purely for fish stock sampling purposes to confirm the presence or otherwise of various fish species.

The heart of the vessel is the operations room into which data from the several laboratories, the acoustic instrument room and the sampling and measuring instruments can be fed and, in some cases, recorded. This is also associated with the data centre, in which is situated the Norsk Data Elektronikk computer, memory store, calculator and echo trace analyzer. The computer will serve a number of functions, from navigation calculations to storage of voyage data from the various laboratories and oceanographic recording instruments. A new concept, however, is its use in conjunction with the highly complex echo sounding equipment. For example, it is possible to examine an area of seabed with three sound beams, each of different frequency and characteristics and by computer analysis and comparison of the traces (which can be greatly expanded) it is hoped to produce a far more accurate count of fish per square metre—and even estimate their size and species.

In order to obtain maximum performance from the acoustic equipment, special effort has been made to keep underwater noise to a minimum by careful stern gear and hull design, by covering shell plates with concrete in critical areas, by fitting the bow thruster with "doors" and by mounting engine units on rubber pads. Although the vessel is fitted with passive stabilization tanks, additional means of stabilizing the transducers have been used, comprising "floating" platforms, gyro stabilized against ± 20° roll and ± 7° pitch from a special Browne reference gyro.

Main dimensions of *G. O. Sars* are length, o.a., 70 m; max. beam, 13·30 m; depth to main deck, 5·15 m; and to shelterdeck, 7.45 m. Gross tonnage is 1500 and displacement, ready for sea, 2000 tons. Maximum speed is 15 knots and cruising range at 10 knots, 10 000 n miles. The design was tank tested at Trondheim and the flume stabilization consultants were John McMullen.

Four Normo LSM turbocharged engines are clutch coupled through a gear-box to the propeller shaft, and any three can deliver the maximum acceptable horsepower of 2500 to the propeller. The fore engines also drive Nebb generators and the after two, hydraulic pumps for winches etc. Current is 380 V 3-phase, transformed where necessary to shore voltage. The engine room console and switchboard is 7 m in length.—*World Fishing, July 1970, Vol. 19, pp. 44–45.*

Sweden's Biggest Tanker

A 227 500 ton turbine tanker ordered by Rederi AB Monacus, Kungsbacka, Sweden, has been named *Brita Onstad* at the Götaverken Arendal yard. She will be the largest ship in the Swedish merchant fleet, the biggest hitherto being *Sea Sovereign* (310 500 dwt) built by Kockums and owned by the Salen Line. *Brita Onstad*, the third in Götaverken's series of eleven tankers of this size, has been built to the highest class of Det norske Veritas and to that society's class for unmanned engine room.

The vessel has a length of 1090 ft, a moulded breadth of 149 ft 7 in, and a moulded depth of 87 ft 6 in. The total capacity of the cargo tanks is 9 992 745 ft^3 and the ballast water tanks have a total capacity of 985 000 ft^3.

The tank subdivision in the cargo space has been carried out with a view to segregation, i.e. the possibility of carrying at the same time various grades of cargo, and has resulted in an agreement enabling part cargoes to be loaded or discharged, while maintaining normal trim and bow stress.

The Götaverken/Stal-Laval steam turbine of AP type develops a maximum of 32 450 shp at 86 rev/min and gives the ship a speed of about 16 knots fully loaded. Steam is generated in two oil-fired watertuve boilers of Babcock & Wilcox type made by Götaverken. Each boiler has a maximum capacity of 69 tons of steam.—*Shipping, July 1970, Vol. 59, p. 37.*

Torsional Strength and Rigidity of "Open" Ships

During the last few years, many papers have been published on this topic and most of the authors used the theory of restrained torsion of prismatic box girders with open cross-section. The deck structure has been replaced by an equivalent thin sheet without hatches to calculate the St. Venant torsional constant, and suitable boundary conditions have been assumed or derived from experimental results. The torsional rigidities so calculated agree with the experimental results satisfactorily, but the calculated torsional strengths agree only by chance, or do not tally with the experimentally determined ones. Unfortunately, the more commonly used finite element method is not, as such, applicable to this problem.

It is well known, that the "open" ships are nowadays built for high speeds. As a result, the ship hull is no longer prismatic. Therefore, it will be treated in this paper as a non-prismatic girder.

The following areas are considered:

a) the region of the hull having an open cross-section;
b) the deck transverses; and
c) the bow and the stern (i.e. closed cross-sections).

A theory of restrained torsion of non-prismatic beams has been developed by the author for these three cases and the method of solution of the mentioned problem briefly presented.—*Vajravelu, P., Shipping World and Shipbuilder, August 1970, Vol. 163, pp. 1145–1147.*

Doxford Seahorse Engine

The Seahorse 580 mm bore 300 rev/min opposed-piston engine of 2500 bhp/cylinder is a joint venture by Doxford and Hawthorn Leslie to develop and build marine propulsion machinery from 10 000 to 70 000 bhp using one size of cylinder. A four-cylinder prototype is being built.

The requirement that the design should not be penalized by the need to provide a multiplicity of cylinders led to the basic parameter, being 2500 bhp/cylinder at 300 rev/min. This is twice the output of any other medium speed engine presently projected or under construction. The opposed-piston configuration is an essential feature of the realization of this performance from engines with from four to seven cylinders. The opposed-piston engine's inherent freedom from thermal difficulties and valve problems are a major contribution towards this result. The frame is fabricated and incorporates cast steel diaphragms for supporting the main bearings.

The materials saved in building an engine of these proportions, when compared with that of the direct-coupled engine, is sufficient to compensate for the additional cost of the gearing. The operating parameters, cylinder working pressures, piston speeds and thermal stresses are well within known and accepted limits. The diaphragm gland separating the cylinder skirt area from the crankcase running gear is an essential feature so that there can be no contamination of the crankcase lubricating oil by combustion sludges.

The crankshaft is semi-built, from centre crank and pin forgings into which are shrunk bobbin pieces forming the side rod and main bearing journals.

Doxford Seahorse engine particulars are:

Cylinders ...	4 to 7
Cylinder bore ...	580 mm
Piston stroke:	
Combined ...	1300 mm
Lower ...	880 mm
Upper ...	420 mm
Cylinder output ...	2500 bhp
Speed ...	300 rev/min
Mean lower piston speed ...	8·8 m/s
B.m.e.p. ...	10·9 kg/cm^2
B.m.i.p. ...	12·0 kg/cm^2
Firing pressure ...	106·0 kg/cm^2
Piston area ...	2642 cm^2
Swept volume/cylinder ...	343·51
Bhp/cm^2 piston area ...	0·473
Bhp/l ...	7·30
Cylinder centres ...	1400 mm
Crankshaft main journal diameter	840 mm
Crankpin diameter (centre and side) ...	520 mm
Centre conn rod length between centres ...	1520 mm
Side conn rod length between centres ...	1300 mm
Exhaust crank lead ...	8°

—*Marine Engineer and Naval Architect, July 1970, Vol. 93, pp. 296–300.*

Modular Approach to DSSV Design

The DSSV, Deep Submergence Search Vehicle, is a planned U.S. Navy development for a search and small object recovery manned submersible capable of operation to 20 000 ft depths. A standard approach to the design utilizing realistic state-of-the-art technology results in a submersible vehicle that is compatible with mission and performance objectives but is incompatible with the combined requirements of depth, endurance and weight. Potential weight reduction solutions compound development risks. A modular submarine design concept circumvents these risks, eases the weight handling problem and offers a solution that provides for greater accessibility for maintenance as well as multiple vehicle make-up versatility and secondary mission growth capability.—*Fuller, R. D. Marine Technology Society Journal, November/December 1969, Vol. 3, pp. 49–56.*

Propulsion Systems for Gas Turbine Destroyers

Four helicopter-carrying destroyers are being built for the Canadian Armed Forces. They incorporate an advanced all-gas-turbine COGOG propulsion system supplied by United Aircraft of Canada Ltd, which comprises two shaft lines, each with a Pratt and Whitney Aircraft FT4A-2 main engine and a Pratt and Whitney Aircraft FT12-A-3 cruise engine, a reduction gear-box, propeller shaft and bearings, and a pushrod-actuated controllable-pitch propeller. The associated ancillary systems, controls and monitoring instrumentation are described, with comments on the underlying objectives and design philosophy.—*Sachs, R. M., ASME Paper No. 69-GT-26 presented 9th–13th March 1969; Jnl Abstracts, B.S.R.A., February 1970, Vol. 25, Abstract No. 28 752.*

Upgraded Engine System Using Scavenging-Air Coolers

This paper is presented in two parts, the first dealing with the theory of the use of scavenging-air coolers to upgrade engines by increasing available horsepower, reducing fuel consumption rates, peak firing pressures, and heat rejection rates, and eliminating engine detonation. Part 2 discusses the installation of scavenging-air coolers on a Clark RA-5 engine, and the analysis of engine performance using the relatively new Beta analyzer. Performance of the engine since installation of the coolers, measured over a period of three years, has shown a consistent increase in horsepower per cylinder.—*Haring, J. M., and Berdon, L., ASME Paper No. 68-PET-27 presented 22nd–26th September 1968; Jnl Abstracts, B.S.R.A., February 1970, Vol. 25, Abstract No. 28 770.*

Nuclear Propulsion for Merchant Ships

The recent commissioning of Germany's first nuclear merchant ship, *Otto Hahn* and the launching of Japan's *Mutsu*, again highlight the absence of a U.K. nuclear merchant ship.

The question being asked by many individuals and authorities is why the U.K., one of the major maritime powers, has not followed the trend set by Russia, U.S.A., Germany and Japan in building a prototype nuclear ship.

The purpose of this paper is to give a brief description of the non-naval nuclear vessels launched to date and to summarize some of the work being carried out in the U.K. to develop an economic nuclear vessel and, by doing so, attempt to show why the U.K. has not yet built a nuclear merchant vessel.—*Wilkinson, G. R., Gaunt, I. A. B., and Rouse, J. R., International Shipbuilding Progress, April 1970, Vol. 17, pp. 101–116.*

Dynamic Factors in Spur and Helical Gears

This paper describes a comprehensive test programme for the investigation of the influence of errors and variations in mesh stiffness on the peak stresses developed in spur and helical gears. The study is primarily concerned with speed effects on tooth loads in regions away from resonance. Four sets of specially designed and instrumented gears were tested in an open-loop set-up. The interaction between tooth loads and system shaft torques at different operating conditions is also investigated. The results are utilized in the paper for the development of a generalized dynamic load formula for spur and helical gears.—*Houser, D. R., and Seireg, A., Trans. ASME, Journal of Engineering for Industry, May 1970, Vol. 92, pp. 495–503.*

Ignition of Lubricants and Hydraulic Fluids in the Lagging of High-pressure Steam Pipes

The purpose of the investigation described in this paper was to compare the self-heating and ignition properties of a mineral oil and some fire-resistant hydraulic fluids leaking into the lagging of the high-pressure steam mains of turbines at current operating temperatures in power stations. Because of the complexity of the heat and material balances involved in self-heating in this kind of system, especially where relatively cool fluid is applied to the outer surface of the lagging on a hot pipe, the experimental approach has consisted of a full-scale simulation of practical conditions in preference to the use of small-scale tests.

Test conditions were chosen to cover the most extreme conditions likely to be encountered in practice. —*Powell, A. W., Wilkinson, B. A. and Bowes, P. C., Jnl Inst. Petroleum, May 1970, Vol. 56, pp. 155–161.*

Discrete Frequency Noise Generation from an Axial Flow Fan Blade Row

An analysis is presented which treats the noise generation from an axial flow fan row by given forces including the effects of a moving medium. The linearization of Euler's equations to yield tractable problems for fan noise is discussed. The three-dimensional problem is decomposed into several two-dimensional problems. Finally, full details are given of a two-dimensional analysis to predict the amounts of acoustic energy, at the blade passing frequency and its harmonics, radiated up and down-stream of a blade row due to its interaction with a neighbouring row.—*Mani, R., Trans. ASME, Journal of Basic Engineering, Vol. 92, March 1970, pp. 37–43.*

Performance of Three Model Axial Flow Turbines Tested Under Both Steady and Pulse Flow

Three turbines, covering a range of reaction of +20 per cent to −10 per cent at their blade roots, have been tested under both steady and pulse flow conditions in order to assess their degree of suitability for turbocharger application.

The results have been analysed in a way which enables efficiencies to be plotted against a mean non-dimensional blade speed irrespective of the level of pulsation. This method of presentation enables the results to be used directly for design purposes and in matching calculations. Thus the efficiency penalty due to the pulse flow, the effect of pulse flow on the choice of turbine size, and the preferred degree of reaction under pulse flow may be assessed.—*Paper by Daneshyar, H., Edwards, K. J., Horlock, J. H., Janota, M., Pearson, R. D. and Shaw, R. submitted to Inst. Mech. E. for written discussion; Paper No. P61/70, 1970.*

Stress and Displacement Analysis of a Shell Intersection

The problem of two normally intersecting cylindrical shells subjected to internal pressure is considered. The differential equations used for the shells are solved subject to the boundary conditions imposed along the intersection between the two cylinders. Details of a procedure for obtaining a numerical solution are given. Numerical results for a radius ratio of 1 : 2 are presented. Problems encountered in the numerical computation are discussed and the results of the analysis are compared with experiment.—*Pan, K. C., and Beckett, R. E., Trans. A.S.M.E., Journal of Engineering for Industry, May 1970, Vol. 92, pp. 303–308.*

High-pressure Water Jets for Undersea Rock Excavation

The paper reviews past research on pulsed high-pressure water jets and their application to breaking rock. Experiments are reported showing the input energy per unit volume required to fracture various types of rock as a function of the pertinent variables. The stagnation pressure of the liquid jets was varied from 50 000 to five million lb/in^2. The theories of jet penetration through water and rock are compared with experimental results. The feasibility and potential advantages of using pulsed water jet equipment for ocean-floor excavation are discussed.—*Cooley, W.C., and Clipp, L.L., Trans. A.S.M.E., Journal of Engineering for Industry, May 1970, Vol. 92, pp. 281–287.*

Solution of Thermal Stress Problems in Tube Sheets

This paper describes the application of the boundary point least squares approach to the plane stress analysis of tube sheets with either mechanical or thermal loads. The paper includes a derivation of appropriate stress functions, a discussion of the point matching and boundary point least square methods, and a description of the application of the method to the analysis of different hole configurations in tube sheets. It concludes with numerical results obtained from the analysis of the thermal stresses near the divider lane of a tube sheet from a two-pass heat exchanger.—*Hulbert, L. E., Trans. A.S.M.E., Journal of Engineering for Industry, May 1970, Vol. 92, pp. 339–349.*

Patent Specifications

Cargo Ship with Liquid Tanks

The invention relates to a cargo ship with tanks arranged in its holds and slidably guided at the sides on vertical retaining elements provided in the hull of the ship, the tanks being designed to hold liquids, in particular gases liquefied at low temperature.

According to the present invention there is provided a cargo ship having in its holds tanks for holding liquids, the tanks being slidably guided on hollow vertical shafts which extend in the hull of the ship from an opening in the region of the deck to a compartment in the bottom of the ship, projections on the sides of the tanks engaging the outsides of the shafts to locate the tanks.

The embodiment shows the application of the invention to a ship with only an outer hull which is formed by two outer side walls (1 and 2), and upper deck (3), an outer bottom (4) and an inner bottom (5). The outer and inner bottoms enclose a double-bottom space or compartment (6). This construction with only an outer hull is not of vital importance for the invention, however, because it is also possible to apply the invention with equal advantage in so-called double-hulled ships.

The total cargo space of the ship is divided into a plurality of holds (8) by means of transverse bulkheads (7) arranged at intervals from one another. Arranged in each hold is a tank (9) which rests on the double-bottom structure of the ship. Any thermal insulation which is required in a given case may be provided on the outer surface of each tank (9) or may be provided, for example, on the inner surface of the ship's walls defining the holds. To support the tanks (9) laterally, retaining elements, in the form of hollow vertical shafts (10) are fixed to the hull. Such a shaft (10) is arranged in each transverse bulkhead between two adjacent holds (8), so that it is employed simultaneously as a retaining element for two adjacent tanks. Moreover, shafts (11 and 12) are associated with the two opposite sides of the tank which are on the outside, these shafts being mounted on the inside of the sides of the ship. The last-mentioned shafts are in the form of double shafts and serve with advantage as loading and unloading shafts for adjacent tank compartments arranged in the double bottom. The retaining elements on the tank or container which co-operate with the shafts (10, 11 and 12) consist of projecting lugs (13), the arrangement of which can be seen from Fig. 1 in conjunction with the shaded areas in Fig. 2. Together with the shafts, these lugs form guides inter-engaging for longitudinal movement in the vertical direction and preventing tilting movements of the tanks as is known *per se*.

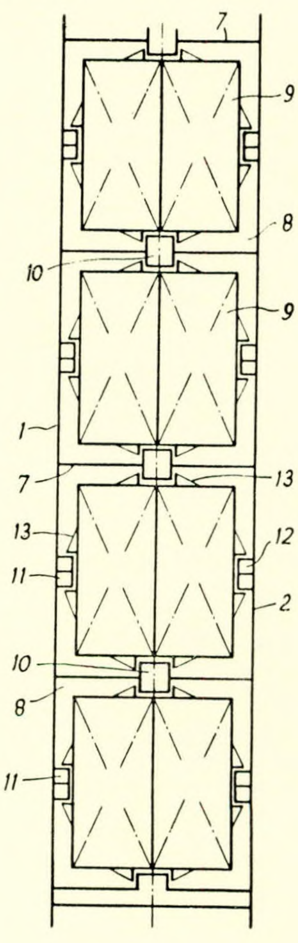


FIG. 1

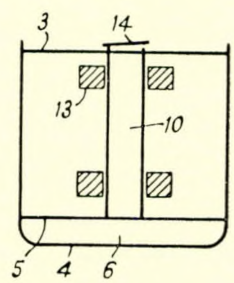


FIG. 2

Fig. 2 shows hatch covers (14) which are provided for closing the shafts (10) at the top.—*British Patent No. 1 189 043 issued to Aktien-gesellschaft "Weser". Complete specification published 22 April 1970.*

Improvements in or Relating to Marine Propulsion Machinery

The figures show a ship's propulsion machinery consisting of two internal combustion V-engines. Fig. 1 is longitudinal section through the engine room. Fig. 2 shows cross sections through the same at the forward and the aft end, respectively, of the engines. Fig. 3 is a horizontal view of the engine room, and Fig. 4 shows the gear-box as viewed from the fore end of the engine room.

Below the engine room (1) there is a double bottom (2) and the engine room is towards the aft peak bordered by a bulkhead (3). The engine room contains two multi-cylinder V-engines (4) and (5) respectively, by which way of a reduction gearing (6) are connected to a propeller shaft comprising one (or possibly several) intermediate shaft parts (7) and the tail shaft (8). The propeller shaft is surrounded by a tunnel (9) having a roof part (10) on which the engines (4) and (5) are mounted.

Bed structures (11) for the two engines are built up integral with the roof of the tunnel. The bed structures are oil tight and serve as oil sumps. The engine entablatures are mounted directly on the bed structures without any intermediate recipient for the lubricating oil. Lubrication oil for the engines is stored in tanks (13), arranged between the forward ends of the side walls of the tunnel and the ship's hull, and each bed structure is by a pipe (14) connected to the adjacent oil tank.

FIG. 1

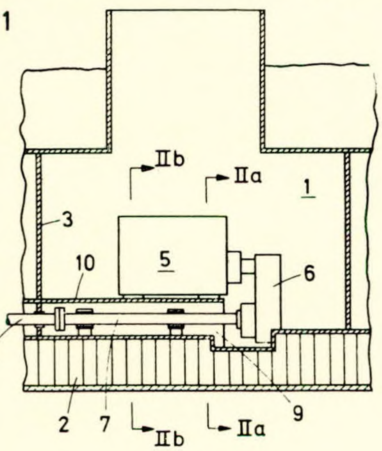


FIG. 2

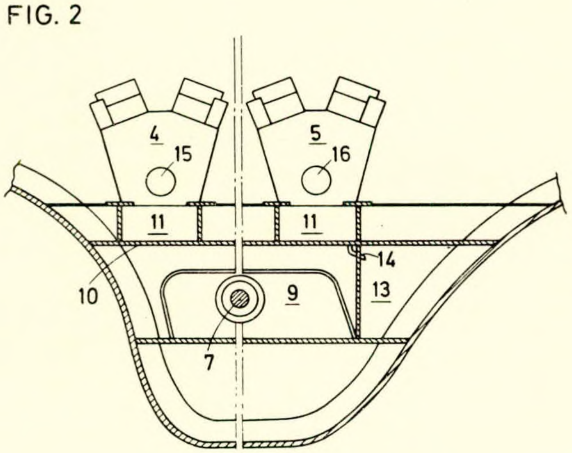


FIG. 3

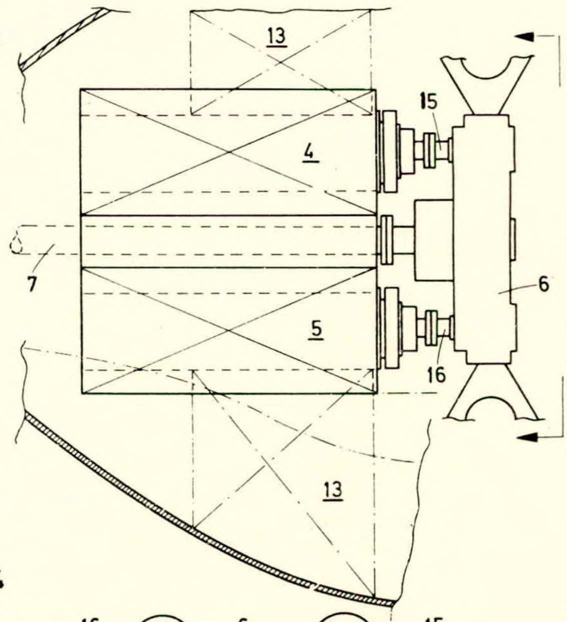


FIG. 4

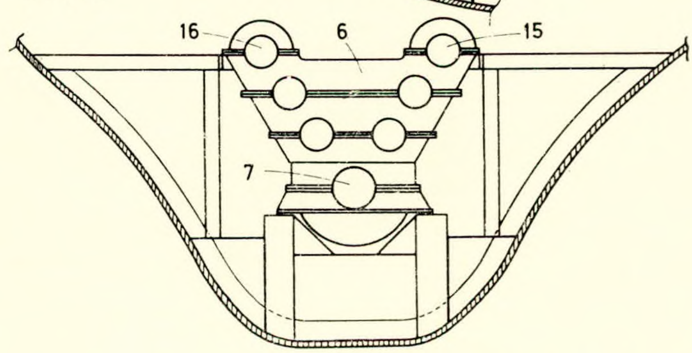


Fig. 2 clearly shows how rapidly the shape of the ship changes within the aftmost part of the hull. The right hand part of Fig. 2 shows the ship at line IIa–IIa of Fig. 1 at the forward end of the engines, and the left hand part of Fig. 2 shows the ship at line IIb–IIb of Fig. 1 at the aft end of the engines. At the forward part of the engines there is sufficient space between the propeller tunnel and the side plating of the ship to make possible the arrangement of the lubricating oil tanks (13), but at the aft end of the engines the propeller tunnel will cover practically the whole breadth of the hull at this part. The location of the engines above the propeller shaft as shown on the drawing has made it possible to arrange the engine room as far aft as here shown. The distance between the crank shafts (15) and (16) of the two engines (4) and (5) is about the same as the distance between either of the crank shafts and the propeller shaft.—*British Patent No. 1 185 976 issued to Aktiebolaget Gotaverken. Complete specification published 2nd April 1970.*

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