FACTORS IN THE CHOICE OF MATERIALS FOR MARINE ENGINEERING

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An analysis of failures in marine engineering components is used by the author to illustrate the factors determining the choice of materials. Fatigue is shown to be the most frequent mechanism of failure and the influence that this has on material choice is discussed and illustrated with reference to various components including crankshafts and gearing.

Materials for parts working at elevated temperatures are also considered e.g. turbine components, bolting and superheater parts. The ever-present problem of sea-water corrosion is discussed and illustrated with reference to the choice of materials for components in marine circulating systems e.g. heat exchanger, circulating pipes, pumps and valves.

INTRODUCTION

Marine engineering is not normally regarded as a field in which the availability of suitable materials is a factor limiting progress, such as is the case in certain other branches of engineering. Nevertheless, the industry's requirements of long life (20 years is normally regarded as the life required) and reliability pose considerable problems to the metallurgist, especially as some of the parts involved are among the largest which the metal working industry has to produce.

Because of the time scale involved and the size and value of the individual parts, laboratory tests have not, until recent years, played a large part in material developments. Accelerated tests on small parts may be meaningless when scaled up to the normal dimensions, and advances have been dependent on a gradual and steady development, involving both engine designers, engine operators and metallurgists. Failure investigation has played an important part in material development as, in a sense, almost every new main engine is a prototype and modifications are continually being made as experience in service builds up. The classification societies are an important source of information on service experience and, where appropriate, changes are made in their rules to incorporate this experience.

Economic factors play an important part in material selection and whilst first cost is usually important, engineers have always to look at the long term economics in making their material selection. The steady advance in the usage of more expensive materials is evidence of the willingness of marine engineers to solve some of their problems by proper selection of materials, both on a technical and an economic basis.

Although this paper deals primarily with materials, it is divided into sections dealing with different components and the material requirements for these components are considered. Some of these requirements are defined in Table I[1].

This table relates to failures investigated by the author over a five-year period, and, while it highlights the importance of fatigue and corrosion fatigue as the main modes of failure, it does not fully emphasize the importance of this type of failure as the table relates only to failures investigated and not to all those occurring. There may be many identical components in the engine, e.g. studs or bolts, in which fatigue failures often occur, and it is not usual to examine every individual failure but only a few representative of the group. Hence, the percentage of failures involving fatigue is in fact considerably higher than that given in Table I.

TABLE I.—FAILURE INVESTIGATION

Mode of fracture	Engine room and propeller, per cent
Brittle fracture	2·5
Fatigue, including corrosion fatigue	68·5
Stress corrosion cracking	14·5
Creep and other high temperatue phenomena	14·5
	100·0

Total number of cases investigated-171
Only those which could be definitely classified have been included

All the failures considered involved mechanical stress. Simple corrosion failures are not considered in the table, but are obviously of importance in practice. The importance of corrosion varies with the component under consideration, but is an ever present factor and must be considered for most applications.

STRUCTURAL COMPONENTS

The main requirements for the structural components of the machinery, e.g. bedplates, entablature, gear cases, are:

1) adequate mechanical properties to provide the strength and rigidity needed to withstand the static and dynamic loads imposed by the working of the machinery;

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2) ease of fabrication, i.e. ability to be cast, welded, flame cut, and hot or cold worked to shape;
3) low cost.

For these reasons cast iron and mild steel are the materials chosen, the latter being more favoured in recent years. Cast iron is still used, but, due to its lower mechanical properties (tensile strength, ductility and toughness) and its lower modulus of elasticity, a cast iron structure is usually heavier than a welded steel fabrication of equivalent strength and stiffness. Mainly for this reason, and also because of the difficulty of manufacturing iron castings of the size required, steel fabrications are now widely used, particularly for large parts.

Some of the earlier steel fabrications suffered from fatigue failures and these have been described by several authorities, including Pemberton[2] and Linden[3]. These failures illustrate the problem of low fatigue strength in weldments. Whilst it is fairly straightforward to obtain tensile properties in a welded joint equivalent to those of the parent metal, it is more difficult to obtain similar fatigue properties as is illustrated by Table II based on work by Newman and Gurney[4]. They found that if the reinforcement was machined off, and the weld polished, and there were no included defects in the weld, then the fatigue strength was equivalent to the rolled plate with mill scale left on.

TABLE II—FATIGUE TESTS ON PLAIN PLATE SPECIMENS AND TRANSVERSE BUTT WELDS

All tests carried out under axial loading with pulsating tension (lower limit zero). Material used ½ in thick mild steel plate to B.S.15

Sample	Maximum stress at 2×10^6 Cycles ton/in^2	Percentage reduction compared with 1
1) Surface machined and dressed by filing	19	—
2) As-rolled plate (mill scale left on)	16	15·5
3) "Good quality" manual transverse butt weld (reinforcement left on) ("Good quality" refers to freedom from significant defects and good reinforcement shape as produced by a skilled welder working in the downhand position.)	10—11½	47—40
4) Automatic transverse butt weld (reinforcement left on) Reduction was attributed to the shape of the reinforcement	6½—11	66—42

They found no benefit in stress relieving at 650°C (1202°F) and in some samples where the stress relief was prolonged, the fatigue strength of both plate and weld samples dropped significantly because of surface decarburization. Other investigators have, however, reached different conclusions, e.g. Wiene[5] showed a 20 per cent increase in fatigue strength by stress relieving at 650°C (1202°F). Also, Gurney[6], in an investigation on the effect of stress relief on the fatigue strength of transverse butt welds containing defects, found that stress relieving gave a pronounced improvement of 40 per cent in welds made with rutile electrodes, although those made with low hydrogen electrodes showed an increase of only 18 per cent.

These results suggest that the effect of stress relieving is complicated by the presence of hydrogen and defects in the weld deposits and, whilst stress relieving may be beneficial, the avoidance of stress raisers, such as poorly profiled weld reinforcements, is of equal importance. Furthermore, stress relieving avoids the risk of distortion and consequent misalignment in service, and fabrications, such as engine bedplates, are usually stress relieved. This, together with greater attention to weld design, for example by avoiding fillet welds (which have very low fatigue strength) in areas of high fatigue load, has resulted in greater reliability and freedom from cracking in service.

Where gear cases are concerned, stress relieving is essential to ensure freedom from distortion and misalignment in service.

DIESEL ENGINE CRANKSHAFTS

The crankshaft of a large marine Diesel engine is too large to manufacture in one piece and is usually built up of a number of smaller forgings or of forgings and castings. Even so, the design and size of some of the individual components impose great difficulties on the foundry and forge. Considerable advances have, however, been made in recent years in the manufacture of these parts, and some of these will now be described.

Forgings

The shaft forgings used to join together the crank webs or crank throws are normally simple and straightforward to manufacture. In recent years it has become fairly common to use vacuum degassed steel for forgings of this size. The use of such steel may not be strictly necessary from a technical point of view, but some steelmakers find it advantageous to use this process as it ensures better mechanical properties and less risk of defects in the forgings, in particular, cracking because of hydrogen in the steel.

While it might appear attractive to reduce the dimensions of these parts by using high tensile steels, the benefit to be gained is limited by the fatigue properties of high tensile material and of the very thick sections of these materials.

For plain unnotched bars tested under fatigue loading, there is a good correlation between tensile strength and the fatigue limit in air. For carbon steels and low alloy steels, the fatigue limit is about 50 per cent of the ultimate tensile strength[7]. Unfortunately, crankshaft changes of section are a type of notch and in this case the fatigue strength is no longer simply related to the tensile strength, and the extent to which the fatigue strength is reduced by the presence of a notch is dependent on the strength of the steel, severity of the notch, type of loading and section thickness.

TABLE III—ENDURANCE LIMIT AT 10^7 CYCLES

Bar diameter in	Steel A		Steel B	
	Unnotched ton/in^2	Notched ton/in^2	Unnotched ton/in^2	Notched ton/in^2
0·125	15·2	9·85	17·4	12·05
0·25	14·3	8·93	17·4	11·15
0·50	12·5	8·02	15·6	8·93
1·00	12·95	8·02	15·6	—
2·00	12·5	7·13	15·6	8·45

Steel A — 0·22 per cent Carbon 0·46 per cent Manganese
0·20 per cent Silicon 3½in diameter hot rolled bar
U.T.S. 27·8 ton/in^2

Steel B — 0·34 per cent Carbon 0·78 per cent Manganese
0·24 per cent Silicon 3½in diameter hot rolled bar
U.T.S. 39·3 ton/in^2.

Table III is derived from results of fatigue tests carried out by Moore[8]. These results show that for a 42 per cent. increase in tensile strength in the two steels considered, the gain in notched fatigue strength in the largest size tested was only 19 per cent.

This table also illustrates the effect of size on fatigue strength. In one steel the notched fatigue limit is reduced by 27 per cent and in the other by 29 per cent when the bar size is increased from 0·125 in to 2·00 in diameter.

For the sizes of crankshaft needed for large marine Diesel engines, the cost of producing the shaft in material of high tensile strength will rise more rapidly than the gain in fatigue strength obtained. For such shafts, therefore, there is unlikely to be a need for materials with a tensile strength of more than about 40 ton/in^2. For smaller engines higher strength material may be used if generous fillet radii are provided and good alignment maintained.

Similar considerations apply to most of the forged components of marine Diesels, e.g. connecting rods, piston rods, eccentrics, and carbon steels of quite low tensile strengths are normally used, except in smaller engines.

Details of the types of steel used for marine crankshafts and other forged parts are given by Dorey[9].

Crank Throws

As the length of a crank pin on a large marine Diesel engine is small compared with the crank throw, it is not possible to forge down the pin from a block in a single operation, and special forging sequences must be used if a sound crank pin is to be obtained. Atkinson and Jackson[10] have described defects in large crankshaft forgings due to unsatisfactory forging procedures but in recent years such defects have been avoided by improved forging sequences[11]. Also, developments in non-destructive testing, notably in ultrasonic testing, have enabled the engineer to place considerably more confidence in the soundness of the forging.

Large numbers of crank throws and crank webs are made by casting and difficulties can arise because of casting defects.

As the manufacture of a large crank throw may require the pouring of over 50 tons of steel, it is not surprising that on occasions defects occur. Large defects can readily be detected by magnetic and ultrasonic test methods and can be dealt with by either cleaning out and welding, where this is acceptable, or by simply grinding away, if the reduction in section size is acceptable. The most difficult type of defect, however, is the very fine porosity which often appears during final polishing of the crank pin. This porosity, which is interdendritic in origin, is to be expected in steel cast into very heavy sections and appears even in vacuum degassed steel. Provided it is well dispersed, and the underlying steel is shown to be sound by ultrasonic testing, there seems to be no reason to reject castings because of this porosity. Evidence in support of this is provided by Carlsson[12] who carried out fatigue tests on full size crankthrows (crank pin diameter 460 mm) and obtained endurance values 2×10^6 cycles of $\pm$ 13·5 kg/mm^2 (8·55 ton/in^2). This correlates well with values reported by Langballe[13] on 65 mm diamter test pieces cut from large castings where endurance limits in the range $\pm$ 13·5 to 18·5 kg/mm^2 at 2×10^6 cycles were found.

Langballe also reports some results on artificial defects and weld defects on the fatigue strength of steel. The lowest value recorded is $\pm$ 10·5 kg/mm^2 (6·65 ton/in^2) for a weld deposit containing defects. This is of interest in relation to weld repairs on large castings. In many cases, the original defect size is so small in relation to the section thickness that it is much safer to grind out the defect leaving a shallow depression than to risk the large reduction in fatigue strength indicated by the given data. Fatigue failures due to unsatisfactory weld repairs are commonly reported in literature, but the author is not aware of any reports of failures resulting from defects treated as just described.

CYLINDER COMPONENTS

The main components of interest are cylinder liners, pistons and cylinder covers. The requirements for these parts are different although related. The three main factors involved are mechanical loading, thermal loading and wear. The cylinder cover is normally not subject to wear but must withstand pressure and thermal loading. Pistons and cylinders must withstand all three conditions but the temperature of the piston crown is generally much higher than that of the cylinder liner.

Pistons

Cast iron is a cheap material with good castability and wear resistance and fulfils most of the requirements of a piston material, provided the thermal loading is low. Its ability to withstand thermal stresses is limited by its low ductility and failure, usually by cracking of the crown, restricts its usefulness.

Failures in cast iron pistons have been explained by Fitzgeorge and Pope[14] as due to creep on the hot face under the steady compressive thermal stress during normal running. On stopping the engine, the surface subjected to the creep deformation is unable to accommodate itself to the undeformed parts and comes under tensile stress. Continued cycling on starting and stopping causes fatigue of the material.

Steel is more resistant to this type of failure as its ductility is much higher than that of cast iron and various types of steel are used, including carbon steel, molybdenum steel (0·4 per cent molybdenum) chromium-molybdenum steel (usually about one per cent chromium and 0·5 per cent molybdenum) and 13 per cent chromium stainless steel. The main advantage of molybdenum and chromium-molybdenum steels is their high creep strength which limits the amount of deformation at high temperatures. The 13 per cent chromium steel has high scaling resistance and is used where burning away of piston crowns is a problem.

Under very severe conditions, steel crowns will also fail and when this occurs it normally takes the form of a wide-mouthed fissure such as illustrated in Fig. 1.

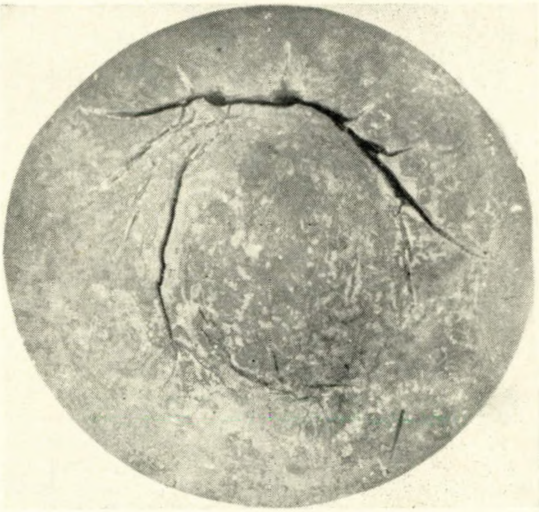

FIG. 1—*Cracked steel piston crown*

Where steel crowns are used, a compromise piston comprising a steel crown and a cast iron body may be needed to provide the necessary wear and seizure resistance for the surfaces in contact with the liner and the piston rings.

Cylinder Covers

Considerations similar to those which apply to pistons, apply to cylinder covers. In small engines, cast iron covers will work satisfactorily, but as thermal loading increases, thermal stress failures may occur and on larger engines steel is normally used, carbon steel and chromium-molybdenum steel being chosen almost always.

Cylinder Liners

The surface temperature of the cylinder liner is normally too low to produce thermal fatigue failures of the type already described. However, the temperature is often high enough to induce severe thermal stress in the material and cracking may occur. Fig. 2 shows a crack commonly found in an opposed piston two-stroke Diesel engine cylinder liner. These cracks start at the inside of the liner, usually on the lower side of a fuel valve or starting air valve pocket, and slowly progress

FIG. 2—*Cracked cast iron cylinder liner*

down and through the liner. In time, similar cracks may form above the pocket and at other pockets and grow in a similar manner.

The appearance and growth of these cracks is similar to fatigue cracking under mechanical loading, but, as they form in an area of high compressive stress due to thermal loads, it is unlikely that they are due to fatigue. Also, as they occur very commonly, it is unlikely that they are due to casting defects or variation in liner thickness. It seems clear that they are caused by thermal stresses and their direction is governed by the hoop stress generated by the combustion pressure. Recent work has shown that a small fatigue stress superimposed on a creep stress will often greatly increase the creep rate. This seems to be the operative mechanism in this case and produces this type of cracking. The problem is avoided by using a steel centre-piece in the hottest zone of the cylinder and similar materials to those used for cylinder covers are chosen.

Cylinder Liner Wear

Cast iron is the material almost universally used for cylinder liners. For large marine engines, a vanadium iron is normally used as this has better wear resistance than vanadium-free irons of similar composition. Table IV shows the wear rates of vanadium-free and vanadium-containing irons in marine Diesel engines operating on heavy fuel oil and using conventional cylinder oils.

The wear rates in liners C and D, which have small vanadium additions, are less than half of those in the vanadium-free liners. The reason for this effect of vanadium has been attributed to the carbide forming properties of the vanadium, producing large amounts of free carbides in the microstructure of the iron, but measurements of carbide content as given in Table IV shows that the vanadium-free irons had a slightly higher carbide content. The effect of the vanadium would, therefore, appear to be to alter the properties of the matrix of the iron and it is possible that free carbide, by retaining the vanadium in the carbide, may in fact be detrimental. Alloy additions, such as nickel and copper, which would help decompose free carbide, might be beneficial in these liners and give improved wear resistance.

TURBINES AND BOILERS

Modern marine turbines operate at temperatures at which creep becomes important and, as can be seen from Table I, this factor must be taken into account or failures may result. Carbon steels and cast irons are used for the lower temperature parts but above about 343°C (650°F) consideration must be given to the use of alloy steel of greater creep resistance.

Various criteria are used for comparing the creep properties of materials, e.g. the stress to produce specified creep deformation at a specified time and temperature, the rupture stress, the

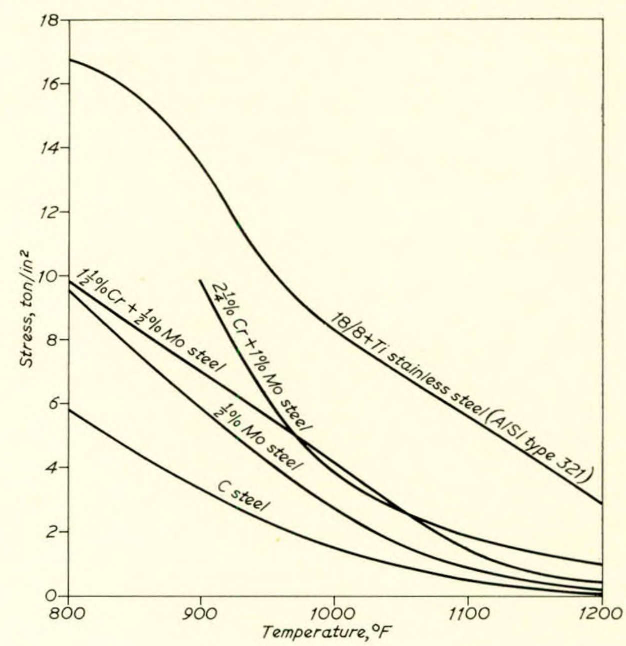

FIG. 3—*Stress temperature curves for creep resisting steels*

TABLE IV—WEAR RATES OF VANADIUM-CONTAINING IRONS

Liner	Carbon	Silicon	Manganese	Sulphur	Phosphorus	Titanium	Vanadium	Wear Rate, mils/1000h	Carbide in structure, per cent
A	3·12	0.72	0·95	0·105	O·16	0·01	0·01	20·5	1·4
B	3·45	1·00	1·03	0·083	0·12	0·01	0·01	34·6	1·1
C	2·98	1·35	0·83	0·108	0·13	0·015	0·08	9·2	1·00
D	2.85	1·36	0·92	0·096	0·10	0·015	0·14	9·0	0·7

ductility at a specified time and temperature. The designer must look at a range of properties around the conditions in which he is interested in order to ensure that the material he selects is not close to its limits of usage. For example, the stress to produce one per cent creep in, say, 100 000 h at a particular temperature may be acceptable for design purposes, but it would be unwise to use the material if its stress rupture ductility under the same conditions was only one per cent as no warning of failure would be apparent from deformation of the material.

The alloying elements used in steel to improve creep resistance are molybdenum, chromium and vanadium added either singly or in combination.

Fig. 3[15] shows typical stress (for one per cent creep in 10^5 h) temperature curves for a range of steels such as are used in marine service. There is considerable overlapping in the range of applicability of these steels so that it is not possible to define exact limits for the use of any particular steel. The austenitic steels, such as shown in Fig. 3, have much better creep properties than the ferritic steels but have not been used very much in marine service as temperatures are usually too low to make use of their high creep strength. They have been proposed for tubes for reheaters[16] and limited use has been made of them in superheaters[9].

Turbine Casings, Valve Chests and Other Cast Components

Steels of the types shown in Fig. 3 are used for these components. These parts are normally subjected to very rigorous non-destructive testing before acceptance and almost invariably weld repairs have to be carried out, thus weldability of the steel is an important consideration and the carbon content is usually kept low.

These parts are usually heat treated by homogenizing at 850-950°C (1562-1742°F) normalizing and finally tempering at 600-650°C (1112-1202°F). Where weld repairs are required, the castings should be stress relieved after welding to avoid the possibility of distortion in service.

Turbine Rotors

Similar steels to those used for turbine casings are used for turbine rotors and shafting. Sometimes more highly alloyed steels are used to take advantage of their higher strength. It has recently been shown that some types of alloy steel are more susceptible to "wire-wool" type bearing failures than others and Dawson and Fidler[17] have shown that chromium and manganese are the most important elements in determining susceptibility to "wire-wool" type failures. Nickel, molybdenum, copper and vanadium did not seem to influence the failure. The acceptable limits in a steel for all these elements have not been determined, but it would seem to be prudent to choose steels with low chromium and manganese contents to use a combination of the acceptable elements to give the desired strength.

Turbine Blading

The most commonly used material for marine turbine blading is stainless iron (0·15 per cent carbon 13·0 per cent chromium). This material has good corrosion resistance in wet steam and its creep resistance makes it acceptable for use up to 900°F (482°C). For higher temperatures, the steel can be alloyed with molybdenum (0·5 per cent molybdenum) or titanium (0·5 per cent) and its use extended up to 1000°F (538°C).

Austenitic stainless steels are also used, e.g. (0·12 per cent carbon, 18 per cent chromium, 11 per cent nickel, 1·2 per cent columbium and 0·2 per cent carbon, 11 per cent chromium, 35 per cent nickel)but, in general, steam temperatures in marine practice do not require the creep strength of these alloys.

High chromium ferritic stainless irons may suffer progressive embrittlement on exposure to temperatures around 475°C (887°F). This phenomenon is referred to as "475°C embrittlement" as it occurs rapidly at that temperature. The 13 per cent chromium stainless irons do not appear to be affected by this phenomenon, but the 17 per cent chromium alloys are and, whilst blades made of this material may perform satisfactorily, any attempts to straighten distortion will result in cracking.

High Temperature Bolting Materials

A feature of marine turbines is the extensive use of bolted joints in the steam piping and valves to enable survey work to be carried out periodically. The materials used are normally chromium/molybdenum and chromium/molybdenum/vanadium steels, but these may fail at high temperatures (i.e. above 900°F) by creep or by embrittlement. Bolts and studs work under different creep conditions to, say, turbine casings. They are initially stretched to a certain stress but at high temperature this stress decreases with time. They are also subject to periodic re-tightening so that the creep relaxation process is repeated throughout their life. If the bolt is frequently re-tightened, as for example a bolt on the cover of a steam strainer, then it will spend a large portion of its life at a high creep stress and creep cracks may form. If it is tightened infrequently, then the percentage of its life at high stress will be much lower and cracking is less likely.

Apart from the possibility of creep cracking, structural changes occur in the alloy and often there is a decrease in notch ductility due to these changes. The author has reported[1] failures in studs to B.S.1501-6 from a valve chest operating at 950°F (510°C). Many studs broke during removal and showed Charpy V-notch values of 3 ft lb. Studs from the steam inlet to the chest, although brittle, showed no creep failures, whereas studs from the cover to the strainer all showed creep cracks, thus illustrating the effect of re-tightening described.

All ferritic steels are liable to show embrittlement with time at high temperature, and the use of high steam temperatures aboard ship is likely to require an improved bolting material. Use has been made of a nickel-base alloy in land turbine practice.

This alloy, Nimonic* 80A, has a coefficient of expansion similar to ferritic steels so that no problem of differential expansion arises on heating and cooling.

Table V based on the work of Buchan *et al.*[18] shows how the expected life of this alloy is influenced by the number of re-tightenings and shows that for an interval between

TABLE V—CALCULATED SERVICE LIVES OF BOLTS SUBJECTED TO SPECIFIC RELOADING CONDITIONS (INITIAL STRAIN 0·15 PER CENT)

Temperature C°	Service life (hours) to accumulate 0·4 per cent total strain at specified reloading intervals			
	1000 h	5000 h	10 000 h	25 000 h
550		70 000	115 000	230 000
565		60 000	100 000	195 000
575		50 000	80 000	175 000
600	11 000	35 000	60 000	125 000
625	9000	30 000	50 000	100 000

TABLE VI—EFFECT OF LONG TERM EXPOSURE AT 500 °C ON THE IMPACT PROPERTIES OF NIMONIC ALLOY 80A

Soaking time at 500 °C (932°F)	Charpy V-notch value in ft lb	
	At room temperature	At 500 °C (932 °F)
0	41	62
30	45	56
100	46	62
300	39	61
1000	34	58
3000	37	—

Sample from 500 °C creep test after 13 725 h (0·15 per cent initial strain)—65 ft lb Q2 Hounsfields Specimen

*Trade mark.

re-tightenings of 10 000 h a life of 115 000 h could be expected at 550°C (1022°F). At temperatures more typical of modern marine turbine practice, e.g. 950°F (510°C), longer life would be obtained for the same interval between re-tightenings.

Table VI shows that the material is resistant to embrittlement, even under stress, for times up to 13 725 hours.

As electron microscopy showed no change in microstructure of the alloy after this test, a much longer life than this could be expected.

High Temperature Scaling

Marine boilers normally operate on heavy fuel oil and this often contains small amounts of vanadium compounds. During combustion the vanadium is oxidized to vanadium pentoxide which may form further compounds with the sodium and sulphur present in the marine atmosphere and in the fuel. These compounds are usually of very low melting point (V_2O_5-660°C) and when molten, are highly corrosive to metal parts. The parts mainly affected are uncooled metal parts in the superheaters, such as baffles, support beams and dampers.

In recent years, much research has been carried out to investigate the behaviour of metals in contact with molten ashes. Because of the complexity of the ash constituents and of the variety of metals used under these conditions, there is no one metallic material which provides absolute resistance to all the possible conditions encountered, but one can generalize and make a reasonable prediction of behaviour for a given application.

In a marine boiler, the ratio of sodium sulphate to vanadium in the ash deposits is likely to be much higher than in a land boiler. This might be expected in view of the salt content of the marine atmosphere which is used for combustion in the boilers.

It has been shown[19] that a high vanadium to sodium ratio is most harmful to austenitic steels of low nickel content, whereas with a high sodium to vanadium ratio (typical of marine conditions) the worst attack is on alloys of higher nickel content. It has also been found that with the lowering of the resistance to corrosion the attack changes from primarily accelerated oxidation to a form of attack which leaves increasing evidence of sulphide formation, due to the alkali sulphate constituent of the ash. Although high chromium nickel-free materials often show good resistance to sulphate ashes, their creep strength is too low to allow their use for any load bearing parts and it is necessary to have materials of high nickel content in order to obtain the creep properties required. It is logical, therefore, to use a combination of high chromium content to resist attack by sulphur with a high nickel content to provide the required strength and also resistance to oxidation if high vanadium to sodium ratio ashes occur. Table VII, derived from results by Lewis[20], shows the effect of increasing the chromium content on the resistance of nickel/chromium steels and nickel based alloys to an artificial fuel ash of the type likely to be encountered in marine boilers. The excellent resistance shown by the 50/50 and 60/40 alloys has led to their widespread use in marine and land boilers and their service performance shows a considerable improvement over alloys with lower chromium content. Although, in severe environments, crucible and field tests show the 60/40 chromium/nickel alloy to have somewhat better resistance to attack than the 50/50 alloy at temperatures above 900°C (1652°F), the 50/50 alloy has adequate resistance for most applications and is usually preferred because of its greater room temperature ductility.

The Properties of Gearing Steels

The gear material should have:

1) high fatigue strength to resist tooth breakage;
2) wear resistant surface to resist normal wear, scuffing and pitting.

Pemberton[21] records that 54 per cent of reported cases of gearing damage concern pitting and scuffing, whilst a further 24 per cent involve tooth fracture.

Chesters[22] has shown that the pitting resistance of a series of gearing materials, increases with their tensile strength. The fatigue strength will increase with the tensile strength also, but because of tooth shape, no gain in tooth fatigue strength is obtained by increasing the strength of the steel beyond about 55 ton/in². Therefore, although the tensile strength of steels could be raised by heat treatment and by the addition of suitable alloying elements, such as nickel, chromium and molybdenum, it would be impracticable to raise the strength to the levels required to give high resistance to surface pitting. A good compromise can be obtained, however, by using a surface hardened gear to give a very high strength surface layer together with a tough core.

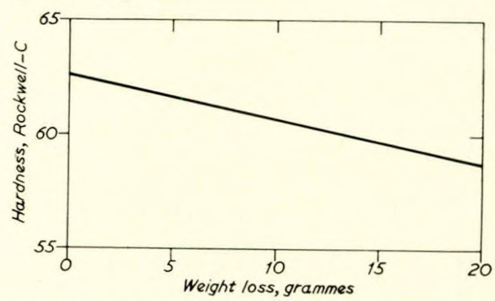

FIG. 4—*Effect of hardness on surface wear of gears*

TABLE VII—ARTIFICIAL FUEL ASH CORROSION TESTS

Tempera-ture, °C	Weight loss in mg/cm² in 300 h in 80 per cent Na_2SO_4 + 20 per cent V_2O_5 ash: 27 per cent chromium steel	8 per cent nickel 19 per cent chromium steel	20 per cent nickel 22 per cent chromium steel	37 per cent nickel 18 per cent chromium steel	20 per cent chromium balance nickel	35 per cent chromium balance nickel	50/50 chromium/nickel	60/40 chromium/nickel
700	21	62	69	105	53	11	1	3
800	37	60	53	1537	70	21	3	5
900	18	139	44	1376	39	53	26	36
1000	45	634	553	1818	1267	55	70	93

Surface hardening, either by case hardening or induction hardening, increases the fatigue strength of steel. The effect is due partly to the increased strength of the surface material and partly to the compressive stresses produced in the surface by these processes. These compressive stresses fall to zero about the case-core boundary, hence the thickness of the case should be sufficient to ensure that the fluctuating Hertzian shear stresses are within the case. Buckingham and Talbourdet[23] claim that the case depth should be at least twice the depth of the maximum shear stress.

Fig. 4[24] shows the effect of surface hardness on wear resistance.

The main requirement seems to be hardness *per se* and Chesters found only a small difference in pitting resistance between induction hardened and carburized steels and showed them to be slightly superior to nitrided steels. However, other tests show that a high carbon content in the carburized surface is beneficial and gears carburized to 1·1-1·3 per cent carbon showed greater pitting resistance than those of similar hardness with 0·7-0·8 per cent carbon. Frederick[25] has pointed out that spalling may occur in a carburized gear if a network of cementite is formed particularly in a high carbon case when additional care would be needed. The same author has also pointed out the desirability of avoiding the use of steels of similar mechanical properties and composition in mating gears as this will increase the susceptibility to scuffing.

To achieve high fatigue strength and prevent tooth breakage, the gear tooth form is important and the maximum radius permitted by the tooth geometry should be provided at the root. The presence of stress raisers such as scratches and machining marks should be avoided if maximum resistance to fatigue failure is to be attained.

Materials for Pinions and Wheel Rims

As the problem of obtaining high strength in a steel part increases rapidly with the size of the part, it is usual to manufacture the pinions from steels with good hardenability and to quench and temper them. The wheel rims are usually made from a lower alloy or plain carbon steel in the normalized condition and these are shrunk on to a cast iron or a fabricated centre.

Up to 1945, the steel most commonly used for pistons was a 3½ per cent nickel steel hardened and tempered to a tensile strength of 40-50 ton/in^2. These were mated with 0·30 per cent carbon steel wheels having a U.T.S. of 30-35 ton/in^2 in the normalized and tempered condition. The use of more highly alloyed steels became essential with the adoption of double reduction gearing and the transmission of higher powers in gear-boxes of reduced dimensions.

En.25 (2½ per cent nickel/chromium/molybdenum) pinions mated with En.8 (0·4 per cent carbon) wheels have for some time been the standard combination for merchant marine gears and both full scale trials and service experience have shown this combination to be highly successful, with particular reference to resistance to pitting and scuffing.

Steels of lower carbon content are used for case hardened pinions and En.36 (3 per cent nickel/chromium/molybdenum) has been used with considerable success. In order to avoid excessive distortion heat treatment after carburizing is restricted to a single quench from 820-840°C (1508-1544°F) followed by tempering at 150-220°C (302-428°F) to give a case hardness within the range 650-775 V.P.N.

Normally some distortion will result from the carburizing and heat treatment processes so that it is necessary to grind the gears after hardening. Too high a surface hardness can result in surface cracking on grinding.

Case depths of 90-100 mils are used but the grinding operation will reduce this slightly.

Nitrided gears are also used. In this case a steel containing alloying elements which form hard nitrides must be chosen. Chromium and molybdenum are the alloy elements normally added for this purpose. The distortion resulting from the nitriding process is much smaller than that produced by carburizing and it is often possible to omit any grinding after nitriding according to the gear size. Epicyclic gears which are now widely employed in marine turbines, use nitrided sun and planet wheels. Case depths are much shallower than for carburized gears, 25 mils being a normal value with a surface hardness of about 700 V.P.N.

Wheel rims are usually made of carbon steels or low alloy steels in the normalized condition. Two commonly used materials are En.8 (0·4 per cent carbon, ultimate tensile strength 40-45 ton/in^2) and En.9 (0·55 per cent carbon, ultimate tensile strength 45-50 ton/in^2). Both these materials can be induction hardened.

As wheel rims, and in particular the main rim, may require large forgings, there is a possibility of ingot defects such as a segregation appearing in the finished forging. It is now usual to sulphur print the radial and circumferential faces of these forgings to ensure freedom from ingot defects.

CORROSION

Corrosion is a factor which has to be considered in nearly all marine engineering components. The cost of corrosion to the shipowner cannot be overemphasized. Recent surveys, quoted by Kenworthy[26], have shown that at least 40 per cent of ship repair costs are attributable either directly or indirectly to corrosion damage.

Whilst corrosion in boilers, air heaters, fresh water and lubricating oil systems is often a problem, sea water corrosion is of particular interest to the marine engineer and this section will be confined to a consideration of corrosion of metals in sea water.

TABLE VIII—E.E.S. EROSION TESTING APPARATUS

Tip velocity 28 ft/s
Water temperature 30°C
Duration of test 60 days

Material	Natural sea water I.P.Y.	Synthetic sea water I.P.Y.
70/30 copper-nickel (low iron)	0·037	0·007
70/30 copper-nickel (high iron)	0·005	0·001
"G" bronze (88/10/2 copper/tin/zinc)	0·046	0·005

Many factors influence corrosion in sea water, but the main factors of interest to the marine engineer are velocity effects and galvanic effects. Stress corrosion cracking, dezincification and graphitization are often encountered in marine corrosion and will also be considered.

In using data from corrosion tests, it is important to know whether the tests have been carried out in once-through sea water taken directly from the sea; recirculated sea water, which may have been stored for a considerable time and has become biologically inactive or synthetic sea water. Table VIII illustrates the difference between natural and synthetic sea water.

Most of the data quoted in this paper are taken from work carried out at a corrosion testing laboratory with which the author's company is associated at Harbor Island, North Carolina and refer to tests carried out in water taken directly from the sea.

Effects due to Water Velocity

The effects of velocity on corrosion behaviour in sea water can be summarized as follows:

1) Local corrosion cells may be set up on the metal surface by differential aeration effects. At low velocities, i.e. 0-3 ft/s, these can continue to function and pitting may occur. Slight motion tends to make the environment more uniform thereby reducing local attack.

   Marine fouling may occur and give rise to oxygen concentration cells. The metal beneath the growth is deprived of oxygen and will become anodic to the surrounding metal exposed to aerated sea water. This can cause severe pitting particularly on the more noble alloys which depend on surface films for their nobility.

2) At moderate velocities, i.e. 3-20 ft/s., quiescent layers at the metal surface are removed and marine fouling organisms can no longer settle on the surface. The corrosive medium in contact with the surface is continously renewed and local cells tend to be eliminated. In the case of carbon steel, the corrosion rate increases with velocity, but in the case of metals which form a passive film, such as stainless steels, the surface films become more protective. On copper alloys, above a certain critical velocity, the protective films are stripped away and a very rapid attack, known as impingement attack, occurs.

3) At higher velocities, i.e., 20 ft/s and upwards, only those alloys such as stainless steels, which can form protective films under these conditions, have low corrosion rates. In most other alloys, the corrosion rate increases with velocity.

TABLE IX—EFFECT OF VELOCITY ON THE CORROSION OF METALS IN SEA WATER

Alloy	Quiet sea water 0-2 ft/s		27 ft/s	120-140 ft/s
	Average corrosion rate in mils/year	Maximum pitting, mils	Corrosion rate, mils/year	Corrosion rate, mils/year
Carbon steel	3*	80	—	180
Grey cast iron	22* (graphitized)	196	176	530
Admiralty gunmetal 88/10/2 Cu-Sn-Zn	1·1†	10	35	43
85/5/5/5 copper tin zinc lead	0·7†	13	72	53
Mn-Al-bronze (B.S.1400 CMA 1)	—	—	21	63
Austenitic cast iron type AUS 101 b	0·8§	nil	8	39
Ni-Al-bronze (B.S. 1400 AB2-C)	2·2‡	45	9	39
90/10 cupro-nickel alloy with iron	<1*	8	18 (cast alloy with 0·84 per cent iron)	33 (Average 12 tests)
70/30 Cu-Ni alloy with iron	<1*	10	5	59 (Average 9 tests)
18/10/3 chromium/nickel/molybdenum stainless steel	<1*	72	<1	0·2
Nickel-copper alloy 400	<1*	52	0·6	0·4
Nickel-copper alloy K500	<1*	55	0·5	0·4
Nickel-chromium-molybdenum-copper-iron alloy 825	<1*	1	<1	0·3
Titanium	<1*	nil	<1	0·2

*3-year test at Harbor Island, N.C.

†42-month test at Freeport, Texas

‡442-day test at Kure Beach, N.C. Alloy 10·6 per cent aluminium, 2·5 per cent iron, 5 per cent nickel, 0·75 per cent manganese

§6 year test at Kure Beach, N.C.

All the above data are taken from actual test results and are thus not exactly reproducible. This is particularly true of the maximum depth of pitting which may vary widely from test to test.

Cavitation conditions can easily arise under these velocity conditions and corrosion rates in the area of cavitation can be extremely high.

Table IX gives a summary of corrosion behaviour of a number of materials used in marine engineering.

Examination of these data shows that the alloys can be divided into four groups as follows:

1) alloys which have appreciable corrosion rates and pitting rates in quiet sea water and whose corrosion rate increases with velocity e.g. carbon steel and cast iron;
2) alloys such as the gunmetals and cupro-nickels which have low general corrosion rates and pitting rates in quiet sea water;
3) alloys such as 18/10/3 stainless steel which have a low general corrosion rate under all velocity conditions, but which are liable to pit severely in quiet sea water;
4) alloys with high resistance to sea water corrosion under all velocity conditions e.g. titanium and nickel/chromium/molybdenum alloys.

The corrosion of carbon steel in sea water is usually governed by the availability of dissolved oxygen and its diffusion to the corroding surface. In normal aerated sea water in fully immersed conditions and low velocities, the corrosion

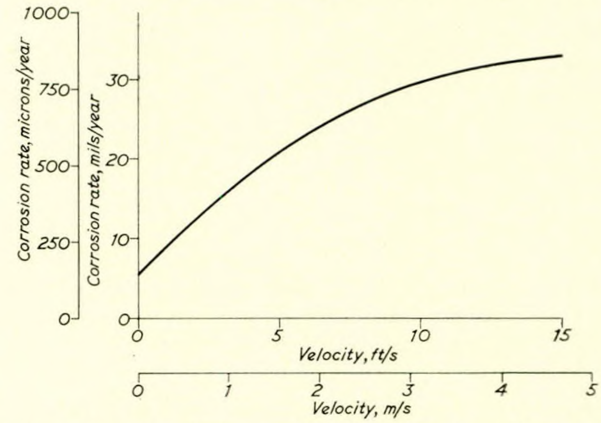

FIG. 5—*Effect of velocity on the corrosion of steel in sea water*

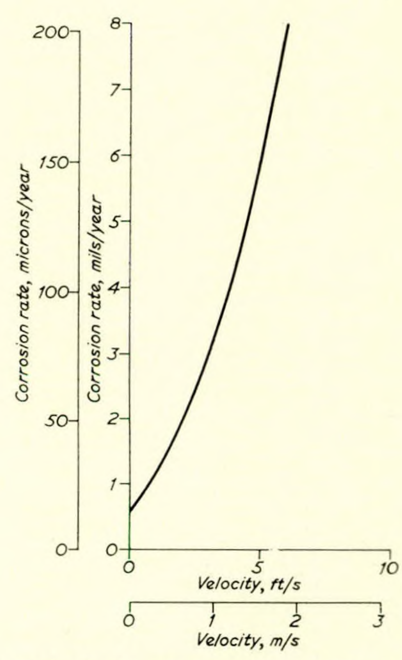

FIG. 6—*Effect of velocity on the corrosion of zinc in sea water*

rate is usually in the range 3-6 mils per year. Cast iron behaves in a similar way to steel, although where graphitization corrosion occurs, corrosion rates can be considerably higher.

If the sea water is flowing past the steel surface, the supply of oxygen is improved and the corrosion rate increases. Fig. 5[27] shows the variation of corrosion rate with velocity. At 10 ft/s the corrosion rate is 30 mils/year, compared with 6 mils/year under static conditions.

Galvanizing will reduce the corrosion of steel in sea water, but as Fig. 6[28] shows, its effect will only be of short duration in flowing sea water. Thus, the use of galvanized steel or cast iron in flowing sea water can only be tolerated where large corrosion allowances can be provided.

Heat Exchanger Tubing and Piping Velocities—(4-12 ft/s)

An important velocity range missing from Table IX is that relating to heat exchangers, i.e. 4-8 ft/s. Impingement resistance is of prime importance in heat exchanger tubing

TABLE X—COMPARISON OF RESULTS OF TESTS OF CONDENSER TUBE ALLOYS IN THE BRITISH NON-FERROUS METAL RESEARCH ASSOCIATION LABORATORIES AND AT HARBOR ISLAND

Testing Conditions

Velocity of jet, 15 ft/s
Air added, 3 per cent by volume
Duration, 28 days
Water recirculated at British Non-ferrous Metal Research Association
Not recirculated at Harbor Island

Material	Average depth of attack (mils)	
	B.N.F.M.R.A.	Harbor Island run 12
Arsenical Admiralty brass	13·5	11·0
Asenical copper	12·0	—
70/30 Cupro nickel, 0·04 per cent Fe	4·5	9·0
Aluminium brass	1·6*	8·0
70/30 Cupro nickel, 0·8 per cent Fe	0·8	4·0†
90/10 Cupro nickel, 2 per cent Fe	0·0	6·0

*One specimen out of 20 pitted to a depth of 26 mils. No other specimen greater than 8 mils.
†Iron content, 0·45 per cent.

alloys and these alloys are usually compared on this basis, although other factors have also to be taken into account.

Only three alloys are in general use for these tubes, viz. aluminium/brass, 70/30 cupro/nickel and 90/10 cupro/nickel. The cupro/nickels contain small amounts of iron as this has been found to improve their impingement resistance.

Table X[29] compares data obtained from tests in recirculated and natural sea water and also illustrates the beneficial effects of adding small amounts of iron to 70/30 cupro/nickel. While these data show that the attack in natural sea water is, in general, much more severe than in recirculated sea water, it should be noted that the B.N.F. jet test using recirculated sea water has provided a reliable guide in establishing the relative merits of condenser tube alloys. The increased severity of the test, when using natural sea water, may disguise the difference in alloy behaviour which may be significant.

Impingement resistance is of prime importance in choosing a tubing material, but other factors must also be considered, notably the effect of polluted sea water on the behaviour of the alloy. All the data presented above relate to tests in clean oxygenated sea water. Sea water normally has about 8 ppm of oxygen dissolved in it, but if the water is contaminated with excessive amounts of organic matter, either of natural origin or from sewage effluent, then the decomposition of this matter will cause a reduction in the oxygen content of the sea water. In some cases complete de-oxygenation occurs and hydrogen sulphide and organic sulphide compounds can form in the sea water. The presence of these substances in the sea water changes the nature of the protective film formed on copper-base alloys and severe corrosion or impingement attack may occur. As the polluted environment varies, it is difficult to present corrosion data which are meaningful and which rate materials in relation to their resistance to polluted conditions. General experience in service conditions is a more reliable guide and indicates that 70/30 cupro/nickel is more resistant than 90/10 cupro/nickel under polluted conditions and that both these alloys perform better than aluminium brass. This, however, is not invariably true, and in some cases aluminium brass may perform better than the other alloys.

The effects of polluted sea water can be reduced in shipboard coolers by ensuring that units are drained when not in use in port. Those units which have to operate in port should perform satisfactorily for considerable periods as the protective

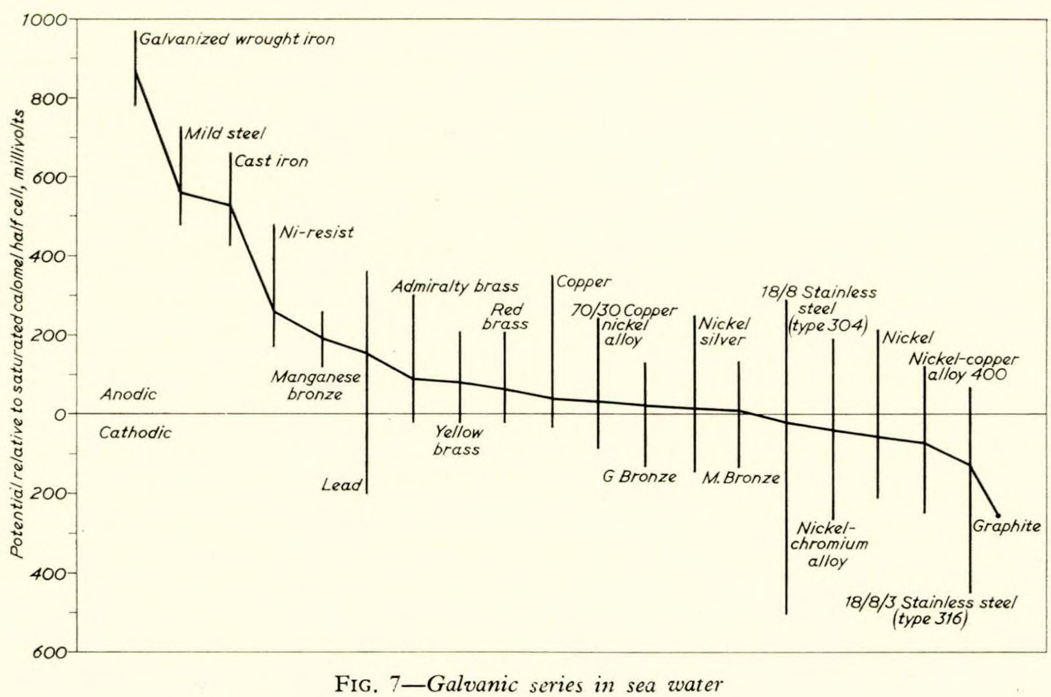

FIG. 7—*Galvanic series in sea water*

film formed in clean sea water will continue to give protection even under polluted conditions.

Galvanic Effects

Galvanic corrosion in marine engineering is well known and has been recognized since the days when the copper sheathing on wooden ships caused corrosion of the iron nails fastening the sheathing to the hull.

It is often necessary to use different materials, in contact, in sea water and care must be taken to ensure that the galvanic corrosion which may occur is acceptable. This can usually be done by choosing materials which have similar electrode potentials in sea water, by making the key component of a more noble metal and by providing a large area of the less noble metal so that, although its corrosion is increased, it is spread over a larger area. Fig. 7[30] shows a galvanic series in sea water for a number of materials commonly used in marine engineering. Of particular interest in this series is the position of graphite. Graphite is often used in packings and greases and may cause severe corrosion of the metal in contact with it. It also occurs on the surface of cast iron which has suffered graphitization and may stimulate corrosion on alloys which one would normally expect cast iron to protect.

Stress Corrosion Cracking

Some materials when used in marine environments are susceptible to a type of failure known as stress corrosion cracking. This occurs under certain conditions of stress and corrosive environment and is due to their conjoint action. In marine engineering the cracking is most commonly observed on brasses, although it may also occur in aluminium alloys, high strength steels and stainless steels.

Table I shows that stress corrosion cracking is an important cause of failure in marine engineering and precautions must therefore be taken. Most of the failures given under this heading in Table I occurred in tubing and were usually due to internal stress in the tubing, although failures can also be produced by applied loads[31]. Breckon and Baines[32] have shown that it is possible to draw tubes so that they are crack resistant, even though they contain high internal stresses, provided that the stresses on the surface of the tube are compressive.

The usual test for stress corrosion cracking tendency in a tube is the mercurous nitrate test and this is specified for copper-base alloy tubes. Whilst this test provides a useful check for susceptibility to stress corrosion cracking, it is unfortunate that tubes which pass the test can still sometimes crack in service. The reason for this appears to be that the substances in sea water which produce the cracking, usually assumed to be ammonia compounds, can cause cracking at a lower stress level than mercury. Where failure occurs in tubes which have passed the mercurous nitrate test, it is usually after several years service, and samples cut from cracked tubes will usually crack in ammonia vapour, although they will still pass the mercurous nitrate test, thus showing that the stress level has not increased in service.

Stress corrosion failures have also become troublesome in recent years in propellers. Heat is often used to enable a propeller to be drawn from a shaft and, provided the heating is carried out slowly and uniformly, no trouble will result. However, the use of high intensity heat sources, such as compressed air/paraffin or compressed air/propane flames, can lead to overheating of local areas on the propeller boss which are left in a state of high internal stress when the boss cools down. This can lead to stress corrosion cracking when the propeller is re-immersed in sea water. Failures of this type have been reported in high tensile brass and manganese/aluminium/bronze alloys.

As the conditions to produce stress corrosion cracking require a combination of tensile stress, corrosive environments and susceptible material, it is possible to avoid cracking by altering one of these factors. The corrosive environment cannot usually be changed, but the stress can often be reduced, e.g. by stress relief heat treatment to reduce internal stresses; and materials resistant to cracking in the environment can be used, e.g. the cupro/nickel alloys for tubes and piping and nickel/aluminium/bronzes for propellers.

Dezincification

Brasses containing up to 37 per cent zinc are subject to dezincification unless inhibited by additions of arsenic. Brasses so inhibited are normally immune to dezincification and this type of failure is now very rare in single phase brass alloys.

Brasses with over 37 per cent zinc have a two-phase microstructure and are prone to dezincification. Arsenic is not effective in inhibiting dezincification of these alloys, although the addition of tin will reduce the severity of the attack.

The mechanism of dezincification is believed to be dissolution of the alloy and precipitation of the copper from the corrosion product. This results in a porous mass of copper which occupies the same space as the original alloy. Thus the metal part does not change appreciably in section, but the copper, being porous and weak, causes a severe reduction in strength, particularly as the dezincified layer is often thick. This type of attack is commonly found on Muntz metal, naval brass, manganese bronzes and high tensile brasses used for such parts as tube plates, valve stems and pump shafts. A thin layer of dezincification on a tube plate is often acceptable and will only give rise to difficulties if, for example brazing is required for fixing tubes. Some aluminium bronzes are prone to a similar type of attack called de-aluminification. This can be avoided by adding 4-5 per cent nickel to the alloy, although difficulty may still occur in the heat affected zone of welds.

Denickelification is a similar phenomenon found in the cupro/nickel alloys, but it is rare in marine environments when it is usually associated with high operating temperatures and low water speeds. If the water speed is maintained at normal levels, i.e. over 6 ft/s, the trouble can be avoided.

Graphitization

Cast iron contains about 3·5 per cent carbon together with silicon and other elements. Its microstructure consists usually of graphite flakes in a metallic matrix and in sea water the matrix corrodes leaving the original graphite network intact. The surface of the metal often appears free from corrosion, but scraping with a tool will reveal a thick layer of graphitic material. This type of attack often occurs on cast iron in sea water and the layer of graphite formed, being highly cathodic, can produce corrosion on surrounding metals.

CHOICE OF MATERIALS FOR THE COMPONENTS IN A COOLING SYSTEM

A typical ship's cooling system contains valves, piping, heat exchangers, filters and pumps. When in use, water flows through the system, but when not in use, the system may be left full of sea water and corrosion in static conditions must also be considered.

Heat Exchanger Tubing

The main factors governing the choice of heat exchanger tubing have already been considered. In clean sea water and at moderate velocities (i.e. up to 8 ft/s) the three commonly used alloys, viz. aluminium brass, 90/10 cupro/nickel and 70/30 cupro/nickel all perform well. Failures under these conditions are rare and are usually due to partial obstruction of the tubes, for example, because of barnacle growth or pieces of rust and scale. If polluted conditions are expected, or where great reliability is required, the cupro-nickel alloys are preferred.

Tube Plates

Tube plates are usually of heavy section so that some corrosion can be tolerated. The most commonly used materials are Muntz metal (60/40 copper/zinc) and naval brass (60/39/1 copper/zinc/tin). Although both alloys dezincify in sea water, this is not normally a problem. The tin content of naval brass reduces its rate of dezincification to about one third of that of Muntz metal. Aluminium brass and 90/10 cupro-nickel are also used as tube plate materials and the latter is used where tube to tube plate welds are needed.

Water Boxes and End Covers

Ferrous materials, such as grey cast iron, spheroidal graphite iron and fabricated steel are the most commonly used materials for these components. As can be seen from Table IX, the corrosion rate will be high unless the water box is protected and the usual protection provided consists of non-metallic coatings, such as paints and plastic or rubber coatings. The performance of these coatings is variable and if the coating contains pin holes, failure can be very rapid.

Water boxes should have adequate depth and be of such a shape to provide a uniform flow of cooling water to all tubes. All turbulence raisers should be avoided and where protector plates are provided care should be taken to ensure that they do not cause undue turbulence. Venting of the water box to remove excess air from the water is also desirable.

A ferrous water box normally provides some cathodic protection to the tube plate and tube ends and its corrosion provides a supply of iron salts in the water which is beneficial in the formation of protective films on the tubes. Therefore, when the water box is coated, protection plates, preferably of iron or steel, should be provided in the inlet water boxes. Zinc protection plates may be used in the outlet water boxes[33].

Non-ferrous water boxes are sometimes used, and these are usually of gunmetal. Whilst this performs well, it is expensive, and steel protection or steel spray coatings should be provided as in coated water boxes. A recent development has been the use of water boxes fabricated from relatively thin 90/10 cupro-nickel strengthened externally with steel ribs to provide the corrosion resistance of a copper-base alloy more economically than in a relatively thick casting.

Pumps

Water velocities in pumps are much higher than in heat exchangers so that the corrosion behaviour of materials at high water velocities is important. The casing is usually of intricate shape and is made by casting. As it is of relatively heavy section, some corrosion can be tolerated and casting alloys, such as gunmetal, Ni-Resist iron, aluminium bronzes and cast cupro-nickels, can be used.

Where relatively light sections are needed (e.g. for the impeller), materials such as stainless steel (18/10/3 chromium/nickel/molybdenum) and Alloy 400 (70 nickel/30 copper) are indicated. These alloys are more noble than the casing materials (see Fig. 5) and they will receive some cathodic protection from them. As their area is less than that of the surrounding metal, they will not unduly increase the corrosion on adjacent parts. Gunmetal and aluminium bronzes are also used, but are likely to suffer erosion if the water velocity is high.

The pump shaft requires high strength and corrosion resistance and materials such as nickel/copper alloy K 500 or 400 are suitable. Similar materials can be used for wear rings.

Valves and Filters

The requirements for valve and filter bodies are similar to those for pump casings and the same types of casting alloy are used.

Cast iron and steel valves are sometimes used, but their life expectancy is short, unless made of massive section. Spheroidal graphite iron is used for some large valves as its higher strength and shock resistance is of advantage.

Various copper-base alloys are used for valve trim, viz. brass (in cast iron and steel valves), nickel/copper alloy 400 (for bronze valves). 18/10/3 chromium/nickel/molybdenum stainless steel trim is usually used for austenitic cast iron valves. The protection afforded to the stainless steel by the valve body is sufficient to prevent pitting attack and this combination performs well in service. Austenitic cast iron, with nickel/copper alloy 400 or K 500 discs, is also used for butterfly valves.

Valve stems are normally made from wrought materials, such as naval brass, aluminium bronze and stainless steel (18/10/3) also nickel/copper alloys 400 and K 500. Naval brass may fail by dezincification in less than two years and should be avoided in valves designed for long life.

Piping

The various parts in a cooling system are connected by piping, often of large diameter. Steel provides piping at the lowest initial cost but, as can be seen from Figs. 5 and 6, the life of steel piping is likely to be short and, as pipe repair costs are high, this material is probably the most expensive one to use over the life of the ship.

Non-ferrous materials such as copper, aluminium brass and 90/10 cupro-nickel are used for sea water piping. The main requirements for this application are:

1) high resistance to impingement attack;
2) high resistance to pitting in static sea water;
3) ease of fabrication and welding.

Copper has low impingement resistance and if copper piping is used the water velocity must be kept low: about 4 ft/s is the maximum which this material will accept.

Aluminium brass and 90/10 cupro/nickel can be used at much higher velocities; 8 ft/s and 10 ft/s respectively, are the design velocities widely used. This means that smaller pipe sizes can be used and this makes the use of copper uneconomic for large pipes. Aluminium brass and 90/10 cupro/nickel are

TABLE XI—PROPERTIES OF PROPELLER ALLOYS

Material	Tensile strength ton/in^2	Yield strength ton/in^2	Elongation on 2-in per cent	100 million cycle endurance limit in sea water* ton/in^2	Cavitation resistance in high velocity jet cavitation test at 120–140 ft/s		Weldability	Stress corrosion properties in sea water
					Corrosion rate mils/year	Maximum cavitation attack mils		
Cast iron	13 to 22	—	nil	—	480	—	Poor	No failures recorded
Manganese bronze	29	14	20	$\pm 2\cdot7$	58	2	Moderate	Stress corrosion cracks
Cast stainless steel Type 304L	31	13	35	$\pm 2\cdot9$	0·1–0·4	1–5	Excellent	Laboratory tests indicate susceptibility but no failures recorded
Ni-Al bronze (B.S. 1400 AB2-C)	38	16	15	$\pm 5\cdot6$	13–19	1	Moderate	No failures recorded
Mn-Al bronze (B.S. 1400 CMA-1)	45	18	20	$\pm 4\cdot0$	54	3	Good	Stress corrosion cracks

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both used in merchant ships and the pipe costs are similar when designed to the velocities given above. The good weldability of 90/10 cupro/nickel and its high resistance to stress corrosion cracking and polluted water conditions have produced a growing use of this material.

In designing and manufacturing piping systems, turbulence raisers, such as close radius bends, misaligned flanges and partially throttled valves should be avoided wherever possible. Where silver brazing alloys are used for making joints in piping systems and where these joints are in contact with sea water, the silver content of the brazing alloy should be at least 50 per cent. Alloys of lower silver content may fail by dezincification and such failures can be very rapid in the corrosive conditions developed in the narrow crevices which the brazing metal normally occupies.

PROPELLERS

The material requirements of propellers vary with their size and the type of service in which they are to be used. For example, a highly powered oil tanker steaming for most of the year at full power will have different propeller requirements to a vessel of low power steaming for relatively few hours per year. Thus the former will require a fully machined propeller made of a high strength alloy to enable thin blade sections to be used so as to give high efficiency, while the latter may only require an as-cast surface in a relatively cheap material.

The desired properties of a propeller alloy are high tensile and proof strength, good corrosion resistance (including stress corrosion and corrosion fatigue resistance) good castability, weldability and machinability. Unfortunately, not all of these properties can be obtained in one alloy at a price acceptable to the shipowner and various materials are used which offer different combinations of these properties. Table XI is a summary of some of the more important properties of propeller alloys.

For the largest propellers, alloys of the aluminium bronze type are almost universally used as these materials give high strength and high corrosion resistance with good casting properties. They are, however, relatively expensive alloys and for smaller propellers cheaper materials can safely be used, e.g. manganese bronze alloys, stainless steels and sometimes cast irons.

Austenitic stainless steels have not been used very extensively in Europe for propellers, but they are widely used for coastal and river vessels in the U.S.A. The main reason for their use is their excellent weldability and repairability which enable propellers exposed to physical damage to be repaired and kept in service for many years.

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Discussion

Dr. P. T. Gilbert (Member) said that in so far as the matters with which Mr. Todd dealt were within his competence to judge, he was in agreement with almost everything.

In the section on stress corrosion cracking, it was stated that the majority of the 25 cases of stress corrosion failures referred to in Table I were due to internal stress in the tubing. Could Mr. Todd give a little more detail on the type of tubing and the mode of failure? It would be surprising if such a large number of failures were due to residual manufacturing stresses, as seemed to be implied. Residual manufacturing stresses were usually a problem only in hard drawn condenser or heat exchanger tubes that were required to be fixed by packing at one or both ends. No significant residual stress remained in tubes that had been heat treated as a final operation to soft or half hard temper (whether condenser or heat exchanger tubing which was to be fixed by rolling-in, or pipelines that were required to be suitable for bending).

It was true that hard tubes could be drawn so that they contained compressive stresses and therefore were not susceptible to stress corrosion cracking. Also the stresses in such tubes could be changed to dangerous tensile stresses by certain straightening operations, and it was therefore important that a stress relief heat treatment be applied after such straightening processes. Tubes could fail in ammoniacal environments after passing the standard mercurous nitrate test but most tube manufacturers were aware of these matters, and although cases of failure due to residual stresses were encountered, they were rare. In his experience in the marine field most of the stress corrosion failures that had occurred in copper alloys had been in pipes that had not been adequately stress relieved after cold bending, or had been strained into position during installation. Would the author describe such failures as being due to "internal stress" or to "applied loads?"

He asked for some clarification of the terms used to describe the various bronze alloys referred to, since this could cause confusion. It was generally accepted that the term "manganese bronze", as given in Fig. 7 and Table XI, referred to a material which was in fact a two phase brass of fairly high tensile strength with relatively small additions of elements such as manganese, aluminium, iron, tin, etc. Being essentially two phase brasses, such materials were susceptible both to dezincification and to stress corrosion cracking. The term "aluminium bronze" was, on the other hand, normally used only for binary copper aluminium alloys containing no zinc, and whilst such alloys could suffer selective corrosion or stress corrosion cracking under some circumstances, they obviously could not dezincify. The alloys referred to in Tables IX and XI as "nickel aluminium bronze" were presumably aluminium bronzes, as defined above, with substantial additions of nickel. Was the alloy, referred to in Table IX as "manganese aluminium bronze", a manganese bronze as defined above with an aluminium addition, or an aluminium bronze with a manganese addition? It could be a copper-manganese-aluminium alloy of the type covered by designation CMA1 in B.S. 1400 and this same specification could cover the material referred to in Table XI as "nickel aluminium manganese bronze". Some appropriate specification or nominal composition should be quoted referring to a bronze alloy.

In view of a recent report from the Battelle Institute that both unalloyed titanium and the common titanium alloys could suffer severe crevice corrosion in hot sea water, was Mr. Todd being too sweeping in including titanium in the category of alloys with high resistance to sea water corrosion under all velocity conditions?

Finally, commenting on Table X, the author said: "The attack in natural sea water is, in general, much more severe than in re-circulated sea water." This was only true of the materials of high resistance. One interpretation of the results was that the conditions in the Harbor Island once-through test did not allow the formation of protective films even on the materials of high resistance and this test was, therefore, of limited use for evaluating alloys that formed protective films under normal service conditions.

Mr. S. H. Frederick said that the author referred to the importance of economic factors in material selection and of first cost. For the shipbuilder, first cost was of prime importance when fighting to obtain an order. Many shipowners with a small number of ships did not specify in detail the materials to be used for the multitude of components required, and in such cases the builder could not always afford to use expensive materials, even though he knew that they would last the life of the ship. His first job was to secure the order, and having done so he must make a profit. In such circumstances he could not be expected to install, for example, a copper nickel salt water piping system in place of one fabricated from galvanized mild steel. Many shipowners were not prepared to pay the extra cost of materials that would last the life of the ship, even though they could be proved to be more economic. His particular method of financing might make a low first cost more attractive than low maintenance costs. In the case of salt water piping, there were many shipowners who could not accurately estimate what their annual maintenance costs were.

With regard to structural components, Mr. Frederick was pleased to see attention drawn to the low fatigue strength of welded fabrications where the welds were undressed. The presence of severe porosity had virtually no effect on the fatigue strength of such welds and too much time and money was expended on eliminating porosity. It was another matter in the case of dressed and polished welds, but these did not exist in ships' structures.

Referring to the author's comments on gearing steels, he suggested that in addition to the two property requirements listed at the beginning of the section, a third and important requirement should be added—the materials selected for mating gears should be compatible. It was well known that resistance to scuffing was largely dependent upon material compatability. Experience had shown it to be inadvisable to mate through-hardened gears when both gears had been made from nickel-containing steels, regardless of the fact that they might be of different hardness. The reason for this was not clear, but there was no doubt that such gears would be prone to scuffing and this had been proved in rig tests and in service.

In the case of materials for propellers, low alloy steels containing up to two per cent nickel had found favour with some shipowners where their ships operated in ice laden waters owing to their greater resistance to damage and relative ease of repair. If such propellers were faced with austenitic stainless steel on the suction side of the blades and if fitted with impressed current cathodic protection, they could be successfully used as working propellers. He had examined such a propeller after five years' service and found it to be in excellent condition.

MR. A. D. LAMB quoted the author's statement that the surface temperature of the combustion zone of a cylinder liner was normally too low to initiate cracking due to residual thermal stress. However, in his experience cracking of this type began to be troublesome if surface temperatures exceeded 350°C (662°F). In the range 350°C-400°C (662°F-752°F) cracking began to be a serious problem, and above 400°C (752°F) it was almost inevitable. Most engines were designed so that the surface temperatures in the combustion zone should not exceed about 300°C (572°F) but a great deal depended upon the efficiency of the water cooling in that region, if the flow was restricted and film boiling took place, very high temperatures could be reached on both the flame and water sides, and in these circumstances rapid failure usually occurred. This also applied to piston crowns if the coolant flow was restricted or there was insufficient total flow in the areas of high heat flux. Investigation was at present being undertaken at B.C.I.R.A. on the effect of directed water flow, and the results so far indicated that much could be done to reduce hot face temperatures by means of higher coolant throughput and directed flow by means of internal jets.

In his opinion the deterioration of cylinder liners was about fifty per cent wear and fifty per cent corrosion. He agreed with the author that vanadium additions to an iron of the type used for large bore Diesel cylinder liners might modify the bearing properties of the matrix pearlite in some way, but it also produced a different form of carbide from an iron which contained vanadium at trace levels. He himself had no evidence to suggest that the carbides present in vanadium-containing irons were detrimental to the wear life. His interest in cylinder liner wear began some years ago when it was indicated that vanadium-containing irons gave greatly improved wear lives irrespective of the type of lubricating oil used. A recent study of the wear lives of a large number of liners had failed to confirm this and it became more clear that chemical composition of the liner was one of the least important factors in the life of a cylinder liner. Evidence suggested that the way in which the engine was operated was by far the most significant factor in determining the life of a liner.

At the moment B.C.I.R.A. was co-operating with one of its members who had produced a number of cylinder liners to a controlled chemical composition and microstructure and also B.S.R.A. who had arranged for the liners to be installed in a tanker. In addition to controlling liner composition, all the piston rings to be run against the liners were to be produced in a known chemical composition and B.S.R.A. was to supervise the installation of those components and the running of the engines; it was hoped it would be possible to obtain reliable information of the lives under known conditions and to determine the most significant factor in the life of a liner under those particular conditions.

MR. G. P. SMEDLEY said that the author had attempted to classify the types of failure of items of machinery which he had investigated during a five year period. Such statistics reflected service conditions rather than any inability of material to withstand them. Materials were reasonably tolerant to static loading and failures were usually few under that condition. One surprising feature of the data was the fairly high proportions of failures by stress corrosion cracking, 25 in 171. The proportion was high considering the various other causes from which failures could arise.

Repeated or cyclic loading could search out weaknesses of design, manufacture and construction. Moreover, fatigue failures could occur at very low stresses under fretting corrosion, sea water corrosion, or by crack propagation from flaws and other very sharp notches. Fatigue failures were expected to predominate, therefore selection of material had only a minor influence.

It was surprising that the author neglected cases involving accelerated corrosion. These were a significant proportion of the total failures of components during service and should exceed the last two items listed in Table I. In the consideration of propeller materials, the author gave the results of corrosion fatigue tests of 100 million cycles duration. The results for manganese bronze, stainless steel and aluminium bronzes were lower than manufacturers of the materials had led them to believe. The values were more in keeping with service experience and the order of merit was about right. It was assumed that there was no significant wastage of the test pieces during the corrosion fatigue tests. In this respect had the materials behaved as expected? With reference to stress corrosion cracking, aluminium bronzes might suffer such failure in a severe moist sulphide atmosphere. Table XI indicated, however, that nickel aluminium manganese bronze was particularly susceptible to stress corrosion cracking in normal marine and other atmospheres and might have failed from such cause during service. Details would be appreciated therefore of the tests or service data used for the assessment of that material.

Reference was made to the strength of welded steel structures. In general such structures displayed poor fatigue strengths. This was because of the qualities and types of weld frequently used in the construction. For success, it must be recognized that full penetration welds made under good workshop conditions were essential. Fillet welds must be kept to a minimum. Where used, they should have good leg lengths and be finished to a smooth contour.

Experience showed that any weld made under restraint was liable to be weak in fatigue. Such welds were more prone to defect, and, unless stress relief heat treated, could contain high residual tensile stresses. While such stresses could be relieved by overstrain, they reduced the strength for large numbers of cycles, the high mean stress being a vital factor. There was adequate evidence to indicate that surface cold working of annealed weldments by controlled shot peening, for example, could raise appreciably the fatigue strength. The advantages were so marked that further research appeared to be justified to study the implications and techniques of surface cold working of weldments intended for applications involving cyclic stress.

The matter of "size effect" in fatigue and its effect on crankshaft design had been discussed thoroughly at a meeting held at the Institute a few weeks ago. The large sizes of marine crankshafts were of built or semi-built construction. Restriction on the tensile strength of the material was not due to the limitations on the notch fatigue strength of steel but entirely to the weakness of the shrink fit in regard to fretting fatigue and to limitation because of "slipping torque". With crankshafts for medium speed Diesel engines which were generally of solid manufacture, increase in fatigue strength was extremely important without any change of geometry, and engine builders were raising the tensile properties of the steels to obtain almost a direct increase in working stress which permitted a marked increase in the output of the engine.

MR. J. NEUMANN, B.Sc. (Associate Member) said the author had referred to the effects due to water speeds in the low region, 0-3 ft/s and in particular to the two actions which occurred: the local corrosion cells set up by differential aeration effects, and marine fouling, both of which resulted in pitting. The advice seemed to be to keep water speeds through tubes above 3 ft/s. In the design of tubular heat exchangers, particularly those with oil on one side and water on the other, it was unrewarding to push the water speed up from aspects of heat transfer. Therefore one would like to run the water through the tubes at about 2 or 3 ft/s. The lower water speed limit for pitting seemed to be somewhat vague, could the author please enlarge on it? Was it a sharp drop-off? Was everything bad up to 3 ft/s and fine from 3·1 ft/s onwards, or was it a gradual change from bad conditions at 1-2 ft/s to reasonable conditions at 3-4 ft/s?

Mr. Neumann asked about the reference to fabricated water boxes, made of 90-10 copper nickel alloy, and reinforced externally with ribs. In what sizes were such boxes constructed, and what was the user experience? What were the author's views on the cladding of steel water boxes with copper nickel, as was done in a power station in Los Angeles which had been in operation since 1958? Did he see any economic future for such practice?

Table IX showed the behaviour of various metals under low water speeds and high water speeds so far as pitting and general corrosion were concerned. The footnotes at the bottom of the table showed that the test durations in the first column, showing *pitting*, varied up to a year or two. Did the other high speed test results shown in the table also relate to the same durations? Further, could the author please indicate which of the tests were done on once-through water and which on recirculated water?

MR. A. W. O. WEBB said that the author had commented on the difficulties of relating the results of accelerated tests on small specimens to the performance of actual components in service. This was illustrated to some extent by the figures in Table XI relating to propeller materials. Only one composition was given for a copper alloy, and that related to nickel aluminium bronze. The particular composition given was not one used for propeller manufacture: the aluminium content was rather higher and the iron content lower. Was the same material used to produce the results given in Table XI?

From the section "Cavitation Erosion" one got the impression that cast stainless steel of the austenitic type was one hundred times better than the nickel aluminium bronze in its resistance to cavitation erosion. His own direct experience with stainless steel propellers, was small, but he believed this was an unfair reflection on the two materials as they behaved in service. The results were quite different from those obtained in other forms of test, such as the magneto striction test or the jet impact test. There was a large amount of published information on cavitation erosion test results obtained on these types of material, and they generally indicated that this type of stainless steel was somewhat less resistant to cavitation erosion than nickel aluminium bronze. For example, some tests which he had conducted on these two materials using the magneto striction type of equipment gave a loss rate in mg/h of 1·1 for nickel aluminium bronze and 2·7 for the stainless steel. Could the author comment on these results?

The figures for the other two copper based alloys, manganese bronze and nickel-aluminium-manganese bronze, which he took to be twelve per cent manganese, also contradicted other results obtained, and the difference between these two materials was considerably greater than indicated in the table. The performance of the manganese nickel aluminium bronze generally was very close to or slightly lower than that for nickel aluminium bronze as used for propellers.

The title of the two columns in the table dealing with cavitation erosion may have been a little misleading. The attack occurring on the specimens in the tests could possibly have been mainly of the impingement type, with the specimens losing a lot of metal by impingement attack, and just a little by cavitation erosion, which produced the deeper pits. If so, this would account for the superior figure for the stainless steel, which was virtually completely resistant under these conditions to impingement attack. All the copper alloys were subject to impingement attack. This was not often a serious problem, but occurred on the leading edges and tips of some propellers. The figures in the paper gave a fair relationship between manganese bronze and nickel aluminium bronze, and correlated well with the different service performance of propellers in the two materials. He used two forms of rotating disc type of test for assessing impingement resistance, and tests in constantly changing sea water indicated that the nickel aluminium bronze gave about three to four times better performance, i.e. it lost about one third of the weight of the manganese bronze. In a large number of tests made at various times of the year, involving various temperatures, and at different speeds, these tests, unlike the Harbor Island tests, had shown virtually no difference between the performance of nickel aluminium bronze and manganese nickel aluminium bronze. Fairly extensive experience with propellers which were regularly seen in damaged condition indicated that the figures obtained with the disc tests were more realistic than those given in the table. In fact, few propellers suffered attack at the rates indicated in the table. He believed that this was almost certainly because the propellers were to some extent cathodically protected by the ship; laboratory tests had shown that impingement attack could be completely prevented by cathodic protection. Possibly the differences shown in the results obtained at Harbor Island and by other people were related to some difference in test conditions, such as a difference in the temperature of the water; the temperatures given in one of the earlier tables indicated that the test liquid was at quite a high temperature, about 26°C (78·8°F). Could the author comment on the reason for these differences in the various tests?

An interesting feature in Table XI was the low level of limiting fatigue stress obtained for all the materials. He agreed with the order of merit in which the tests placed the various materials, but nearly all the figures were about half those which he and other people had obtained. There was an outstanding difference between the tests conducted at Harbor Island and those conducted by other people. Making the tests in constantly changing sea water was expensive and difficult to set up and to maintain, and most people carried out their tests in circulating sea water or in small quantities of water held in a container in the laboratory.

The level of stresses would give propeller designers some degree of concern. The figure of $\pm 2{\cdot}7$ tons/in^2 for manganese bronze did not allow of much in the way of the introduction of residual stress raisers to cause some possible trouble. It should be remembered that these figures were obtained on small separately cast pieces, and the properties of these materials in the form of propellers would be somewhat lower. The author had indicated that manganese bronzes were quite satisfactory for small propellers though not for large ones, but Mr. Webb pointed out that large propellers of the order of thirty tons were regularly being made in manganese bronze and giving satisfactory service.

MR. A. McCONNELL referring to bearing failures on turbine rotors, pointed out that the author had indicated that certain types of steel were more susceptible to "wire wool" type failures than others. His company had had very satisfactory experience over many years with small auxiliary turbine shafts chromium plated at the journal bearing positions. These were very lightly loaded, but they operated at high speed. The bearing material was the normal Admiralty type tin-based white metal, and a combination of the chrome plate and Admiralty white metal was found to be fairly resistant to the passage through the bearing of small quantities of dirt and also to the presence of water in the lubricating system. Had there been any successful experience of chromium plated bearings in main propulsion machinery on ships?

Some information would be useful on the preferred type of lubricating oil for turbine bearings, since most ships did not want to carry any more types of oil than they had to. There were certain types of gear machinery, where E.P. additives were essential in the oil, and it seemed that in some cases there had been a tendency to use these oils on ordinary plain bearings. Under normal service conditions there was a tendency to pick up a lot of water in the oils; this seemed to react with some of the E.P. additives, and bearing failures had occurred on which this mechanism was believed to have had some influence.

Nitrided gears were mentioned in the paper, and alloys of this type were of great interest if by using them one could avoid a grinding operation on the finished wheel. What was the preferred nitriding alloy for these nitrided gears, and was it necessary to remove the thin white layer from the nitrided surface before the gears were put into service?

It was surprising that in Table IX the pitting rate for cast iron was quoted as two and a half times greater than that of mild steel. His experience was almost the reverse—the general impression over many years was that cast iron was generally less prone to severe pitting attack than mild steel. Condenser water boxes were fairly obvious examples of this.

Finally, exfoliation attack on 70-30 cupro-nickel in high temperature heat exchangers was an almost universal experience in large power stations, although he personally had not seen it on a marine installation. But since steam temperatures were tending to go up on ships, and there would be a tendency to use cupro nickel tubing or nickel alloy tubing in high temperature heat exchangers, a word of warning would be appropriate: it seemed inadvisable to use 70-30 cupro nickel tubing for these high temperature steam applications.

MR. H. CAPPER agreed with the choice of materials because they were based on the established properties of materials matched to the performance required of them. The operative factors were the "established properties" on which the choice was made. Another factor was that it was no use designing a component on properties of the materials which were not achieved in practice by virtue of the poor manufacturing techniques of the component from which it was made. There should be an additional column in Table I headed "Cause of Fracture" which would be enlightening.

Regarding fatigue in structural components; there were many instances of cracks in bedplates, entablatures etc. of Diesel engines. A large proportion of these were from welds. In the more modern designs, prepared fillet welds were required, but in some cases the design was such that it was virtually impossible to make a sound weld due to inaccessibility, and even more difficult to chip out the back of the weld before putting down the backing run. Here there might be the correct materials, but there was a poor component due to lack of appreciation of the practical limitations of the fabrication technique.

With cast steel components for cylinder covers and piston crowns, there were failures due to unsoundness or poor surface conditions. In some cases these castings were complicated and almost impossible to cast sound. An awareness was required of the limitations of the founder and a better appreciation of the possibilities of fabricating cast and wrought components by welding them together, to ease founding.

Would the author split the failures attributed to fatigue into those resulting from fatigue without corrosion and those with corrosion? Mr. Capper would suggest that in marine engineering about eight per cent would have failed in uncontaminated fatigue and about sixty per cent resulted from corrosion fatigue, making the total of sixty eight per cent in Table I. This meant that designers must be very careful when basing their requirements on the fatigue limit of a material tested only in laboratory air.

In the case of stress corrosion cracking, by far the most prevalent failures which he had found recently, since the virtual disappearance of high tensile brass from marine applications, had been in aluminium brass tubes and pipes. These had been caused by externally applied stress due to bad assembly of the system in the ship, or internal stress due to failure to anneal the material after cold deformation of one sort or another. In this case there was the right material choice, but failures arose due to poor fabrication or assembly.

Mr. Capper suggested that the best treatment for steel castings was high temperature annealing followed by either normalizing or stress relieving, or both. The high temperature annealing should be carried out at 900°-950°C (1652°-1742°F) followed by slow cooling. In the case of steels containing chromium, stress relieving at 580°-620°C (1076°-1148°F) was essential after high temperature annealing or normalizing because of the possibility of martensite formation.

As regards case hardening gear teeth, the author mentioned that hardness *per se* was the main requirement, and then said that nitriding was inferior to other methods. Good nitriding should result in V.P.N. hardness of some two hundred points above other case hardening methods. He suggested that there were other reasons than hardness, if nitriding was inferior.

Noticing the preponderance in Table IX of nickel-containing materials, he, like Dr. Gilbert, was not sure what the composition of many of them was. However, he drew attention to the excellent properties of titanium, certainly in the context of the requirements in that table.

MR. S. G. CHRISTENSEN (Associate Member) said that the comments on the fracturing of cylinder liners were interesting; whilst he would agree that thermal stress was the cause of cracking, he felt that the analysis following was not wholly correct and that other important factors had been missed.

As many of the type of cast iron cylinder liners shown had a normal useful life, it was felt that other factors were present apart from the choice of material. In the few cases of this failure that he had examined, there had been some case history of slackness of flame tubes or leakage of cooling water from the valve pockets. In each case the roller expander had been used to effect a cure and it was thought that the use of this tool had overstressed the material around the valve pocket and this overstressing had initiated failure or contributed to it. In the case of the flame ring the coefficient of expansion of the ring material was greater than the cast iron of the liner and when the engine was cold, these rings often appeared slack but tightened themselves when hot, due to the difference in thermal expansion. If this factor was not recognized, then continually tightening the flame ring could be a cause of trouble.

In the case of valve pocket leakage, most chief engineers were cognizant of the damage that could come about if leakage occurred and caused an "hydraulic lock" when starting, possibly resulting in a slip of the shrink fits of the crankshaft. Again in such cases, over-use of the tube expander to prevent serious damage from water leakage could set up high localized stresses in way of valve pockets.

There was a strong possibility where leakage had occurred, that the run down of cooling water when the engine was stopped caused a high tensile stress due to thermal contraction in the region of contact between cooling water and liner. This, together with high localized stress caused from deformation due to the use of the tube expander, no doubt accelerated failure.

With regard to failure of cylinder covers and also water cooled pistons, it appeared in the cases that he had examined, that failure was due to high thermal stress set up by overheating of material, caused by interference with heat transfer from oil films formed on the cold surface from leakage of oil at cylinder lubricating oil quill joints.

The operating engineer was often more fortunate than the metallurgist as he could examine things more closely at the time of dismantling. By the time the metallurgist arrived at the scene of failure, parts were often cleaned up and the true causes of failure were erased in the cleaning process.

The author mentioned stress raisers and lowering of fatigue limits due to inclusion in welds, this was certainly an important point. Whilst certain welds might be photographed to keep welding up to a good standard, some point was missed if, for example, the top edge of a sheer strake was left rough and poorly finished.

He felt some remarks should be made regarding cases where material was perhaps doubtful but design was also at fault or design was correct and manufacturing procedures wrong or at worst drawings had not been worked to during machining processes. Recently he had examined a main engine crankshaft which had oil fillets incorrectly machined to drawing dimensions and drilled oil holes full of stress raisers caused by the use of blunt drills or entrapment of cuttings. In each of these cases, it would appear that in the effort to speed production good engineering practice had not been carried out.

Correspondence

MR. A. ROSE, B.Sc. (Associate Member) wrote that the author had referred to the susceptibility of certain rotor materials to "wire wool" type failure and indicated that, whereas, the presence of chromium and manganese in steels was likely to result in this type of bearing failure other materials, including molybdenum, were unlikely to have this effect. Although Dawson and Fidler came to this conclusion Darling and Isherwood* reporting on tests at B.S.R.A. came to the conclusion that the ½ per cent molybdenum rotor steel was very prone to failure. Much of the stimulus for these tests was from the bearing failures of *Northern Star*† which also had a ½ per cent molybdenum steel shaft.

In a later paper‡ these results had been analysed together with Dawson and Fidler's and with service failures, producing a table which showed ½ per cent Mo steel, one per cent CrMo steel and three per cent CrMo steel as being susceptible to bearing failure under certain conditions. The presence of one per cent CrMo steel in the table was questioned by Dawson in the discussion to the paper and further investigation showed that the inclusion of this material had been based on false service information.

These points were significant in that the addition of chromium to a molybdenum steel not only improves its creep properties but can also render the rotor less prone to bearing failure. Unfortunately the amount of chromium that could be added had not been determined and although one per cent CrMo steel was safe bearing material, three per cent CrMo steel is very likely to suffer a "wire wool" failure.

In discussing these failures, however, it must be noted that the lubricant also seemed to have an effect and that the use of sulphur and/or sulphur phosphorus E.P. additives in the lubricating oil could prevent failure of the "wire wool" type.

The author had discussed only metals and his comments on the use of plastics as engineering materials would be welcome.

With the notable exception of glass reinforced plastic hulls, plastics have found little acceptance as engineering materials in the marine industry. "Engineering" was used here in a wide sense to cover any stressed component.

For normal engineering design in metals the properties of the material were readily available in a compact form. Since these properties were constant over a fairly wide range, yield stress, Poisson's ratio and Young's modulus were often all that was required and elastic theory adequate for the calculations. In the case of plastics this was not so, since the properties of plastics varied over the potential range of usage.

Single point properties were not generally acceptable in design calculations and a more complex stress-strain-time-temperature relationship must be used.

This was unlikely to be more difficult than rotor strength calculations, since the main point which must be considered in any stressed plastic part was creep. As far as the engineering plastics were concerned there was already considerable information on mechanical properties to which more information was being added continually.

Would the author give his views on the future of plastics in marine engineering, bearing in mind some of their obvious disadvantages, such as high expansion rates and low yield points?

*Darling, R. F. and Isherwood, T. 1967. "Engineering Tests for Marine Turbine Lubricating Oils". *Trans.I.Mar.E.*, Vol. 79, p. 25.
†Jackson, G. S. and Winyard, C. 1964. "*Northern Star*: Evolution and Operation". *Trans.I.Mar.E.*, Vol. 76, p. 229.
‡Rose, A. 1967. "Marine Bearings". *Trans.I.Mar.E.*, Vol. 79, p. 233.

Author's Reply

Replying to the discussion, Mr. Todd said that he was grateful to all the contributors to the discussion and felt the value of the paper had been greatly enhanced by the useful information provided.

Several contributors had asked for verification of the data on stress corrosion cracking in Table I. Several of these failures related to propeller boss cracks, such as described under "Stress Corrosion Cracking" and two related to longitudinal cracks in aluminium brass piping. However, most of the failures were stress corrosion cracks in heat exchanger tubes. These cracks were predominantly longitudinal, thus suggesting internal stress in the tube rather than applied stress. In most cases the tubes had performed satisfactorily for several years before failure occurred and sections from the uncracked parts of the tube successfully passed the standard mercurous nitrate test, although they cracked readily in ammonia vapour. This suggested that the level of internal stress in the tubes was relatively low. As several years were needed for these failures to occur it could be that tube manufacturers were now more familiar with this problem and tubes supplied over the last few years would be satisfactory.

Mr. Smedley had asked about the stress corrosion susceptibilities of the nickel aluminium manganese bronze alloy (B.S. 1400 CMA1) noted in Table XI. This observation was made on the basis of service experience. Several propellers made from this material suffered cracking on the boss. This was attributed to uneven and excessive heating to remove the propeller from the shaft for survey purposes. This would set up high internal stresses and when the propeller was exposed to sea water again the stress corrosion cracking would occur. This type of cracking had become more prevalent in recent years with the growing use of powerful heat sources such as oxy-propane flames. The cracking might be so severe as to cause scrapping of the propeller.

He pleaded guilty to Dr. Gilbert's complaint of lack of clarity in describing the various bronze alloys and had tried to correct this in the published paper. The manganese bronze alloys of Fig. 7 and Table XI were, as Dr. Gilbert rightly supposed, high tensile brasses.

Regarding the corrosion behaviour of titanium in sea water, this should be qualified to read "in aerated sea water at ambient temperatures". Dr. Gilbert was correct in stating that titanium and some titanium alloys were prone to crevice corrosion in hot de-aerated sea water and brine.

He agreed with Mr. Frederick's view that first cost was of prime importance to the shipbuilder and to stay in business he must use materials which were as cheap as possible, consistent with a reasonable life. However, often when a shipowner specified a better material he was quoted an unrealistic extra charge and he thought the view expressed by McNee and McNaught* on the pricing of "optional extras" was very true and would give a shipyard a good opportunity of increasing its business and profit.

He was also grateful to Mr. Frederick for pointing out the importance of compatibility on the steels chosen for mating gears.

His information on cathodically protected low alloy steel propellers was also interesting. The author would have some misgivings about the fatigue life of such propellers if the cathodic protection failed for any appreciable time. However, their low cost and ease of repair would certainly be advantageous in ships operating in ice laden waters.

Regarding Mr. Lamb's contribution on liner cracking, the point the author wished to make was that the temperature on the cylinder wall was too low to give thermal fatigue failures of the type found in pistons but was sufficiently high to give a crack of the type he had described i.e. by fatigue-accelerated creep. He agreed with Mr. Lamb's view that any restriction in cooling water flow would increase the risk of cracking and would add that risk of failure by seizure also increased rapidly.

In the case under consideration some of the earlier liners might have suffered from scaling due to water side corrosion, but the use of an inhibitor on new ships resulted in essentially clean conditions. This did not, however, reduce the incidence of cracking, thus suggesting that the normal operating temperatures were sufficient to cause cracking.

Mr. Lamb's view on liner wear were interesting. The author would not be prepared to accept that in a liner using an alkali additive oil, the wear was about 50 per cent due to corrosion. If this were so, then liners using conventional lubricants must have a much higher proportion of wastage due to corrosion. However, he would agree that cylinder liner wear was in part a corrosion problem.

The evidence for free carbide being detrimental in vanadium iron liners was indirect and stemmed from the observation in the paper that the effect of vanadium would appear to be to alter the properties of the matrix. If this was accepted then free carbide, by taking up most of the vanadium present in the iron, would denude the matrix of vanadium and reduce its beneficial effects. It would follow, therefore, that any alloy elements which decomposed free carbides without producing ferrite in the matrix would be beneficial.

The data quoted for vanadium-free liners represented only two out of 20 liners. In every case this large group of liners showed wear rates at least twice those of vanadium iron liners in the same engine. Whilst he agreed with Mr. Lamb's view that engine operation could have a marked effect on liner wear, the data he had seen from liners operating in conditions as closely controlled as possible would not support the view that liner composition was unimportant. Liners showing high wear rate due to poor operating conditions were an argument in favour of improving operating conditions rather than downgrading liner materials.

Mr. Smedley asked that corrosion failures be included in Table I. Unfortunately, as he had indicated when presenting the paper, corrosion failures were often accepted as normal and it would be difficult to separate acceptable and unacceptable corrosion failures. He would agree, however, that if all corrosion failures were included in Table I this would probably be the largest single item.

The corrosion fatigue data were determined on bars completely bathed in a stream of aerated sea water. The specimen would be free to corrode but, as far as the author was aware, no excessive wastage occurred. The cause of the difference in corrosion fatigue data in these tests and tests by other authorities was at present being investigated but, as Mr. Smedley pointed out, these data were more in keeping with service experience.

Mr. Neumann raised the question of the effect of low water speeds on corrosion. There was no sudden transition at 3 ft/s, but this was normally regarded as the limit above which fouling organisms were unable to attach themselves to the tube surface. Hence it was advisable to maintain the water speed in excess of this to avoid partial blockage and possible impingement failure. An upper velocity limit was usually set by pumping costs and one did not normally have to consider velocities over about 8 ft/s in tubes. In the case of oil coolers, which seemed to be Mr. Neumann's main concern, another problem arose due to the increase in oil viscosity with fall in temperature. High water speeds could cause excessive cooling and result in a layer of thick cold oil on the tube exterior with a pronounced fall in heat transfer. For such coolers as low a velocity as possible should be used with as much turbulence as possible on the oil side. If deposits on the tube surface were troublesome then vertical mounting of the cooler might help.

Fabricated 90/10 cupro-nickel water boxes had been made in a wide range of sizes from small coolers up to atomic power main condensers. He knew of no difficulties with those which had been fitted.

Clad water boxes could also be manufactured, but those mentioned by Mr. Neumann used a cladding of the order of thickness now considered suitable for an externally reinforced 90/10 cupro-nickel water box. When linings were being considered, thinner material could be used and the problem became an economic one, as the cost of welding a thin lining into a water box was relatively high and in many cases it would be cheaper to fabricate a steel reinforced cupro-nickel water box.

Todhunter† described the experience with clad water boxes in the Los Angeles Power Station and stated that these boxes had operated with no maintenance for nine years and none should be required for 30-40 years.

Mr. Webb's contribution on propeller materials added greatly to the value of the paper and pointed out the complexities of some of the problems which the author had perhaps over simplified. The problem of defining the cavitation erosion behaviour of a material was a case in point. It was fairly easy to comprehend pure cavitation damage where the material was being physically torn from the surface of a propeller. The behaviour of the material in fast flowing sea water was also reasonably straightforward to test. However, in service the material might be subject to both these effects; the relative importance of each varied and it was not always easy to define which property of the material was more important in resisting particular conditions. The data given in Table XI measured both the uniform wastage due to high speed flow and the cavitation effect down stream of a hole in the specimen. The high resistance shown by stainless steel to fast flowing sea water was shown in the first column and the cavitation effect in the second column. These data confirmed those from many similar tests on stainless steel and indicated that it had good resistance to fast flowing sea water but was prone to cavitation damage, presumably due to its relatively low strength. Service experience with pump impellers confirmed these data as stainless steel usually performed better than aluminium bronze in such applications. The generally good performance of aluminium bronze alloys in ships' propellers might be due in part to the cathodic protection provided by the ship's structure and to the cathodic protection often provided around the propeller aperture.

The aluminium bronze analysis quoted in Table IX referred only to the quiet sea water corrosion tests. The high velocity data were obtained on alloys conforming to B.S.1400 AB2C.

The author's statement that for the largest propellers alloys of the aluminium bronze type should be used and for smaller propellers manganese bronze etc., could safely be used, was, he

*McNee, G. and McNaught, J. 1967. "Operating Experience with Large Modern Turbocharged Oil Engines". *Trans.I.Mar.E.*, Vol. 79, p. 305.

†Todhunter, H. A. 1967. "Condenser Waterbox Service". *Materials Protection*, Vol. 6, July, p. 45.

thought, correct but Mr. Webb was also correct in pointing out that "smaller'" might be over 30 tons in weight.

Mr. McConnell had asked whether the author knew of any successful experience with chromium plated bearings in main propulsion machinery. The answer was yes, he knew of many Diesel engine crosshead bearings repaired and returned to service by nickel plating to just under size and finished with a layer of hard chromium plate. These had performed successfully for many years.

He thought, therefore, that a chromium plated journal and a ferritic steel containing chromium would provide quite different bearing surfaces so that the susceptibility of chromium steels to "wire wool" type failure was not necessarily applicable to chromium plated bearings.

Whilst he was aware that oil additives had an effect on susceptibility to "wire wool" failures, a discussion of this seemed beyond the scope of the paper. He agreed with Mr. McConnell's view that E.P. additives were beneficial for some gear applications and that for convenience the same oil was often used in turbine bearings. The occurrence of turbine bearing problems such as tin oxide corrosion and "wire wool" failures in recent years were often associated with the use of E.P. oils.

The high corrosion rate for cast iron quoted in Table IX was perhaps a little unfair but grey cast iron could certainly show rapid graphitic corrosion in sea water and where this happened the corrosion rate might be much higher than that observed on steel. Mr. McConnell's observations on the use of 70/30 cupro-nickel for feed water heaters were generally correct. This alloy was prone to exfoliation attack in power stations working in peaking conditions. In base load stations the 70/30 cupro-nickel alloy usually performed satisfactorily. As base load conditions were more akin to ocean going ship operation it was unlikely that similar trouble would be experienced in marine service. However, alternative alloys such as 90/10 cupro-nickel and Alloy 400 had good resistance to exfoliation attack and could be used in place of 70/30 cupro-nickel.

The essential constituents of nitriding steel were nitride-forming elements such as chromium, molybdenum, aluminium and vanadium. Molybdenum was usually present as it counteracted temper brittleness at the nitriding temperature and in this country chro-molybdenum steel was often used. The white layer on nitrided surfaces was due to iron nitride and might be up to two mils deep. As this layer was brittle it was normally recommended that it be removed.

Regarding Mr. Capper's remarks on corrosion fatigue, the author would agree that corrosion is a factor often influencing fatigue behaviour in marine engineering. However, a large proportion of the failures in Table I were concerned with such things as eccentric studs and similarly threaded components working either in oil or in the engine room atmosphere. Whilst these environments might have exerted some slight corrosion effect on the failure the basic cause of such failures was usually incorrect pre-tightening. Hence he would not accept the ratio suggested by Mr. Capper, but would consider a 50:50 ratio more appropriate.

The authors remarks on surface hardened gears related to the investigation by Chester which he noted "seemed" to show that hardness was the important factor. The author did draw attention to other investigations which showed high carbon content to be important. The basic problem was that when a gear was carburized several new factors were introduced. Notably, an increase in surface hardness, high compressive stress in the surface and a change in composition of the surface. Many investigations had been carried out to determine which of these factors was important in relation to gear performance. The work of Chester seemed most relevant to the paper as it provided data on pitting resistance, an important cause of trouble in marine gears.

Mr. Christensen's remarks regarding "second-hand" information being given to the metallurgist were unfortunately often correct. It was unreasonable to expect an accurate answer based on inaccurate information and the author hoped that engineers would realize that only by close co-operation with the metallurgist could the full benefit of his services be obtained.

In the case of cylinder liner cracking under consideration, liners were examined as soon as possible after the engine had been stopped but in most cases there was no evidence of water leakage even where cracks were present. In addition cracks occasionally commenced on the upper side of the pocket, thus precluding the possibility of water leakage as water would tend to run down the liner.

Regarding tube expansion, strain gauge tests were carried out with the co-operation of Mr. Lamb of B.C.I.R.A. and a valve insert was grossly over-expanded into a liner pocket. The stress developed in the pocket was found to be very low i.e. of the order of 500 lbs/in^2—much too low to account for the cracking.

Mr. Rose's contribution on "wire wool" type failure was interesting and indicated that the addition of one per cent chromium to a molybdenum steel reduced its susceptibility to this type of failure. The author's remarks on the effect of molybdenum on the susceptibility of steels were based on laboratory investigations which showed that the addition of molybdenum did not increase the susceptibility of carbon steels although these steels did show some susceptibility. As there were probably more ½ per cent H.P. rotors in marine service of molybdenum steel than any other material, this steel would not appear to be very prone to this type of failure. Darling and Isherwood's results seemed to indicate that susceptibility increased with increasing alloy content.

The plastic materials were not treated in the paper as he had interpreted marine engineering in the strictest sense and plastic materials were not used to any important extent for engine room components. Mr. Rose had himself pointed out some of the problems in using plastics. The one factor which the author would add was the fire hazard. Most plastics have low resistance to fire and to high temperatures and this precluded their use in any component vital to the operation of the ship. This factor would he considered, limit the use of plastics to relatively minor applications in ships' engine rooms in the foreseeable future.

Annual Dinner 1968

ANNUAL DINNER

The Sixty-fifth Annual Dinner of the Institute was held on Friday, 8th March 1968, at Grosvenor House, Park Lane, London, W.1, and was attended by a record number of members and guests.

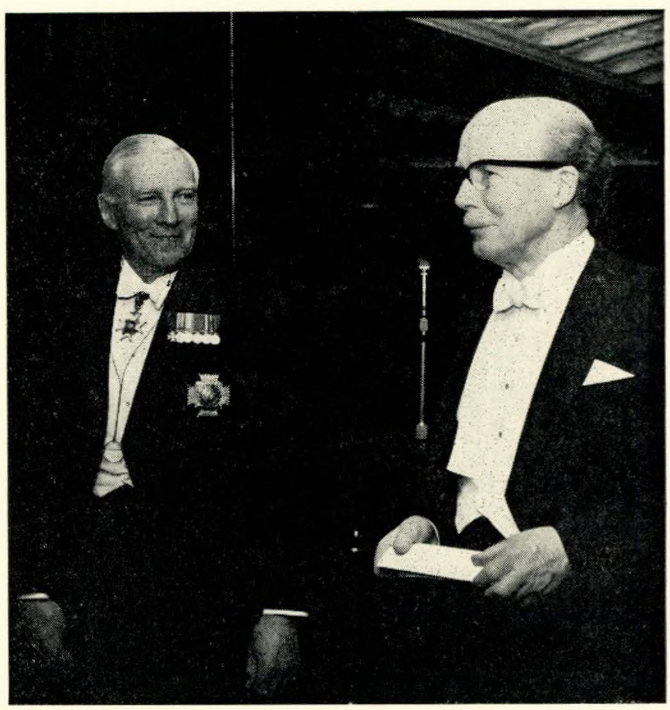
The President, Vice-Admiral Sir Frank Mason, K.C.B., was in the Chair. He was supported by the Chairman of Council, Mr. J. McAfee, Captain W. S. C. Jenks, O.B.E., R.N., Vice-Chairman of Council, and the Chairman of the Social Events Committee, Mr. I. W. Robertson.

The official guests included: His Excellency El Marques De Santa Cruz, The Spanish Ambassador; His Excellency Mr. Erling Kristiansen, The Danish Ambassador; His Excellency Mr. J. Axida, M.B.E., The High Commissioner for Malta; His Excellency Mr. A. P. Rajah, The High Commissioner for Singapore; His Excellency Mr. Shanti S. Dhavan, The High Commissioner for India; Georg Knap Thestrup, Esq., The Norwegian Chargé d'Affaires; J. L. Knott, Esq., C.B.E., The Deputy High Commissioner for Australia; Dr. Heinz Naupert, First Counsellor (Economic Affairs), representing His Excellency The Ambassador of the Federal Republic of Germany; Rear-Admiral B. D. Yashin, Soviet Naval Attaché, representing His Excellency The Soviet Ambassador; Knut H. Staubo, Esq., Past President, The Norwegian Shipowners Association; Rear-Admiral The Earl Cairns, C.B., President, The Navy League; The Right Honourable The Lord Nelson of Stafford; The Right Honourable The Viscount Simon, C.M.G., President, The Royal Institution of Naval Architects (Past President); Sir Frederick Hoare, Bt., Honorary Member; Sir Gilmour Jenkins, K.C.B., K.B.E., M.C. (Past President); Major-General Sir Leonard Atkinson, K.B.E., B.Sc., President, The Institution of Radio and Electronic Engineers; J. P. W. Mallalieu, Esq., M.P., Minister of State, Board of Trade; Rear-Admiral A. F. Turner, C.B., D.S.C., Chief of Naval Supplies and Vice-Controller (Ministry of Defence); Sir John Hunter, C.B.E., B.Sc., D.L. (Past President); Sir Stewart MacTier, C.B.E., B.A. (Past President); L. Drucquer, Esq., Chairman, The Council of Engineering Institutions; Sir Stanley Brown, President, The Institution of Electrical Engineers; A. C. Grover, Esq., Chairman, Lloyd's Register of Shipping; F. I. Geddes, Esq., M.B.E., Chairman, The British Shipping Federation; Captain L. W. L. Argles, C.B.E., D.S.C., R.N., Captain Superintendent, The Thames Nautical Training College; B. E. Bellamy, Esq., Under-Secretary (Marine), Board of Trade; F. B. Bolton, Esq., M.C., President-elect; C. M. P. Brown, Esq., C.B., C.M.G., Second Permanent Secretary, Board of Trade; R. C. Bryant, Esq., Under-Secretary (Shipping Policy Division), Board of Trade; J. Calderwood, Esq., M.Sc., Honorary Member; R. Cook, Esq., M.Sc., Honorary Treasurer; The Reverend Maurice Dean, B.A., The Rector, St. Olave's, Hart Street, London, E.C.3; J. E. Denyer, Esq., President, The Institution of Mining and Metallurgy; Captain W. E. B. Griffiths, C.B.E., Master, The Honourable Company of Master Mariners; Stewart Hogg, Esq., O.B.E., Past Chairman, Social Events Committee; V. H. F. Hopkins, Esq., President, The Diesel Engineers and Users Association; G. G. Howard, Esq., Managing Secretary, The Salvage Association; Dr. R. Hurst, G.M., Director of Research, The British Ship Research Association; J. Leckie, Esq., C.B., Deputy Secretary, Ministry of Technology; Dr. D. M. A. Leggett, M.A., Vice-Chancellor, The University of Surrey; A. Logan, Esq., O.B.E. (Past President); W. McLaughlin, Esq., Member of Council, Institution of Engineers and Shipbuilders in Scotland; A. J. Marr, Esq., President, The North East Coast Institution of Engineers and Shipbuilders; Dr. D. C. Martin, C.B.E., Executive Secretary, The Royal Society; Colonel G. P. Mason, C.B.E., T.D., D.L., J.P.; G. L. Mitchell, Esq., President, The Institution of Plant Engineers; C. C. Pounder, Esq. (Past President); A. Rose, Esq., B.Sc., Denny Gold Medallist 1967; H. Shirley-Smith, Esq., C.B.E., B.Sc., President, The Institution of Civil Engineers; R. Stockdale, Esq., President, The Society of Consulting Marine Engineers and Ship Surveyors; D. S. Tennant, Esq., C.B.E., General Secretary, The Merchant Navy and Air Line Officers Association; F. G. West, Esq., Institute Silver Medallist 1967; G. Yellowley, Esq., Chairman, The National Association of Marine Enginebuilders.

The Loyal Toasts having been duly honoured:

MR. KNUT H. STAUBO, Past President, Norwegian Shipowners Association, proposed the toast of "The Royal and Merchant Navies of the British Commonwealth."

He said: I felt indeed honoured when I was invited to



At the Annual Dinner held on Friday, 8th March at Grosvenor House, London, W.1: The President, of the Institute, Vice-Admiral Sir Frank Mason, K.C.B. (left), with Mr. J. McAfee, Chairman of Council attend the annual dinner of your distinguished Institute and was asked to propose a toast to The Royal and Merchant Navies of the British Commonwealth. Your Institute covers a vast field. One of your greatest contributions lies in the inspiration you promote through always trying to look ahead. This inspiration is needed if shipping is to stand up to the challenges ahead of us.

In my speech tonight I will try to turn my eyes towards the future. When looking for an opening remark to my speech my thoughts lead me however to the past to the opening scene of "The Merchant of Venice" by Shakespeare, where Antonio tells his friends that he is worried and sad without knowing why. His friends tell him: "Your mind is tossing on the ocean", and they continue by saying that: "Had they such ventures forth the better part of their affections would be with their hopes abroad wondering what sort of wind their ships would have and fear misfortune".

Most of us here tonight are engaged in ventures taking the better part of our affections. But the whims of nature are no longer the most important hazards. Antonio's ship and her crew were subject to the perils of the sea. In merchant shipping today we are faced with new and different types of peril which may cause misfortune and worry. They certainly may disturb the commercial pattern as much as the wrong winds did. I am thinking in terms of governmental interference, whether they take the form of subsidies, grants (I cannot see the difference by the way), cargo preferences or outright discrimination. I know that these problems are causing you concern as well, as they are of grave concern to us in Norway. Great Britain has been a main force in establishing the frame of reference for modern shipping. This frame of reference has given equal opportunity for everyone engaged in our venture, and the result has been an expansion and a technical and commercial adaptability which is essential, if shipping is to be able to serve world trade reasonably and effectively. We see, however, tendencies today which may upset this pattern.

Mr. President, of the perils mentioned, discrimination is probably the most dangerous. It is detrimental not only to shipping but also to world trade. All experience shows that it leads to less flexibility, to inefficiency and ultimately to higher costs of transportation.

In this connexion I think it is essential to stress that shipping is an international industry serving all nations.

Subsidy is another peril of modern times, but of a different character. Like discrimination it leads to a distortion of the markets. Surely shipping people in Great Britain could not be surprised when we in Norway were rather shocked when your Investment Grant System was introduced and applied also to shipping. I know that British shipowners would have preferred the previous system, and I agree that quite a number of other nations already had introduced subsidies, but I am worried about the effects. To put it mildly, our feelings in Norway were a little like those you have when a respected virgin turns to vice. Such action may cause pleasure, but unfortunately not in this respect.

The difference between subsidies and cargo preference is—in a word—that the former constitutes a direct form of aid and the latter indirect. Subsidies are paid directly from the taxpayers' pockets. The cost of flag discrimination on the other hand is borne by a number of countries, those using shipping services as well as those providing them. Thus, politically, flag discrimination appeals to certain goverments more than subsidies because the cost to the national budget is not so directly obvious to the taxpayer.

But where do discrimination and subsidies lead us?—they unfortunately seem to lead us to more discrimination and to more subsidies. We have a frightening example within the shipbuilding industry, being subsidized as it is in almost every country, and no yard seems to make money. I have never been charmed by the idea of trying to lift oneself by one's own hair. I am also positive that an industry protected and subsidized can never in the long run uphold its independence. Regulation will be the answer by those providing the means needed and the result may be that the principle of survival of the fittest will only apply to governmental power. The rôle of the individual will vanish, and will a system of government control and regulation be able to provide the means whereby progress is achieved, imagination, the spur of creation, or to develop new ideas?

I am worried when recognizing the outlines of the new international pattern. Are we really entering a future where world trade and world transportation will be regulated by govenments? I hope not. I foresee a vicious circle. One country might say: "Others do it, we have no choice." or "If you can't beat them, join them". Another expression is, however, "Two wrongs do not make a right".

Today still more than 90 per cent of sea transportation is free, and let us look at the achievements of the Merchant Navies of the world.

The far-reaching technological developments that we see under way in shipping today have already showed results. Transportation costs have, the last 10-15 years, been reduced by 25-50 per cent in many trades. Rationalization, automation, new types of ship, the use of computers have opened new fields of business and new lines of thought in our industry. Changes take place at such a pace that I sometimes marvel at the fact that we seem to do our utmost to make our existing ships and traditional ways of transportation not only look old-fashioned, but we deliberately make them old-fashioned. On top of that we boast about it. Shipping is a very uncertain line of business. We do not know what shipping will look like in five or ten years—but that is what makes our industry so fascinating and challenging. We feel we are on the right track, but we risk being run over if we just sit there. The reduction in the costs of sea transportation has taken place at a time when prices on commodities and most other services in the world have increased considerably. The results derived from rationalization and modernization in sea transport have, however, been passed on to the users of our services. It seems to me that we should have been given credit for this development. Instead we are subject to a rather severe international criticism. The results produced should not warrant this. We have, however, no choice. We have to take the criticism seriously. Unfortunately we have to spend time in the defence of our industry, time that would have been better spent on progress.

In all fairness I must admit that so far there have been more words than action in this respect, or to use a different expression—after all is said and done—more is said than done.

If I have sounded rather gloomy I can assure you, Mr. President, I am still an optimist. We are still to a large extent architects of our own future and in this lies our challenge.

I hope you will excuse me for having concentrated my words on merchant shipping, as this naturally is nearest to my heart. However, I would like to stress that through the ages up to our present time the merchant navy would hardly have been able to exist and fulfil its task without the existence of the Royal Navy. We who live in Norway know from our history what it might mean to be on the other side in a conflict with the Royal Navy, but we also do indeed know what it means to side with the Royal Navy, which we have done more or less for the last 160 years. During the last two world wars your Royal Navy has demonstrated not only your old traditions, but that you still were the greatest sea power. All Norwegians certainly salute in gratitude the deeds of the men and the ships of the Royal Navies.

I now propose a toast to The Royal and Merchant Navies of the British Commonwealth; the Royal Navies with their glorious past and the Merchant Navies with their fine tradition and valuable contributions to world trade and their building of bridges between nations. I do this in humble and thankful recognition of their history and in the sincere hope that their high ideals and their fine example will be carried far into the future to the continued benefit of the whole world. (*Applause*)

The RT. HON. THE LORD NELSON OF STAFFORD said: It is a great pleasure and honour for me this evening to propose the toast of "The Institute of Marine Engineers". I speak as an industrialist whose activities impinge in many ways on the activities of the marine industries, and I have the highest regard for their work in all their branches, and particularly for the way they deal with extremely difficult environments in which they have to operate. It is, therefore, a special pleasure to propose this toast for that reason, and also because your President this evening is a very old friend of mine. We have done many things together and I am delighted to be here on this occasion.

I, like you, I am sure, have listened with great interest to what Mr. Staubo had to say, speaking with the authority of a great maritime nation—Norway. Norway is a nation which is such a good friend to this country in both peace and war. He looks to the future in addressing us this evening and points out the dangers of interfering with the normal laws of economics. I am sure many of us would agree entirely with what he says.

Curiously enough, I also look to the future in what I have to say to you this evening. I look to the future from a different aspect, from the technical problems which face so many of us in the engineering business, in the changing of the world in which we live. Engineers have to try and diagnose what is going to be required over a long period ahead, 10 or 30 years maybe, way beyond the 5 to 10 years to which Mr. Staubo referred and, therefore, in the very grey area of the unknown.

The speed of change at the present time presents us all with tremendous problems. Not least in that of marine engineering. This is your sixty-fifth dinner tonight, Mr. President, and I am sure at no time in the history of the Institute has the change been as rapid as it is at the present time. I have no great knowledge of the sea. I am bewildered by the many changes which face the marine engineer, and those responsible for ordering ships which will be in service over a long period of time in the years ahead; the changing economical conditions—wages, material costs, interest rates—changing market requirements, changing political situations, changing technical prosperity, and changing transport patterns around the world. All these are woven together in a most complex and interconnected pattern.

The trouble as I see it is no one of us really understands all the factors involved and this, therefore, makes us very dependent, one upon another. I think your gathering here tonight is very illustrative of that, Mr. President, if I may say so, because I cannot think all the assembled company are marine engineers, here tonight, but I am sure many of them, like me, are extremely concerned with or involved in marine engineering in one way or another. But how to arrive at the right decision over a span of 20 to 30 years ahead? Thinking about this situation, looking ahead in this context, brought to my mind some of the things I do know about. It also brought to my mind a number of things which I know nothing about and, finally, the implications which these have on us all.

Three major situations seem to have affected the area about which I know nothing. First of all, the ever-widening parameters of choice which face us—arising from developments in prime movers, gas turbines, Diesel engines, atomic power, the removal of limitations through choice of fuel, increasing knowledge of hull design, materials and the way to join materials, the increasing availability of a wider range of complex automatic gear and the increasing advantages, in reliability and cost from building standard packages into a self-contained whole.

In another area one sees rapid changes in electronic techniques, and in electronic apparatus—ever-more accurate, ever-more complicated navigational aids, direction finding, communications and radar equipment underwater devices, inertia navigation systems, etc. Today, we have facilities for instant measurements, on-line computer calculations, data logging, visual presentations and on-load closed loop automated systems. Finally we have micro-electronics, that miracle of modern science which gets smaller and smaller every time one looks at it, and consumes less power and becomes less sensitive to the environment in which it operates.

Turning to another area altogether, one sees the increasing complexity of the system in which one operates, starting with the engine control system progressing to the engine room system, the ship's system, the fleet system and finally the transport system, all interacting one with another. These, ladies and gentlemen, are things which are happening all around us which I know something about.

Then I turn for a moment to the things I do not know anything about—the changing transport patterns in which our equipment has to operate; the limitations of crew manning, the rules and regulations which govern these affairs; the possibilities for bridge control, all-purpose crews, docking crews coming aboard with pilots, the problems of re-training, difficulties of government interference, difficulties of changing the established regulations, and the problems of insurance and rates.

I know nothing about these things, but they all affect our choice of parameters; they all affect the point at which apparent economic gain is in fact for one reason or another of no particular interest. These factors have implications for us all, but none of us know the whole story. For this reason there must be the closest contact between us all who are in one way or another in this game and great decisions hinge upon this. These are not easy problems and are not easy decisions. Some of them are extremely painful decisions.

In the shipyards, themselves, for instance, you have the Geddes Report. The shipbuilding industry in this country is right now grappling with the problem of coping with these changes. In the industries supporting the shipbuilding industry we have an area which is very largely untouched up to now. There is a need for the manufacturers of prime movers, auxiliary equipment, etc., to organize themselves to cater for the needs of the yards, and to gear themselves to meet their programmes and requirements.

At the same time, we have the problems of the ports—changes in the facilities to cater for the size of ships, bulk handling container traffic, and faster turn-round times. How important these decisions are for us all, and how important is the timing of them, which is so critical. I was talking only a couple of days ago to someone engaged in the steel industry who was unable to take advantage of the economics of bulk handling of ore, because we have not yet the facilities in this country to unload the largest ore carriers.

Finally, we come to the impact these things all have on the life of our professional bodies, all of whom are well represented at your table tonight, Mr. President. The interfaces between traditional professional areas are all becoming blurred, each one is itself interacting on the other—marine, electrical, electronic and mechanical.

These problems present a great challenge to all professional bodies. I am sure, Mr. President, your Institute, the Institute of Marine Engineers has a great part to play in catering for these essential interfaces which are required between all of these professions, in the marine field. You must be successful in this, it has the greatest impact and greatest importance to us all and we are all vitally involved. Therefore, with this knowledge, Mr. President, and in this spirit it gives me the greatest pleasure to propose the toast tonight of "The Institute of Marine Engineers", coupled with the name of its President, Sir Frank Mason. (*Applause*)

The President, in reply, said: Having listened to these eloquent speeches only strengthens my belief on the need for a Royal Society for the Prevention of Cruelty to Sailors! I am prepared to go into it if anyone wishes to be a founder member tonight. What does an ordinary sailor say to all this? What does an ordinary sailor say to shipowners and captains of industry?

To start with, I have two pleasant duties. One is to thank Mr. Staubo for coming all this way to honour us with his presence tonight. It is very nice of him to come here, if you think about it. King Alfred the Great founded the Royal Navy for the express purpose of exterminating Mr. Staubo's ancestors. They had a regrettable predeliction for visiting these shores for purposes entirely different from those of Mr. Staubo tonight. The second duty is to thank Lord Nelson for finding time in his fantastically busy life—he is off to Paris at 7 o'clock tomorrow by motorcar—to propose the toast of the Institute. You must bear with me if I strike a personal note. It gives me great pleasure for him to be here for two reasons. I had great regard, and indeed affection for his father and mother and, if I may say so, I have the same feelings towards him. It is often said that greatness skips a generation. It is not so in this case.

After all this eloquence I can only offer two quite simple remarks. The duty of the King's ships throughout the centuries has been to protect merchantmen. Organized convoys did not start with the wars of our own day or even the Napoleonic Wars. The history of the Royal Navy has been the history of convoy. The first recorded arrangements were in the reign of Edward III, who commanded that merchantmen has to sail in company under State escort. "The King's well beloved and trusty Admirals at sea, Philip de Courteney and William de Neville" appear to have acted as the modern commodore of convoy and senior escort officer. The object, of course, was to protect the wine trade. The King put a tax of 2/- a tun on it. I am afraid our modern Chancellors of the Exchequer have got their predatory ideas from this. The King did at least give some direct service in return!

It seems to me this duty of protecting merchant ships still remains because it arose from the inescapable fact that we are an island and live by commerce. Today, even more than in past centuries, we are dependent on seaborne trade. I would have thought this required, above all things, unhindered passage across the seven seas. It seems to me unhindered passage also requires peace and it is lamentable that, even in the highly civilized countries, a community has to be protected from piracy and the equivalent by policing. My experience afloat, which was quite a lot really for the Navy, was that the presence of the White Ensign in the many places where British nationals and British trade are to be found is a useful and, indeed, sometimes necessary adjunct to peaceful trading.

If you think that is going too far, it is certainly a great encouragement, as I have been told in many parts of the world at various times.

The second comment I would make concerns the importance of people. As machinery and equipment grows more and more complicated, as Lord Nelson was saying, so the importance of people increases. This applies to every facet, from design through to operation. To cope with this situation the nation needs more highly qualified engineers. This is where the term Chartered Engineer should mean something to industry. I mean by this that the title stands for something quite definite. If you like, it provides a yardstick; it means that, in addition to having recognized academic qualifications, the holder of the title has also undergone training and has shown his ability to accept responsibility to the requirements of a particular professional institution.

That is something more than having an academic qualification.

The second rôle of these professional institutions is to enable their members to keep up-to-date and, as you will have heard tonight, that is no mean task. I hope these two rôles will be recognized by industry and that employers will support the efforts of the Council of Engineering Institutions to provide a steady flow of well-qualified people by making effective use of them.

So far as this Institute is concerned, it was formed to serve those who go down to the sea in ships, and occupy their business in and around waters, great and small. It is still its endeavour to discharge this duty to its far-flung membership. Like other institutions, its strength lies in its branches. I have had the privilege during my year of office of visiting the seven Branches of the Canadian Division, the Branch in New York and the groups of members in San Francisco and Edmonton. Of course, until I arrived in Edmonton I had derived my ideas from G. A. Henty and thought it was an outpost of the Royal North West-Mounted Police, instead it is a thriving city of 440 000 people, and growing. It depends on oil, not buffaloes or Red Indians!

I also enjoyed visiting most of the twelve Branches in the United Kingdom. Everywhere I found members in good heart and getting on with the job of serving the community, which, after all, is the prime duty of the engineer.

If the guests will forgive me for turning to one or two domestic matters, I should like to take this opportunity of thanking all those to whom the membership owes so much, the Branch Offices, the Secretary and Headquarters staff, and for this function tonight, the Social Events Committee, the Chairman of which is Mr. Robertson. And last, but not least, our charming and very efficient administrative officer, Mrs. Sullivan. I will have to get used to calling her Mrs. Baker because she is getting married in April, and I am sure you will join me in wishing her every happiness.

Finally, that lovely word which delights the hearts of all gatherings, meetings and congregations, the word which makes a somnolent assembly sit up and show some semblance of attention. Finally, I should like to say how glad I am to see you all here tonight. This expression of pleasure includes a very warm welcome to all our guests, be they those of the Institute or the private guests of members. Of course there are very special thanks to Mr. Staubo and Lord Nelson for the compliment they have paid us in being with us tonight.

To end, which is another way of saying "finally", I am going to do something which calls for no response from you. To start with I am going to say good-bye: "Good-bye", meaning "God be with ye". So I am going to toast each one of you personally and wish you well in all your undertakings—guests and members. I wish you well, indeed. (*Applause*)

Marine Engineering and Shipbuilding Abstracts

No. 6, June 1968

Fast American Cargo Liner

Ingalls Shipbuilding Corporation has delivered to American President Lines the first of a series of five ships. These are the new APL Seamasters, and they rank as the largest and fastest general-cargo carrying ships operating under any flag. The first of the group is *President Van Buren,* and the vessels following under close order are *President Grant, Taft, Johnson* and *Fillmore.*

Not only are they the biggest of the American-flag break-bulk carriers, but the cruising speed will be 23 knots.

President Van Buren displaces 21 230 tons at a maximum draft of 30 ft 7 in, she stretches for 572 ft and has a beam of 82 ft. She is probably the largest and fastest vessel that ever will be built for the general-cargo trades; there are bigger vessels coming but these are special-purpose container, barge or bulk carriers. She is the first merchant ship in the world to be built almost entirely of high-strength, low-alloy steels.

Stability of the ship was enhanced, by weight reduction above the centre of gravity without loss in deck strength or load hoisting capacity. Steels with a minimum yield strength of 100 000 lb/in^2 were specified in the main deck, second deck, sheer strake, king posts, masts and booms.

Van Buren is equipped to take a total of 143 containers, 104 below decks and 39 on deck. Cargo will be handled by a complete, power-positioned Ebel rig, an improved type gear arranged to provide the maximum versatility required to handle break-bulk, palletized, refrigerated, containerized or heavy-lift cargoes in both modern terminals and in those ports where facilities are less advanced. This Foster-Wheeler steam generator differs somewhat from the conventional marine double-cased construction. Both casings will be held at saturated temperature and will expand equally. The conservative heat release rate is limited to below 70 000 $Btu/h/ft^3$. The design assures that the hottest tubes will be in the areas of coolest gas and maximum superheater metal temperatures will not exceed 1050°F.

Superheater outlet conditions are 870 lb/in^2g and 955°F. All tubes in the boiler-water side have thicker walls and were specified to have two BWG gauges in excess of minimum Coast Guard requirements for longer life. Casings, brick pan, uptakes and ducts are of corrosion resistant steel, with casings increased in thickness over usual marine standards to insure against warping or buckling.

Boiler capacity at normal power is 150 000 lb/h of steam. Superheated steam from the boiler drives a cross-compounded, single-plane, General Electric steam turbine with a maximum continuous rating of 24 000 shp (21 000 shp normal). Boiler combustion air is heated by a single, rotary regenerative air heater (Ljungstrom type) built by Air Preheater Co. mounted in the engine room casing at the upper deck level. This heater will deliver air to the burners at about 460°F, while bringing uptake temperatures down to about 280°F.

Take-home capability could not be ruled out, though there are several steamships at sea without it. This is provided by an 800 kW gas turbine generator set (General Electric LM-100) which would put all its power into a 750 hp take-home motor, geared into the second reduction of the main engine. *Van Buren* will make 8 knots on her gas turbine alone.—*Marine Engineering/Log, August 1967, Vol. 72, pp. 57-64, 121.*

Novel Hull Design

The idea, advanced by Blohm and Voss AG, would reduce shipbuilding to a straight, production line method, cutting building costs enormously, and turning ships out in only four months from time of order.

Unlike other series-production ideas, a shipowner taking advantage of *Pioneer* multi-carrier system would have a large choice in terms of size of ship, types of cargo handling gear and whether or not she will be an ordinary dry-cargo carrying vessel or one with containers or even barges.

Contracts for new ships should no longer be arranged on the basis of a general arrangement drawing, lines plan and model tank tests but rather on the basis of detailed profitability investigations using methods of operations research. *Pioneer* is a practical "modular" ship. The beam and the depth of all *Pioneers,* whether 14 000 dwt 13 knotters or 22 000 dwt, 18-knot container ships, will be exactly the same. Only the length is a variable factor. This means that *Pioneer* midsections can be stored on land, ready to be welded into existing vessels.

This is the first ship to have her hull designed as a polyhedron, that is, constructed entirely from flat plate. The savings in capital cost of hull construction is thought to exceed the slightly higher resistance costs of such a hull.

It is assumed that the designers have allowed for possible cavitation on the ship's edges.

An intestering characteristic of this lines plan is that two intermediate bodies are interchangeable. The starboard part of the forward intermediate body is identical with the port part of the after intermediate body.

From the results of the numerous tank tests with conventional hull lines compared to the flat-sided lines, Blohm and Voss is convinced that, when employing flat plates, the same or similar relations between speed and power can be achieved, provided the inclination of the individual plates to each other is correctly arranged and certain angles are not too sharp.—*Marine Engineering/Log, September 1967, Vol. 72, pp. 65-69.*

Monitoring Turbocharger Vibration

The deleterious effects of non-uniform build-up of combustion residues in the turbine blading of exhaust gas blowers are well known. Harmful vibrations are set up, the imbalance being accentuated if larger deposits should flake off. A serious imbalance can normally be detected audibly and it is often possible to take remedial action. Less severe vibration, however, often goes undetected, only to result ultimately in bearing damage.

On ships where the engine room is unmanned during the night hours the risk of damage is magnified since even the most serious vibrations will remain undetected.

To satisfy the need for a vibration warning sytem, Naval Safety Electronics of Copenhagen have developed equipment to detect vibrations caused by imbalance. The Vibralarm, as it is designated, measures the vibration level on each turbocharger and triggers an audio and visual alarm signal as soon as the preset limit is exceeded. The device is connected to a siren in the engine room and to the normal alarm monitoring equipment. Indicator lamps show which of the turbochargers is affected.

As the system is sensitive only to frequencies corresponding with the frequency of rotation of the blowers, vibration from the main engine or from auxiliary machinery within the engine room does not influence the measurement. It is so designed that the limit may be set individually for each turbocharger and vibration pick-ups can be fitted to sets of any type used with main engines. The equipment is easily installed and requires no regular maintenance.

The system comprises a master unit and a number of pick-ups, of piezo electric type, corresponding with the number of turbochargers. One pick-up is mounted on each blower and is linked with the master unit through normal cabling. Installation work can be undertaken by any shipyard.

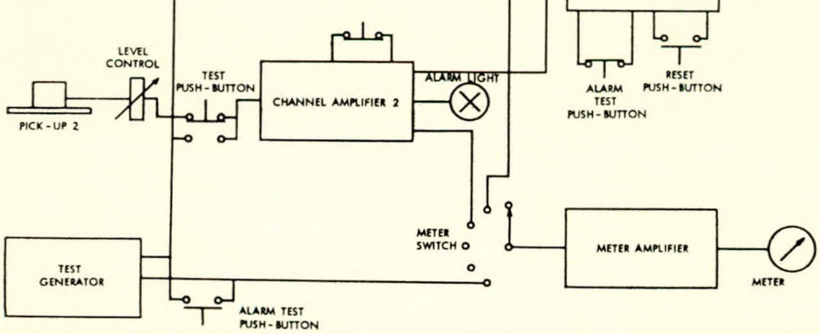
When the turbocharger is running, a signal corresponding with the vibration level is generated. There is a visual display at the master unit. Should the pre-determined level be exceeded the alarms are triggered. The siren can be muted but the indicator lamp remains lit until separately switched off.—*Shipbuilding and Shipping Record, 4th January 1968, Vol. 111, p. 23.*

New Coatings

In a recent test, 14 different zinc-rich coatings were placed in the same marine environment. All were good for two years but after three years, only the post-cured types were holding up. The coal-tar type of zinc-rich paint lasted about 18 months.

It is now thought that the best wash primer for shipyards is the new inorganic zinc silicate especially formulated for this type of work. The manufacturer states that it can produce major savings by virtually eliminating steel corrosion throughout construction and by greatly reducing surface preparation costs.

Steel plates and shapes are blasted clean of mill scale and contaminants by low-cost automatic blasting equipment and the coating is applied immediately prior to the start of fabrication. The zinc silicate wash primer is said to protect the steel from corrosion completely for a year, is immune to damage from abrasion, dries within minutes and can be shaped, welded and fabricated into a finished section with little damage to its protectiveness. Only a minor degree of repreparation of welds and dirt is needed before the application of the primer. Ship-



Turbocharger vibration monitoring system yards say that about one per cent of the surface needs to be redone in this way.

An important innovation in inorganic zinc coatings has been the introduction of the self-curing formulations. They are available either in a water base or a solvent base, permitting the user to select the coating best suited to existing weather and work conditions. For instance, users in many areas prefer a water-based inorganic zinc coating in hot seasons since there is less overspray and related problems; on the other hand, a solvent-based inorganic zinc is preferred for cold weather applications.

As a result, a goodly number of naval architects now are specifying several of the different yet compatible zinc silicate coatings as alternatives, allowing the shipyard to use whichever formulation is best for current application requirements.

Lockheed Shipbuilding and Construction Co. has a new steel processing facility which not only will be utilized for processing its own steel but also steel for other firms in the area. High speed gantry cranes lift the flat plates by means of 33 magnets and transport them to the feed end of the conveyor line. After passing through an oven which dries the plate by heating it to body temperature, the plate is shot blasted simultaneously on both sides with small steel shot. Lockheed says this cuts to one-twentieth the time required to sand-blast by hand.

Brushing and cleaning then takes place, after which the plate is sprayed by primer with 12 guns, six on top and six on the bottom. The rate is at 55 gal/h. One of the important sources of saving is the ability to recover over 90 per cent of the paint not adhering to the plate.

The plate dries as it moves down the remainder of the conveyor system at 16 ft/min. It is then picked up by a second crane which stacks it for future use, loads it for movement to the plate shop or the slabs or into the new automatic flame planer and cutting facility. — *Marine Engineering/Log, November 1967, Vol. 72, pp. 56-58.*

Fighting Barnacles

The traditional killer of organisms such as grass and barnacles on ship bottoms has been copper and its compounds. The effective anti-fouling paints of today slowly leach out cuprous oxide to prevent the growth of barnacles. Incidentally, the old sailing ships were sheathed in pure copper plate, an expensive but effective anti-fouling agent.

The cuprous oxides can be found on various types of anti-fouling paint. They can be suspended in coal tar resins, vinyls or oleoresinous type paints. According to the Society of Naval Architects and Marine Engineers (Tech. Res. Bull. No. 4-6) the film-life failure rates run from one year for the oleoresinous to a maximum of four years for the vinyls.

In the meanwhile, new pigments of a higher toxic quality in newer vehicles are coming in. The Naval Research Laboratory has actually raised cultures of barnacles and is seeing how they stand up not only to the cuprous oxides but to the newer tin, lead and mercury compounds.

Aside from the tributyl tin oxides, which are being used with success on several installations, research on the use of the organolead compounds has led to the development of promising marine paints for protecting ship bottoms and other underwater installations.

Two lead compounds, triphenyllead acetate and tri-n-butyllead acetate, have shown the most promise to date as effective, long-life toxicants in anti-fouling paint formulations.

The research sponsored by the International Lead Zinc Research Organization, Inc. (ILZRO), is being conducted by the Columbus Laboratories of Battelle Memorial Institute.

Conclusions that can be drawn from the research thus far are:

1) effective, noncorrosive anti-fouling paints can be prepared from certain organolead acetates;
2) organolead in combination with cuprous oxide produces anti-fouling paints superior to those containing cuprous oxide alone. At present, copper and cuprous oxide are the most widely used toxicants in anti-fouling paints.

Additional research is under way to develop optimum paint formulations to make the best use of organolead compounds. A large group of vinyl-rosin-type paints has been prepared and placed on exposure at Battelle's Florida Marine Research Laboratory.—*Marine Engineering/Log, November 1967, Vol. 72, p. 58.*

Electronic Control System for Ships' Stabilizers

The electronic control system for ships' stabilizers developed by the Hydraulic Power Division of the Vosper Thornycroft Group in collaboration with the Group's Industrial and Marine Controls Division, dispenses with control rod linkages to the stabilizer fin assemblies, connexions being electrical only.

The system is housed in a box which can be mounted on transverse or fore-and-aft bulkheads. Input is 24 V d.c., a current of only 3 amp being required which is converted to 400 c/s a.c. in a static inverter consisting of two solid-state plug-in modules, an oscillator and a thyristor unit. The a.c. voltage then passes to the signal and feed-back units.

The normal signal generator is for roll rate and consists of a high-response miniature rate gyro. If required, roll angle input can be included, the signal coming from another miniature gyro in the box. Roll acceleration from accelerometer mounted further outboard and connected by electrical wiring only to the control box can also be included. Other special signals, such as helm angle compensation, can be included at this point if needed. All these additional signals can be incorporated at relatively little extra cost.

The signal, now representing the sum of inputs from the various sensing devices, passes to a pre-amplifier unit, again a solid state module within the control box. Provision is made at this stage for inclusion of a compensating unit which can be manually or automatically adjusted, to reduce fin angle at the higher speeds when the necessary stabilizing forces are developed at smaller angles of incidence. At this point the signal leaves the control box and is passed down the cable to the main amplifier, one of which is mounted by each fin unit.

A feed-back transmitter attached to the fin stock and supplied from the inverter in the control box sends a negative signal proportional to fin angle into this main amplifier. This is summed with the input from the control box to give an output direct to the coils of the electro-hydraulic servo valve in the fin's hydraulic actuation circuit. The result is that at any time fin angle is proportional to the control signal.

The new Vosper control system can be used with other hydraulic stabilizer gears having variable-delivery pumps by incorporating an additional feed-back element giving a signal proportional to the swashplate angle of the pump. This feed-back would form another input to the main amplifier unit.

The new system is claimed to improve stabilizer performance because of the faster response of the sensing units and because system stability is improved by the reduced lags stemming from the use of high-response components throughout the system. Other advantages are ease of installation and lower cost than other control systems currently available.

The elimination of rod linkages not only facilitates installation but also removes constraints on fin movement.—*Marine Systems, November/December 1967, Vol. 2, p. 115.*

Paint Systems for Hulls and the Problems of Frictional Resistance

The resistance which limits the speed of a ship in water consists of two components:

1) Residual resistance. The power which the water imposes on certain parts of the hull depends on the shape of the underwater hull and its speed. A turbulent flow of water is not favourable, and this should be carefully checked and eliminated. Certainly the torpedo form offers far less resistance than the caisson shape of cargoes and tankers (compare Navy ships

and modern passenger liners). This is a problem for the engineers and naval architects.

2) Frictional resistance. This is due to layers of water which adhere to the surface of the hull.

The total friction resistance (Rf) is a function of the friction factor which is due to the surface conditions (Cf), to the kinematic viscosity (v) of the water, the total immersed surface (S) and the speed (V).

$$Rf = Cf. \frac{v}{2} . S . V^n \text{ (n about 2)}$$

The friction resistance affects the layers of water adhering along the surface of the hull. These layers move according to the speed of the vessel dependent on the distance from the surface: the speed of the layers is equal to the speed of the vessel only in the immediate vicinity of the hull.

So, all other things being equal, the friction resistance is proportionate to the mass of water which moves with the hull. This resistance is also responsible for the loss of speed of fluids in pipelines.

The only factor which interests us here is the friction factor (Cf), because this factor depends immediately on the condition of the surface of the bottom. The ideal case would be a perfectly smooth surface, not wetted by water, which would always stay in this initial state. For practical reasons a number of facts and possible variants should be considered:

a) roughness immediately after the paint system has been applied to the hull;
b) nature of the surface and initial wetting;
c) action of the flowing water on the surface (swelling and softening of the coating, deterioration of the paint surface);
d) blistering, rusting, peeling and flaking of the paint in certain areas;
e) attachment of marine growth, shell and grass.

Blast-cleaned steel, which fortunately has been generally adopted as the method of removing millscale, introduces a surface roughness, sometimes considerable. This profile, if examined under a microscope, shows amplitudes of from 50 to 150 microns.

However, the systems applied in thick coatings, such as TRO. 2·03-metallic primer high build, TRO. 2·10-2·12 coal tar epoxy, TRO. 2·54-chlorinated rubber barrier coat, will completely cover these surface irregularities and result in smooth surfaces for the anti-corrosive.

It should be noted that the degree of fineness of pigments in the paint for the final coating should be such as to allow a smooth surface. For maximum efficiency the surface of the coating should not show the well-known deficiencies, met too frequently, of orange peel, racking, crunning and sagging.

Many studies have been made to estimate the coefficient of friction of materials. These friction coefficients (Constant of Tidemann) are for surfaces 30-ft long as follows: varnish—f0,010524; painted steel—f0,010570; smooth zinc or copper—f0,009900.

McEntree has published this constant for shellac binders modified in different ways: shellac pure—f0,00878; shellac with graphite—f0,00866; shellac with oil—f0,00484; shellac saponified— f0,01045.

Actually, for smooth and hard surfaces the coefficients of friction are about equal.

Sea water, with a mean pH of eight, hydrolyses or saponifies a number of classical binders, particularly those used in anti-fouling paints such as rosins, resins, shellac, oils. All constituents of binders with hydroxy, ester and acid groups absorb water, swell up and are hydrolysed or saponified.

Some weeks after immersion such coatings lose gloss and become rough due to surface attack. It has been found that after three months the coefficients of friction will increase by about 45 per cent. The water penetrating the coating produces an increase in volume by as much as 50 per cent sometimes and at the same time the coating becomes soft and fragile.

For primers and anti-corrosives binders which do not hydrolyse or saponify are preferred, and these possess a very high resistance to permeability of water and irons.

Special bituminous binder, coal tar epoxy, chlorinated rubber, epoxy polyurethane and so on come into this category.

The pigments and laminar fillers increase the path along which the irons penetrate which means a lower permeability.

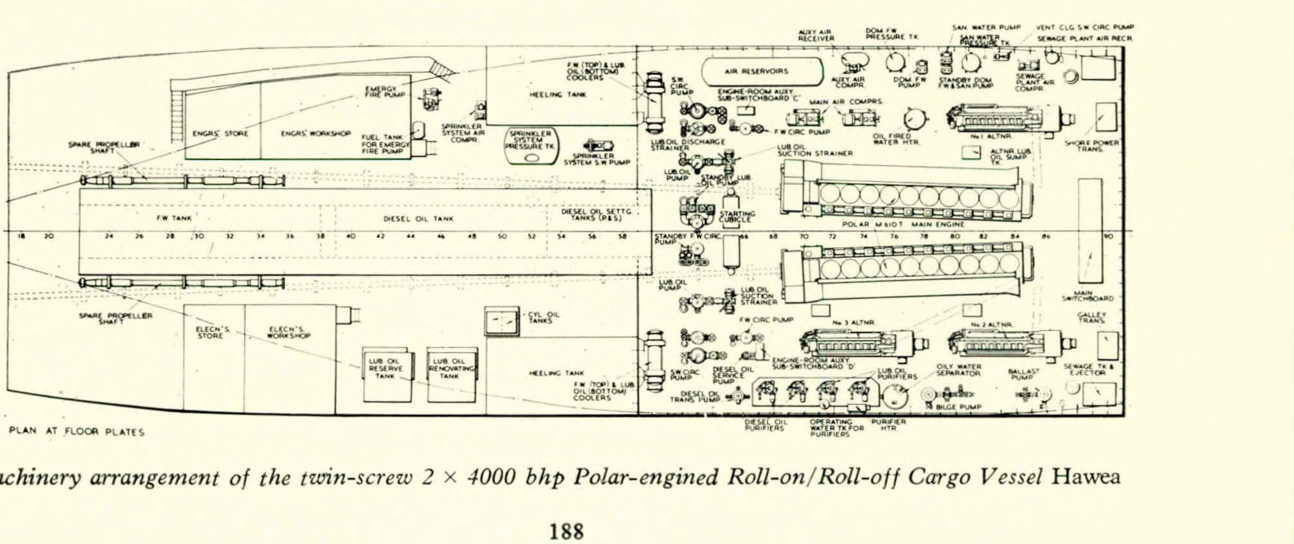
As far as anti-fouling paints are concerned the problem has not yet been completely solved but more and more improved binders are being introduced, as also are paints with a higher toxic pigment loading, to eliminate the surface roughness.—*Bouveret, L. M., Marine Systems, November/December 1967, Vol. 2, p. 116.*

Hong-Kong-built Roll-on/Roll-off Ship

The 2489 dwt roll-on/roll-off cargo vessel *Hawea* has been built by the Taikoo Dockyard and Engineering Co. of Hong Kong Ltd. for the Union Steam Ship Co. of New Zealand to inaugurate its Union Seacargo Express Service between Auckland and Lyttelton/Dunedin on the South Island of New Zealand.

Principal particulars:

Length, o.a.	366 ft 4 in
Length, b.p.	338 ft 0 in
Breadth, moulded	56 ft 0 in
Depth to upper deck ...	35 ft 6 in
Depth to main deck ...	19 ft 0 in
Maximum mean draught ...	16 ft 3 in
Deadweight	2489 tons
Displacement	5407 tons
Gross register	2926 tons
Service speed	16·5 knots



Machinery arrangement of the twin-screw 2 × 4000 bhp Polar-engined Roll-on/Roll-off Cargo Vessel Hawea

Machinery		
Main engines	... ...	2 × Polar M610T
Output	... ...	2 × 4000 bhp

A hold has been arranged under the main deck aft and it is used for carrying small containers weighing up to 3 tons and measuring 6 ft by 6 ft by 4 ft. This container hold is served by a 15-ton Clarke Chapman crane via flush hatches on the upper and main decks.

If required, cargo in *Hawea* can be handled in unitized loads on Sea Freighter pallets which are 14 ft 5 in long by 8 ft wide and which are capable of carrying a load of 12·5 tons. The load height is normally 5 ft but can be adjusted up to a maximum of approximately 12 ft 3 in.

Main propulsion is provided by two Polar M610T engines each coupled individually by shafting supported in Cooper roller bearings to a four-bladed S.M.M. bronze propeller. Each engine develops 4000 bhp when operated continuously at 250 rev/min.

An interesting comparison of the main machinery in the twin-fixed propeller installation on *Hawea* can be made with that of the twin controllable pitch propeller installation on *Ulster Prince*. For main propulsion, *Ulster Prince*, a roll-on/roll-off vessel of 4389 tons displacement and 53 ft beam, operating at 17 knots service speed, has two 12-cylinder, direct-coupled S.E.M.T.-Pielstick engines developing a total of 6000 bhp at 310 rev/min. The *Hawea* installation, of four cylinders less, gives a total of 8000 bhp at 250 rev/min for a service speed of 16·5 knots.—*Motor Ship, December 1967, Vol. 48, pp. 389-393.*

Advanced-design Motorship Machinery Plant

The principal objective of the study was to investigate, through the application of technological advancement and modern plant techniques, a Diesel power plant for a general dry cargo ship which will have low overall cost, a high degree of reliability, safety and minimal maintenance.

At the direction of the Maritime Administration, a machinery-aft propulsion plant for the general dry cargo vessel design PD 108 powered by two Fairbanks Morse Model 38A20 engines, each rated at 10 500 bhp, was developed. The Fairbanks Morse Model 38A20 engine is a residual fuel-burning Diesel engine requiring distillate fuel for start-up, low-power manœuvring and shut-down.

The power plant consists of the main propulsion units, auxiliary systems in direct support of the propulsion units, ship's service auxiliary systems in support of ship's operations and a cargo auxiliary system in support of the ship's principal function.

The main propulsion units consist of two direct-reversing Diesel engines, in parallel, driving a fixed-pitch marine propeller through clutches and a twin-pinion, horizontally offset, single-reduction gear.

The power plant has been designed to provide remote start-up, operation, control and shut-down from a central control station located in the engine room. Manual operations to be performed outside of the central control station are kept to a minimum consistent with ship safety, reliability and maintainability. Pilot house control of the main engines, which can be transferred from the central control station in the engine room to the bridge, is provided as an optional feature.

The advanced-design Diesel plant has a design life of 25 years, with the main system capable of operating 20 000 h between major overhauls. Maintenance of components which can be performed in a 24-h period is accomplished after 8000 operating hours.

All components, systems and the assembled integrated power plant developed are based on operation in the following environmental conditions:

a) Roll and list: 30° roll each side of the vertical at a period of about 8½ seconds continuously, with occasional rolls of 45° each side of vertical. Continuous operation with a list of 15°.

b) Pitch and trim: 10° pitch at a period of four seconds continuously; continuous operation at 5° trim by the bow or stern.

c) Atmospheric air 75°F 85 per cent RH (correction data are furnished for higher and lower conditions).

d) Sea temperature 75°F (correction data are supplied for higher or lower conditions).

From the results of this study, it can be generally concluded that the utilization of a Diesel propulsion power plant on a U.S. Flag merchant dry cargo vessel can be economically justified. For plants of equal power and capability, the initial invested cost by the owner for the motor ship is 2·92 million dollars for a centrally controlled, automated vessel as compared with approximately four million dollars for the equivalent steam-turbine-propelled ship.—*Butler, E. A., Kaufman, R. and Pedersen, T. V., Marine Technology, October 1967, Vol. 4, pp. 362-387.*

Model Studies for an Oceanographic Ship Derived from an Offshore Supply Vessel

During the last few years, a number of operators of oceanographic research ships have considered the Gulf Coast offshore-oil well-supply vessel type for possible conversion to a research vessel. Such a ship of 155-165 ft length offers a number of attractive features, chiefly, reduced operating costs and space for mounting portable or interchangeable equipment. The vessel's principle disadvantages reputedly are limited stability and a lack of seakindliness. An investigation of such vessels has recently been conducted by the Scripps Institution of Oceanography and the Department of Naval Architecture of the University of California. Model experiments were conducted to determine the resistance and powering characteristics, the motions in head seas and rolling in beam seas. The transverse stability was computed for calm water and following seas. Four different hull forms were considered in these studies:

1) a typical 155-ft-long single-chine supply boat;
2) a round-bilge version of the same;
3) a single-chine design developed to overcome several of the suspected deficiencies of form (1) while still retaining the principal proportions;
4) an affine variation of form (1) produced by multiplying transverse dimensions by 0·75 and increasing vertical dimensions by the same amount.

It is concluded that the most serious deficiency of the vessel type is the tendency to slam in a head sea so limiting the maximum speed. The small saving in resistance exhibited by both the narrower and round-bilge versions is insufficient to justify choosing either of these on this basis alone. In addition, both of these forms roll more severely than the two wide, single-chine forms. Finally, the increased freeboard of the third form results in an ample margin of stability. It is felt, therefore, that form (3) provides a suitable basis for developing a research ship design.—*Paulling, J. R. and Silverman, M., Marine Technology, October 1967, Vol. 4, p. 403.*

Controllable Pitch Propellers

All propellers are subject to cavitation and the resultant production of undesirable underwater noise under certain loading conditions. Except for the effects of a slightly larger hub and a generally thicker blade root section, a controllable pitch propeller will perform at or near its nominal design pitch essentially the same as a fixed pitch propeller. The hub size should be minimized as much as possible and the strut provided with a fairing of the same diameter. To permit the blade root section thickness to be minimized the connexion to the hub should be of a type providing the maximum possible strength. With bolted-on controllable pitch propeller blades it is usually necessary to cut into the root fillet. This later type construction will usually require a thicker blade section and/or larger hub to provide the necessary strength.

Thicker blade sections have a greater tendency to cavitation although it can be partly alleviated by reducing the design pitch of these sections.

Some naval installations with controllable pitch propellers have been reported to have excessive cavitation noise at relatively low ship speeds. In tracing some of these reports it has been readily apparent that the basic cause has been the necessity to operate at much less than nominal design pitch because of machinery limitations requiring high shaft rev/min. Model basin cavitation tunnel tests have confirmed that operation of a controllable pitch propeller at pitches progressively less than design induces progressively more cavitation. If minimum cavitation is required with a controllable pitch propeller it is necessary that hub size, root thickness and root loading be minimized. The machinery must also be selected to permit operation at optimum pitch setting and rev/min throughout the ship speed range where cavitation noise is of major importance.

The requirement that the controllable pitch propeller operates at near or greater than its design pitch over the majority of the ship speed range, on ships where propeller noise is a consideration, has caused changes in some of the earlier concepts of combined type power plants.

Full scale tests conducted about 14 years ago on a submerged submarine fitted with a pair of experimental controllable pitch propellers showed that by increasing pitch over normal design and lowering rev/min, a speed increase of about 40 per cent was obtainable before the onset of cavitation. *—Boatwright, G. M. and Strandell, J. H., Naval Engineers Jnl, August 1967, Vol. 79, pp. 537-554.*

Testing Marine Coating

A new marine coating system using metallized aluminium is being given its first commercial test by a leading oil company.

Based on an experimental coating system that is showing an exceptionally long life in sea-water tests, the new coating protects steel with only three to four mils of metallized flame-sprayed aluminium and three organic top coats. The first top coat is a polyvinyl butyral wash primer that seals the aluminium, the second, a thin coat of clear vinyl, which prevents leaching of chromate in the primer and the last, a coating of tributyl tin oxide (TBTO), which wards off foulants.

The new coating system is claimed to be a "highly effective, long-lived, and easily maintained system for protection against corrosion and fouling". Resistance to scraping and abrasion is excellent, far better than that of any system based wholly on organic material. Most important of all, perhaps, is ease of maintenance. Even without the TBTO, foulants do not attach very strongly to sprayed and sealed aluminium. So it is only necessary for maintenance to scrape off foulants lightly, hose down the surface, let it dry and apply TBTO paint.

The new coating is being tested on the hull of a tug working New York harbour and the Hudson River. Ideal for a tough test, the harbour area with its corrosive salt water, and winter ice puts the average tug into dry dock for coating maintenance almost every 12 to 18 months.

Conducted by the Metallizing Committee of the American Welding Society, a current test has shown that coatings similar to the one used on the tugboat, but containing only three mil of flame-sprayed aluminium and no TBTO, are in perfect condition after 12 years in the ocean off the North Carolina coast.

Equally resistant under half-tide and full immersion conditions, these aluminium-based coatings are showing no pitting or blistering and no measurable change in thickness.

When protecting against corrosion, a sprayed-aluminium base offers a form of cathodic protection that differs from the conventional in several respects. By distributing the anode over the surface and utilizing it as a barrier, protection is afforded without electrolytic sacrifice. If small areas of steel are exposed by severe mechanical damage, there is still a very high anode-to-cathode ratio, and the exposed steel is protected with negligible sacrifice. With this system there is no problem of current distribution, nor is drag added and frequent replacement is not required as with expendable anodes.—*Welding Jnl, October 1967, Vol. 46, pp. 852-853.*

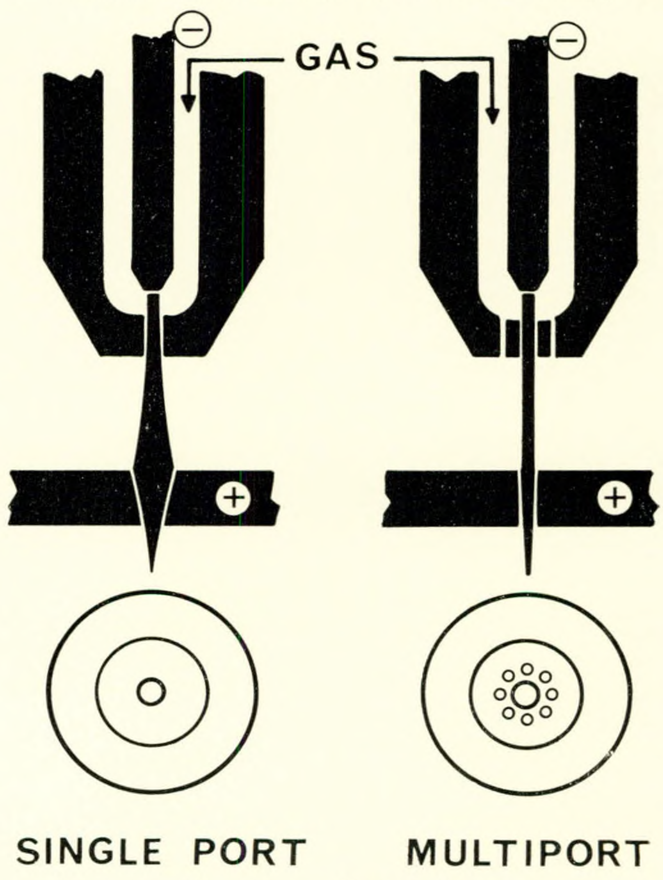
Plasma Arc Cutting

In plasma arc cutting, a suitable selection of several operating and equipment variables is necessary to achieve the best results. This involves the use of properly shaped arc chamber and nozzles, selection of the correct gas flow rates and best cutting gas composition and selection of the optimum cutting speeds, the optimum arc current and the optimum nozzle elevation above the work. Frequently, an improvement in one cut characteristic is obtained at the sacrifice of another. For example, the angle between cut faces in stainless steel can be reduced by increasing the hydrogen concentration in the cutting gas but dross becomes more tightly attached to the bottom edge. In most plasma arc cutting applications the heat-affected zone is narrow and can be neglected, the cutting of aluminium is an exception. Information presented in earlier papers shows that heat-treated aluminium alloys may be subjected to microcracking as a result of plasma arc cutting, while the strain-hardenable alloys are not.

As a result of recent improvements, the plasma arc process now has the capability of producing exceptionally high quality cuts on stainless steel, aluminium and many other metals in the thickness range from $\frac{1}{8}$ to 1 in. Top and bottom kerf corners are sharp, the included angle between cut walls is no more than a degree or two, and dross is only lightly attached or completely absent.

The improved cut quality results chiefly from the use of multiport cutting nozzles with unusually high arc current density and gas velocity in the nozzle.

Changes in arc chamber gas dynamics and enthalpy and



Plasma arc cutting nozzles

in arc shaping beyond the nozzle result in improved performance. Multiport and single port nozzle geometries are shown. The multiports are auxiliary gas passages located around the central passage. In a typical multiport nozzle, the central passage may be 75 per cent of the total gas discharge area. All of the arc plasma passes through the central passage.

High quality cuts are readily made in stainless steel, aluminium and other metals varying in thicknesses from $\frac{1}{8}$ to 1 in. In this range no changes in any of the operating parameters or equipment are required, except speed of cut.—*Welding Jnl, August 1967, Vol. 46, pp. 657-659.*

Spanish Tanker Construction

The largest ship ever to be built in Spain went down the ways at the yard of Astilleros de Cadiz S.A., at Cadiz on 2nd December 1967. She is 98 000-dwt tanker *La Rabida* which will be owned and operated by the Compania Maritima Rio Gulf S.A. to supply crude oil to the Rio Gulf de Petroleos S.A. refinery at Huelva. The sponsor at the launching ceremony was the Marquesa de Villaverde, daughter of General Franco.

The construction of the tanker began in 1966 and delivery is due for summer 1968. The vessel will have an overall length of 266m (875 ft), length, b.p. 252m (827 ft), breadth 39m (128 ft), and draught 14m (45 ft 10 in). She will be powered by a Sulzer-Manises Diesel engine of 20 700 bhp constructed at the Valencia factory of Astilleros de Cadiz. Her loaded speed will be 16·2 knots.

The Cadiz firm has installed an IBM 360/40G computer for shipbuilding and this is identical with one used by Det norske Veritas in Oslo. The latest projects for ships of 100 000, 151 000, 200 000 and 232 000 dwt were developed by this computer. The Cadiz yard has also negotiated a technical assistance agreement with the Central Institute of Industrial Research, Oslo, for the application of the Autokon system which develops computer programmes for automatic fairing of ships' hulls etc; this system also facilitates the application of numerical control to the flame-cutting processes.—*Shipbuilding and Shipping Record, 4th January, 1968, Vol. 111, p. 22.*

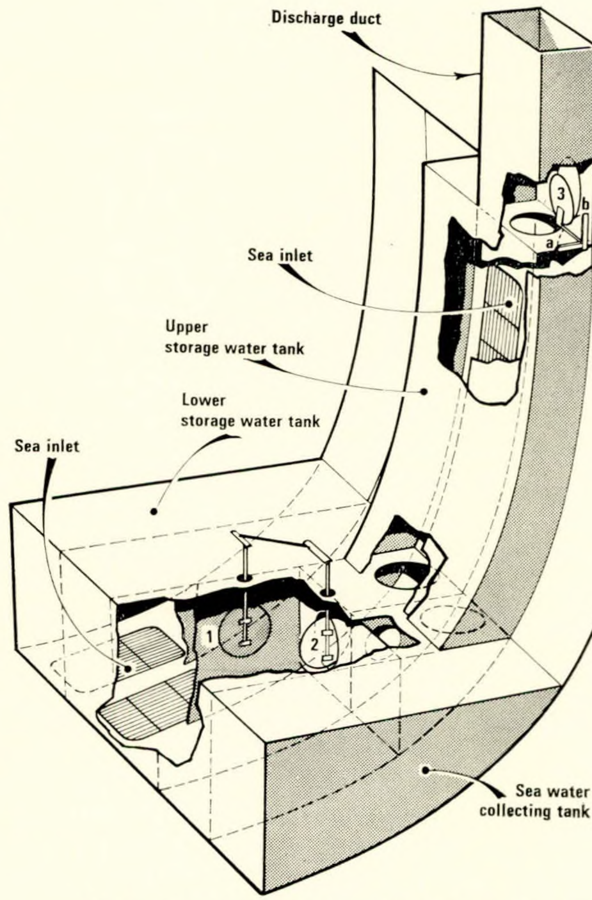
Lead Ship of Six Refrigerated Cargo Vessels

Following extensive trials Blohm-Voss AG, Hamburg, recently delivered the refrigerated cargo vessel *Polar Ecuador*—the lead ship of a series of six—to the Hamburg-Südamerikanische Linie.

Principal particulars are:

Length, o.a.	485 ft $2\frac{3}{8}$ in
Length, b.p.	436 ft $4\frac{3}{8}$ in
Breadth, moulded	64 ft $3\frac{5}{8}$ in
Depth, moulded	
(to U.D.)	39 ft $10\frac{3}{8}$ in
(to M.D.)	31 ft $4\frac{3}{8}$ in
Draught	
open	21 ft $6\frac{7}{8}$ in
closed	26 ft $11\frac{11}{16}$ in
Deadweight	
open	4728 tons
closed	7832 tons
Capacity	
grain	444 185 ft^3
bale	437 525 ft^3
Power output	2 × 7440 bhp at 500 rev/min
Speed, open/closed	$23\frac{1}{2}/21\frac{1}{2}$ knots

The selection of propulsion machinery, the design and the outfitting of the machinery spaces, was essentially governed by the owner's stipulated requirement that *Polar Ecuador* be powered by twin medium speed Diesels geared to a single shaft, the engines, together with auxiliaries, being integrated into a



Patented Tank Sea Inlet and Discharge System

comprehensively automated installation. This was to result in the selection of 16-cylinder S.E.M.T./Pielstick engines running at 500 rev/min and developing 465 bhp/cylinder—an aggregate 14 880 bhp. The engines were built under licence by Ottensener Eisenwerk.

For increased manoeuvrability and to avoid any limitation in speed stages, couplings of electromagnetic slip type have been adopted. The engines work within a range lying between approximately 200 and 500 rev/min which, at a 1:3·84 reduction ratio, corresponds with a 50-130 rev/min range of shaft speeds. Lower shaft revolutions are achieved through the slip couplings.

In the manoeuvring mode the engines run in opposite senses and are clutched in and out as required.

To guard against total immobilization, and to permit maintenance work and repairs to be carried out while at sea, each engine is served by a wholly independent ancillary system—sea, as well as fresh water, fuel injector cooling, lubricating and fuel oils. Pumps and coolers have been arranged as close as possible to the engines in order to avoid long pipe runs and cross connexion is not possible.

Vee-form eight-cylinder DV8TCWM Rolls Royce Diesel engines (developing 595 bhp at 1800 rev/min) were selected as prime movers for the 400kW Siemens alternators—provided to meet electrical power demands when in port and in coastal waters. Through the adoption of a new sea water injection system it has proved practicable to realize a measure of rationalization in the main sea circulating lines. The injection system involves a pair of tanks, the lower arranged on the ship's bottom and the upper extending up the ship side. Both are open to the sea and are inter-connected by means of butterfly valves with a supply or "collecting" tank (see diagram).

The sea circulating pumps may draw via the supply tank from either upper or lower tank according to prevailing requirements. The overboard discharge is also connected with the supply tank and, by means of a third valve, circulating water may be readmitted to the system, rendering unnecessary heating in cold climatic conditions.—*Shipbuilding and Shipping Record, 7th December 1967, Vol. 110, pp. 805-812.*

Metal Corrosion in Deep-ocean Environments

This investigation was undertaken to determine whether unusual phenomena that are not present in water near the surface exist at great ocean depths. Knowledge of the corrosion behaviour of metals in the deep ocean is of interest to the U.S. Navy because of the need for constructional materials in the conquest of hydrospace.

In this article the deep ocean is defined as a depth of about 200 ft or more. Some of the most significant features of deep marine environments which differ greatly from shallow water are low water temperature, variation of oxygen content with depth and high hydrostatic pressure. These and other environmental factors have been described and discussed in more detail elsewhere.

Studies of metal corrosion in deep-ocean environments were approached from two aspects:

1) a corrosion survey of materials representing various important alloy classes;
2) an investigation of crevice corrosion in selected passive metals.

The general corrosion tests were designed to study the total effect of all environmental parameters on the behaviour of typical classes of alloys susceptible to various forms of attack in surface water.

Changes in oxygen content with depth and crevice size were the main variables examined in the crevice-corrosion tests. Crevice corrosion and pitting in passive metals is initiated when anodic areas form by the combined action of differential oxygen concentration cells and the chloride ion. This situation is established when one area of the metal is freely exposed to the environment and the other shielded. Localized attack then proceeds by galvanic action between active oxygen deficient areas and passive areas on the metal surface. Other investigators have shown that the extent of attack is a function of the size relationship between the active and passive surfaces. The degree of attack decreases with increasing crevice area and vice versa. Since crevice corrosion in passive metals is mainly an oxygen-dependent process, it was postulated that the phenomena could be better characterized by studying the additional effect produced by varying the crevice size.—*Wheatfall, W. L., Naval Engineers Jnl., August 1967, Vol. 79, pp. 611-618.*

Magnetostrictive Torquemeters

When considering the physical basis of operation of magnetostrictive torquemeters, it should be appreciated that an unmagnetized ferromagnetic crystal comprises several domains magnetized to saturation along one of the crystal axes of easy magnetization, the size of the domain being such that the net magnetization of the crystal is zero. The application of an external magnetic field causes the domain boundaries to move, so that there is a net magnetization in the direction of the applied field and, as the field increases, the boundary movements progress until, at a point corresponding to the knee of the magnetization curve, the crystal essentially comprises a single domain. Magnetization of the crystal is still along its easy axis, which will not, in general, coincide with the external field magnetization but the macroscopic behaviour of the solid, consisting of an enormous number of randomly oriented crystals, will show a net magnetization in the direction of the applied fields, with no magnetization in any other direction. At this stage, further increase in the magnetization of the material can proceed only by rotation of the domain magnetization vectors away from the easy crystal directions, resulting in a much more difficult process than the initial process of domain-boundary movement and hence accounting for the greatly reduced incremental permeability above the knee of the magnetization curve.

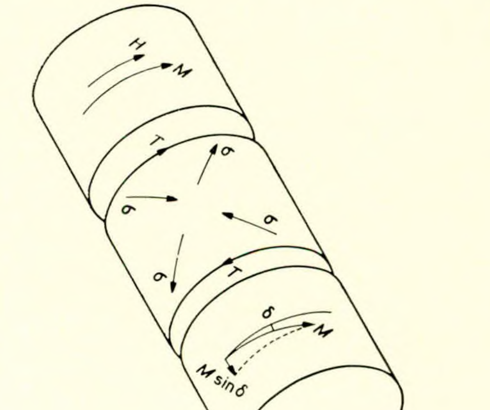
Stress affects the magnetization of the crystal because of magnetomechanical coupling, one familiar effect of this phenomenon being magnetostriction. Tensile stress gives rise to domain-wall movements, tending to increase magnetization in the direction of, or perpendicular to, the stress direction, depending on whether the magnetostrictive constant is positive or negative. These effects are not observed in a non-polarized material, as stress produces equal but opposite magnetizations. However, polarization by an external field makes them apparent, if the polarization and stress directions coincide, by the stress dependence of the magnitude of the magnetization vector. If the directions do not coincide, rotation of the magnetization vector also occurs and this is the effect which is utilized to measure torque.

The effect of stress on the direction of magnetization is shown schematically representing the wall of a ferromagnetic tube magnetized by an external field of strength H. With no torque applied, the resulting magnetization H coincides with field strength H but, under the influence of a torque T, equal and opposite tensile and compressive stresses σ are set up along the principal axes at an angle of 45° to the axis of the tube, causing rotation of M through an angle δ. The component $M \sin \delta$, perpendicular to H, is then a measure of the applied torque T.

Any practical magnetostrictive torquemeter based on the preceding considerations must incorporate various features consistent with ferromagnetic phenomena. Thus, insofar as present designs are concerned, torque-induced magnetization cannot be detected directly but is manifested in terms of an e.m.f. induced in a signal coil requiring an alternating applied field and producing an output signal of the suppressed-carrier, amplitude-modulated type familiar in a.c. servo-systems. Inherent in such a system is a limitation of the signal frequency to less than half the carrier frequency.

Hysteresis, i.e., the pinning of domain walls by crystal imperfections, will be apparent unless all "memory" of this pinning by the stress effect can be obliterated. For this purpose, the external magnetization must be sufficient to sweep the domain walls completely through the crystal during each half-cycle, the response of the signal then corresponding to the mean of the hysteresis loop.

If a linear response is desired, the diversion angle δ of the magnetization vector should be sufficiently small to enable the value of $\sin \delta$ to be taken as being approximately in direct proportion to δ. In materials having low magnetostrictive con-



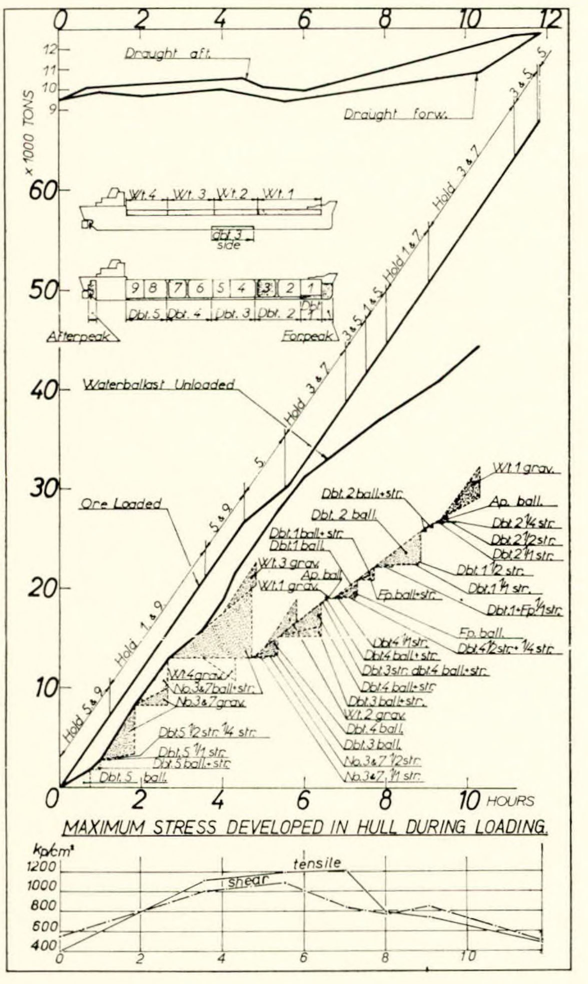
Schematic representation of the effect of stress on the direction of magnetization in the wall of a ferromagnetic tube stants, such as steel, this condition can be fulfilled by stress loads as high as 5000 lb/in^2. In materials having high magnetostrictive constants, such as nickel, the achievement of linearity requires restriction of stress loads to much lower levels.

Practical realization of these principles can be obtained in three different ways, the first of which can employ a solid shaft and utilizes a system of stationary excitation and pick-up coils isolated from the shaft by a small air-gap, while the other two require a tubular shaft having a relatively thin wall thickness and have either both coils rotating or a rotating excitation coil and a stationary pick-up coil. The main features of magnetostrictive torquemeters incorporating these three different types of system are discussed.—*Barton, T. H. and Solar, L., IEEE Trans. on Industry and General Applications, U.S.A., July/August 1967, Vol. IGA-3, No. 4, pp. 310-314; Engineers' Digest, October 1967, Vol. 28, pp. 68-71.*

Hydrodynamic Considerations Regarding Ballast Piping on Bulk Tankers

The frequent and excessive demands for rapid deballasting of bulk carriers which yards are required to meet within acceptable cost limits, has necessitated a more rigorous evaluation of the bilge and ballast pumping arrangement than commonly applied to vessels previously built.

The hydrodynamic considerations involved in this evaluation are basic and adaptable to many other piping systems. The conditions to which they are applied, however, may in many respects be more severe than generally encountered in other ship systems.



Theoretical deballast and loading diagram for a 67 000 ton bulk carrier

A bulk carrier is commonly provided with sufficient ballast capacity to secure a proper submergence of the propeller, a reasonably small metacentric height and an acceptable trim. This ballast water is usually carried in double bottom tanks, peak tanks and topwing tanks. For larger ships ballast water is also carried in one or more holds to improve stability or to allow for provisions of minimum scantlings.

The sequence and rate of deballasting of a bulk-carrier is of particular significance for ships having restrictions in trim and draught.

The figure is a theoretical deballast and loading diagram for a 67 000 ton bulk carrier. This ship was specified to be deballasted completely from heavy ballast condition within a period of 10 hours, while simultaneously being served by two conveyors each loading 3000 tons of ore per hour. During this procedure the ship was restricted forward and aft to a minimum draught of 9·4 metres in order to keep the ship clear of conveyor. The maximum stresses occurring are given in the diagram.

The pumping arrangement of this ship was designed to allow loading and deballasting so as to fulfill the said requirements without at any time subjecting the ship to stresses beyond the maximum permissible. At the same time restricting unnecessary shifting of the conveyor to an absolute minimum. The demands on the ballast system were rigorous and a satisfactory solution could not be found by following empirical methods and the classification requirements alone. In cases like this a theoretical approach is a necessity.—*Sverdrup, C. F., European Shipbuilding, 1967, Vol. 16, No. 5, pp. 88-95.*

Pulsating-arc Fixed-position Automatic Pipe Welding System

The metal fabricating industry has long needed a reliable automatic technique for joining pipe or tube systems having stationary members. Often, joint locations afford limited access and require three dimensional orientation in all conceivable positions. The development of the pulsating-arc fixed-position automatic pipe welding system results from a concentrated programme aimed at solving these problems.

The original concept of an automatic pipe welding system grew out of the frustrations encountered by many people engaged in the fabrication of high quality piping complexes such as those found in food and chemical processing plants, refineries, power generating stations, aerospace components and modern ships and submarines. The fundamental system requirements were:

1) It should be as automatic as possible, i.e., minimizing operator judgment and skill factors.
2) It must operate in any position with the pipe fixed at one or both ends.
3) The equipment must be of rugged design and extremely compact, enabling it to survive rough use and to overcome accessibility limitations.

With these objectives in mind, the pulsating-arc fixed-position automatic pipe welding system was developed.

The automatic welding of pipe and tube joints in any fixed position has been accomplished through the use of a unique automatic pipe welding system. A pulsating-arc gas tungsten-arc process in conjunction with precision made torch or electrode holder carriages and positive control of all essential variables, makes it possible to weld for 360° around a joint using uniform welding conditions. This system has been used successfully for the welding of critical piping systems on materials such as Inconel, copper nickel, Monel, stainless steel and carbon steel. The size range covers from ½ through five in in diameter with the present system and will be extended through 16 in with the model currently under evaluation.

The system's capabilities include fusing consumable inserts, depositing fill and cover passes using cold wire, electrically neutral filler metal, addition and single-pass, square butt joint welding of thin walled materials automatically in any fixed position.

Some of the benefits realized through use of the automatic pipe welding system are:

1) reduction of weld defects;
2) higher production rates;
3) elimination of interpass grinding;
4) uniform weld shrinkage;
5) virtual elimination of distortion.

Mauskapf, L., Welding Jnl, November 1967, Vol. 46, pp. 922-926.

Arc and Transfer Characteristics of the Steel/Co_2 Welding Process

The basic principles of the pulsed current technique for achieving synthetic spray transfer for consumable electrode wires in inert gas shields are briefly surveyed to provide a background to a feasibility study of the control of transfer for a 3/64 in diameter mild steel wire in a Co_2 gas shield.

Bead-on-plate deposits are compared for both steady d.c. and modulated current supplies at 25 and 50 pulses per second for pulse currents up to over 600 amp peak, and for a range of wire feed speeds from 100 to over 500 in/min. The burn-off characteristic under pulsed current is compared with that for a steady current, and from oscillograms of the arc current and voltage, data are obtained on the natural transfer characteristics and the effects of the pulsed current supply.

The limitations of the pulse technique for control of free flight transfer are discussed both for the CO_2 arc and in relation to aluminium in argon and it is shown that in general for CO_2 gas shield there is no advantage over steady current operation.

The results also imply that there is no real "gap" between short circuit and full spray transfer welding, and that in the flat position a restricted form of free flight transfer can be used in the current range 200-300 amp for a 3/64 in diameter mild steel wire.—*Needham, J. C. and Carter, A. W., British Welding Jnl, October 1967, Vol. 14, pp. 533-549.*

400 000-dwt Tanker Design

Simultaneously with the construction of their 500 000-ton class building dock at Chiba, work on the experimental design for a 400 000-dwt tanker was started at Mitsui Shipbuilding and Engineering's Ship Initial Design Department. Joint research in the field of hull strength and structural arrangement is being undertaken with Lloyd's Register of Shipping.

The main basic requirements in the design of a 400 000-dwt tanker, as envisaged today, are as follows.

The research results of a European shipowner indicated that, in order for a tanker on a one-way load basis from the Persian Gulf to Europe via the Cape of Good Hope to be as operationally economical as a 200 000-dwt transitting the Suez Canal, it would require a deadweight of some 400 000 tons.

As it was believed that the design study of a vessel of this would easily lead to a 500 000-dwt design if desired, the 400 000 figure was chosen for the experimental " super mammoth " design.

A decisive factor in determining the most economical hull form is the draught. After giving special considerations to the depth of the Persian Gulf and the existing conditions in the major European ports of discharge, the external draught of the experimental 400 000-dwt vessel was set at 80ft, the same as that of the Lloyd's Register of Shipping's experimental 500 000-dwt design.

With due consideration for safety and in view of the output limitation of a single shaft propulsion system, the twinscrew, twin-engine system has been selected for the experimental ship. The question of choosing between the Mitsui B and W K98FF type Diesel engines or non-reheat or reheat steam turbine engines as the main propulsion plant to give the ship a service speed of 15 knots at full draught is still under study.

The basic requirements in designing an economical hull form are the deepest possible draught and the minimum length/breadth ratio. However, in a ship like this, which has a relatively deep draught, it becomes difficult to maintain the length/breadth ratio below 5·5, because of length/draught restriction as well as limitations in the engine room length.

Thus, the following hull dimensions were arrived at after studying many hull forms to keep the length/breadth ratio between 5·5 to 6·0 and yet keeping the hull steel weight as light as possible:

Length, b.p. ...	... 1990 ft
Breadth, moulded	... 204 ft
Depth, moulded	... 110 ft
Draught, external	... 80 ft
Block coefficient	... About 0·83
Deadweight ...	... about 400 000 long tons
Main engine ...	... Mitsui B and W Diesel engine 2 sets × 8K98FF type
Maximum cont. output	59 200 bhp at 103 rev/min
Cont. service output ...	53 600 bhp at 100 rev/min
Speed	... about 15 knots under full draught at CSO with 15 per cent sea margin

Shipbuilding and Shipping Record, 4th January, 1968, Vol. 111, p. 20, 22.

Hydrogen-stress Cracking

For many years, it has been recognized that the presence of hydrogen in steel may lead to a number of undesirable conditions and the problems that result have challenged metallurgists working in many fields.

The nature of this problem can be illustrated best by an example. A steel bar that has been heat treated to a tensile strength of 230 000 lb/in^2 supports a tensile load that stresses the bar to 175 000 lb/in^2. Many engineers would expect the bar to support that load far beyond the life of the structure, since the service stress-strength relationship is the basis upon which design engineers usually make their calculations. However, if conditions are introduced that allow entry of large amounts of hydrogen into the steel while it is under stress, the bar will support the load for a relatively short period of time (probably 10 minutes or less) and then fail suddenly in an entirely brittle manner, i.e., without evidence of any plastic deformation.

If an identical bar is subjected to a somewhat lower stress, the time to failure will be longer but the bar will still fail. As the applied stress is progressively reduced, the time to failure will increase until finally a limiting load is reached where failure will not occur even after very long times under stress, regardless of the quantities of hydrogen present. This type of behaviour is shown by a curve which illustrates the hydrogen-stress-cracking behaviour of steel under very severe conditions where large quantities of hydrogen are present. The stress below which failure apparently does not occur is frequently called the "minimum stress for failure".

Experiments similar to the one described above, conducted on bars of the same steel, heat treated to progressively lower strength levels, show that the minimum stress for failure for a given hydrogen content increases as the strength level of the steel is decreased.

Various investigators have presented many conflicting ideas about the mechanism of hydrogen-stress cracking and a number of theories have been advanced to explain the phenomenon. None of the proposed mechanisms has been accepted by the majority of the investigators in the field. However, there seems to be rather general agreement that the hydrogen-stress-cracking failure process consists of three separate and distinct stages:

1) the incubation period;
2) a period of relatively slow crack growth;
3) sudden rupture with extremely rapid crack growth.

Whatever the nature of the mechanism, the phenomenon appears to be dependent upon the diffusion of hydrogen and possibly upon diffusion to regions of high tensile stress.—*Fletcher, E. E. and Elsea, A. R., Battelle Technical Review, December 1967, Vol. 16, pp. 10-15.*

Aspects of Radiography and Ultrasonic Testing of Welds in Steel with Thicknesses from 100-300 mm

With increasing steel thickness not only does the handling, forming and welding become problematic, but also the non-destructive examination.

When thinking of nondestructive testing in heavy industry, images of ultrasonic testing of big forgings or radiography of heavy castings come to mind. Radiography of forgings is seldom undertaken because the presence of cavities is very rare. During forging possible discontinuities in the original ingot are flattened and difficult to reveal radiographically.

On the other hand ultrasonic testing is not such an obvious technique to discover flaws in castings because cavities in castings are often irregular in shape. They mostly tend to scatter sound instead of producing clear signals. Structure as well as shape and surface condition are of appreciable influence. Here radiography offers advantages, providing ultrahard radiation is used for thick sections, in particular when a variety of thickness calls for a great latitude of the radiograph.

In the case of welds in thick plates one is inclined to consider the weld as an elementary casting flanked by forged or rolled members. However, when built up by many individual welding runs, cavities and inclusions will generally not have dimensions like those in real castings but will have dimensions in accordance with the thickness of the welding runs. Large defects may occur but are generally two-dimensional discontinuities like lack of fusion and cracks.

There are therefore good reasons to consider both radiography and ultrasonics in the domain of heavy welded structures.

For thicknesses of steel over 100 mm, normal X-ray equipment with high tension transformers cannot be used any longer. Other machines enabling acceleration of electrons to higher velocities before they hit the X-ray target are required. Machines developed and used for radiography are the resonance transformer, v.d. Graff generator, betatron and linear accelerator.

These machines work according to different principles but the ultimate aim is the same, to produce high energy radiation from 1-30 MeV or more by means of accelerated electrons.

Radioactive sources for high energy radiation can also be used; the most popular one is ^{60}cobalt, emitting a radiation energy of slightly more than one MeV.—*de Sterke, A., British Jnl Nondestructive Testing, December 1967, Vol. 9, p. 94.*

Gas Tanker Development at AG Weser, Bremen

After discussing the importance of natural and petroleum gases as an energy source and examining the need for transporting them by sea, the author reviews the development of LNG and LPG tankers at AG Weser. Since the yard delivered its first gas tanker in 1962, a second gas tanker has been delivered and two others are being built. The review is particularly concerned with the various aspects of the design and construction of cargo tanks for carrying the liquefied gases at atmospheric pressure (at the corresponding boiling temperature). The review is also concerned with multi-purpose gas carriers which handle most cargoes at atmospheric pressure but can, when necessary, carry them at higher pressures and temperatures. A brief account is given of tests on tank construction and materials, including insulation materials. "Independent" tanks are preferred to the "integrated" type.

Descriptions are given of the following AG Weser designs:

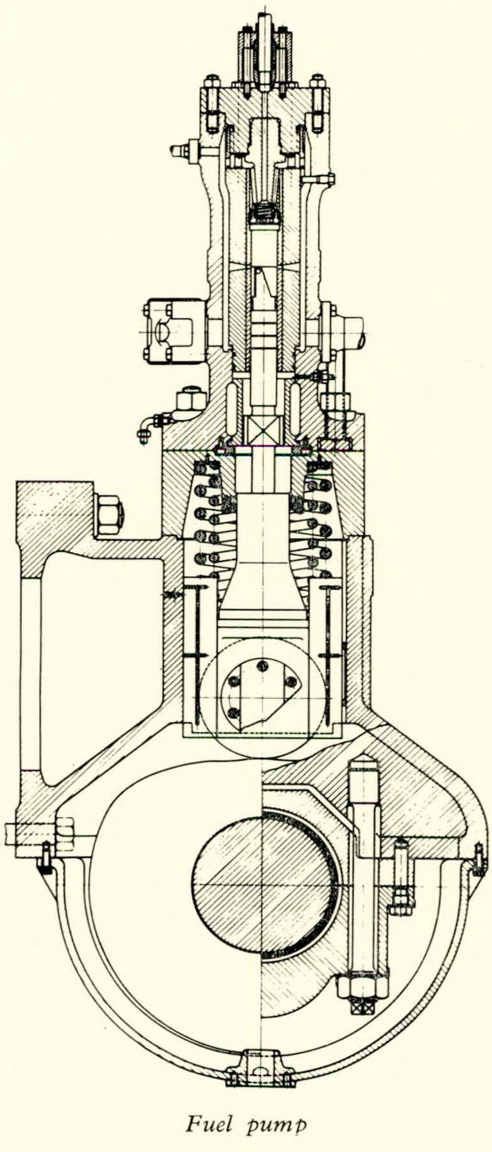
a) Two LPG carriers with capacities of 18 000 and 30 000 m^3 (635 660 and 1 059 440 ft^3); the larger one is due for delivery early in 1968.
b) A 1963 project for a 40 000 m^3 (1 412 590 ft^3) capacity LNG carrier.
c) Two multi-purpose gas-tankers. One of these two is *Lincoln Ellsworth,* delivered to Norwegian owners in September 1966; she has a tank capacity of 810 m^3 (28 605 ft^3), and is designed for ethylene, propylene, propane, butane, and anhydrous ammonia. The other, which is under construction, has a capacity of 6000 m^3 (211 890 ft^3). General arrangement drawings are given of all five ships.

Volger, M., Hansa, 1966, Vol. 103, No. 22, pp. 1839-1852; Jnl. B.S.R.A., July 1967, Vol. 22, Abstract No. 25 484.

Super-large Bore Engine

A new Burmeister and Wain Diesel engine with a cylinder bore of 980 mm and a piston stroke of 2000 mm will be the first super-large-bore engine to go into service. The engine, of a new type designated K89FF, will be installed in a 160 000 dwt tanker now being built by A/S Rosenberg Mekaniska Verksted in Stavanger, Norway, for the Norwegian owner Sig. Bergesen d.y. and Co. It has seven cylinders and is contracted for a continuous service rating of 25 000 bhp at 103 rev/min and a maximum continuous rating of 27 500 bhp at 106 rev/min. However, following workshop tests it has been decided to quote the following rating—continuous service output 3500 bhp/cyl at 100 rev/min and a maximum continuous rating of 3800 bhp at about 103 rev/min.

The engine is a two-stroke, single-acting, high pressure,



Fuel pump

turbocharged, crosshead Diesel engine with poppet valves and uniflow scavenging. It is built in units of from six to twelve cylinders, covering a maximum continuous output range of 22 200-44 400 bhp at a maximum continuous rating of 3700 bhp/cyl at 103 rev/min and an m.e.p. of 10·7 kg/cm². The corresponding service rating, at 100 rev/min, is 20 100-40 200 bhp. However, as stated above, it was decided after workshop trials, to rate the engine as follows: Continuous service rating 3500 bhp/cyl at 100 rev/min and 3800 bhp/cyl at 103 rev/min at m.e.p. = 10·45 and 11·00 kg/cm², respectively, the maximum continuous ratings are 22 800-45 600 and the maximum continuous service ratings 21 000-42 000 bhp. The overall length (with built-in thrustblock) is from 16 570-27 330 mm, while the dismantling height is 13 000 mm, but can be reduced to 11 850 mm.

The fuel pump and its drive is of a new design. The fuel pump barrel is separated from the housing and, in case of wear, the barrel and plunger form an easily exchangeable unit. Adjustment of injection timing is effected by moving the pump barrel to a higher or lower position, which is done from the outside without dismantling. The fuel pump drive now only consists of a roller guide with roller and a cam on the camshaft. To enable the fuel pump to be rendered inactive during operation, a pneumatic cylinder has been incorporated, which ensures that the fuel pump roller is engaged and disengaged gently.

The camshaft is supported by an underslung bearing in a housing which contains both the fuel pump and the exhaust valve drive. The housing is rigidly secured to the cylinder jacket. Cams and bearings can be easily inspected. Cams and coupling flanges are shrunk onto the camshaft by the SKF oil pressure method which facilitates adjustment or replacement. —*Holland Shipbuilding, December 1967, Vol. 16, pp. 60-63.*

Vosper Thornycroft Moving-weight Stabilizer

After protracted theoretical analysis, a moving weight system was devised. The mathematical model for the system having been decided, a most extensive simulation was then performed on the PACE analogue computer at Glasgow.

With the use of data supplied by the National Institute of Oceanography comprising sea characteristics recorded at Sevenstones Light during some 3000 ten-minute periods of observation extending over a year, it was possible to construct a prevalence distribution of root mean square wave slope, averaged over each of these periods. By presupposing that a ship was moving through this sea in a large circle at a known speed, the associated period of wave encounter and root mean square roll amplitude could be constructed. Thence, the equations of motion of the ship and of the moving weight and control system could be altered in order to arrive at the optimum parameters, with respect to both performance and physical practicability.

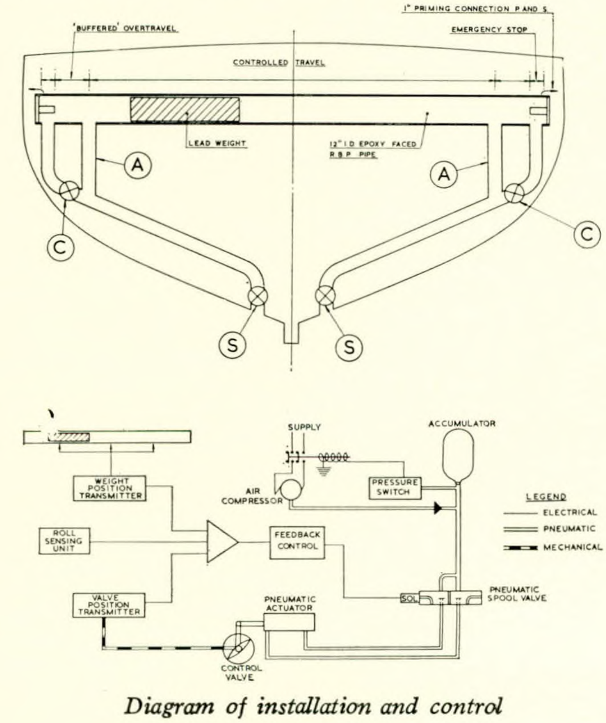


Diagram of installation and control

Of a number of models visualized, the system outlined here presented the most promise for further development.

A cylindrical metal weight, mounted on end rollers, slides inside a plastic-faced tube placed athwartships (Fig. 1). The ends of the tube are connected by pipework to the hull side below the water line. The system is filled with sea water. A further pipe link is made (A) and two control valves and two stop valves are fitted (C) and (S).

Energy input from the waves causes the ship to roll, which in turn imparts potential energy to the weight, causing it to accelerate. The parameters of the control system are chosen so that the weight is in a position to give the largest restoring torque at the optimum time.

Thus, the weight is either (*a*) allowed to slide without restriction and accelerate or (*b*) braked after its motion has started or (*c*) its initial acceleration is hindered when it is already at a potential maximum. Furthermore, if the weight builds up sufficient speed to pass the junction with pipe (A), then a much greater deceleration is applied over the remaining short length. An emergency stop device is incorporated in the weight and tube end fittings in case of pipe failure.

When the system is not in use it is effectively locked by the closing of the stop valves. The control valves are actuated by a signal from the control system (Fig. 2), which measures and processes derivatives of deck angle and weight position.

An electronic summing unit, similar to the new Vosper stabilizer control system used for active fin control, accepts inputs from a roll-sensing gyro and from transmitters of weight and control valve position. The output from this electronic unit controls a solenoid-operated spool valve in the pneumatic circuit to the valve actuator.

In a typical 100-ton vessel, the plastic tube would be 12 to 15 in in diameter and the weight would be approximately one ton. The tube can be fitted at any convenient frame position provided that the maximum beam is available. The pipework and valves can be fitted in any convenient position.

For a vessel of 6000 tons displacement, the most practical configuration would consist of two tubes, each 36 in diameter and each carrying a weight of 20 tons.—*Marine Systems, September/October 1967, Vol. 2, p. 93.*

Integrated Container Handling System

For transhipment in seaports, DEMAG has developed the first European version of a container ship loader which has been specially designed to suit local working conditions and safety regulations and takes into account the load-bearing capacity of normal quays for handling cargo. The loader is in operation in the Margriet docks in Rotterdam. A second loader is in the construction stage and will be used in the Hamburg port.

They are designed for loading or unloading containers of all standardized dimensions and weights in two minutes. Within three minutes they can perform a combined cycle during which one container is loaded and another unloaded on the return leg of the cycle. The ship loader has a portal of box construction with a single spar tubular lattice boom which is braced by a tie rod and can be raised almost vertically. To keep the wheel pressures low, the hoisting gear with Ward-Leonard control is accommodated in a machinery house arranged over the landside leg. The four hoisting ropes are routed via the rear end of the boom to the trolley, run over the equalizing pulleys of the load spreader and terminate in a mechanism at the tip of the boom. This enables the container to be inclined for trimming, switches off the hoisting gear in the event of over-loading and permits the hoisting ropes to be changed easily.

The relatively light trolley is equipped with drives for all track wheels, so as to render possible accurate, no-swing positioning of the load. It also carries the driver's cabin and thus affords an unhindered field of view in all working positions.

The automatic spreader is the heart of the loader. Very special attention has been devoted to its development. Box construction is employed for the load-carrying structure proper, in order to give a high degree of rigidity combined with minimum obstruction of vision. The twist locks and flippers at the four corners are actuated independently, so that no coupling linkages are needed; actuation is hydraulic and each element can therefore be properly set without any complications. Power is supplied by two hydraulic pumps, one of which is sufficient for maintaining operation at reduced speed. Additional safety features ensure proper locking of the container in the spreader and prevent premature release. To permit handling of various container dimensions, the spreader proper can be released from the intermediate suspension gear and replaced by a different spreader.—*DEMAG News, 1967, No. 185, pp. 18501-18503.*

Gas Turbine Auto-starting Sequence Controller

Rolls-Royce Marine Olympus gas turbines are being installed in a number of vessels for overseas navies. Among these are two Finnish CODOG corvettes of 600 tons displacement now being built at the Wartsila shipyard, Helsinki. These vessels will have press-button panels for complete start-up sequence control supplied by Vosper Electric, the Industrial and Marine Controls Division of the Vosper Thornycroft Group. Vosper have also supplied the ship stabilizer equipment.

Before press-button start-up from the machinery control room console or the bridge can be effective a number of checks must be made before the starting sequence is initiated and as it takes place. Should any of the pre-start checks reveal a malfunction, the starting sequence cannot be initiated and an appropriate indication is given on the console. If the checks are satisfactory, pressing the button initiates a 30-second start-up cycle which proceeds automatically until the engine is running at normal idling speed. If any of the steps are not completed satisfactorily, as proved by appropriate sensing devices, the start-up sequence is automatically terminated.

The actuation system is electro-pneumatic which can be expressed as "pneumatic muscles and electric nerves". The functions are generally tested by microswitches, pressure transducers and tachogenerators. The Vosper-built panel itself operates with conventional electro-magnetic relays of approved design. When the starting sequence is complete and the engine is running satisfactorily, it comes under manual throttle control and interruption of and subsequent establishment of the electric supply does not affect its operation. This arrangement, appropriate to operate by unskilled hands, and, where unwarranted shut-down under emergency conditions might be disastrous, can be described as a "fail-set" system. The start-up sequence cannot be initiated again until after a stop button has been depressed.

Should the gas generator fail to light up within 30 seconds the attempt to start is automatically abandoned and the effect is the same as if a stop button had been depressed.

While the gas turbine unit is running, a number of important values are monitored continuously. Faults may be indicated in the form of a master flashing light and audible warning, with display of an individual fault indicator light when pre-set limits are exceeded. There is provision for alarms and trips to be accepted, in which case the master warning light stops flashing and becomes steady and the audible warning is muted, while the fault is investigated. If the fault is cured the reset button is operated and the panel returns to a dark presentation. Operation of the reset button against an accepted but unrectified fault will cause the fault to show again.—*Gas and Oil Power, January 1968, Vol. 64, p. 13.*

Generation of Noise by Flow Through Pipes

In order to obtain detailed information on the mechanism of sound radiation from water supply systems (containing flow without cavitation), the author performed measurements of frequency spectra and spatial distributions of wall pressure fluctuations in a smooth pipe and behind sharp changes of cross section.

The barium titanate transducers used had a size of 8 × 8 mm^2 and were mounted with their faces flush with the wall. Their frequency response was flat within $\pm$2dB between 40 Hz and 10 kHz.

In the smooth pipe of one inch diameter, the root mean square value of the wall pressure fluctuations was found to be 0·007 of the total pressure head at the axis of the pipe, independent of the rate of flow which was varied between 90 and 185 l/min. With good approximation, these fluctuations measured behind a sudden expansion of the pipe also proved to be proportional to the total pressure head at the axis of the pipe; further, they were shown to be proportional to the pressure loss caused by the expansion and to the ratio of the cross sections before and behind the expansion. Maximum values of pressure fluctuations were found at a distance behind the expansion where the exhausting jet touches the wall.

Based upon existing knowledge about the structure of pressure fluctuations in boundary layers and in jets and about the sound radiation from such pressure fluctuations, the author concludes: The results of the measurements lead to a pressure correlation radius of about 0·13 r and a downstream convection velocity of the spatial pressure distribution of about 0·8 U in a straight pipe where r equals the radius of the pipe and U equals mean flow velocity at the axis of the pipe. The sound pressures which are caused directly by turbulent wall pressure fluctuations or by a sudden expansion are, in both cases, only a small fraction of the turbulent wall pressure fluctuations. Sound emitted from a turbulent pipe flow, therefore, can be assumed to be caused only by forced flexural vibrations of the pipe wall stimulated by the turbulent wall pressure fluctuations. *Boerner, H., Maschinenbautechnik, February 1967, Vol. 16, pp. 83-87; Applied Mechanics Reviews, November 1967, Vol. 20, p. 1126.*

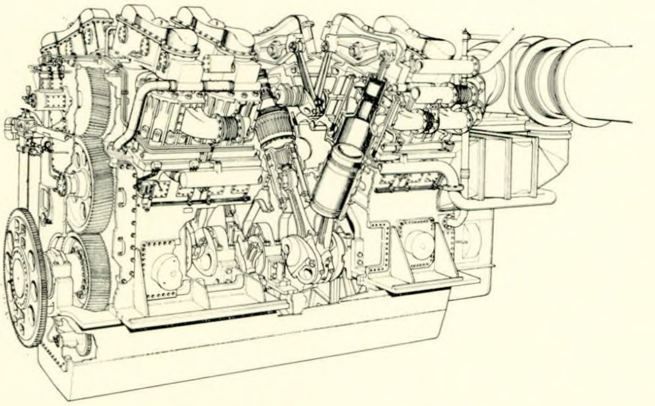
American Opposed Piston Diesel Engine

Designated the Fairbanks Morse Model 38A20, this opposed piston engine utilizes welded steel for the majority of its structural parts with the cylinder block composed of one compact weldment. The opposed piston design has well-established advantages, giving high output within a compact engine and simplification of parts by the elimination of valves.

The full range of engines covered by this development comprises stationary Diesel and dual-fuel units together with reversible and non-reversible marine engines capable of operating reliably on residual fuels. The range includes six to nine-cylinder in-line and 12 and 18-cylinder vee form engines covering a power range from 6000 to 22 500 bhp at 400 and 450 rev/min.

The engine is turbocharged, and scavenging is accomplished on the uniflow principle. Inlet air is controlled by the main pistons and the air inlet ports are slanted to give optimum mixing of the fuel and air. The flow of exhaust gases is controlled by the upper pistons which, as explained, are of smaller diameter and stroke than the lower main pistons. For quick discharge of the exhaust gases, exhaust ports are cast into the upper cylinder at an angle conforming to the rotation of the charge.

Combustion chambers are formed between the pistons in the upper and lower cylinders, the openings for fuel injection nozzles, starting air and cylinder relief valves being arranged



Artist's sectional drawing of a 12-cylinder vee-configuration Fairbanks Morse engine of the 38A20 design. This is rated at 15 000 bhp at 450 rev/min and at a b.m.e.p. of only 10·19 kg/cm². It has a maximum overall length of 8·2 m and weighs less than 130 tons

in the upper cylinder. Two fuel injectors, located opposite each other in the upper cylinder, are supplied by independent fuel pump plungers for each cylinder which are contained in a single housing adjacent to each other and actuated by cams which for the non-reversing version of the engines are fitted direct to the upper crankshaft. An engine-driven fuel pump supplies fuel to the injection pump headers through engine-mounted filters.

Engine-driven pumps are fitted for fuel, lubricating oil, jacket water and fresh water circulation.

Other design features include individually cast cylinder liners and housing for upper pistons and exhaust ports and passages. Another distinctive feature is that the upper and lower crankshafts are cast from high tensile ductile iron alloy. As the crankpin and main journals are cast hollow, rotating weight is reduced considerably and main bearing loads are lessened. Aluminium-tin alloy bearings are utilized throughout the engine. Both upper and lower cast alloy iron pistons incorporate the "cocktail shaker" principle of oil cooling, the reciprocating action of the pistons producing a motion which repeatedly scrubs the underside of the piston crown with a constantly changing oil supply. The piston rests on an insert and is free to rotate to distribute uniformly the heat load on the piston top and wear on the piston skirt.—*Motor Ship, January 1968, Vol. 48, pp. 466-468.*

Repair of Dredge Pumps

Buildup and repair of dredge pumps and parts can be profitable, whether they are made of austenitic manganese steel or carbon steel.

Pump shells, pipeline fittings, impellers, cutterheads and other dredging equipment parts may be repair welded and their worn areas built up. This process will make these components wear longer at a savings in both material costs and downtime. In many cases hardfacing is necessary and recommended.

Welding is not recommended for parts cast of alloy irons such as the commonly used Ni-Hard alloys. Consult the pump manufacturer if there is any doubt as to the type of base metal.

To determine if a pump is made of carbon steel or manganese steel, test with a common magnet. Manganese steel is non-magnetic. Carbon or low alloy steels have a strong magnetic attraction.

Close attention to temperatures in manganese steel casting welding is most important. Rule of thumb procedures indicate that welding should not be continued in any area of the casting unless the temperature of that area is below 750°F (400°C).

Very little or no ductility is exhibited in the hardfacing materials generally used to hard-surface pump parts. Instead of weld beads stretching there will be superficial cross checking of the bead deposits. This cross checking is beneficial since it will relieve some of the shrinkage stresses and reduce possible distortion. This normal checking pattern should be no cause for concern.

Most pump welding problems are encountered in welding pump shells. Welding heat causes the base metal to expand and as it cools it contracts. In complex shapes, such as pump shells, all sections will not cool and contract at the same rate. This, and the shrinkage forces of the weld beads, if not controlled, will cause distortion of the pump shell.

Screw jacks should be set tight approximately 30 in apart around the inside circumference of the pump shell and crosswise between the side plate gasket fits.

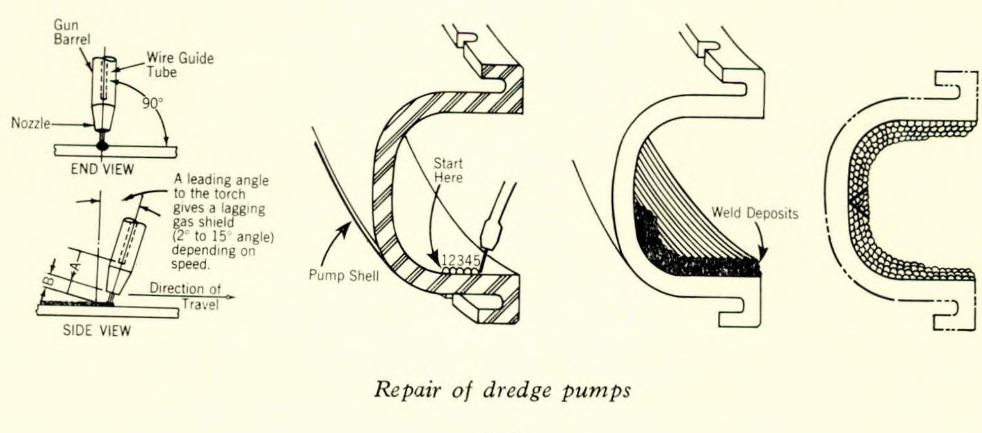
Distortion can be partially controlled by peening the weld deposit.

Weld beads should run parallel to the flow of material and care must be taken not to overheat the casting. Use Tempilstiks or stop welding in an area when you can no longer hold your bare hand on the casting six inches away from the weld area.

If manganese steel filler bars are being used they should be tack welded to the base metal and then welded on each side to the base metal. Bars should be spaced approximately one and a half to two in to allow for electrode manipulation and raised from the base metal by a distance at least equal to the diameter of the electrode being used.

Normally, a direct current constant potential generator or rectifier is used with welding grade CO_2 gas to shield the arc. Normal operating range is 500-700 amperes.

The welding gun should be inclined in the direction of travel at an angle of about 15° as shown. The distance between the base metal and the end of the wire feed gun nozzle should be approximately ¾ inch.—*World Dredging and Marine Construction, November/December 1967, Vol. 3, pp. 18-22.*



Repair of dredge pumps

7025 Operating Hours Between Cylinder Overhauls

Following their experiments with extended operating periods between cylinder overhauls on the Hitachi-B. and W. 1274-VT2BF-160 main engine of the 33 900 dwt tanker *Yamatomi Maru,* the Yamashita-Shinnihon Steamship Co. Ltd, Tokyo, have been endeavouring to lengthen further the operational running hours between such overhauls. The ship concerned this time was the 120 000 dwt tanker *Kaho Maru* jointly owned by the Yamashita-Shinnihon Steamship Co. Ltd and the Nissho Steamship Co. Ltd, and propelled by a Hitachi-B. and W. 1284-VT2BF-180 main engine.

Originally it was the owners' intention to run the engine for at least 7000 hours before withdrawing any of the pistons but, due to ring breakages, numbers 3, 5, 6 and 10 pistons had to be withdrawn after 5443, 5824, 6228 and 6228 running hours respectively. Subsequently all 12 pistons were withdrawn during a recent docking for repairs and servicing to the ship. The remaining eight cylinder units had, by then, completed 7025 operating hours and inspection showed them to be in good condition. The piston crowns were free from fusions and there was very little trace of combustion gases having passed the top rings. All piston rings were free and unbroken and showed signs of adequate lubrication. The cylinder liners displayed a clean and glossy surface and the maximum wear rate recorded was 0·0828 mm/1000 h—well within the acceptable limits.

During the 7025 running hours, the engine was operated at approximately 80-87 per cent of the full-load rating of 27 600 bhp, on fuel with a maximum viscosity of 590 sec. Redwood No. 1 at 100°F, and with a cylinder oil feed rate of 0·36 g/bhp/h. The interval between examination of the exhaust valves was originally 1600 operating hours but due to the frequency of exhaust valve seats burning, this period was reduced to 800 operating hours. This burning of the exhaust valve seats was eventually traced to faulty atomization at the fuel valves and the fuel valves were consequently withdrawn more frequently for checking; the exhaust valve operating hours were then extended back to 1600.

With only the minor troubles mentioned above, the advanced techniques employed in Diesel engine design and construction and the use of good quality cylinder lubricants, the owners feel that long periods between cylinder overhauls can be accomplished relatively easily. However, they conclude that prospects of such long operating periods will be worsened by deterioration of the ship's hull which in itself affects the engine performance and efficiency and, therefore, to offset this they feel that a larger engine with a greater reserve of power to overcome hull deterioration later in the ship's life, is necessary.—*Motor Ship, January 1968, Vol. 48, p. 484.*

Hydrodynamic Aspects of Contra-rotating Propellers

The gain from reducing the propeller rev/min varies according to vessel type and size. For a 150 000-ton tanker there is a gain of about three per cent per ten rev/min drop of propeller speed, while for a container-ship the corresponding gain is limited to less than one per cent. To be able to realize this gain the speed drop must be accompanied by an increase in propeller diameter. For container-vessels a limit is soon reached depending on the needs for clearances etc. which do not permit too large a propeller. If the rev/min is dropped without increasing the propeller diameter the efficiency also drops.

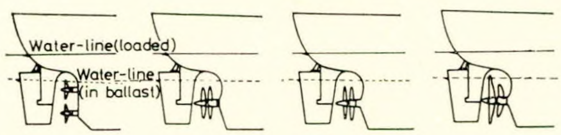
For older vessels of normal types, a transition to conventional twin-screw is not favourable from the hydrodynamic aspects. This is mainly due to the propellers not being able to utilize the wake behind the hull. For a 13 000-ton cargo vessel, the investigation shows that even for the most favourable twin-screw the power requirement was higher at all speeds than for the single-screw alternative. A similar result was obtained for a 70 000-ton tanker, although here the difference was less. For the highest speed, 18 knots, the requirements were equal.

The largest tanker of 300 000 ton, showed an opposite tendency. Here, the twin-screw alternative was most advantageous within the speed-range tried.

Different alternative arrangements for placing two propellers in the wake behind the vessel have been suggested.

The first suggestion, with two propellers over each other, is mentioned by van Manen and is interesting in principle. It allows an effective use of the wake but as it involves a considerable limitation of the propeller diameter it is of limited use.

Tandem propellers have been suggested as an alternative



1) two-shaft propeller arranged on centre line
2) contra-rotating propeller, low speed
3) tandem propeller
4) propeller arrangement with free-running turbine-propeller (Grim)

Various proposals for multi-propeller arrangements

to contra-rotating propellers. Two propellers are mounted one behind the other on the same shaft, but American experiments as well as theoretical analyses show that this arrangement is hydrodynamically undesirable. Certain Russian experiments with tandem propellers are reported, however, to have given good results. SSPA do not have any experience of this type.

Grim's suggestion of a free-running turbine propeller (Propeller und Leitrad) is perhaps the most interesting. On the same shaft, behind a conventional highspeed propeller, is mounted a free-running propeller of large diameter. The inner portion of this operates as a turbine and is driven by the main propeller's slipstream. The outer part acts as a low speed propeller of large diameter and contributes to the forward thrust. It reduces the loading on the main propeller and counteracts the rotation in the propeller wake. The whole arrangement is, in principle, a form of hydrodynamic gear. A disadvantage is that the stern propeller's diameter must be very large if any real efficiency is to be brought about.—*Tanker and Bulk Carrier, January 1968, Vol. 15, pp. 492-494; 504.*

Developments in Gyroscopes

The essence of an inertial system of navigation is that it detects the accelerations of the vehicle on which it is mounted develops an electrical signal proportional to the acceleration and then, by a process of double integration, continuously deduces current position in terms of any required co-ordinates, for example, latitude and longitude. Since its operation is entirely internal and thus independent of any external references such as landmarks, sun, stars, radio or radar transmission, it cannot be jammed, thus offering obvious military advantages, and being unaffected by a lack of external visibility it is equally operational in fair weather or foul. In addition, its self-contained character lends itself well to guidance as well as to navigation, particularly in regions where ground or shore based facilities are meagre or non-existent. Inertial navigating systems have found widespread application in airborne and space guidance and navigation, and have been incorporated in naval ships. Civil airlines, in the interests of automatic navigation and economy in flight crews, are also exploring the possibilities: one system under extensive construction at the present time is that for Concorde.

Although the principles of inertial navigation are simple, its realization in practice has posed extreme technical problems.

In the first instance, the accelerometers which respond to accelerations respond equally well to the gravitational field of the Earth: the two responses are indistinguishable at output and hence it is vital that the local vertical be known with very great precision, either to maintain the accelerometers in the horizontal plane, or alternatively to account, through appropriate resolution, for their component output due to gravity. Another strict requirement for long distance travel is an accurate knowledge of the Earth's figure, so that account can be taken in the system's computer of departures from exact sphericity: together with the internal calculations that must be performed, this demands an elaborate computing facility.

If the journey is so short that it may be assumed to exist in a fixed horizontal plane, many of the difficulties disappear. It suffices then to mount the perpendicular accelerometers on a horizontal stabilized platform, to integrate their outputs, thus providing signals proportional to the two component velocities and to perform a second integration to provide a measure of distance travelled in the two component directions.—*Maunder, L., 22nd January 1968, N.E.C.I.E.S. Paper.*

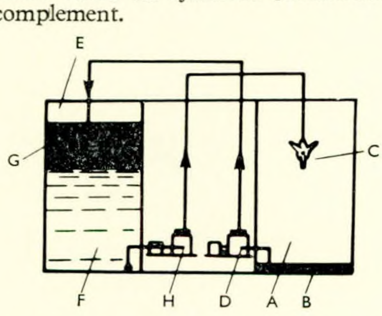
Russian Tank Cleaning Emulsifier

When oil is pumped from tanks, about one per cent of the cargo remains. Exactly how much, depends on the pumping system, the degree of heat, design of vessel and, what is most important, on the viscosity of the liquid. The main function of cleaning is to give this residue fluidity and then to remove it. It is desirable to preserve its commercial quality so that it may be eventually used and, of course, it must not be disposed of by discharge overboard together with the cleansing waters.

According to the requirements of the International Convention on the Prevention of Pollution of the Sea there is a 50 mile zone along the coastline where it is forbidden to eject water containing more than 20 mg/l of oil products. The London Conference which drew up this requirement noted that the only effective method of preventing such pollution was completely to ban the pouring of oil into the sea. It has thus become necessary to avoid discharging overboard cleansing waters contaminated with oil products.

For many years the cleansing of oil carrying vessels was one of the most labour-consuming jobs. Even today certain vessels are washed by extremely primitive methods, employing, in the main, manual labour.

The use of washing machines does not of itself result in sufficiently clean surfaces, especially at the bottom of the tanks. Modern cleansers based on synthetic surface-active agents are a necessary complement.



ML-6 compound is dissolved with warm sea water in one of the central tanks (E). After being heated to 70-80°C, the solution is pumped (H) under pressure of 9-11 kg/cm^2 to the washers (C) in the tank to be cleaned (A), the emulsion (B) is continuously pumped out by the cleansing pumps (D) into the tank (E) where the emulsion breaks down into the cleaned out product (G) and the washing solution (F) which is again pumped up to the washers in the same or next tank

Russian tank cleaning emulsifier

In many countries compounds which work on the principle of emulsifying their water solutions are used. However, the stable emulsions that are formed prevent the re-use of such compounds on the closed circuit principle.

This is why the ML-6 cleanser, recommended for tank washing by the analytical laboratory of the Oceanology Institute of the U.S.S.R. Academy of Sciences, is of interest. This product combines with the remains of oil products and fats to form weak emulsions with a defined brief period of existence but long enough for removal from the tank. After settling for a short while, the emulsion separates naturally into two layers: the oil product (or fat) and the ML-6 cleansing solution which can be used again as many as 10 to 15 times over.

The solution, which is a mixture of surface-active agents, not only makes it possible to salvage the remains of oil products or fats but even improves their commercial qualities by reducing the water content from 35-40 per cent to 0·4-5 per cent and by cutting down the level of mechanical admixtures.

The cleanser is used in the form of water solutions in concentrations from 0·1 to 0·2 per cent. Only one ton of the compound is needed for cleaning a tanker with a deadweight of 30-45 000 tons. Ordinary sea water with salinity more than five per cent is used.

The need for preliminary washing or steaming of the tanks is obviated. And fuel consumption is cut by half if the work is conducted on a closed circuit basis: the cleasing solution which separates from the emulsion has a temperature of about 50-60°C (122-140°F) and only a little fuel is required to raise it to a temperature of 70-80°C (158-176°F).—*Tanker and Bulk Carrier, December 1967, Vol. 14, pp. 458-461.*

Rational Approach to the Selection of Ship Size

The selection of vessel size as measured by cargo capacity is one of the most important decisions affecting the overall economics of a proposed ship. This decision is often a difficult one because the predicted availability of cargo has long term trends upwards or downwards. In addition, seasonal fluctuations may be expected. Other complications arise because of differences which may exist on each leg of the voyage in cargo availability, freight rates and so on. In these circumstances, the selection of ship size has in the past been rather arbitrary simply because the complexities of the problem precluded any sort of rational approach. Electronic computers, however, provide the means to solve problems of this nature. This paper shows how ship size may be selected in such a way as to provide the most economical design for a given forecast of cargo availability. Sensitivity studies lead to a few tentative conclusions as to the relative importance of factors such as sea speed or length of voyage. The influence of cargo availability patterns receives particular attention. Although the example shown here is for a rather simple case, the ideas behind the analysis can be extended to more complicated situations.—*Benford, H., Motor Ship, January 1968, Vol. 48, pp. 469-473.*

Measurement of Dead Load in Steel Structure by Magnetostriction Effect

There is always some stress distribution in steel structure, such as iron tower and bridge, even if there is no external force which may be caused by wind, loading etc. This kind of stress termed dead load, is due both to the constraining force between members of a structure in its construction and to the gravitational force.

There is no simple way to calculate this dead load theoretically, since the equilibrium equations are statically indeterminate by the existence of redundant members.

X-ray measurement of the lattice constant of the surface of a member may give the numerical stress distribution, and a photoelastic method has been developed recently for the same purpose. However, these methods require expensive apparatus and/or rather high techniques. A simple nondestructive method of dead load measurement given in this report enables us to measure stress distribution under the surface.

A magnetic measurement of dead load is made by attaching a U-shaped probe on the surface of a member to be examined. Since the permeability is a function of the stress, stress distribution in a member affects the permeability, which can be detected by an alternating current bridge.

The true stress, σ, is given by

$$\sigma = (1 \pm \alpha)\sigma_{\text{obs}} \pm (\beta) \text{ kg/mm}^2,$$

where σ_{obs} is a stress value corresponding to the unbalance current of the bridge on a calibration chart obtained with a standard test piece which has the composition, size, shape, and heat treatment of the member, α and β are constants which show an error. The second term depends on material. However, this can be decreased by measuring the stress twice at two different values of the bridge current.—*Iwayanagi, J., Yoshinaga, A. and Yoshii, T., May 1967. Ship Research Institute, Tokyo, Paper No. 19.*

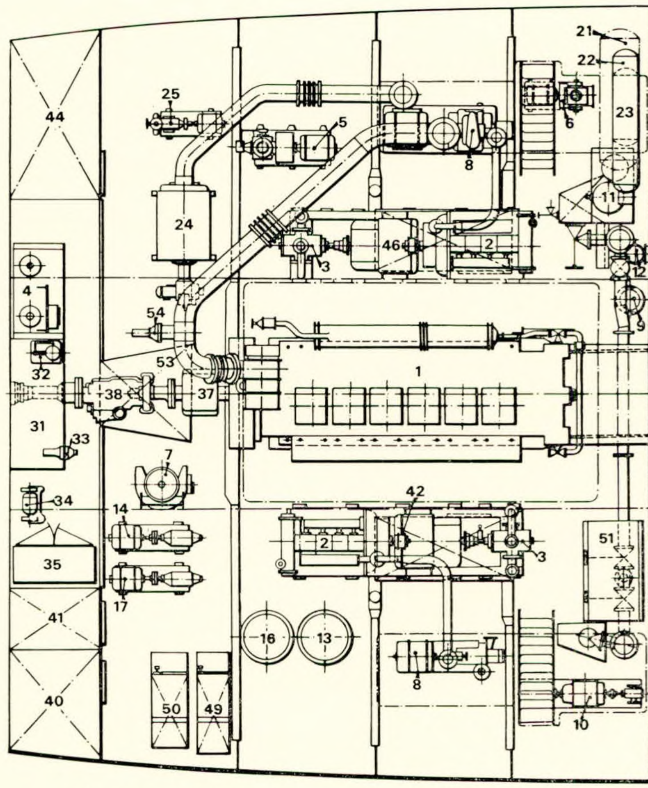
Controllable Pitch Propellers

Recent years have seen many design studies and a few applications of high powered gas turbines, either alone or in combination with other prime movers for ship propulsion. Some of the most pressing needs for such applications are minimum weight, minimum cost, highly efficient and highly reliable means of providing a reversal of propeller thrust with the unidirectional gas turbine. For most applications, the controllable pitch propeller can meet these needs.

Assuming, for a given application, that sufficient overall ship and power plant studies and trade-off analyses have justified a controllable pitch propeller, the next steps are to optimize the type and design. For high power controllable pitch propellers, hydraulic actuated types are considered mandatory. Even with this limitation, there are several types, previously used at about 20 000 shp and below, whose designs might be extrapolated to higher powers. These types vary in many important particulars including weights and dimensions. These variations can have significant effects on cavitation, efficiency, ship installation cost, operating capability, appendage drag, system reliability, maintenance cost and operating costs. Because of these effects, the following should all be considered for inclusion in specifications or as evaluation factors in selection of designs offered:

1) The ship's design speed, power, and propeller rev/min, both ahead and astern, are obviously the starting point. Any required margins over these values should also be determined.
2) The hydrodynamic design requirements of the propeller blades may or may not be determined by the controllable pitch propeller supplier.
3) The proposed combined machinery and propeller envelope of operating modes should be clearly stated.

For example, on some installations, pitch might have to be changed at low or zero ship speed and propeller rev/min whereas on others it might be changed at maximum conditions of power, rev/min and speed or intermediate between these limits. With more than one Diesel engine per shaft, the ability to increase pitch at maximum rev/min but with less than maximum ship speed and power would be a requirement to permit a smooth transition from, for example, two to three engine operation. The type of integrated control between prime mover governors and the propeller pitch may determine the maximum conditions at which pitch must be changed.—*Naval Ship Systems Command Technical News, November 1967, Vol. 16, pp. 26-39.*



1) main engine
2) Diesel alternator
3) hydraulic pump
4) side doors and hatch operating machinery
5) air compressor
6) emergency air compressor
7) fire pump
8) bilge and ballast pump
9) fuel oil transfer pump
10) lubricating oil transfer pump
11) lubricating oil separator
12) heater separator
13) fresh water hydrophore
14) pump for cold water
16) sanitary stagnant water hydrophore
17) stagnant water pump
21) main engine starting air bottle
22) main engine starting air bottle
23) auxiliary engine starting air bottle
24) heating boiler for accommodation
25) circulating pump for accommodation heating
31) working table
32) drill
33) vice
34) grinding tool
35) tool cupboard
37) thrust block
38) hydraulic pitch setting machinery
40) clean lubricating oil tank
41) clean lubricating oil tank
42) lubricating oil circulating tank for main engine
44) fuel oil settling tank
46) fuel oil drain tank
49) oil tank
50) kerosene tank
51) switchboard
53) oil tank for hydraulic pitch setting machinery
54) emergency pump for propeller

Roll-on/roll-off coaster

Roll-on/Roll-off Coasters

Four coastal cargo vessels were placed in service by Det Forenede Dampskibs-Selskab A/S (D.F.D.S.) at Copenhagen. Theye were all built by Cantiere Navale Felszegi, Trieste and were launched in pairs.

Built to the requirements of Bureau Veritas ✠3/3 L.1.1 A and CP, the hull is strengthened to Finnish ice class IB, has a soft-nosed and raked stem, cruiser stern, and an unusual line in that from about midships the upper deck to the stern is raised 6½ ft. As in open shelter-decked vessels a prime requirement was to keep the gross register measurement under 300 tons, the tonnage opening was placed aft on the upper deck. The continuous main deck provides an unobstructed run to the store room bulkhead forward with all hatches made flush fitting and only narrow side casings forward and aft encroach on deck space.

Main propulsion is by a six-cylinder MaK type Mu.45A unidirectional Diesel engine, developing 799 bhp at 350 rev/min, and directly coupled to a KaMeWa controllable pitch propeller. The engine is provided with built-in lubricating oil and cooling water pumps, and while normal starting, stopping and speed regulation is from the engine room, stepless variations of engine speed and emergency stop controls are installed on the bridge. The controllable pitch propeller is ice reinforced and can transmit 800 shp and instead of a fixed pitch spare propeller, spare blades for the controllable pitch unit are carried. The cast iron stern tube is of Deutsche Werft manufacture with white metal lining and Simplex packing.

Electric power at 380V 3-phase 50Hz a.c. is supplied by two 70kVA alternators, each powered by an air started four-cylinder Deutz type A4M.517 Diesel engine rated at 83 bhp at 1500 rev/min which is also clutch coupled to a hydraulic pressure pump. For harbour use there is a 15kVA alternator powered by an electrically started two-cylinder Deutz type A2L.514 Diesel engine rated at 25 bhp at 1500 rev/min. All the auxiliary engines are fitted with fresh and salt water cooling pumps and fresh water cooler with thermostat for water by-pass; and automatic stop for abnormal pressure and temperature limits.—*Shipbuilding and Shipping Record, 11th January 1968, Vol. 111, pp. 58-60.*

Large Bulk Carrier

A bulk carrier of 74 203 dwt has been built in Japan for Elcapitaine Inc. of Monrovia, an affiliate of Ceres Hellenic Enterprises Ltd, Piraeus, which is owned by the Livanos family. This vessel, *Fotini-L*, was constructed by the Hakodate Dock Co. Ltd, Hokkaido, and is the first of six vessels on order at this shipyard for the Livanos interests. The others include a sister ship and four smaller vessels of about 25 000 dwt. She is the largest ship yet built at the Hakodate yard. A special feature of *Fotini-L* is that she is the first Livanos ship to be equipped with fully automated and remote control systems for the main and auxiliary machinery and these enable her to operate without a watchkeeper in the engine room during the night.

Fotini-L is a single-deck ship with a bulbous forefoot, upper wing tanks, machinery and accommodation aft. She has been built to the American Bureau of Shipping classification A.B.S. A1 E and A.M.S. as bulk carrier and "Strengthened for the carriage of heavy cargoes—Nos 2, 4, 6 and 8 holds may be empty". The hull is divided by watertight bulkheads into 12 compartments, comprising forepeak, nine cargo holds, machinery space and after peak.

The propelling machinery in *Fotini-L* consists of an Uraga-Sulzer type 9RD90 turbocharged Diesel engine having a rated output of 20 700 bhp at 119 rev/min, and a service output of 18 630 bhp at 115 rev/min. During sea trials with the engine running at the m.c.r. a mean speed of 18·28 knots was attained at 122 rev/min. Control of the main engine can be carried out either from the wheelhouse or from an air-conditioned sound-proof compartment in the engine room. The propeller is of nickel/aluminium/bronze alloy and has five blades. The diameter is 6400 mm and the pitch ratio is 0·7469.

Electricity for power and lighting is obtained from two 600-kW main Diesel-alternator sets, one 300-kW auxiliary Diesel-alternator set and, when the vessel is at sea, from a 620-kW turbo-alternator set taking steam from the exhaust-gas boiler. The two main sets are each powered by a Daihatsu type 6PTSc-30 Diesel engine of 900 bhp output at 514 rev/min and the auxiliary set by a Daihatsu type 8PSHT-20F Diesel engine having a ratio of 540 bhp at 900 rev/min. The turbo-alternator is powered by an Uraga type UEG-500B steam turbine developing 920 hp at 1800 rev/min.—*Shipping World and Shipbuilder, February 1968, Vol. 161, pp. 403-409.*

Ship Industry Research in Poland

Poland's shipbuilding industry has announced plans to develop a large section of its network of research centres. Apart from the Computer Centre recently set up at Gdynia with an I.C.T. 1904 computer, a research centre is now to be built at Gdansk.

During the first stage of construction, which has just begun, the new centre will be equipped with a deep water tank 250 m long, an auxiliary tank, and laboratory and workshop sections. In the second stage of construction, scheduled for 1971, a wave tank and shallow tank will be built for the examination of passenger ships.

The central design and research centre at Gdansk is to be equipped with a computer section and a new department handling work on ship machinery and mechanism prototypes is to be built.

Research work at the Gdansk centre has been replanned to concentrate on electronics and automation and more than a third of the staff have been instructed to undertake theory.

Ship equipment research work has been transferred from the central design and research centre at Gdansk to the shipbuilding industry's engineering factories. Design offices recently set up at these factories are to be enlarged with a staffing of ten per cent of the total work force at each factory to allow independent research work to be carried out.

The Shipbuilding Institute of Gdansk Technical University will also be enlarged to concentrate research into ship hydromechanics and power plants. Plans have also been prepared for a branch of the Nuclear Institute of the Polish Academy.

Scientific staff at the Maritime Institute in Gdansk and the Sea Fishing Institute in Gdynia will be doubled and provided with research ships. The Sea Fishing Institute is to be equipped with an ocean-going ship laboratory, a trawler for stern fishing and a cutter.—*Shipping, December 1967, Vol. 56, p. 33.*

Comparative Methods for Preliminary Design

The first step in the design of a new ship is the determination of the geometry of the hull, dimensions, proportions and fineness. These parameters must satisfy a certain number of conditions or, in other words, they are the solutions to a certain number of equations. In practice, the number of such parameters is so great that the equations soon become prohibitively complicated if it is desired to take them all into account from the start; for this reason it is preferred to state a reduced number of equations which would yield the main parameters only and leave the adjustment of these, as well as the determination of the others, for subsequent steps of approximation until the final values are reached.

There are of course many methods for determining the basic parameters starting from the main requirements for the ship; it may perhaps be said that every designer has one of his own. Among those published, the author clearly prefers that given by Watson.

This method, however, as well as others, requires a certain amount of trial and error together with the use of several curves and data of a statistical character.

It is felt that, as any method for the first estimation of main parameters is of necessity approximate, it is preferable to have one as direct and short as possible, as well as reasonably accurate, because over-elaboration at this stage is not justified.

With the considerable technical data today available to the designer from widespread literature, it will always be possible to find one, or more ships, very similar to the one to be designed; this should enable him to start from actual figures rather than mean values as given by statistical data. Therefore, the method given is based on the assumption that a parent ship with its basic data is available to design the new ship. This has, furthermore, the advantage of facing the designer from the very start with a design of a similar, actually built ship, avoiding theoretical speculation and uncertainty as far as possible.—*Mandelli, A., Shipping World and Shipbuilder, January 1968, Vol. 161, pp. 69-72.*

Hot Gas Engine

Four specific fields of application can now be met by the hot gas engine as developed over recent years in the Philips laboratories at Eindhoven in Holland from the original Stirling hot air engine principles. They are:

1) hot gas engine;
2) cold gas engine;
3) refrigerating machine;
4) heat pump.

In principle the hot gas engine operates on external combustion. This permits the use of any kind of fossil fuel of almost any quality; moreover the fuel can be changed at any time.

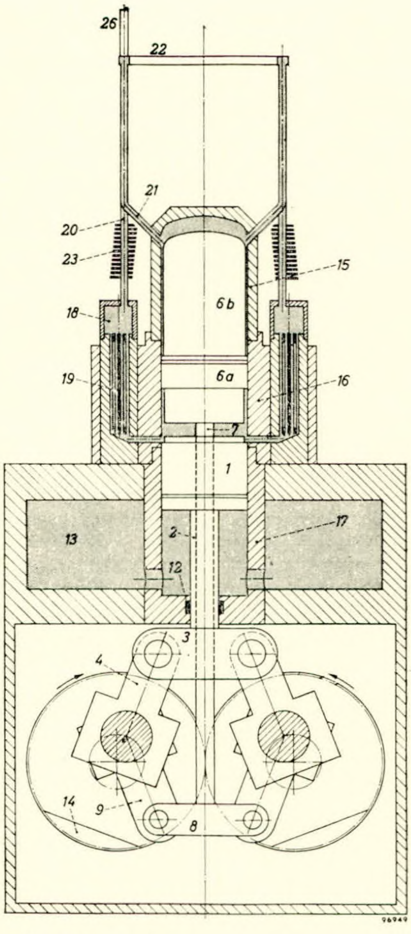
The Philips-Stirling hot gas engine as it is known today is a piston engine comparable in speed with petrol engines and quick-running Diesels but because of a special driving mechanism it is fully balanced, whereas the other types are not. Thermal energy is converted into mechanical energy by first compressing an enclosed quantity of working gas at low temperature and then, after heating, letting it expand at high temperature, around 700°C, (1292°F). As the piston that closes the gas filled working space needs to do less work to compress the gas at low temperature than is released when the gas expands at higher temperature, the engine delivers a net mechanical power.

A section through the essential elements is shown. The required temperature variations of the working gas, usually helium or hydrogen, are obtained by dividing the working space into a cold portion and a hot portion, between which the gas is moved to and fro by a displacer piston. For this purpose the two spacers are joined by a number of channels which are heated on the hot side (heater) and cooled on the reverse (cooler). When the gas flows from the hot to the cold space, the heat in the gas would be lost if there were no regenerator between the two heat exchangers. In the regenerator, which consists of a porous fine meshed metal mass and is a vital part of the whole assembly, this heat is stored and given up again when the gas reverses its direction. In principle, it should only be necessary now to supply heat to the heater in order to compensate the tendency to cool down during the expansion process; conversely, in the cooler, it should be necessary only to remove compression heat.

Cooler, regenerator and heater all are of annular shape, and are mounted round the expansion compression cylinder. Such an arrangement helps compactness and low gas flow losses. The displacer piston consists of the piston base (6a) and a thermally insulating dome (6b). The piston base fits like an ordinary piston into the cylinder (16) at the level of the cooler. There is a small clearance between the dome (6b) and the cylinder wall (15) of the hot space, just sufficient to prevent the two ever coming into contact. Underneath the power piston is the gas buffer space (13).

Two major problems needed solution before the double-acting principle could become effective. They were the drive and the piston sealing. The single-acting displacer system was simple because of the small pressure difference across the displacer piston, and the power piston had a constant low temperature on both sides. The drive question was solved by the invention of the Meijer rhombic drive.

Pressure sealing was a difficult problem because of the combination of moving parts, the need for an absolute seal to prevent escape of helium or hydrogen and the prevention of the



1) power piston
2) hollow piston rod
3) yoke
4) connecting rods
6a), (6b) displacers
7) displacer rod
8) bottom yoke
9) connecting links
12) piston rod seal
13) buffer gas chamber
14) counterweights
15), (16), (17) cylinder wall portions (in which power and displacer pistons move)
18) regenerator block
19) cooler block
20), (21), (22) heater tubes
23) cooling fins
26) tube for temperature probe

Diagrammatic section through Philips-Stirling hot gas engine

slightest leakage of lubricant into the working gas. The solution was found in the roll-sock seal.—*Gas and Oil Power, January 1968, Vol. 64, pp. 8-11.*

Residual Oil Combustion in Gas Turbines

The author gives a brief review of the problems encountered in the combustion of residual fuel in commercial gas turbines. The technical problems have been solved except for turbine deposit build-up, running base loads for long periods. Use of this fuel is economically marginal owing to increased operating complications and maintenance costs. —*Taylor, W. G., 1967, A.S.M.E. Gas Turbine Conference and Products Show, Paper No. 67-GT-19.*

Effects of New International Formulation on Turbine-generator Performance Calculations

For the pressure-temperature range of steam turbine operation, the new International Formulation properties differ from those in Keenan and Keyes Steam Tables primarily in the values of enthalpy and entropy around 1000° F. The deviation is significant at pressures above 2400 lb/in^2 and diminishes as temperature is both increased and decreased. This paper presents the effect of this deviation in steam properties on steam turbine generator heat rate and high pressure turbine efficiency. Some tests data are presented which indicate the new Formulation is more correct at 1000° F.—*Cotton, K. C., Deming, N. R. and Garbinski, E. H., September 1967, A.S.M.E.-I.E.E.E. Power Generation Conference, Paper No. 67-Pwr-9.*

Recent Sea-salt-in-air Implications as Highlighted by Marine Gas Turbine Requirements

Marine gas turbines are vulnerable to damage caused by the induction of salt-laden air. The compilation of data on sea-salt-in-air has been going on for years but these data, except for a small fraction, cover only average concentrations; very little sampling has been done during high velocity wind conditions. The authors describe their field trips on various naval vessels and the difficulties they encountered in attempting to collect sample deposits of heavy concentration. Their findings are given and some proposals are made to facilitate collection of this kind of data in the future.—*Kaufman, R. E. and Pollini, R.J., 1967, A.S.M.E. Gas Turbine Conference and Products Show, Paper No. 67-GT-44.*

Fuels for Marine Gas Turbines

Fuel properties that affect marine gas turbine operation are discussed. Particular emphasis is given to fuel-contamination factors including:

a) fuel contaminants affecting engine performance;
b) test procedures for measuring fuel contamination;
c) sources of fuel contamination;
d) prevention of fuel contamination.

Cabal, A. V., and Warren, J. H., 1967, A.S.M.E. Gas Turbine Conference and Products Show, Paper No. 67-GT-50.

Marine Gas Turbine Regenerator

Gas turbine heat exchangers of the stationary and rotary types are used to increase power plant thermal efficiency. Economic justification for the use of regenerative equipment depends upon the criteria of initial cost and component life. Materials selection and fabrication techniques must be in accord with cost and life objectives. In this regard, the U.S. Navy Ship Systems Command has sponsored a programme to investigate the particular problems associated with marine gas turbine heat exchangers. The results of the investigation obtained to date are given.—*Weber, H. A. and Thorsrud, E. C., 1967, A.S.M.E. Gas Turbine Conference and Products Show, Paper No. 67-GT-51.*

Smoke Elimination in Gas Turbines Burning Distillate Oil

Various methods of evaluating gas turbine smoke are described and compared. Two methods of reducing smoke in distillate-oil-fired units are evaluated. It is predicted that invisible stack plumes will be required in the near future. This can be achieved presently by the use of air atomization combined with a small quantity of fuel additive.—*Taylor, W. G., September 1967, A.S.M.E.-I.E.E.E. Power Generation Conference, Paper No. 67-Pwr-3.*

Some Model Experiments with Transom Flaps to Round-bottom Craft

Recent research in the Ship Division, National Physical Laboratory, has produced design data covering a wide range of high speed semi-displacement craft of round bottom section. Arising from this work, came the desire to examine the effect of fitting flaps to the stern of such craft. Transom flaps are short flat surfaces fitted to the underside of the craft and extending to the full width of the transom. They can either be in the form of solid wedge or they can be movable enabling them to be set to any desired angle. Their action depends on the development of an upward force concentrated over the flap, which significantly reduces the running trim of the vessel by forcing the stern upwards and in certain cases, also results in lower total drag.—*Bailey, D., September 1967, National Physical Laboratory, Ship Division, Ship Report No. 102.*

Cylinder Oil Analysis By Isotope at Sea

A team from the Gdansk Maritime Institute and the Nuclear Research Institute in Warsaw are making a round trip on the B512 bulk-carrier *Metalowiec* to study wear of engine components while under way. They will use the latest isotope techniques pioneered by Shell to investigate the effectiveness of cylinder oil and to establish the percentage of oil burnt while the engine is running. These methods will allow them to shorten the period of examination and eliminate the need to lift covers for gauging wear. They will also determine the percentage of oil which is burned and passes over with combustion gases and the percentage which reaches the crankcase.—*Marine Engineer and Naval Architect, September 1967, Vol. 90, p. 403.*

A New Method of Expressing and Measuring the Sealing Ability of Gasket Materials

This article presents sealometry, a method of establishing the sealing ability of gasket materials by the standarized measurement of the surface deformations under the action of an applied force. The "sealometer" used in the measurements employs standard surfaces which idealize the roughness of the sealing surfaces in an reproducible manner. The sealing ability of the gasket material is expressed in terms of its sealing factor.—*Roth, A. and Inbar, A., Jnl of Materials, September 1967, Vol. 2, pp. 567-580.*

Eddy Current and Ultrasonic Techniques for Inspection of Large Parts

An automatic eddy current scanning and recording system is described. The system is capable of inspecting large parts (26 × 10 ft) in approximately 50 per cent less time than conventional penetrant methods. It has additional advantages in that no etching or excessive cleaning is required. A technique for measuring the length and depth of discontinuities with a rotating eddy current probe is discussed. A brief discussion of automated surface inspection using ultrasonic surface waves is also included.—*Musil, F. J., Jnl of Materials, March 1967, Vol. 2, pp. 65-80.*

Lloyd's Register's Rules for Dry Cargo Ships

The main purpose of this paper is to discuss some of the background to Lloyd's Register's latest Dry Cargo Ship Rules. The previous revision of the Rules took place in 1954 and as several of the new Sections are based on different methods of analysis, it was considered worth while to set down for the record some of the reasoning behind the Rules.—*Roberts, W. J., 5th December, 1967 I.E.S. Paper.*

New Method for Determining the Instantaneous Value of Rate of Pressure Rise in an Internal Combustion Engine Cylinder

If it could be conveniently and accurately measured, the instantaneous rate of pressure rise in the combustion chamber of an internal combustion engine, in units of pressure (e.g. kg/cm^2) per crankshaft degree, might well give a useful quantitative indication of, for example, detonation. The conventional indicator diagram can hardly be used for this purpose because of the difficulty in drawing an accurate tangent.

The author describes in detail the principles of an electronic pressure-rise indicator which he has developed for the continuous determination and recording of this instantaneous rate of pressure rise. It can be used for research purposes and for workshop tests.—*Terada, K., Zeitschrift Verein Deutscher Ingenieure, 1967, Vol. 109, No. 4, p. 128; ibid, No. 5, p. 175; Jnl B.S.R.A., October 1967, Vol. 22, Abstract No. 25821.*

Influence of Rotor Metal on Bearing Failures Generally Classified as the Machining Type

An unusual and often severe type of journal and thrust bearing failure has been encountered in both marine and land-based turbine installations. These failures have occurred with steel rotors running against high-tin babbit metal. From the outward appearance of damage, these failures have been called "machining" or "wire-wool" failures. This paper discusses the results obtained from bench-scale research in which service-type failures can be reproduced. The failure mechanism strongly implies that rotors which are less susceptible to this type of failure can be developed.—*Karpe, S. A. 1968 I.Mech.E. Paper P10/68.*

The Port of the Future

This paper discusses the factors which will govern the type, location and capacity of ports required in the immediate future. These factors are primarily economic such as the changes in cargo and passenger traffic, the costs of the sea and land components of transport and the cost of competing systems such as air transport. Technical factors are also concerned, such as the possibility of building and operating larger and more specialized ships and new ways of cargo handling.—*Johnson, S. 10th January 1968, I.E.S. Paper.*

Radioactive Method for Investigating Anti-Wear Properties of Lubricating Oil Additives under Conditions of Electro-erosion

An experimental unit has been designed to model the friction of a cylinder piston ring block when applying constant voltage to the radioactive rubbing parts separated by a lubricating oil layer. Wear was measured with a radiometric scaler and the readings recorded by an electronic potentiometer. The authors found that at a constant voltage applied to the rubbing surfaces, electro-erosive wear takes place.—*Zaslavsky, Y. S. and Shor, G. I., Wear, 1967, Vol. 10, pp. 223-230; Fuel Abstracts and Current Titles, November 1967, Vol. 8, p. 112.*

Improving Metals under Pressure

There is now unequivocal evidence that physical properties of metals can be improved by working the metals under pressure. Specifically, if a metal is worked sufficiently under high enough pressure, it is found that the resulting metal is stronger and more ductile than the same metal worked the same amount by the same or equivalent conventional (non-pressure) processes. This result may well be general for all metals. Basic reasons for this metal improvement are lacking.—*Mechanical Engineering, June 1967, Vol. 89, pp. 58-59.*

Corrosion Test Methods for Heat Exchanger Application

Laboratory corrosion tests can be used to evaluate the resistance of commercial and experimental, copper-base, heat exchanger alloys to general overall attack, pitting and crevice attack and erosion-corrosion cracking. The results of laboratory tests can be used to predict the relative resistance of copper alloys to service conditions and to obtain a better understanding of the factors controlling corrosion resistance in various service environments.—*Wolfe, W. and Hager, S. F., September 1967, A.S.M.E.-I.E.E.E. Power Generation Conference, Paper No. 67-Pwr-7.*

Smokeless Combustion in Oil Burning Gas Turbines

This paper reviews the mechanism of smoke formation in oil firing gas turbines and the important smoke-controlling combustion parameters, of which the most important is the spatial distribution of fuel and air. Experimental determination of smoke particle size and quantity have been made, and the optical properties of gas turbine exhaust containing such smoke are presented. Proved solutions to the smoke problem lie either in fuel additives which inhibit smoke formation, or in combustor designs which proportion the fuel and air properly for smokeless combustion.—*DeCorso, S. M., Hussey, C. E. and Ambrose, M. J., September 1967, A.S.M.E.-I.E.E.E. Power Generation Conference, Paper No. 67-Pwr-5.*

Numerical Analysis of Crack Propagation in Cyclic-Loaded Structures

An improved theory is proposed for the crack-growth analysis of cyclic loaded structures. The theory assumes that the crack tip stress-intensity factor range ΔK, is the controlling variable for analysing crack-extension rates. The new theory takes into account the load ratio R and the instability when the stress-intensity factor approaches the fracture toughness of the material K_c. Excellent correlation is found between the theory and extensive experimental data.—*Forman, R. G., Kearney, V. E. and Engle, R. M., Trans. A.S.M.E., Jnl of Basic Engineering, September 1967, Vol. 89, pp. 459-464.*

Current Revolution in Overseas Transportation

The authors discuss the recent change in the philosophy of shipping companies engaged in overseas transportation to emphasize the overall movement of cargo from an inland point of origin to ultimate destination rather than merely from pier to pier. A brief historical outline is given of various unitized cargo operations which have been essential to the development of this new philosophy, followed by a summary of some of the problems involved in unitization. The economic significance of new developments is treated.—*Zubaly, R. B. and Lewis, E. V., August 1967, N.Y.A.S.-A.S.M.E Transportation Engineering Conference, Paper No. 67-Tran-33.*

Invention and Innovation in Transportation Overseas

A basis fact of maritime transportation is that three quarters of the globe is covered with water. The necessity for moving people and things overseas has mothered many interesting inventions and innovations. Inventors and innovators adapt the inventions of others to new purposes. In the history of maritime transportation, there have been quite a few originators and very many innovators. Most of the originators are known by name while most of the innovators are not. This paper describes some of the most interesting inventions with their historical context. — *Tyler, D. B., August 1967, N.Y.A.S.-A.S.M.E. Transportation Engineering Conference, Paper No. 67-Tran-46.*

Safety of Structural Components under Conditions of General Yielding

A maximum of safety using a minimum of material is the aim of every designer and, depending on the safety factors given, his design will be developed through variation in shape and/or changes in the choice of material. In the case of statically stressed members, a better utilization of material can be obtained if the safety factors used are related to the elastic strain limit rather than to the elastic stress limit, which latter is the usual criterion.—*Barp, B., Escher Wyss News, 1967, Vol. 40, No. 2, pp. 36-38.*

Considerations on Propeller Layout from the Engine Builder's Point of View

The paper gives the engine builder's point of view on propeller layout, especially in view of the service results obtained with the recent bulk carriers and tankers of large tonnage. The author points out that with a properly laid out propeller more satisfactory service results can be expected, with increased reliability and reducd maintenance cost. This ruling is applicable to all types of propulsion equipment, Diesel and steam alike, although the effects can be somewhat different. The most important factor remains proper maintenance of the hull.—*Smit, J. A., 27th February 1967, I.E.S., Paper No. 1329.*

Application of Wave-making Resistance Theory to Hull Form Design

In recent years, the wave-making resistance theory has shown remarkable progress, especially in connexion with its use in high speed electronic computers. These facts have made it possible to design an ideal ship hull form by applying the theory. This report gives some examples of the application of wave-making resistance theory to hull form designs of a large oil tanker and a high speed cargo ship. However, such a theory has many defects in practical application and further investigations are required for its practical use.—*Abe, M. and Ohkusu, M., Japan Shipbuilding and Marine Engineering, November 1967, Vol. 2, pp. 27-37.*

Electrochemical Study of Aluminium Corrosion in Boiling High Purity Water

The purpose of this study was to ascertain the merchanism by which a protective oxide film is formed and maintained on the surface of a metal such as aluminium when immersed in water. The work involved an examination of the cathodic polarization behaviour of 1100 aluminium during aqueous oxidation. Measurements were restricted to cathodic polarization because anodic polarization is more likely to alter the system under study. The choice of 1100 aluminium rather than high purity material was prompted by the rapid intergranular corrosion which is characteristic of high purity aluminium in pure water.—*Legault, R. A. and Draley, J. E., Corrosion, December 1967, Vol. 23, pp. 365-370.*

Ferrofluid Sensor

An accelerometer sensor small enough to fit in the palm of the hand and applicable to flight control of airplanes, submarines etc. is under study.

The design of the accelerometer incorporates a cylindrical housing filled with a highly magnetic fluid. The accelerometer sensor differs from ordinary instruments of this type by having its only moving part, the proof mass or small body, immersed in the fluid. Supported in ferrofluid, the proof mass is levitated so there is no mechanical friction and any force applied to the proof mass will cause a motion. — *Mechanical Engineering, October 1967, Vol. 89, p. 55.*

Warp Tension Meter for Seine Net (Fly Dragging) Vessels

Warp tension meters, employing an electronic system developed by the White Fish Authority, were installed on the seine-net vessels *Opportune II* and *Altair*. Although the operational advantages were firmly established during a prolonged assessment in commercial fishing, the saltwater environment proved too severe for the electrical system. It was decided to design warp tension meters of a more robust nature and Kelvin Hughes Ltd undertook to produce suitable units. The outcome is a system using an hydraulic cell to measure the warp tension with a pressure gauge to indicate the load.—*February 1968, White Fish Authority, Bulletin No. 27.*

Measurements of the Components of Resistance on a Tanker Model

This report gives details of measurements of local shear stress, pressure, wave pattern and total resistance of a model of raked bow tanker. The shear stress and pressure measurements have been integrated over the hull surface to give the frictional and pressure components of resistance. Combination of these measurements suggests that the method of measurement of skin friction and the hypothesis upon which it is based are sound. The pressure form effect has been deduced and is seen to be large compared with that of a streamlined body.—*Steele, B. N., December 1967, N.P.L., Ship Division, Ship Report No. 106.*

Design Study of Unmanned Engine Room for Stern Freezer Trawlers

This report describes an automatic system to replace the permanent engine room watch at present maintained on a freezer trawler. The report outlines a basic system which is limited strictly to the functions necessary to the unmanned engine room concept as defined in the study specification, but various optional extras and alternatives are put forward to be added to the basic minimum system, depending on the final composition of the engineering team or if further reduction of engineering effort or improvements are desired by the owner.—*January 1968, White Fish Authority, Bulletin No. 26.*

Development of a High Output Four-cycle Engine, Supercharged on the Constant Pressure System

The first part of the paper contains theoretical considerations covering airflow requirements of the engine, turbocharger matching and selection and an estimate of the expected gas pulsation amplitudes in the exhaust manifold using nonsteady-state gas dynamic calculations. The second part contains some design features of the engine followed by actual test results on two modes of operation, Diesel and dual fuel. In the final part of the paper, the test results, are discussed and the advantages and disadvantages of the constant pressure and pulse supercharging system are analysed.—*DeWilde, E. F., 1967, A.S.M.E. Diesel and Gas Engine Power Conference and Exhibition, Paper No. 67-DGP-11.*

Comparison of High-powered Single-engine and Multi-engine Plants for the Propulsion of Merchant Ships

The author discusses the principal factors in the evaluation of high output marine Diesel propulsion plants. The alternate systems discussed are the large, low speed Diesel engine with direct propeller drive, several medium speed Diesels driving through reduction gears and many high speed engines used with electric power transmission. Detailed tables of cost, maintenance, efficiency and so on are included. The author ends with a summary and discussion of the advantages and disadvantages of the various combinations.—*Zinner, K., 1967, A.S.M.E. Diesel and Gas Engine Power Conference and Exhibition, Paper No. 67-DGP-2.*

Piston Engine Heat Recovery Systems

The author describes several of the criteria for design and selection of heat recovery systems. The first section of the paper delineates a number of the considerations involved in designing heat recovery silencers, steam separators, and related equipment. The second section enumerates some of the rules and reasons used when selection of the equipment is made for particular installations.—*Clay, P. E., 1967, A.S.M.E. Diesel and Gas Engine Power Conference and Exhibition, Paper No. 67-DGP-1.*

Connecting Rod Design

The paper describes laboratory measurements of various deflexions within and around the split big end of a certain Diesel engine connecting rod under various forms of static loading. Crude assumptions are made about the magnitude and nature of dynamic forces involved in actual running. The author concludes that the best design will be the one yielding the lightest part with the lowest stress and deformation.—*Demirguc, A. Z., Bulteni Istanbul Teknik Universitesi, 1966, Vol. 19, No. 1, pp. 122-137; Applied Mechanics Reviews, December 1967, Vol. 20, p. 1191.*

Control Systems for and Sea Operation Experience with CODOG-powered Frigates

The Royal Danish Navy frigate *Peder Skram* successfully passed her sea trials and has been in operation for about 15 months, with an accumulated gas turbine operation of 490 h at the end of December 1966. The frigate has twin-screw CODOG propulsion machinery consisting of a 22 000 shp jet engine-fed power turbine and a 2400 shp two-stroke Diesel engine. These two alternative prime movers drive the propeller shafts with controllable pitch propellers through a common reduction gear including free-wheeling clutches. The control system is described.—*Svensson, S. O., 1967, A.S.M.E. Gas Turbine Conference and Products Show, Paper No. 67-GT-43.*

Cylinder Liner Scuffing

Laboratory equipment using reciprocating ring segments on water-cooled liner sections was developed to permit rapid evaluation of the variables which influence cylinder liner scuffing and this gave good correlation with engine results. Chromium-plated cylinder liner scuffing can be controlled by proper specification of the liner finish, ring material and lubricant. Chromium-plated liner surfaces with a random roughness pattern that does not include large land areas were found to be best for break-in.—*Avery, R. W., 1967, A.S.M.E. Diesel and Gas Engine Power Conference and Exhibition, Paper No. 67-DGP-12.*

Acid Cleaning of Heat Exchangers

Cleaning of heat exchangers usually involves shutting down a segment of a gaseous plant which causes many difficulties, such as production losses and equipment failure. However, Goodyear Atomic Corp. has developed a method of cleaning heat exchangers without interrupting production. Acid is injected into the cooling water before the water enters the heat exchanger. — *Koehler, F. A., Materials Protection, December 1967, Vol. 6, pp. 28-29.*

Thermodynamic Design of Combined Steam and Gas Turbine Marine Propulsion System

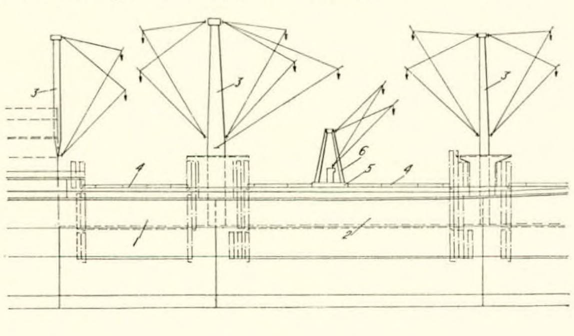
The authors present the thermodynamic design of a 25 000 shp combined steam and gas turbine (COGAS) power plant for marine propulsion. The plant includes a high pressure ratio, aircraft-type gas turbine which exhausts into an unfired, low pressure waste-heat boiler. The paper considers all aspects of the thermodynamic design including selection of the feed heating system, steam conditions and gas turbine cycle. It also considers the different part-throttle operating modes and describes the major components. — *Marwood, R. M., and Bassilakis, C. A., 1967, A.S.M.E. Gas Turbine Conference and Products Show, Paper No. 67-GT-16.*

Patent Specifications

Ship's Supplementary Loading Device

The transhipment or handling capacity of the conventional loading devices is generally not sufficient to carry out the transhipment from long cargo holds as rapidly as is possible with standard long hatchways on the same ship.

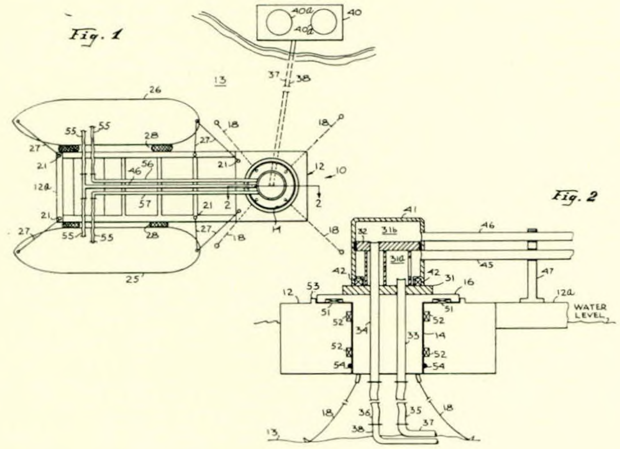
It is the object of the invention, in connexion with long cargo hatchways on ships, to keep free the entire hatchway aperture during the transhipment of long and bulky goods and, during loading operation with normal goods, to accelerate the process by means of supplementary loading devices.



The cargo hold (1) has a standard cargo hatchway, as shown in the illlustration. The cargo hold (2) has, due to its considerable length, a correspondingly long cargo hatchway. Hatchway covers (4) are arranged on the hatchway side coaming close to the cargo hatchway of the cargo hold (1). The long cargo hatchway of the hold (2) is sealed off only partly by the hatchway covers (5) while the remaining portion is sealed off by a hatchway cover (5) which is also connected with it in such a manner as to be fast therewith but is releasable. Securely mounted on this hatchway cover (5), is the supplementary loading device (6), which has a form of king posts, framework with derricks or cranes and can, after it has been released from the hatchway coaming, be lifted by the conventional loading devices (3) and set down at any desired point on the hatchway side coaming or deposited at some other point on board or on land.—*British Patent No. 1 094 090 issued to VEB Warnowwerft Warnemünde. Complete specification published 6th December 1967.*

Floatable Terminus for Tankers

In Fig 1 is shown a terminus (10) which includes a pontoon (12) which is floatable and includes a floatable pier (12a). In the opening (14) within the pontoon, is mounted a buoy (16) which is anchorable to the bottom of the sea (13). When the buoy is anchored, it is prevented from rotating or

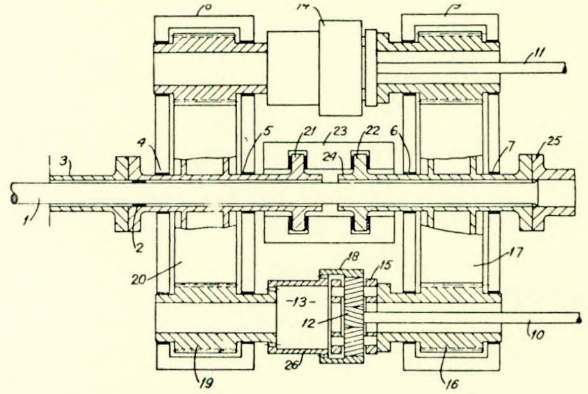


drifting by chains (18), but the pontoon (12) and the pier (12a) are rotatably coupled to it and can rotate freely. The pontoon (12) and pier (12a) include mooring hooks, bollards or bitts (21) used to tie up or moor tankers (25) and (26) to the pier by means of mooring cables (27). The pier (12a) is fendered by guards (28) to prevent damage to the ships coming alongside it and to the pier itself.

Fig. 2 is a cross sectional view along line (2-2) of Fig. 1 and shows an internal multi-chamber unit (31) mounted on top and forming part of the buoy.—*British Patent No. 1 096 080 issued to Imodco International Ltd. Complete specification published 20th December 1967.*

Shafting Arrangement for Two Counter-rotating Propellers

(1) in the drawing indicates an inner propeller shaft journalled in a bearing (2) within an outer propeller tube shaft (3) which is journalled in bearings (4) and (5). Each of the shafts supports a propeller (not shown) and is driven via parallel gears (8) and (9), and (10) and (11) indicate the shafts leading from power units (not shown). Each one is connected to a sun wheel (12) in a planet gear (13) and (14), the planet carrier (15) of which is connected to a pinion (16) engaging one of the parallel gears, (17) while the inner toothed ring (18) of the planet gear is in engagement with a gear (20) of the other parallel gear. Located in the space between the parallel gears and planet gears is a thrust bearing (21) for shaft (3), and



a thrust bearing (22) for the shaft (1) combined with a housing (23).

The front bearing (22) is connected to a tubular shaft portion (24) supporting and secured to the gear wheel (17) and which is connected to the shaft (1) by means of a coupling (25) on the forward side of the parallel gear (9). The shaft (24) is journalled in bearings (6) and (7). (26) indicates a resilient member in the form of a gear coupling between the planet gear and drive.—*British Patent No. 1 094 669 issued to Stal-Laval Turbin Aktiebolag. Complete specification published 13th December 1967.*

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