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TRANSACTIONS (TM)

Paper No. 12

STEAM/DIESEL CONVERSION OF ORE/OIL CARRIER

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Read at 17.30 on Monday 9 March 1981

ISSN 0309-3948
Trans I Mar E (TM)
Vol. 93, 1981 Paper 12

Steam/Diesel Conversion of Ore/Oil Carrier

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SYNOPSIS

The conversion of the main propulsion machinery of a steam-turbine powered ore/oil carrier of some 225 000 dwt to diesel-power is described. The authors mention a serious breakdown of the original steam turbine and gearing and technical and commercial inducements to re-engine the vessel with geared medium-speed diesels. Cost considerations extended over a five-year life cycle. The practical aspects of the conversion design study are discussed, particularly the selection of gearing and alignment. Some of the original plant in the vessel was utilized. The various electricity supply alternatives are discussed and maintenance of a high-powered medium-speed diesel propulsion installation is considered.

INTRODUCTION

In September 1977, *Alva Sea* suffered a major machinery casualty in the Red Sea whilst on passage from Falconara in Italy to the Arabian Gulf. This left the vessel totally immobilized.

The whole of the LP turbine and cylinder had been destroyed, together with its primary transmission system, including the torque tube and associated toothed couplings, the LP epicyclic gear assembly, the LP secondary helical pinion and the main bull wheel itself. The damage to this latter was beyond repair, necessitating its complete replacement. The LP turbine rotor had overspeeded, resulting in severe damage to the turbine itself.

Widespread damage resulted from the centrifugal disintegration of the LP rotor exhaust-end stage wheels where these had ruptured the turbine cylinder attachment to the condenser exhaust casing. Flying debris had pierced various service pipes, electrical cables, engine room walkways, floor plates, electrical fittings, steam pipe and turbine casing insulation generally. Considerable damage had also been suffered by the main condenser, the debris passing through the exhaust casings of the axial-flow condenser and into the tube stack itself. The external damage is shown in Fig. 1.

The extent of the damage necessitated the complete replacement of the main LP turbine, its pedestal bearing (Fig. 2) and foundations, together with the whole of the LP transmission system, including the secondary pinion and main bull wheel.

Extensive repairs would also have been necessary to the main condenser and to the external systems and equipment damaged during the failure.

Therefore it emerged that, to restore propulsive power to the vessel, heavy expenditure would be required in replacing the main items described above and effecting repairs external to the LP turbine and gearing.

The replacement of the main bull wheel was a major element in the estimated cost. Further, the delivery period for a wheel of the size required for *Alva Sea* was extensive, a fact that was to feature strongly in the subsequent decisions.

ECONOMIC AND TECHNICAL APPRAISAL

After due consideration of the turbine plant repair costs involved, together with the corresponding down-time, the possibility of re-engining with diesel machinery was raised, naturally influenced by the rising costs of bunkers throughout the world. The latter fact became a prime consideration in the early part of 1978 when world-scale rates were abnormally low and bunker prices were continuing to rise. Figure 3 shows the statistical trends of both voyage charter rates and bunker prices for the period in question.

The original installation was designed for 32450 shp, giving an average service speed of 15.75 knots. The comparison between the steam and diesel installations was first made on an equivalent power basis. This would have called for a diesel installation delivering 32 000 shp which, using normal margins, would have meant an mcr of some 38 800 shp.

It was debatable at that time as to what power/speed would be most appropriate for a motorship of this size, but it was clear that increases in fuel costs would continue.

Eventually, it was decided arbitrarily to consider a diesel installation of 32 000 bhp mcr for normal operation at 27 200 bhp, i.e. 85 per cent. Table I compares the prime factors of each alternative, reducing the equivalent power output of the turbine to the same basis of 27 200 hp. For the purpose of this initial exercise the gearing and shaftline losses were ignored, i.e. the diesel machinery brake output was considered the same as the turbine plant shaft horsepower.

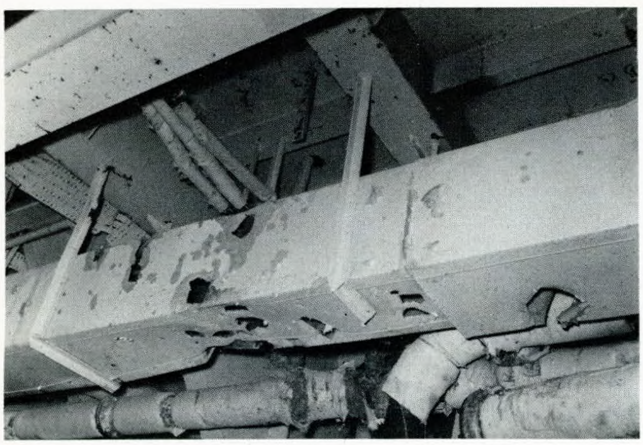


FIG 1 External damage caused by flying debris

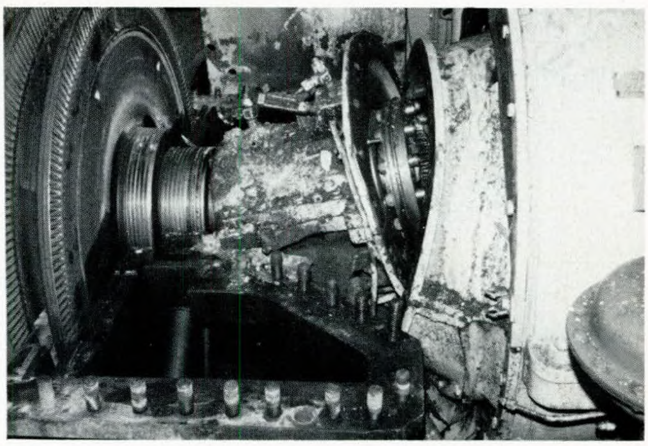


FIG 2 LP turbine pedestal bearing damage

T. C. Roomes trained under the alternative scheme with the Furness Prince Line. He then spent 10 years with P & O as an Assistant Superintendent in sea and shore appointments. In 1969 he joined the Vlasov Group as manager of new construction and is presently the Fleet Manager responsible for the operation of a fleet of eight product carriers.

Takesh Fujii studied mechanical engineering at the Yokohama National University in 1963. He then joined the ship design department of M H I and is currently Project Leader in the ship repair department of Mitsubishi Heavy Industries, Yokohama.

Table I: Basis of cost considerations

PARAMETERS	STEAM TURBINE PLANT Maximum	*De-rated*	DIESEL INSTALLATION
Basis (shp)	32 000	27 200	32 000 mcr 27 000 (85%)
Speed (knots): laden	15.00	14.25	14.25
ballast	16.50	15.75	15.75
Consumption (L.tons/day)	160	136	97
Spec. fuel cons. (g/hph)	About 210		About 150
LO cons. (kg/day)	25		554
List price kg (US$)	0.707		0.6252
Capacity (dwt)	225 000		224 500
Cargo (dwt)	214 000		216 500
Effective days	350		345
R&M annual costs (US$)	base		80 000
Crew requirements	base		+1 Eng.Off. +2 PO/mech.

The $80 000 for additional maintenance and repairs was assessed on the basis of recommended overhaul periods and the estimated consumption of spare parts. Insufficient service experience exists to date to confirm this assumption.

The cost escalation figures used have also varied, particularly in the case of fuel and insurance costs. However, the present indications are that our general assumptions were reasonably accurate.

Cost appraisals were made on the basis of a life of five years which confirmed conclusively the advantages of converting to diesel machinery in lieu of the original steam turbine plant, using as parameters the following basic elements.

1. Rate of return on investment capital 12.5 per cent pa.
2. Loan capital of conversion cost mortgaged at 8.5 per cent over a period of five years.
3. Propulsion power of diesel installation assumed as 8.5 per cent of turbine installation.
4. Fuel quality for the medium-speed diesel installation assumed as 180 cSt. (Although engines designed to burn 380 cSt.)
5. Premium for 180 cSt assumed to be US$2.5/ton above 380 cSt fuel, applicable at that time.
6. Cost escalation for bunkers and lubes 12 per cent pa; crew costs 13.5 per cent pa; maintenance and repairs 20 per cent pa and insurance costs 5 per cent pa.

It was considered that significant advantages could be derived from a conversion to diesel and a full technico/economic study was put in hand.

CONVERSION DESIGN STUDY AND APPLICATION

Design objectives

The general objective was to install an acceptable propulsion system within the existing machinery spaces of the vessel that would allow for a continuous reliable operation within the skills and abilities of the operating personnel. Such a propulsion system was to take full advantage of the economic benefits derived from the reduction in fuel consumption for given ranges of speed. The alternative installation finally selected was to be:

(a) reliable and of proven background with a good service reputation for the powers envisaged;
(b) capable of being maintained within the normal confines of tanker service, i.e. short transit times in port, full vessel mobility during discharging and loading operations;
(c) capable of burning high-viscosity fuel;
(d) capable of being entirely serviced at sea with good maintenance accessibility, to assist sea staff to make scheduled overhauls in as good an environment as possible;
(e) possess a low space/power ratio, to fit the machinery space originally designed for a turbine plant.

Design constraints

Since the time available for taking the final decision was limited, it was necessary to impose, by judgement, certain constraints:

(i) principal dimensions of existing machinery spaces to be maintained;
(ii) original shafting, together with the original fixed-pitch propeller, to be retained;
(iii) utilizing as much as possible of the existing plant and equipment while keeping the converted plant's efficiency as high as possible;
(iv) making 'slow-steaming' possible to potential charterers' requirements;
(v) leaving the original cargo pumping plant and systems unchanged.

Selection of diesel machinery

All available machinery alternatives were reviewed in order to satisfy these objectives and constraints. Slow-speed, directly reversible main diesel engines are normally preferred by Vlasov in its newbuilding policy because they can safely burn highly viscous residual fuels. However, due to the space constraint imposed, this alternative was considered impossible without extensive structural changes to accommodate either a twin- or a single-screw installation.

The possibility of a twin-bank, slow-speed diesel on a common bedplate, driving the propeller via a reduction gearbox, was briefly investigated but abandoned because the available power range was below the continuous-service requirements for a vessel of *Alva Sea*'s size.

The owners' technical investigations led to a medium-speed geared installation, mainly due to the space limitations involved. This also enabled the existing shaft line to be preserved with minor modifications, together with the original propeller.

Intensive investigations were begun to select the most reliable installation capable of satisfying our basic design objectives. A short list of medium-speed engines was prepared. Following discussions on their service reputations a final decision was made in favour of the

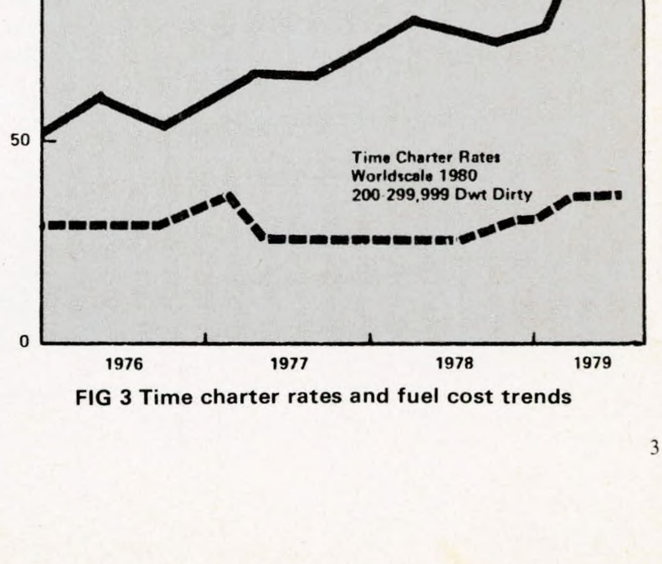


FIG 3 Time charter rates and fuel cost trends

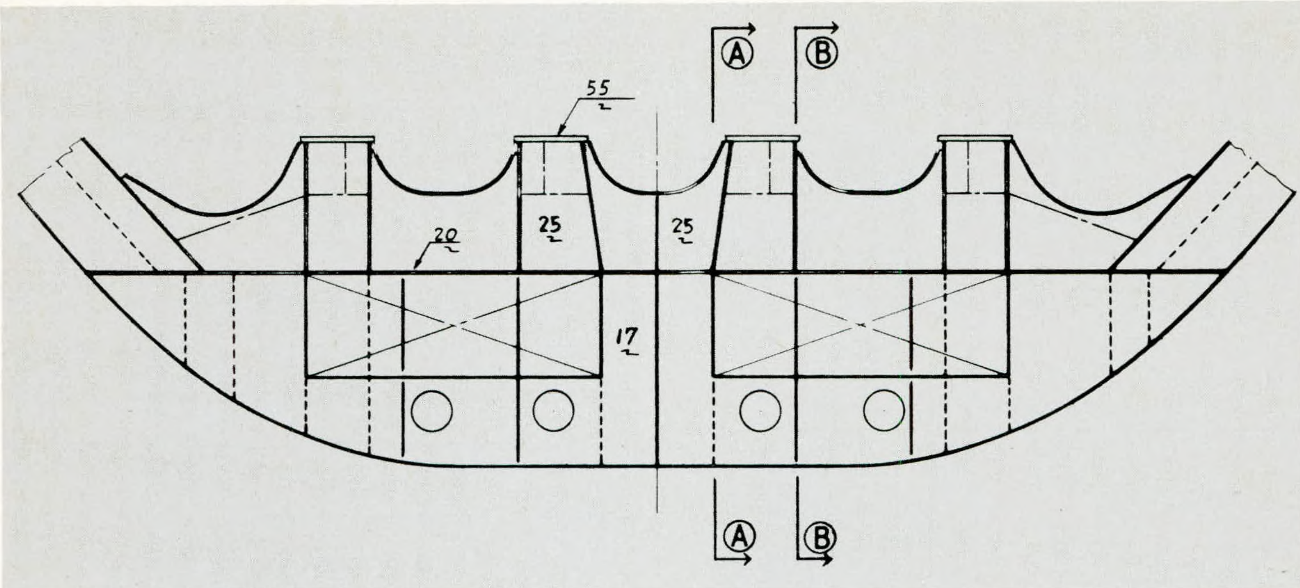


FIG 4 Transverse section of new engine and gearbox foundations at Fr. 47

MAN V52/55; a high point in favour of this engine being its proven service reliability, conservative rating and, above all, the fact that units had been at sea since September 1970.

Particular attention was paid to the bedplate and foundation, which is a single, robust, rigid casting, containing heavy webs and integrally-cast main bearing pockets. Considerable attention was also paid to the suitability of this engine to burn residual fuels, always an area of risk with any trunk engine.

Its piston design has been specially developed to provide an adequate seal between the combustion space and crankcase when under load. The piston is of composite construction having an aluminium skirt and a steel crown. This latter has a barrel-shaped vertical section, and a semi-elliptical horizontal section designed to make the piston tend to assume the shape of the cylinder when under load which assists the gas sealing.

The engine also has independent cylinder lubrication at the upper liner where alkaline cylinder oil is injected at six points; a further precaution against the burning of heavy fuels.

Selection of gearing

Considerable attention was given to the selection of suitable gearing by joint consultation between the shipyard and owner and certain gearing manufacturers, both in Europe and Japan. In addition, the owners compared their own operating experience of certain medium-speed reduction gear installations, operating in smaller units in their Group, with that gained by other ship operators in the industry.

The basic considerations were that designs should accommodate:

1. structural flexing of the vessel due to varying draught conditions and the effects on mainshaft alignment;
2. variations of gear tooth loading due to the effects of weather and hull fouling;
3. the characteristics of the original shaftline and propeller;
4. differences in shafting and engine alignment between initial cold alignment data and hot, full-load running conditions.

It was decided to fit a straightforward, reversible, single-reduction, double-helical gear train with facility for de-clutching either engine.

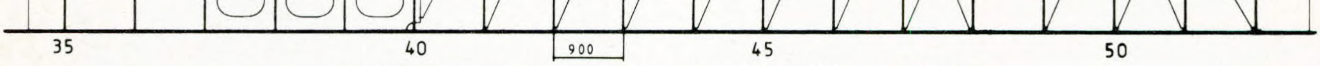


FIG 5 Longitudinal section of engine girder (625 mm off CL)

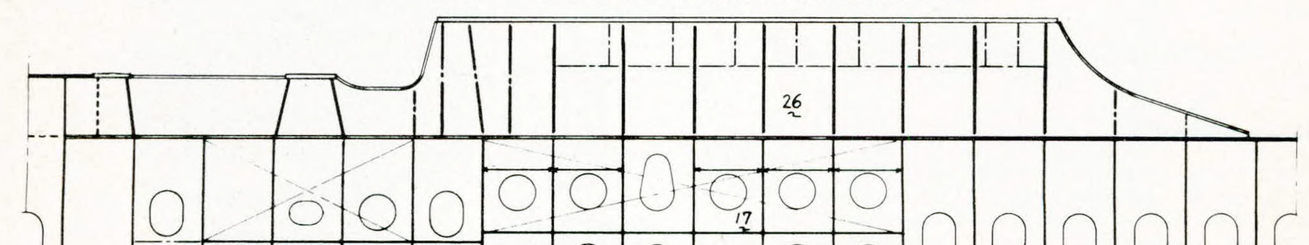


FIG 6 Longitudinal section engine girder (1600 mm off CL)

Structural flexing

Because of the natural flexing of the vessel resulting from draught differences between the light and heavily laden conditions and the related effect of the buoyancy of the machinery space, a heavy, stiff structure was considered necessary for both the new main engine and gearbox foundations. This was to contain the effects of hull deformation in both the longitudinal and transverse planes. The natural frequency of the structures was calculated and the design ensured that this frequency was very different from the propeller-blade frequency.

The heavy seating structure, as shown in Figs 4–6, was designed to carry both the main engine and gearbox foundations integrally. In particular, the depth of the modified hull girder in this area is designed to provide maximum rigidity.

The gear casing was also of rigid design, its stout construction chosen not only for transmitting high powers and resisting torque effects, but also to accommodate a large, heavy main wheel and the corresponding unusually large span between the centrelines of the new twin engine installation.

Variations of gear tooth loading

Increases in hull resistance due to weather or hull fouling are substantial in vessels of the size under consideration. In order to allow for this, the shipyard paid considerable attention to the gear parameters, allowing ample margins beyond classification requirements for gear-tooth strength, surface finish and specific tooth loading.

The gears installed consisted of high-quality, carburized, case-hardened pinions, driving a through-hardened wheel. Both pinions and wheel were profile-ground to a minimum surface finish quality of 3 μm.Ra.

Table II shows the parameters employed were very conservative, particularly in the 'K' values and the tooth module.

In addition to mechanical, torsional vibration-damping at the end of each engine, resilient rubber couplings were also provided between each main engine and the gearbox input shafts. The combined effect of these two devices rendered the plant free from torsional vibration throughout the whole of the operational range.

Whilst this type of flexible rubber coupling will dampen a substantial part of the torsional vibrations, a certain proportion may be transmitted, particularly if the engine is operating off balance due to poor combustion. In calculating the specific loading of the gear teeth, due consideration was, therefore, given to the stiffness of the resilient couplings. For further protection of the gearing against excessive torsionals resulting from engine imbalance, an electronic instrument has been permanently installed. This is capable of measuring mean effective and maximum pressures, as well as indicated horsepower of each main engine cylinder. This instrument has subsequently proved a valuable feature during the vessel's operation in enabling ship's staff to monitor engine balance regularly and accurately.

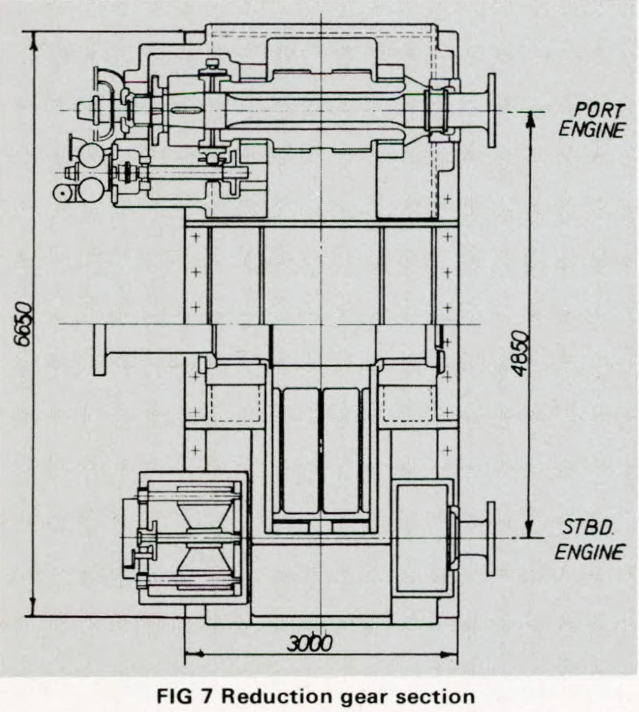
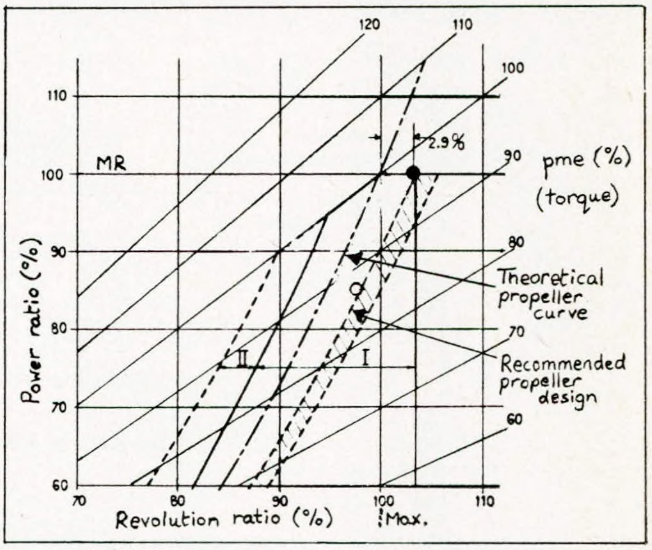


FIG 7 Reduction gear section

Table II: Particulars of reduction gear

PARAMETERS	PINION	WHEEL
Type	single reduction	
Transmitted horsepower	2 × 16 000 PS (2 × 11 768 kW)	
Input revolution	430 rev/min	
Output revolution	82.3 rev/min	
Reduction ratio	5.222	
Kind of teeth	Double helical	
Module	14	
No. of teeth	54	282
Breadth of teeth	420 mm × 2	
Helical angle	28° 45″	
Pitch circle diam.	862.7 mm	4505.1 mm
Pressure angle	25°	
Material	Ni–Cr–Mo steel	
Heat treatment	Case hardening	Through hardened
Hardness HB	577–634	280–290
Surface finish	Profile grinding	
K-value	10.2 kg/cm² (IMPa)	
Centre distance	2683.9 × 2	



MR : maximum rating
I : continuous running permitted
II : can be run for restricted periods
● : aimed proving point at completion of conversion
○ : aimed service operation point

FIG 8 Propeller revolution design point

Original shaftline and propeller

After careful consideration, it was decided to retain the original shaftline and fixed-pitch propeller. It was then necessary to consider the effects of hull resistance upon engine and gear loading. This was allowed for by arranging the reduction gear ratio so as to afford a margin of protection against possible thermal overloading of the engine due to excessive torque arising within the theoretical power/propeller-revolution relationship.

In the conversion a power of 32 450 shp at 86 rev/min was reduced to 32 000 bhp at 430 rev/min.

In order to estimate the available shaft horsepower after conversion, losses were deducted from the diesel engine's maximum power as follows: new reduction gear loss and thrust and shaft line losses (2 per cent); shaft-driven pumps (1 per cent). The available shaft horsepower then becomes $32\,000 \times (1{-}0.03) = 31\,040$ and the corresponding propeller speed is

$$86 \times \sqrt[3]{\frac{31\,040}{32\,450}} = 84.7 \text{ rev/min}$$

However, an allowance was needed for hull deterioration. A review of the vessel's operating data since she had been commissioned some four years previously revealed that the propeller revolutions had fallen about 2 per cent from their original design point. Taking this into

Table III: Calculation results of bearing reaction force and shaft offset

NAME OF BEARING			AFT STERN TUBE BRG *Aft end*	AFT STERN TUBE BRG *Fore end*	FOR'D STERN TUBE BRG	INTERM. BRG	RED. GEAR AFT BRG	RED. GEAR FOR'D BRG
Cold condition	Reaction force	kN	657	55	226	290	198	279
		(kg × 10^3)	(67.0)	(5.6)	(23.1)	(29.6)	(20.1)	(28.4)
	Mean surf. Pressure	Mpa	7.13	1.31	0.36	0.95	0.55	0.77
		(kg/cm^2)	(72.7)	(13.4)	(3.7)	(9.7)	(5.6)	(7.9)
	Off set	mm	0.45	0.0	0.0	3.1	6.7	6.7
Hot condition	Reaction	kN	662	42	245	263	259	234
		(kg × 10^3)	(67.5)	(4.3)	(24.9)	(26.8)	(26.4)	(23.9)
	Mean surf. pressure[a]	Mpa	7.18	1.00	0.39	0.86	0.72	0.65
		(kg/cm^2)	(73.2)	(10.2)	(4.0)	(8.8)	(7.3)	(6.6)
	Off set	mm	0.45	0.0	0.0	3.1	6.1	6.1

[a]Mean surface pressure is calculated dividing reaction force by bearing area where pressure actuate and not by total bearing area.

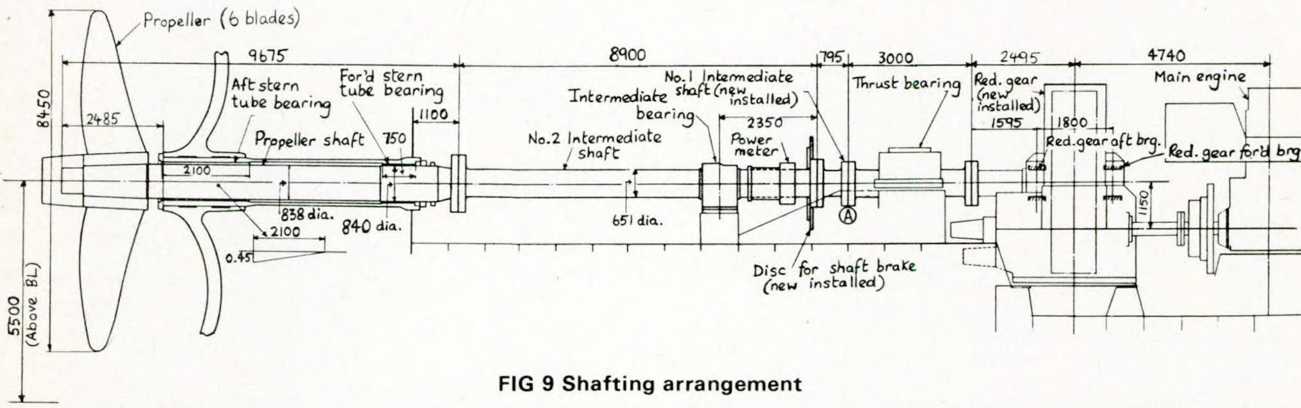


FIG 9 Shafting arrangement

account, together with an allowance for future deterioration, the reduction gear ratio was designed to be

$$\frac{430 \times (1 + 0.03)}{84.7} = 1:5.23$$

The above reduction in propeller speed was the maximum possible, considering the power to be transmitted and the existing classification rule for shaft diameters. The layout of the reduction gear, as illustrated in Figs 7 and 8, shows how the propeller speed was arrived at.

The remainder of the original shaftline system was also utilized, with modifications to the forward end only. Here, the original thrustblock assembly was moved aft to accommodate the new, heavier gearbox seatings in the conversion design and one section of the intermediate shafting was shortened to suit.

Although the original main thrust assembly was utilized, the designers preferred to arrange for the thrustblock to be independent from the gearbox structure: problems have been encountered with integral thrust assemblies, due to the increased complexity of the forces to which such a gear casing design is subjected.

Fresh torsional vibration calculations for the new main engine and adapted shafting and propeller system were of course prepared, as were new shaft alignment and bearing reaction calculations for the amended shaftline.

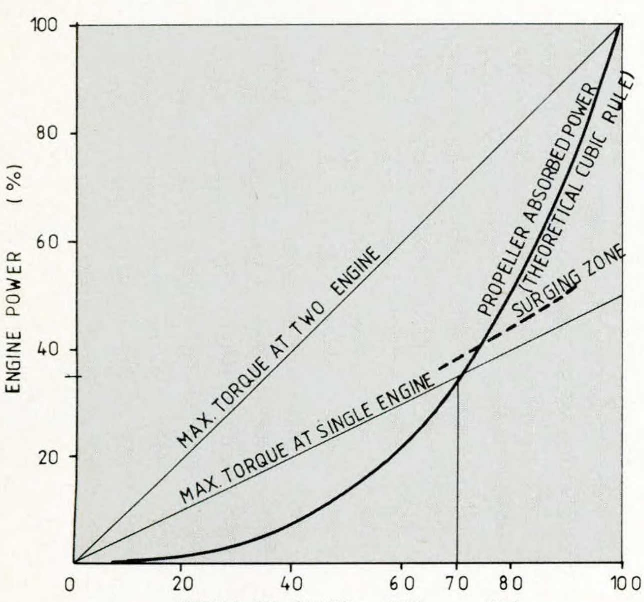


FIG 10 Single engine operation

Shafting, gearing and engine alignment

In view of the high power to be transmitted and the heavy gearing system selected, the alignment of shafts, gears and engine was considered to be crucial to the successful operation of the ship. Careful calculations were made with a standard computer program at the shipyard to derive the shaftline system bearing reactions and coupling off-set values in the static hot, and cold, conditions. Particular attention was given to ensuring that the loadings of the main gear wheel bearings were approximately equal in the hot condition.

The calculated static condition results, shown in Table III, were later applied to alignment work by checking bearing reactions with hydraulic jacks, together with the coupling offsets as previously calculated.

The assembled reduction gear had been given a no-load running test in the manufacturers' shop, up to full speed, with the gear case bolted down. Gearwheel and pinion alignment, together with meshing, were verified.

In order to check that the gear case structure was true, selected spots were milled on its surfaces and accurate reference measurements taken. The data recorded were carefully related to the subsequent gearing installation and alignment procedures on board. Indeed, the jigs used to measure them have been left on the vessel and the original reference points are well protected in case there is any future occasion when alignment is in doubt.

After accurate checks to ensure that the new engine/gearbox seatings are parallel, main engines and gearing were aligned relative to the tailshaft centreline by the use of optical instruments and both engine and gearbox foundations chocked to a high standard.

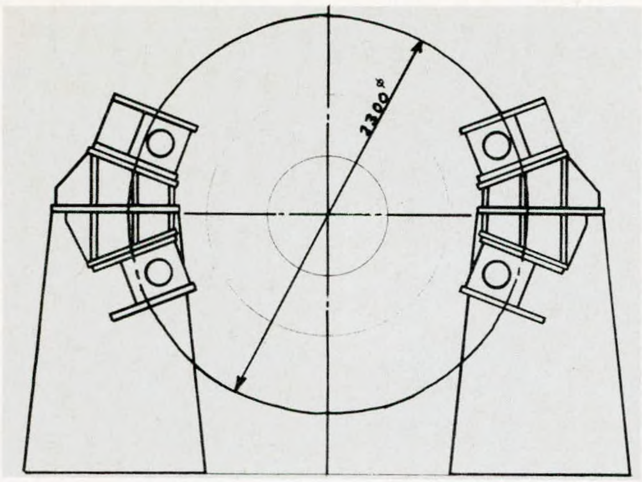


FIG 11 Arrangement of shaft brake

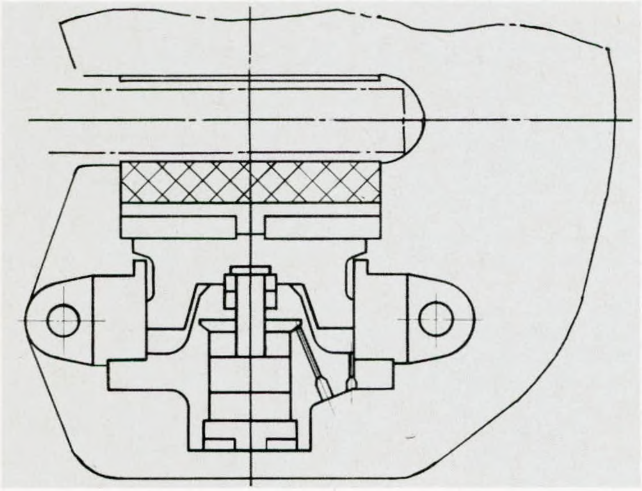


FIG 12 Section of brake clamp

Engine and reduction gear alignment relative to the vessel structure was checked by means of crankshaft deflections, which help not only to verify basic crankshaft bearing alignment but also to confirm the rigidity of the modified engine/gear case foundation structures, relative to any hull deformation that might be due to flexing or engine heat.

A number of readings were taken under cold and hot conditions (the latter being induced by the use of a heater to simulate the circulation temperature of each engine) and at varying hull conditions, in a drydock and afloat, at light and full draughts. In all cases the foundation alignment was seen to be uniform.

Table IV: Plant configuration alternatives

ALTERNATIVE	ARRANGEMENT[a]	STATUS	SLOW STEAM	FEATURE
I	T/G 1500 kW	Existing		Poor economy
	D/G 1450 kW	Existing	Only possible with D/G operating	High mdo cons.
	D/G 1050 kW	New		
II	T/G 1500 kW (waste heat)	Existing partly	Possible down to 53% M/E power	Good utilization existing plant
	D/G 1450 kW	Existing		High economy
III	T/G 1500 kW	Existing		Poor utilization
	D/G 1450 kW	Existing		Existing plant
	S/G 1050 kW	New		Low economy
	Provided either with:			
	a) Directly coupled	New	Impossible	High cost
	b) Thyristor control	New		Complex and high cost
	c) Torque converter	New	Possible but with high M/E power loss	Low efficiency

[a] T/G Turbo generator; D/G Diesel generator; S/G Shaft generator.

The final axial displacements between main engine crankshafts and gearbox input-shaft coupling were confirmed as being well within a value the flexible couplings are designed to tolerate. Figure 9 shows the general arrangement of the shafting.

Dynamic alignment was assessed by the conventional method of measuring the degree of gear-tooth contact by the close examination of selected teeth that had been previously coated with a hard lacquer. The results, after sea trials at full power at varying hull draughts, were excellent, indicating prime alignment of gears under load, with optimum tooth contact.

Main engine clutches: single-engine running

Clutches were provided between each main engine and gearbox input shaft to permit single-engine running.

Each engine's crankshaft, designed to transmit 16 000 hp (11 768 kW), would be driving a propulsion transmission system designed to absorb 32 000 hp (23 536 kW) when running singly. It was therefore protected against overtorque and exposure to high thermal loading. Each main engine hydraulic governor was equipped with a torque-limiting device that limited engine power to a pre-determined level, relative to its revolutions.

The simplified diagram in Fig. 10 is based upon the theoretical cubic relationship of rev/min and the power absorbed by the propeller. It indicates that the maximum propulsive speed supportable by a single main engine is around 70 per cent of engine revolutions or 34 per cent of total installed power.

In order to keep the turbo-blower surging zone beyond the intended single-engine operating range, well-matched and efficient turbo-blowers had to be selected.

Actual results from a single-engine running test, carried out subsequently on sea trials, confirmed that a propeller speed of 67 per cent of engine revolutions could be attained: so that 55 rev/min of the mainshaft gave a ship's speed of 9.34 knots.

The engine clutches employed were kept simple, being barrel-toothed, gear-type couplings, disengaged by mechanically operated hand gear with the engines stopped. They also act as flexible couplings, providing further alignment tolerance between main engine and gearing, via quill shafts arranged within the bores of each input pinion. The barrelled-tooth feature allows for a maximum offset of 1.5 mm and an angular displacement of 1.04 deg.

The coupling teeth were surface-hardened by nitriding, having been previously hobbed. Particular attention was paid by the shipyard and its sub-contractors to the internal forced lubrication of the clutch gear teeth.

Astern running performance

A most important point to consider in converting from steam turbine to diesel propulsion is the latter's effect upon astern manoeuvrability.

When the propulsion is stopped, a ship of the *Alva Sea*'s size maintains headway for a considerable period due to its inertia. During this period, the propeller and shafting continue to rotate in an 'ahead' direction, due to the flow of water over the propeller. In order for a propulsion engine to start astern, this torque must be overcome.

Turbine engines, whilst limited in astern power, have high starting torque. In order to assist the low starting torque inherent in any medium-speed diesel engine to improve crash-stop performance, a shaft brake was provided.

The brake installed is of the disc type, pneumatic Ferodo-lined brake clamps being arranged radially around a heavy disc, bolted to an intermediate length of shafting, as illustrated in Figs 11 and 12. The braking torque necessary to arrest the turning moment at the propeller, due to counter-flow at nominal ahead speed, was theoretically estimated and the brake designed accordingly. Immediately fuel is cut off from the engines, the brake is actuated to arrest the main shafting; and released after the pre-determined time when 'astern' power can be applied.

Confirmation of the shipyard calculations in respect of shaft-brake design and astern manoeuvrability was obtained during the official sea trials (Fig. 13).

Auxiliary power requirements

With the exception of the machinery that had suffered the casualty described earlier, the condition of the plant in the vessel was good. In particular, the condition of the main boilers was excellent, their heat transfer surfaces and casings being sound. The large economizers were also in a very good state. With the exception of some high-pressure steam pipes and certain control elements damaged by the machinery casualty, the entire steam and condensate systems were available for utilization.

Both the turbo- and diesel-alternators originally installed in the vessel were also fully available.

Table V: Waste heat loads supportable at slow steaming powers

MAIN ENGINE LOAD (%)	SEASON	STEAM FLOW AT T/G (kg/h)	EQUIV. ELECT. LOAD (kW)	VESSEL SPEED (knot) *Full*	*Ballast*	REQUIRED ELECT. LOAD (kW)
90	winter summer	8670 9580	1070 1190	14.2	15.9	728.5 730.8
80	winter summer	7550 8310	920 1025	13.8	15.3	728.5 730.8
70	winter summer	6510 7180	780 870	13.2	14.7	728.5 730.8
60	winter summer	5730 6090	670 725	12.5	14.0	569.4 571.7
55	winter summer	5130 5650	590 660	12.1	13.2	569.4 571.7

1. Summer condition: air temperature 35°C, sea temperature 30°C.
Winter condition: air temperature 20°C, sea temperature 10°C.
2. At 53% engine load (est. lowest limit) V_L = 11.8 kts, V_B = 13.25 kts.

Electric power

Within the constraint imposed by the retention of the original main shafting system and fixed-pitch propeller, the options in Table IV were examined with a view not only to the cheapest conversion cost but economic operation, including at 'slow steaming'.

Alternative I was rejected because of the high cost of marine diesel oil. Alternative III was rejected because of the difficulties in maintaining generating frequency over the operating range; the low plant efficiency and high absorption of engine power.

Alternative II was selected as the most economic; it provided the best utilization of existing plant and was also the most efficient operational solution. The original economizers were transferred from the boiler flue gas uptakes and made to enter the main engine exhaust-gas path to act as waste-heat recovery boilers, with only the addition of new pre-heater and superheater sections. This greatly contributed towards minimizing the overall conversion cost.

Waste heat recovery system

The plant arrangement made full use of the unusually high exhaust gas volume of the high-powered diesel installation. Its principal features are described below.

A waste heat boiler in each main engine's exhaust-gas uptake is designed to generate steam to support electrical loads even during slow steaming. From Table V, it was expected that this could be done at 55 per cent of the main engine's mcr, corresponding to a laden ship speed of 12 knots. This is approximately the speed attainable by a steamship, slow-steaming on one boiler.

In order to minimize the electrical demand at sea the main engine and gearing LO pumps were shaft-driven. In addition, some six new auxiliary pumps with two-speed motors were installed. The finalized waste heat recovery system is as shown in Fig. 14. The original turbo-alternator was used in the system, provided with a new multi-stage low-pressure turbine, underslung condenser and condensate-extraction plant.

As the capacity of the existing LP steam/steam generator was insufficient, a new steam separator was added, as were new waste-heat boiler circulating pumps, together with new feed pumps for the system. Apart from the addition of a feed heater, the remainder of the plant in the system was utilized from the original secondary condensate system existing in the ship.

The turbo-alternator can also operate on saturated steam in port or during manoeuvring, supplied from the LP steam/steam generator. In order to reduce flashing-up time, and for additional security after long shut-down periods at sea, a heating coil was installed in the water drum of one main boiler which was served by the waste heat recovery system.

Modifications to boiler plant

With the elimination of the need to supply 60 bar/513°C steam for the main propulsion turbines, the boilers could be simplified and their rating reduced, with consequent gains in ease of operation, improved safety when flashing up, and extended superheater life.

To achieve this, the primary superheaters were removed and the

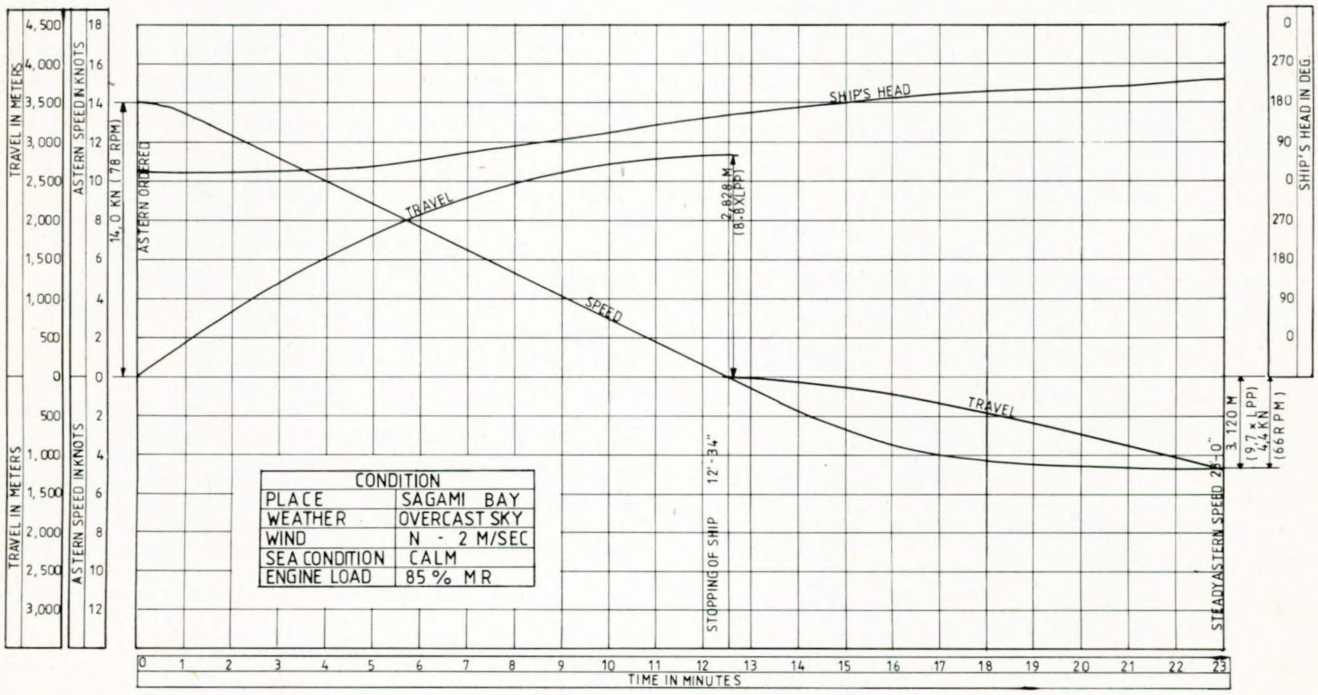


FIG 13 Manoeuvring record at sea trial

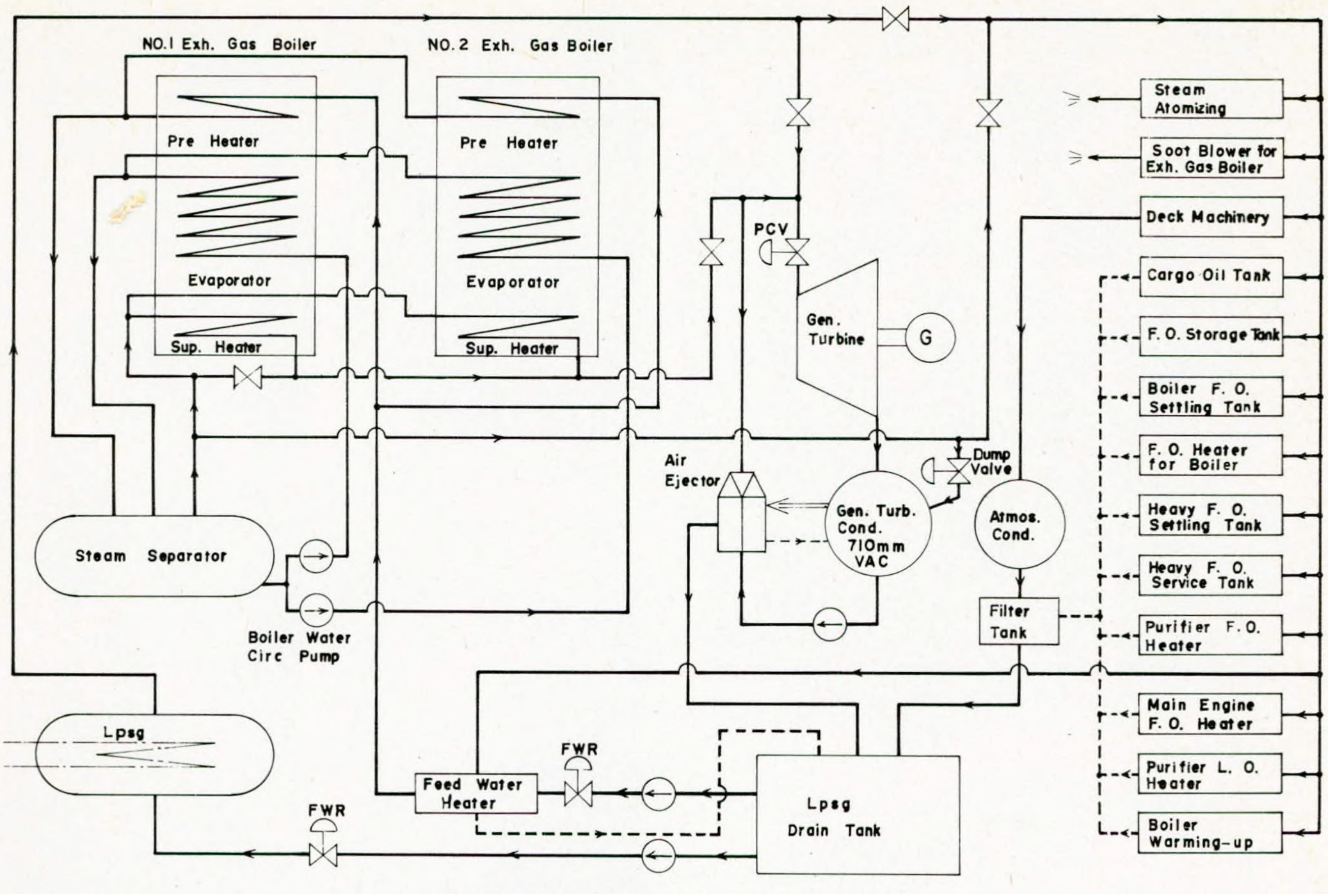


FIG 14 Waste heat recovery system

surface of the secondary superheaters reduced to provide revised steam conditions of 60 bar/350°C to match the original requirements of the remaining steam cargo pumps. The existing steam attemperators in the water drums of both boilers became redundant and were, therefore, removed, together with their related service piping and control elements.

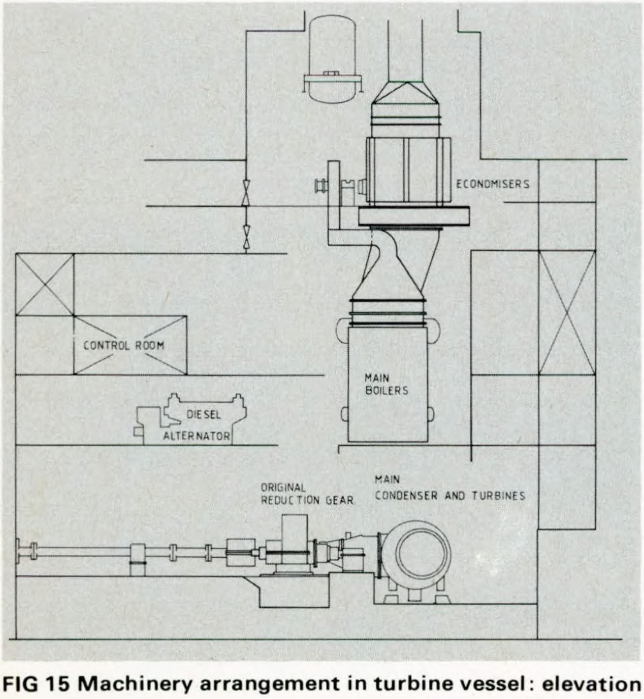


FIG 15 Machinery arrangement in turbine vessel: elevation

Machinery arrangements

The above considerations resulted in the machinery arrangement being revised as shown in Figs 15–18.

It will be noted that the original boiler site has been retained. Also noteworthy is the change in the inner bottom line, immediately underneath the engine foundations, where the original hull structures had been totally revised. No other structural changes were made.

FINALIZATION OF DESIGN AND CONTRACT AWARD

During the design a very close technical liaison was established between the owner and the shipyard so that, when the contract was finally placed, all areas of the conversion work had been fully considered and full specifications described in detail the conversion outlined in this paper. Of particular note was the extensive work carried out by the shipyard in matching the existing electrical and

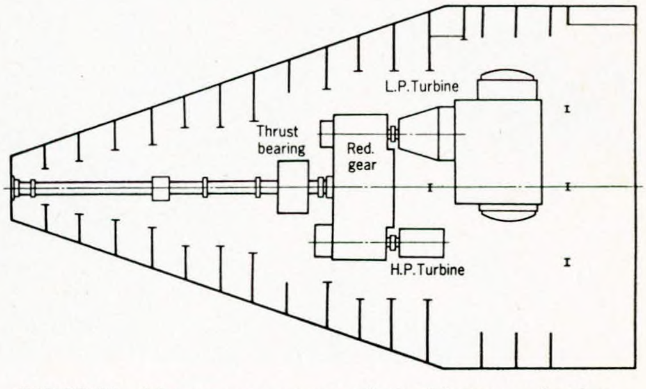


FIG 16 Machinery arrangement in turbine vessel: plan

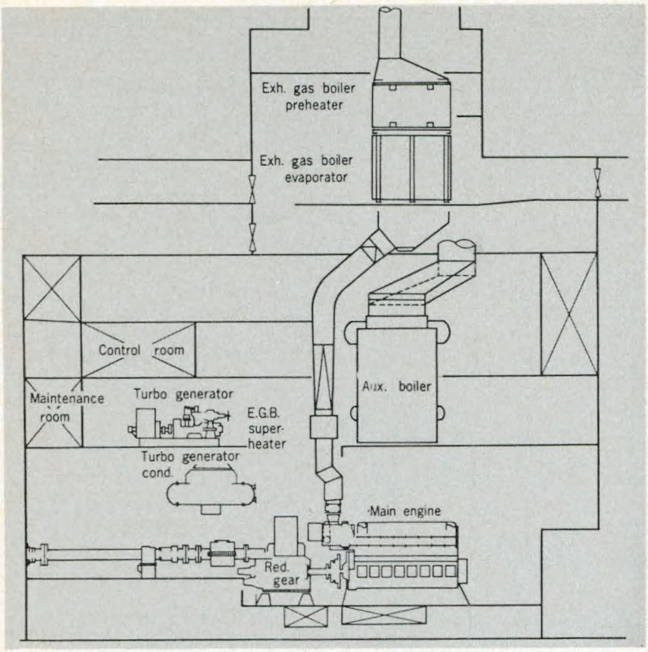


FIG 17 Machinery arrangement in motorship: elevation

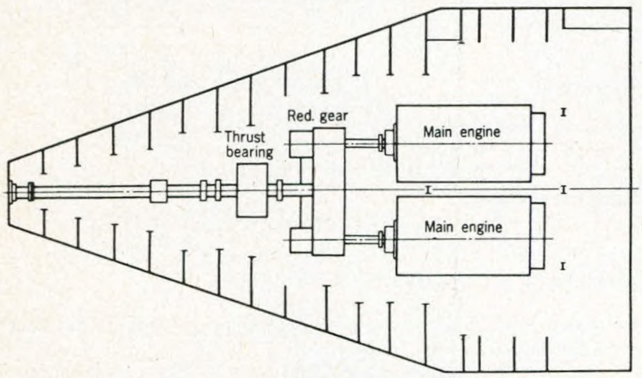


FIG 18 Machinery arrangement in motorship: plan

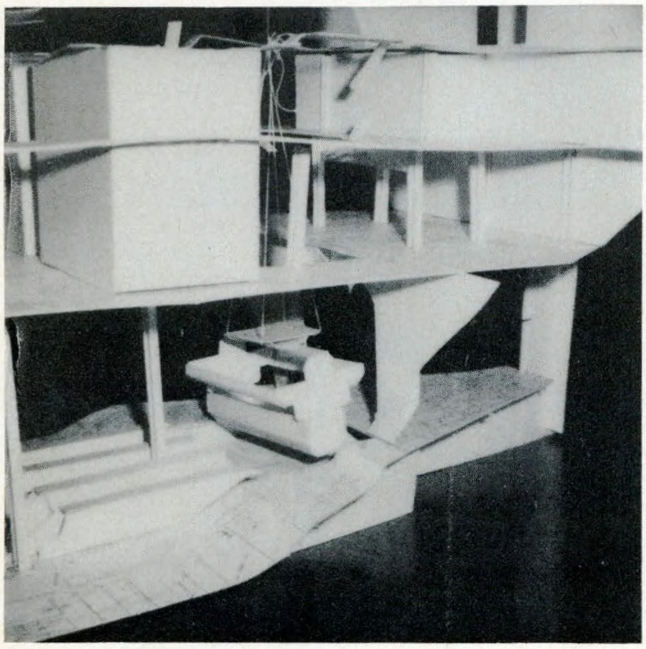


FIG 19 Model used for determination of access

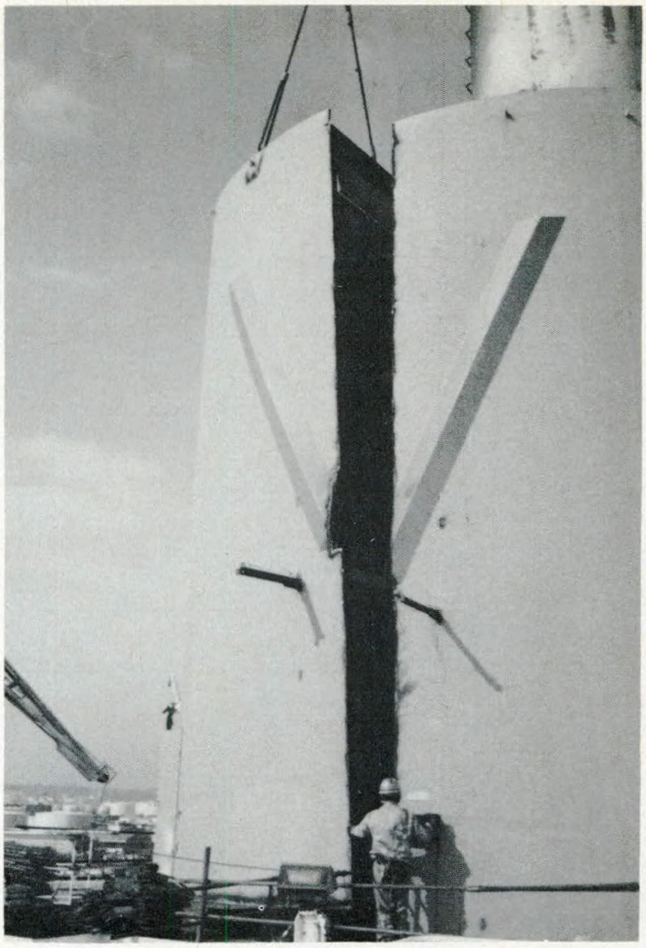


FIG 20 Preparing initial access

control systems to the new installation required as a result of conversion. Even the original control consoles, instruments and re-arranged mimic panels were matched on completion.

Special consideration was given to the scheduling of the work, including access for the installation of the main engines with minimal structural and other disturbance. Models of the engine room section were made for this purpose, one of which is shown in Fig. 19.

EXECUTION OF WORKS

The vessel was taken from Bahrain, where she had been since the machinery incident occurred, and towed to Yokohama, arriving in February 1979. The conversion work began immediately and

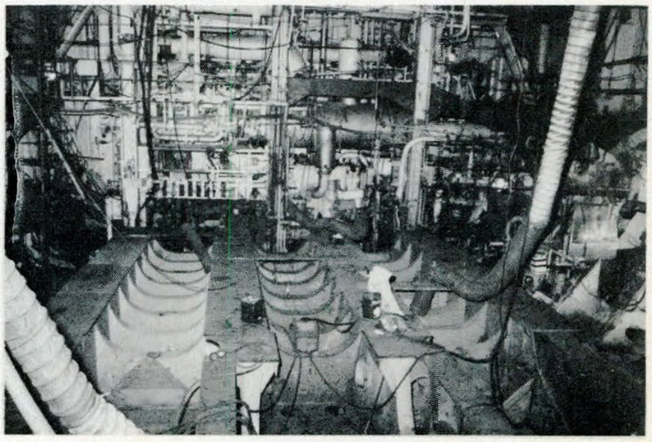


FIG 21 New propulsion machinery foundations

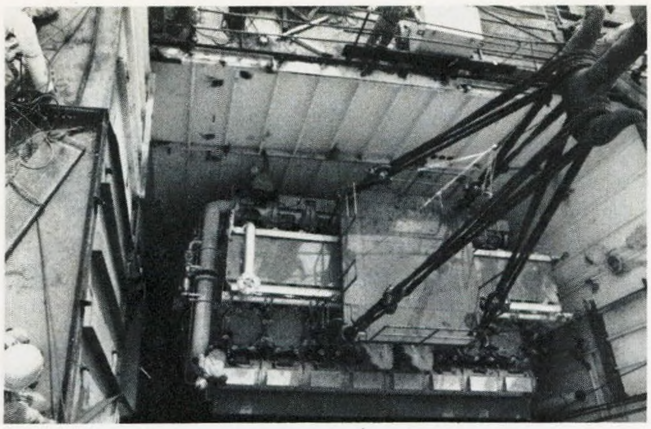


FIG 22 Lowering new engine into place

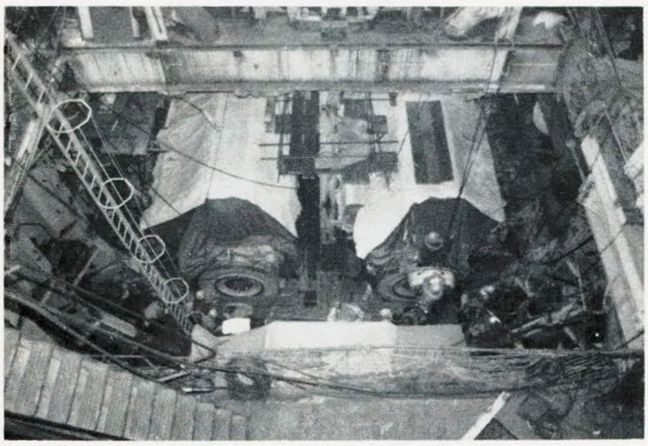


FIG 23 New engines installed in place

progressed according to schedule, with surprisingly few difficulties.

Most of the problems which did arise concerned the condition of machinery unrelated to the conversion, which had suffered during the long lay-up period. In particular the bottom shell suffered severe and tenacious build-up of barnacles.

Figures 20–23 show some stages of the conversion work.

OPERATIONAL ENVIRONMENT, TRAINING AND MAINTENANCE

It was clear, that, unless adequate provisions were made, certain problems would emerge from the new machinery space environment that could impose potential discomforts on the operating personnel. It was foreseen that the following problems would directly influence the operational environment from the human aspect.

- Increased noise levels from the new high-powered medium-speed diesel installation.
- Higher ambient temperatures, resulting from increased radiant heat emanating from exhaust trunkings of the new main diesel engines and the new exhaust gas boiler.
- Restricted air flow in lower parts of the engineroom in way of the new main engine site, further aggravating the high temperatures.
- High manual work-load in servicing and overhauling a large multi-cylinder, medium-speed diesel installation,

Environmental problems

The potential problems resulting from the machinery space environmental changes, such as noise, heat and restricted ventilation, were closely examined during the conversion design with the objective of minimizing their effects on the operating staff.

As far as noise levels were concerned, values measured on sea trials in areas remote from the main engine were below expectations and, surprisingly, only marginally above those originally recorded with the vessel as a steamship. In certain areas of the engineroom, however, such as between the main engines at the lower platform, noise levels were unavoidably higher.

The design could not be altered in any way to minimize this effect but staff are provided with approved ear protection devices which are worn at all times in the machinery spaces when the main engines are running at power. In addition, refuges from excessive noise have been provided by the addition of a new sound-proof air-conditioned workshop and the retention of the original engine control room.

The predicted high ambient temperatures were offset by increasing air flow to the lower areas of the engineroom through addition of trunkings, where possible; and cutting openings in the structures that form No. 2 platform level to improve local air flow.

The remaining problems that had to be overcome were the potential operational complexities that inevitably resulted from the conversion itself, and the high manual work-load in servicing and overhauling the engine. These restrictions were minimized by the preparation and training of operating staff, together with the provision of a special maintenance facility in the converted vessel.

Personnel preparation and training

Engineer Officers, with the background and experience in the operation of a high-powered motorship that also possesses the hybrid features of a large generating plant, were carefully selected by the Technical and Personnel Departments of the owners. The policy has been established that Chief Engineers appointed to the vessel must have a combined steam and motor certificate of competency. Senior 2nd Engineer Officers should hold a superior motor certificate whilst the Junior 2nd Engineer Officer must have a steam certificate. Two 3rd Engineers are appointed to the vessel, one steam and one motor. The 4th Engineer carried has motorship experience.

Preparatory training courses for Engineer Officers designated for service in *Alva Sea* were also conducted both in Europe and in Japan.

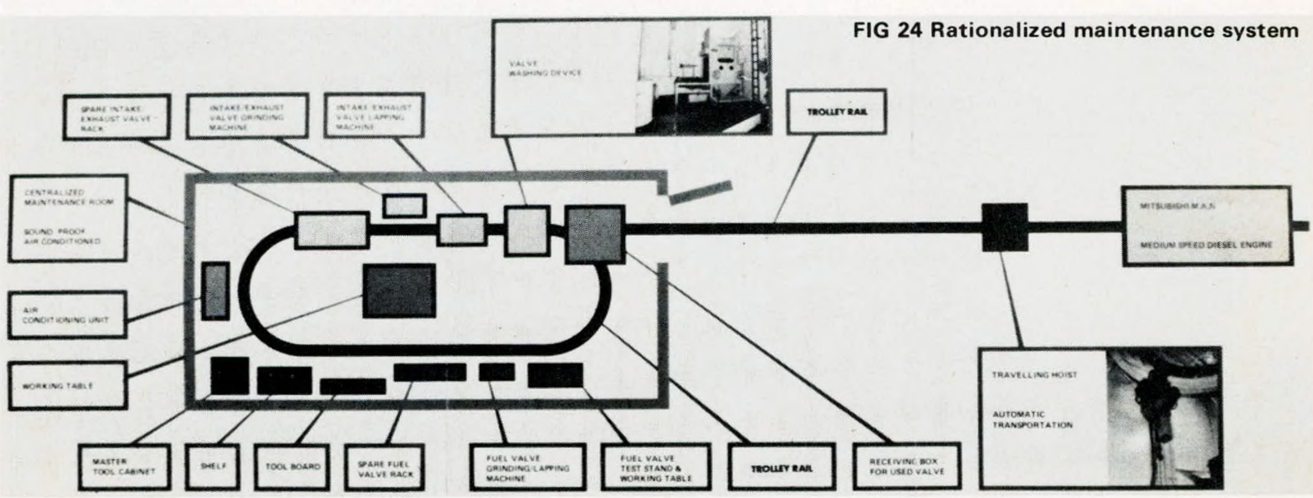


FIG 24 Rationalized maintenance system

Whilst it was intended to provide operational continuity by designating fleet staff for service in *Alva Sea*, it was obvious that there would be some future wastage of sea-going personnel, so that some of the effects of the training would be lost. It was, therefore, decided to evolve a method of ongoing shipboard training to provide Engineer Officers subsequently appointed to the vessel with the opportunity of receiving instruction as to the ideal, and often easiest, manner in which to operate and maintain the new plant.

This was achieved by the provision of a special video training film, developed in close consultation between owners and shipyard, and professionally produced. The film contains a narrative that describes the routine overhaul of the engines and turbo-blowers, together with instruction as to the safety features and functions of the bridge control system and the correct use of a rationalized maintenance system that was developed as part of the conversion design.

Standing instructions to the vessel are that the training film be shown on each major engineering staff change, or prior to an intended overhaul of main engines. Reports from the Chief Engineers on the usefulness of this training feature have been extremely positive and its use has become a regular routine.

Special maintenance facilities

Mention has been made of the heavy workload foreseen in routine maintenance. In order to ease this load, and at the same time endeavour to ensure good shipboard servicing and repairs, a rationalized maintenance system, outlined in Fig. 24, was developed by the shipyard.

A central air-conditioned workshop (Fig. 25) is arranged with a material flow path. Cylinder head components removed from the main engine pass through a blasting cabinet to clean them (Figs 26 and 27) after which they are subjected to overhaul by grinding, lapping, reassembly and testing, before being returned to stock ready for the next scheduled overhaul.

Outside the workshop, close attention was paid to the necessary cranage and guide rails, and to the collection, lifting and transfer of heavy parts and cylinder-head components from both engine sites up to the maintenance room described.

The advantages of these maintenance facilities are mainly twofold. First, from the human relations aspect, the improved working environment in the air-conditioned workshop, together with the specialist equipment provided there, go a long way towards offsetting the arduous work-load. Routine maintenance, essential to the effective trading of a large, hard-run tankship, are carried out under controlled circumstances in what is intended to be a 'tool room' atmosphere, thereby doing much to improve working conditions and personnel morale.

Second, there is no doubt that the use of these facilities by adequately trained personnel greatly enhances operational efficiency by both improving the quality of repairs and reducing maintenance times.

In both these aspects, however, supplementary to any special facilities or equipment, the key to success lies with the proper training (and re-training) of experienced and well-motivated personnel.

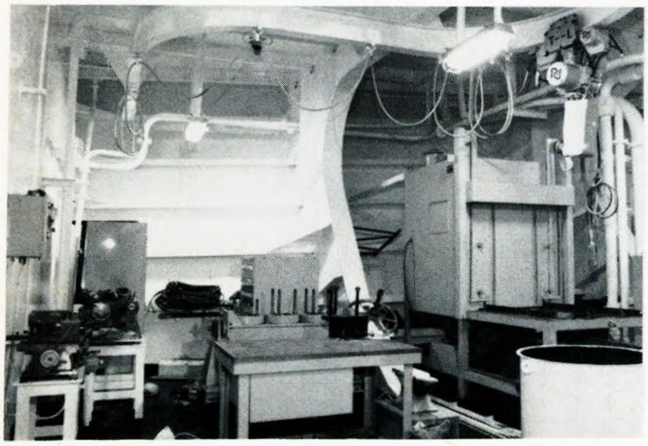


FIG 25 Air-conditioned maintenance room

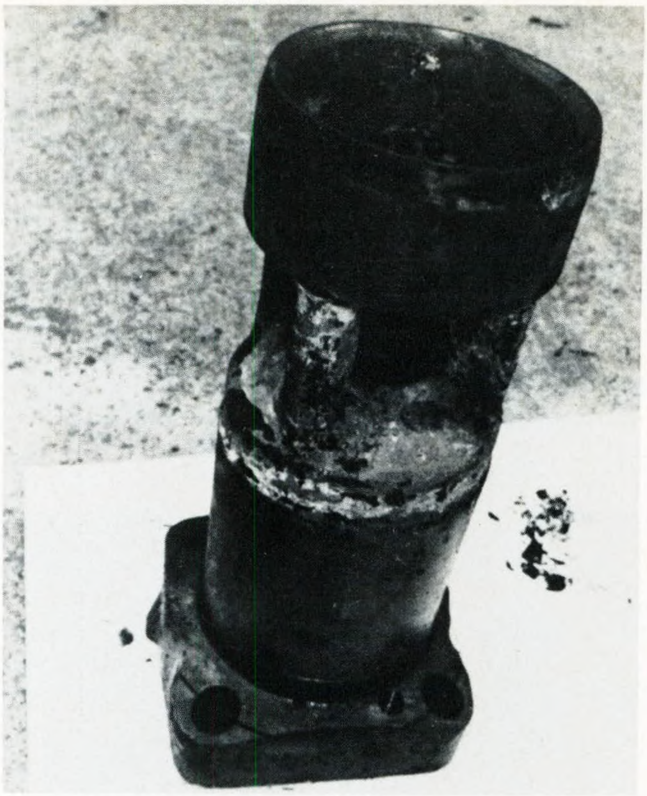


FIG 26 Prior to blasting in cleaning cabinet

COMPLETION AND TRIALS

In view of the nature of this conversion, a full commissioning and trials programme was developed, covering not only the converted main engine and other items, but the remaining original plant.

The results of the trials fulfilled the design predictions: the guaranteed fuel rate being confirmed at 141.6 g/hph, with the engines running on diesel oil. Tests carried out when burning heavy fuel oil resulted in a specific consumption of 137.8 g/hph. Both figures are

FIG 27 After blasting in cleaning cabinet

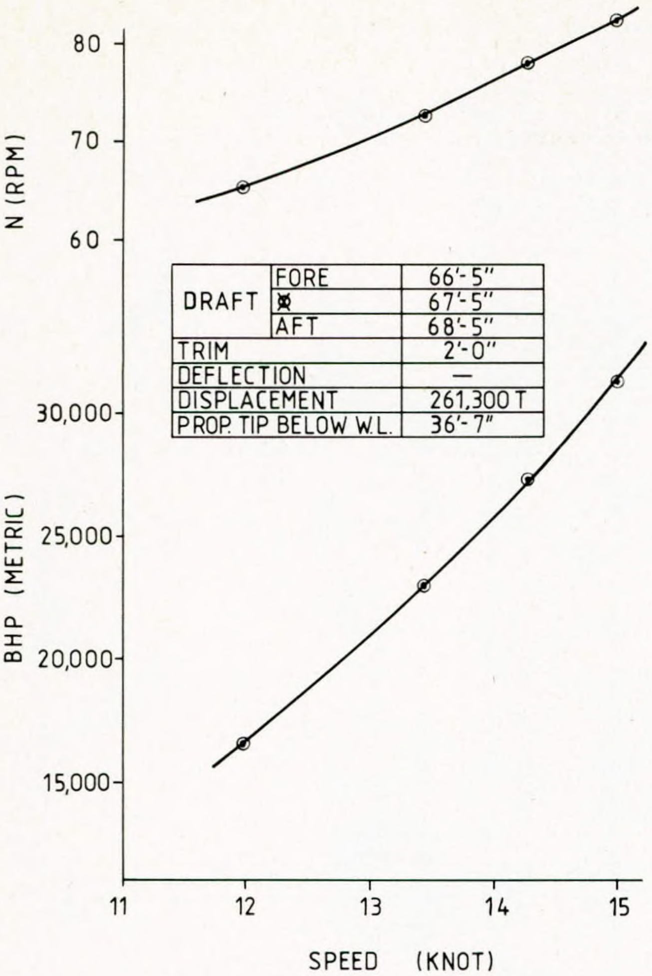


FIG 28 Results of speed trial

corrected to correspond to a standard fuel having a lower calorific value of 10 200 kcal/kg (42 698 kJ/kg). The results of the speed trials are shown in Fig. 28.

SUBSEQUENT SERVICE EXPERIENCE AND PERFORMANCE

Considering the magnitude and complexity of the machinery conversion, the subsequent operational service experience of the ship has been satisfactory and the design objectives have been achieved.

Early problems with the forced-lubrication pumps provided with the new main engine were overcome by minor modifications to the related service pipework and re-adjustments to the regulating valves, enabling these units to operate normally.

Main engine overhaul times, which were initially longer than expected, have now been substantially reduced by a combination of experience and time and motion study carried out by the ship's staff and the Technical Operations staff responsible for the vessel.

The main engine's exhaust valves suffered heat corrosion sooner than expected, causing some operational disruption, but no extensive delays. This problem has been investigated jointly by the owners and the shipyard and certain design modifications were made at the recent conversion guarantee docking, aimed at restoring exhaust valve life to a normal length.

The problem was attributed to a series of items with a cumulative effect. First, it became clear after re-commissioning, particularly in hot areas of the world, that air supply to the lower part of the engineroom was below design expectations. This fact was particularly evident in the vicinity of the main-engine's turbo-blowers, where elevated ambient temperatures were recorded in service.

The second observation was that the predicted revised propeller revolutions were not quite met; the shaft revolutions being some 3 per cent below estimate, due to the hull being far rougher than anticipated. Removal of accumulated barnacles proved particularly difficult during the conversion docking and it is probable that de-scaling left the hull rougher than originally estimated, thus increasing the hull resistance accordingly.

To alleviate these problems, the air supply to the engines was improved by the provision of larger supply fans and modifications to the supply trunking in the lower part of the engineroom; whilst the thermal loading of the engine was reduced by cropping the propeller to raise revolutions and lower transmitted torque.

Reports from the vessel following these modifications indicate that they have been successful.

CONCLUSIONS

The current state of the world's economy has had a profound effect upon our industry, an industry particularly sensitive to both political and economic fluctuation and change. This recession in international trade has led to an overall contraction in the total number of ships operating; and the ships remaining in full commission are operating in extremely competitive trading climates. If a shipowner is to remain competitive he must be assured of reliable modern ships and machinery. Above all, the escalating price of fuel dictates that plant efficiency is more important today than it has ever been.

The conversion described above, by its very nature, necessitated a close consultative relationship between the shipowner and the shipyard. A high level of co-operation existed throughout the whole of the project, from the initial design preparation to the final testing, re-commissioning and sea trials. At these latter stages, as in any large repair or conversion contract, owners' and builders' staff jointly tested and re-commissioned the ship. In this particular case, because of the complexity of the converted systems, this co-operation was of prime importance.

In the future, detailed consultations between the shipowner and the shipbuilder on operational and design problems will become even more important, both for conversions and new construction, in order to combat the effects of economic restrictions in our industry by effective and economic ship design.

A further contributory factor in ensuring the success of this project has been the human element. The diligence and professional interest of the ship's staff responsible for re-commissioning and subsequent operation of the ship has been of a high order. We consider that the emphasis during the design stage on minimizing the environmental discomforts to the operating staff, as well as their preparation and training, was time well spent.

Close to one and a half years have elapsed since *Alva Sea* was re-commissioned to commence a new trading life in her converted form, and during this period the owners' operational objectives have been realized by the shipyard's design solutions described above. In hindsight, there are, as is often the case, certain features that could have been improved. Overall, however, the conversion of *Alva Sea*'s machinery is generally regarded as both a technical and a commercial success.

Discussion

DR S. ARCHER (FIMarE): Although the main interest of this paper lies in the geared medium-speed diesel conversion, could the authors state whether the cause of the LP turbine run-away was clearly established? Having regard to the extent of the resulting damage, were there any fatalities or injuries to personnel? Hopefully, they were safely in the control room, or in some other suitable 'bolt hole'; or, perhaps, the vessel's machinery had a UMS class notation and thus there was nobody actually present in the engine room at the time of the accident?

Overspeed trips are, of course, required by the classification societies. In the case of cross-compound turbines, Lloyd's, for example, will permit a single trip to be provided and this must be fitted to the LP turbine. One might expect such trips to be required on both HP and LP turbines, but the argument appears to be that if the HP turbine (not protected by its own trip) loses its load, it is less likely to 'go through the roof', owing to its inherent self-choking characteristic when attempting to overspeed. Of course, it is also much smaller and is protected by a much stronger casing. If the trip were fitted to the HP turbine alone, this would not, of course, control an LP turbine run-away. Despite this, even with an extremely prompt shut-off on loss of LP load, there must be a sizeable store of potential energy, still left in the HP turbine and cross-over pipe, which is available for accelerating the LP turbine.

It is an analogous problem to the control of racing in steam reciprocators, where probably the stored steam potential may be even greater and, in addition, the masses to be accelerated are much smaller. Could it be then that, in theory at least, the high inertia of the LP turbine is normally able to protect it from excessive run-away speeds? It would be useful to know what overspeed protective devices were fitted and whether there was evidence that they had functioned correctly.

Some manufacturers of larger medium-speed marine diesels, including MAN, claim that their engines will accept heavy residual fuels up to 4000 Redwood No. 1 at 100°F (1000 cSt). However, probably most, in common with MAN, recommend that, in view of the well-known problems associated with such fuels, viscosities not exceeding about 1500 Redwood No. 1 (380 cSt) represent the best compromise today. It is noted the authors have, perhaps wisely in view of the maintenance problems encountered after the conversion, adopted a fuel with a viscosity of only 180 cSt (750 Redwood No. 1), although they point out that the engines are designed to burn 380 cSt fuel (1500 Redwood No. 1). The extra premium of US$2.5/ton for the lighter grade may well be worthwhile, balanced against the probable reduced maintenance costs.

It is noted that MAN type V52/55 engines were finally selected on account of their proven reliability and conservative rating. On the basis of an mcr of 16 000 bhp per engine and with 16 cylinders, this is in line with published claims by MAN and, thus, the normal 85 per cent rating 850 bhp/cylinder) would indeed seem adequately conservative.

On the design of the reduction gears, the authors state this is a reversible gear train with facility for de-clutching either engine. The reversing mechanism in the gearbox is not at all clear from the section in Fig. 7. Could the authors please supply details? As regards the de-clutching of the engines, using gear-type couplings, is care taken when re-connecting to preserve any particular phase between the firing angles of corresponding cylinders in the two engines? Assuming identical crank arrangements and firing orders in the two engines, the node-at-gears, or 2-node, mode will be non-excitable by all harmonic orders for zero phase angle firing, i.e. simultaneous firing of comparable cylinders in each engine, although for other reasons not necessarily a desirable arrangement.

For other modes ('normal' modes), such as 1-node and 3-node, in which the whole of the system—including line shafting and propeller—participates, all integral odd orders and $\frac{1}{2}$ orders will be non-excitable for phase angles of, respectively, 180 and 360 deg, and certain even integral orders for other phase angles. The authors state that a mechanical damper is fitted to each engine and resilient rubber couplings between each engine crankshaft and gearbox input shaft. They claim that these devices 'render the plant free from torsional vibration throughout the whole of the operating range'. This is a confident claim and was no doubt substantiated by measured records. Does this suggest, perhaps, that the damping from these two devices is such that no regard need be taken to engine firing phase angles, in either the 'node-at-gears' modes or the 'normal' modes? Could the authors also give details of the engines' firing order?

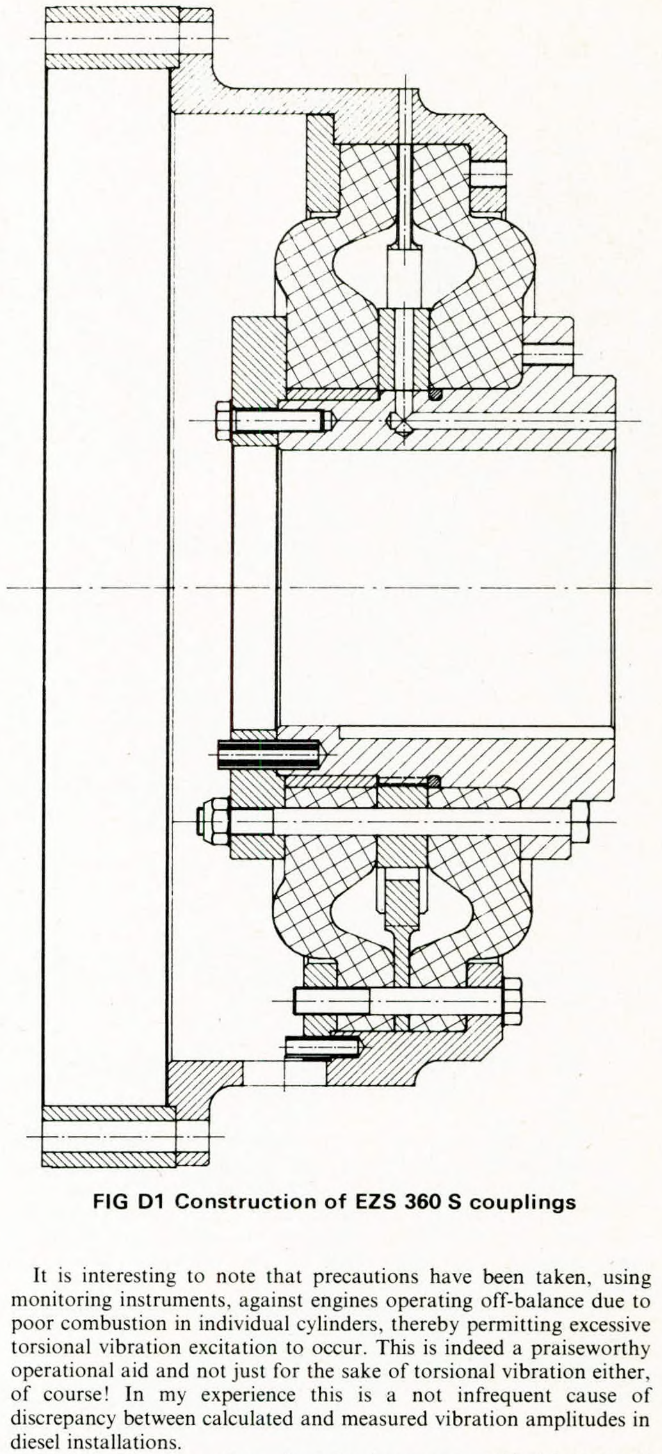


FIG D1 Construction of EZS 360 S couplings

It is interesting to note that precautions have been taken, using monitoring instruments, against engines operating off-balance due to poor combustion in individual cylinders, thereby permitting excessive torsional vibration excitation to occur. This is indeed a praiseworthy operational aid and not just for the sake of torsional vibration either, of course! In my experience this is a not infrequent cause of discrepancy between calculated and measured vibration amplitudes in diesel installations.

As claimed by the authors, the reduction gear data listed in Table II indicate a conservative design which allows adequate margins beyond classification society requirements in respect of gear tooth specific loading, bending strength and surface finish. These margins would be expected to take care of the effects of increased hull resistance and additional loading due to any torsional vibration which might develop.

As a check on the design, the gear data have been compared with the rule requirements of Lloyd's Register of Shipping. The results indicate margins of 43 per cent on surface contact pressure (*K*-value) and 172 per cent on bending strength based on the wheel material. Even if a vibratory torque at full load equal to one-third full transmission torque (maximum allowed by Lloyd's) were applied to the gears, these margins would still be 7.5 and 104 per cent respectively. Bearing in mind the extra damping devices incorporated in the system, such an assumption of vibratory torque would be thought pessimistic; can the authors confirm this?

The graphical results of the crash stop astern test (Fig. 13) are of much interest. They would, however, have been even more informative if the corresponding results for the turbine machinery had been provided. Recent tests in Japan, after the conversion of the S/T *Mobil Hawk* (285 440 dwt) to geared medium-speed diesel machinery (mcr 36 000 bhp), showed that, even without the shaft brake as fitted on *Alva Sea,* but using the de-clutching facility, the diesel installation nearly halved the head reach and time to dead-in-the-water. This must largely be ascribed to the fact that the diesel installation has its *total* power available in the astern mode (assuming, of course, the propeller can absorb it) compared with the usual 40–50 per cent only of the turbine installation. It would be interesting to know whether comparative tests were carried out on *Alva Sea,* without the use of the brake, to assess its contribution to the manoeuvre.

Incidentally, it would seen the figures given for astern travel at the lower right-hand edge of Fig. 13 should perhaps be 1120 m and not 3120 m. If this is the case, this would be equivalent to 3.5 LPP and not 9.7 × LPP.

Finally, congratulations to the authors on a most informative and detailed account of the conversion of *Alva Sea,* a highly topical subject in these days of stringent fuel economy.

MR W. S. WAYNE (Shell International Marine Ltd): Two factors to be considered when examining re-engining are the increase in maintenance load and the reliability of the new plant. Both would appear to be best served by minimizing the number of cylinders in the installation. With the benefit of hindsight, would the authors have chosen a third-generation medium-speed diesel with less cylinders, available at the time of the conversion?

From Table I, we see that allowance was made for three extra staff in the engine room department. Assuming that these men each work 8 hours per day for a 300-day year, this generates approximately 7000 man hours of extra maintenance, or more than 200 man hours per cylinder installed. This would seem excessive and I would like to know if the extra staff have been found necessary.

My final point relates to the steam plant. Recently, my company undertook a basic re-engining study of a VLCC. Our thinking at that time was to utilize as much of the existing main condensate systems as possible, in order to avoid long shut-down periods for the main system components which would otherwise only be used in port. We felt that these long shut-down periods could lead to excessive corrosion problems. By incorporating the waste heat recovery system with the existing auxiliary steam and condensate system, the deaerator could be utilized at all times.

In the authors' system, keeping the steam system separated requires an extra feed heater in the low pressure system, presumably to reduce cold and corrosion in the waste heat unit, whilst there is no effective means of deaeration of the feedwater other than that effected in the turbo-generator condenser. I should appreciate the authors' comments on this point.

MR K.H. GEIGER and MR J. R. R. BAIKIE (Vulkan Kupplungs und Getriebebau B. Hackforth GmbH): In the paper it was pointed out that, if an engine was operating off balance due to poor combustion, the vibratory loads on the gears could be increased. Simultaneously, the vibratory load on the flexible coupling increases. As manufacturers of the highly flexible couplings used in this installation, we have, in reaction to the comments made in the paper, studied the effect of engine imbalance on the vibratory load at the gears and in the couplings.

Between engine flywheels and gearbox input shafts, our couplings EZS 360 S are installed. These are ply-reinforced, highly flexible rubber couplings with a progressive torsional stiffness characteristic. Figure D1 illustrates the construction of these couplings.

Figure D2 shows the coupling vibratory loads for the case of two engines driving the propeller, for the following engine conditions:

Curve 1 : all cylinders equal mep (ideal).

Curve 2 : 5 per cent mep difference per crank on both engines.

Curve 3 : same, but 10 per cent mep difference.

The line T_{KW} represents the allowable mechanical loading of the coupling. The line P_{KV} represents the allowable thermal loading of the coupling. It should be noted that the worst combination of cylinder imbalance has been chosen. The curves represent the vector addition of orders 0.5–12.

Figure D3 shows the vibratory loading at the gears for the same engine operating conditions. The vibratory torques have been added to and subtracted from the mean transmission torque. For torque-transmitting gears, the upper limiting value represents 133 per cent of the maximum transmitted torque as the maximum permissible value. The bottom limiting value indicates zero torque (for indication of gear hammer).

It is clearly seen that an increasing engine-imbalance will result in an increasing vibratory load in the coupling and at the gears. In addition, we have made calculations with one cylinder misfiring, i.e. one cylinder has only compression and no combustion.

Figure D4 shows the vibratory torque in the coupling of engine no. 1 for two engines driving one propeller, and for cylinder no. 1 of engine

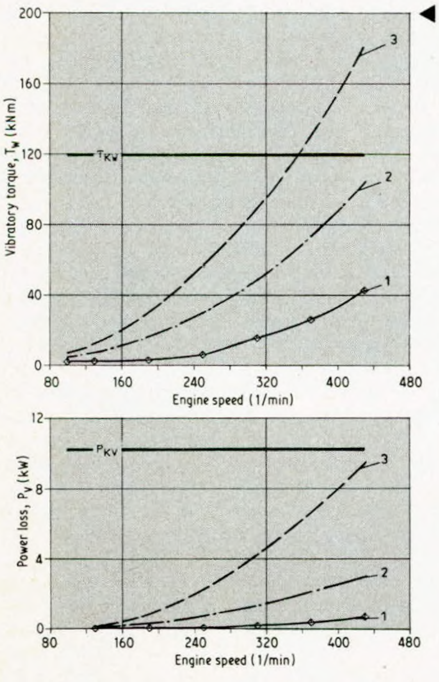


FIG D2 Coupling vibratory loads: two engines driving the propeller

1 : all cylinders equal mep (ideal)
2 : 5 per cent mep difference/crank (both engines)
3 : 10 per cent mep difference/crank (both engines)

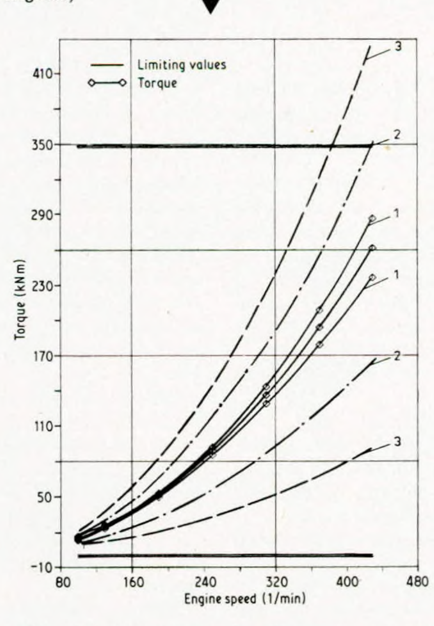
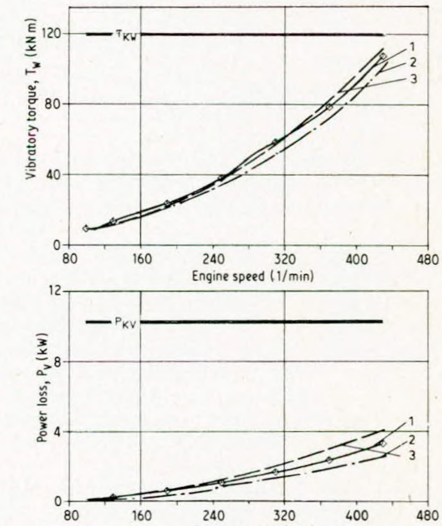


FIG D3 Vibratory loading at gears (engine conditions as Fig. D2)



1 : all other cylinders equal mep
2 : all other cylinders ± 5 per cent mep
3 : all other cylinders ± 10 per cent mep

FIG D4 Vibratory torque in engine no. 1 coupling for two engines driving the propeller; and cylinder no. 1 of engine no. 1 misfiring

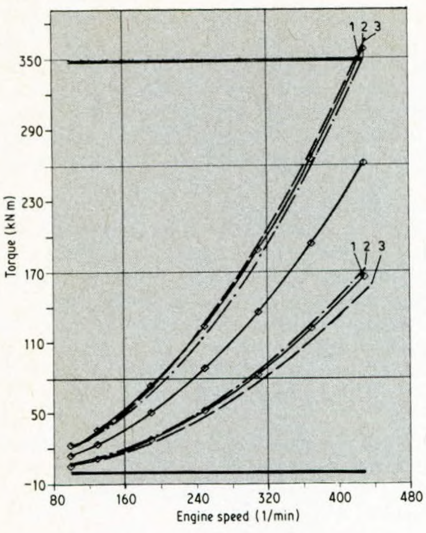


FIG D5 Corresponding vibratory load at gearbox pinion of engine no. 1 for two engines driving propeller; cylinder 1 of engine 1 misfiring

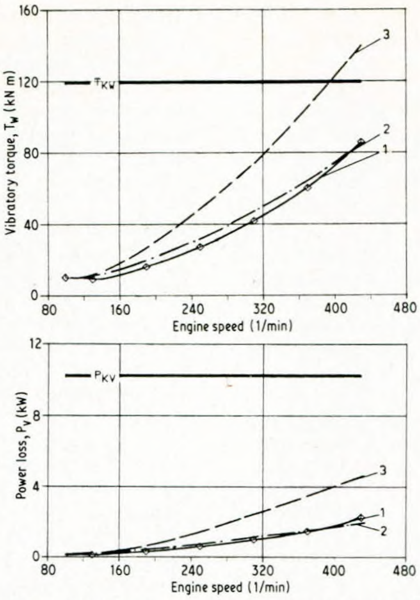


FIG D6 Vibratory torque in coupling of engine no. 2 (conditions as Fig. D4)

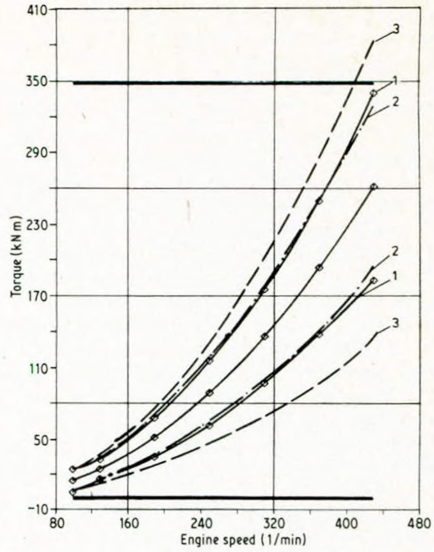


FIG D7 Vibratory torque at gearbox pinion of engine no. 2 (engine conditions as Fig. D4)

no. 1 misfiring.

Curve 1 : all other cylinders equal mep.

Curve 2 : all other cylinders ± 5 per cent mep.

Curve 3 : all other cylinders ± 10 per cent mep.

As in the previous case, we have considered only the worst combinations of imbalance.

Figure D5 shows the corresponding vibratory load at the gearbox pinion of engine no. 1.

Figures D6 and D7 show the vibratory torque in the coupling and at the gearbox pinion of engine no. 2. The engine conditions are the same as for Figs D4 and D5.

As it is possible to declutch one of the engines, we have also calculated for the case of one engine driving the propeller. The results of these calculations are shown in the following figures.

Figures D8 and D9 show, respectively, the vibratory loads in the coupling and at the gears for the engine conditions:

1. all cylinders equal mep.
2. all cylinders ± 5 per cent mep.
3. all cylinders ± 10 per cent mep.

Figures D10 and D11 show the influence of a misfiring cylinder in combination with:

1. ideally balanced;
2. 5 per cent mep difference;
3. 10 per cent mep difference.

The theoretical analysis indicates that, when the imbalance of the engines is too big, the vibratory loads induced can be even larger than with one misfiring cylinder and otherwise properly balanced cylinder.

MR G. VICTORY (FIMarE): Like Dr Archer, I would be very interested to hear further details of the cause and effects of the breakdown which initiated this interesting project. In addition to the risk to personnel from flying metal, there is the danger that by penetrating fuel tanks or fracturing valves or pipes, an escape of oil could occur and lead to a disastrous fire. It would appear that those on board were indeed fortunate to escape on both counts. These are the sort of occurrences about which the Members of the Institute would like to know; perhaps more details of such cases could be given regularly in *Marine Engineers Review*.

Some of the factors in the economic appraisal on which the decision to re-engine the vessel were taken were, in my opinion, loaded against the diesel alternative. However, I note that the author has said this was done intentionally, so that if the decision was in favour of the diesel alternative then there would be no doubt as to its correctness. It is to be expected therefore that the advantages will be somewhat greater than those anticipated!

The authors mention that the engines are designed to operate on 3500 s fuel (Redwood at 100°F) but that it is intended for the time being to operate on 1500 s fuel. As the quality of an economic fuel will undoubtedly deteriorate in the years to come (there are already signs that it is doing so), perhaps the authors would give some information on the additional equipments fitted, or foreseen, in order that the vessel may operate without trouble in the years to come.

The use of a disc brake to overcome the very high engine inertia and propeller turning moment during stopping and reversing is interesting. Perhaps we can hope that, having disc brakes in the engine room, we might soon get them on windlasses—the antiquated brakes of which are so suspect that many Masters of large tankers are unwilling to 'drop' the anchor, in case they are not able to stop the anchor and cable from running out and being lost.

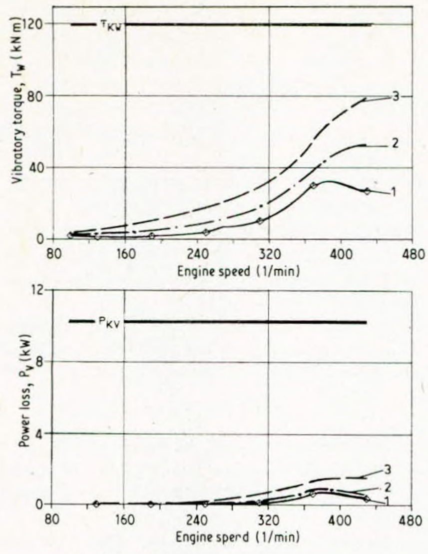
In respect of 'stopping ability', the authors state that the shaft brake design and expected manoeuvrability was confirmed during official sea trials. A graph showing results of a 'crash stop astern test' is given, presumably to support this statement. At first sight this appears very satisfactory—vessel stopped in the water from 14 knots in 12 min 34 s, with a 'travel' of 2829 m. Having seen similar claims for large tankers at IMCO it was evident what had happened. A closer look confirmed that the claims of the part played by the brake and engine in stopping the ship were exaggerated; as they are always likely to be unless the actual track of the ship is studied.

The curve of the 'ship's heading' shows that, before stopping, the ship had turned through about 160 deg; i.e. it was pointing almost in the opposite direction. Before it had reached 5 knots astern the ship had turned through another 90 deg. This raises two questions. What part had the brake and engines played in stopping the ship? What would have happened had it been necessary to attempt an emergency astern stop in a narrow channel or in congested waters?

First, these large tankers tend to be directionally unstable. They can unpredictably swing to either side or perhaps even go reasonably straight when the engines are stopped or put astern. If the vessel swings then its head comes inside the line of motion of the ship, i.e. the ship 'skids' into the turn, and the very large area of the ship's side acts as a more effective break than does the propeller. Second, if this manoeuvre is carried out in confined waters, the swing of the vessel could result in it finishing a mile or more to one side of its original course, which could be fatal in a narrow channel or result in a collision with any nearby ship.

Perhaps the authors could tell us (i) whether the ship's behaviour is predictable when the engines are stopped and put astern with the rudder in the amidship position; (ii) whether it is possible to ensure that the ship will stop in a straight line by using the rudder, assuming that it can be used, and (iii) what the stopping distance is if the ship is kept straight during the manoeuvre and not allowed to turn or zigzag from side to side with the use of the rudder. Otherwise I am afraid that the information given is at worst misleading and at best useless!

Finally, would the economics have shown it to be worthwhile re-engining the ship if it had ten or, say fifteen years of useful life ahead?



1 : all cylinders equal mep
2 : all cylinders ± 5 per cent mep
3 : all cylinders ± 10 per cent mep

Fig 8 Vibratory loads in coupling: one engine driving propeller

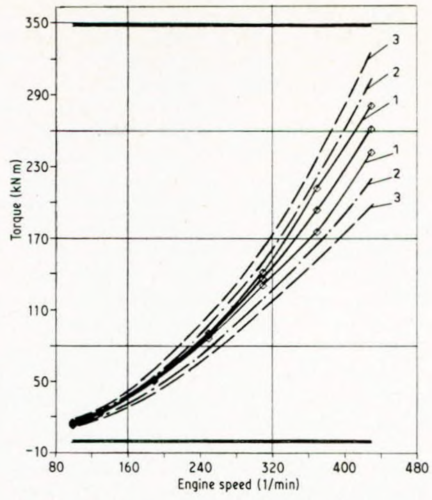
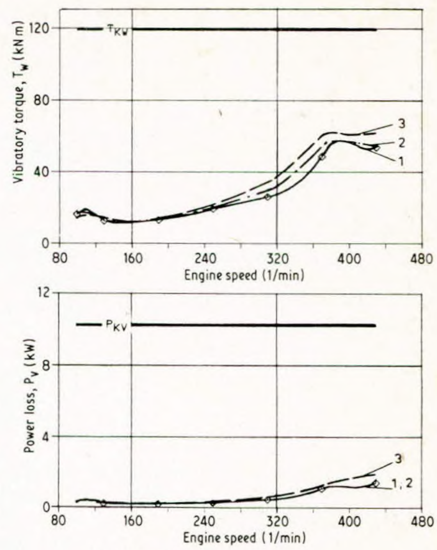
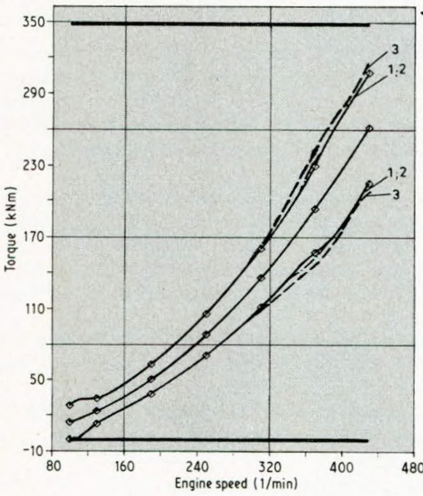


FIG D9 Vibratory loads at the gears (engine conditions as Fig. D8)



1 : ideally balanced
2 : per cent mep difference
3 : 10 per cent mep difference

FIG D10 Vibratory loads at the gears: cylinder no. 1 misfiring



◀FIG D11 Vibratory loads at the gears with cylinder no. 1 misfiring (engine conditions as Fig. D10)

A. F. HODGKIN (Babcock Power Ltd): The substance and content of this paper is somewhat distasteful to me. This is no criticism of the authors' expertise or skill. It has become fashionable for high-powered steam ships, built in better times, to be downrated and converted to diesel drive; it is the exclusion of steam from its rightful position that causes my distress. Some owners of high-powered steam ships have been able to secure worthwhile economics by retaining steam propulsion in downrated form using better gear ratios and lower propeller speed. Was this considered as an option?

It is readily understood that the accident on *Alva Sea* precipitated action on the part of the owners and doubtless there were good reasons and arguments for the solution adopted. A consideration of some importance in this matter, however, is fuel quality. It was very notable, at a recent vocal meeting of this Institute when fuels were being discussed, that great concern was expressed by several engine designers over the ability of their engines to digest some of the fuels included in a recent draft British Standard Specification, without increasing operating difficulties and maintenance. I have no wish to cause an argument over the suitability of the engines in question for the proposed service, as I am unqualified to do so; I would merely remind all concerned of the relative fuel flexibility of steam propulsion plant. Could the authors say if the conversion would have taken place had the turbine accident not occurred?

At least I can take comfort from the fact that the boilers have not been displaced from *Alva Sea*. Indeed, it appears that their present role is only marginally less important than their former one. Originally designed to provide 72.5 tonnes/h of steam at 63 bar and 513°C, each boiler is now required to produce 66.5 tonnes/h of steam, of which 3.5 tonnes/h is taken fully superheated to the feed pump, the remainder being de-superheated for the cargo pumps and steam/steam generator. Each boiler has been simplified and its operational tolerance improved by removing the primary section of the superheater together with the attemperator and the steam temperature control system.

The remaining secondary superheater, a 4-row inverted loop unit, was calculated to give about 410°C steam temperature when the feed temperature was 147°C and combustion resulted in 5 per cent excess air. After de-superheating the steam temperature to the cargo pumps was expected to be about 330°C. Not required to provide a high boiler efficiency in their cargo pumping role, the stud tube economizers were removed and pressed into service as exhaust gas heat recovery boilers. This is considered to be a very wise move as good service from the stud tube economizer in this application can be anticipated from experience gained by Gotaverken, who for many years used a similar studded tube in the construction of their own exhaust gas boilers.

I am pleased to learn from Mr Roomes that the performance of the boiler plant has been good. As a general concluding comment I should like to express the view that if we are to learn in future of anything burning out, I would rather it be engine exhaust valves than any part of the boiler system.

MR A. GALLOWAY (The Salvage Association): I should like to ask the authors the following questions.

1. Has the shaft brake proved satisfactory in operation?
2. Has it been found in practice that the waste heat recovery system supports the normal electrical loads at 55 per cent of main engine mcr?
3. Great care has obviously been taken in regard to the setting up of a maintenance system. With the passage of time has it been found that an increasing element of outside assistance is necessary?
4. Can the author indicate, with the benefit of hindsight, what factors he would have altered, and in which way they could have improved (a) the economics of operation; (b) maintenance, and (c) crew acceptability?

DR J. F. SHANNON: The failure sequence of the turbine and gearing as described by the authors points to a sun pinion-torque tube failure, freeing the LP turbine, which overspeeded causing severe damage. The machinery was obviously a Stal Laval design, which had the main wheel gearbox spring mounted and tilting under load, thereby putting the fixed turbine out of line with the epicyclic gear. A long torque tube, with gear tooth couplings at either end, connected the sun pinion and turbine. In the first generation of these machines, the gear tooth couplings had little or no clearance and, with the torque tube stiff in bending to avoid whirling, there was little flexibility to accommodate misalignment.

In the new design with diesel engines, the gearbox and engines are chocked to the integral foundations. Each engine shaft is connected to its hollow pinion by means of a quill shaft which has a resilient rubber coupling at the engine and a gear tooth coupling at the aft end of the pinion. The gear tooth couplings are barrelled and give, with the quill shaft and resilient rubber coupling, sufficient angular and radial flexibility to give a good cardan shaft effect. The resilient rubber couplings also give axial flexibility between the engine and gear. Thus there is sufficient flexibility between the engine and gear to cope with misalignment likely to occur between them.

The alignment of the propeller shafting and main wheel, to give about equal loads on the main wheel bearings, shows that the intermediate bearing is the most suitable and effective bearing to adjust to changing ballast conditions. The analysis given by the authors is for hot and cold static conditions.

When both engines are running at equal power, the forces on the main wheel from the pinions cancel, so no additional stability is gained. With only one engine running, the main wheel is stabilized when the forces are downward. However, when the pinion force on the main wheel is upward, the main wheel is lifted to the top of the journal bearings. With normal circular bearing clearances, this would be of the same order as the difference between the hot and cold settings. At the higher loads the wheel would be stable with substantial loads on each main wheel journal bearing.

However, when the power from the one engine is such as to balance the dead weight, the main wheel would be floating and conditions could be unstable and very much susceptible to adverse ballast conditions. Have any adverse effects been observed at this condition?

Would the authors state if circular bearings are used or, for example, were close clearance–taper land journal bearings used to give better control of the position of the main wheel?

Another interesting feature of the gear design is the use of carburized, case-hardened and ground pinions on a through-hardened heel rim, also profile ground, giving ample margins in gear tooth strength and specific loading. As the gears probably were ground on Maag machines, the surface finish would be of the order 0.3 μRa (CLA), not 3.0 μRa as recorded. Presumably this is a typographical error. It must always be noted that, to get the best from the 'hard on soft' tooth combination, the 'hard' pinion must have a very good surface finish on which the 'soft' teeth will polish. Otherwise, the wheel tooth surface will deteriorate.

In earlier studies of main shaft brakes for stopping the propeller, it was preferable to have at least two discs, giving a more compact arrangement and increased reliability.

Reference to the torsional vibration characteristic with its damping and freedom from critical speeds, with monitored power balance in each cylinder, reflects a thorough investigation of this aspect.

The success of this machinery is very creditable and the authors' review of the subsequent service experience is very just, showing the improved fuel efficiency but also the defects of excessive noise and increased maintenance of the engines in comparison with the previous turbine machinery.

MR A. GATSIADIS (Tarponship SAM): The knowledge and experience gained from this project must have been extremely useful for similar conversions that followed the re-engining of *Alva Sea*.

It is interesting to note that, once again, the allowance for hull wetted surface deterioration has proved to have been underestimated. There is more than sufficient evidence today to accept a figure of 5–6 per cent as more realistic.

In view of the importance of the project, would it not have been advisable to modify the propeller for optimum efficiency and apply underwater urethane high gloss coatings to a suitably prepared steel surface? The resulting increase in propeller efficiency and decrease in frictional coefficient would have allowed a reduction in power for the same rev/min and the same speed. The combination of these two factors would have produced a further saving in fuel oil consumption and main engine maintenance costs.

CAPT. J. A. SMITH (FIMarE): The authors' account of the fitting of a shaft brake recalls the Elder Dempster Lines 'A' Class twin-screw passenger vessels, where, in the 1950s, pneumatically-operated calliper brakes were fitted to both shafts, but for somewhat different reasons.

In those ships the problem was not the size and momentum of the vessels, but the excellent balance and very high mechanical efficiency of the Doxford LB two-stroke opposed piston engines. These took a relatively long time to slow to a shaft speed at which the astern air could kick the engine into reverse sufficiently for the fuel to catch the engine in reverse before it began to turn ahead under the influence of the propeller in the ship's wake.

Of course, a skilled and determined operator, prepared to use the necessary starting air, could and always did achieve a good start astern, but a timorous whiff of astern air and a dilatory change to fuel could indeed find the engine turning ahead, albeit slowly, when it fired. The brakes fitted to these ships were quite effective in preventing the consequences of such maloperation.

However, later designs of engines proved to have less angular momentum, and, as far as I know, such brakes had not been fitted until the authors perceived their value in securing a prompt start astern in what might be described as a bigger league.

PROF. S. IZUMI (Nagasaki University): I find the paper particularly interesting because the authors describe in a practical way the complete conversion procedure, from the economical feasibility study to the results of the operation.

I feel that special consideration of engine bottom structure and shafting alignment would provide very valuable material for another, future project. I should like to ask the authors to comment on the following two points.

(1) The difference of astern performance with shaft brake and without shaft brake.
(2) The method of load sharing between the two engines.

M. J. GALLOIS (SEMT): About one year prior to the *Alva Sea* conversion, we were involved in the conversion from steam to medium-speed diesel propulsion of a tanker of 285 000 t/dwt by our licensee, Messrs I.H.I. at AIOI. We are thus well aware of all the problems which have to be solved in such a case, and congratulate the authors on this comprehensive report.

Nevertheless, we have a few questions and remarks. First, regarding the amount of labour caused by such conversion: could the authors give some figures regarding the time necessary for removal of the old, and installation of the new propulsion machinery; and for pre-fabrication of modules? How many tons of steel had to be used to build up the new seatings, and so on?

Regarding the selection of the medium-speed engines; the authors refer to the 'proven service reliability' of that engine as well as to 'the fact that units had been at sea since September 1970'. To our recollection, this type of engine had, at least initially, suffered from repeated casualties, as reported in several marine journals. What had been done to achieve the present reliability?

Although not mentioned in the paper, we presume *Alva Sea* has two 16-cylinder V-engines. If so, there are 32 cylinders to maintain. Could the authors comment on the justification for this choice, especially since medium-speed engines are available which can provide more output with only $2 \times 12 = 24$ cylinders?

One major reason to make such a conversion is the low fuel consumption of medium-speed engines. The figures for specific fuel consumption in Table I, upon which the calculation of the financial gain must be based, do not seem to be very precise according to our own experience.

For the steam turbine, about 210 g/hph can only be reached at optimal conditions, when the turbine runs at the design point. Considering manoeuvres, part-load, etc., a good average figure is about 230 g/hph. Slow steaming, as many tankers do these days, can even mean 270 g/hph.

On the contrary, the specific fuel consumption for the diesel installation of about 150 g/hph is higher as we would expect, based on practical experience with 22 medium- and large-size tankers in service with 15 000–36 000 hp medium-speed machinery. Could the authors give us the real consumption figures in service?

Note: Unfortunately the author was unable to reply to the above contributions.