THE INSTITUTE OF MARINE ENGINEERS
76 Mark Lane, London EC3R 7JN *Telephone: 01-481 8493 Telex: 886841*

TRANSACTIONS (TM)

ECONOMIC POWER GENERATION AT SEA: THE CONSTANT-SPEED SHAFT-DRIVEN GENERATOR

G. G. Pringle

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Economic Power Generation at Sea: The Constant-speed Shaft-driven Generator

G. G. Pringle CEng, FIMarE, MIMechE
Vickers Shipbuilding and Engineering Limited

SYNOPSIS

Escalating fuel costs and forecasts of a future deterioration in quality of fuels are together challenging shipowners and designers to develop alternative methods of achieving economies in ship operation. This paper describes a new concept in power generation from variable-speed propulsion engines, which offers a practical solution to the problem of economic generation of power at sea. The author describes a power take-off system from a variable-speed prime mover which, by virtue of the speed control capability inherent in a variable-ratio epicyclic, delivers a constant-speed output for efficient power generation. A number of alternative arrangements are discussed. The possibility of retrofitting to existing ships is considered.

INTRODUCTION

Since the oil crisis of 1973 and the massive fuel price rise in 1979, a great deal has been said and written about the effect on ship operating costs. Some authorities claim that, at present, fuel is absorbing about 60% of ship operating costs. Ten years ago the figure was 20%.

Predictions about the future escalation rates for marine fuels vary between slightly and extremely pessimistic. The only point of agreement is that conditions will not improve!

It would be bad enough if this was the only problem. However, another more serious problem has been predicted. The landward demand for more distillates is increasing and the inevitable consequence is that the residual fuel currently available to the shipping industry will deteriorate in the not-so-distant future.

The price differential between marine diesel fuel and 380 cSt residual fuel is presently about 2:1. It is unlikely that this gap will close significantly. Energy conservation has become a growth industry, not least in the number of papers written on the subject.

Shipowners have been forced to look at ways of reducing fuel costs. With existing vessels there are few choices available. Slow running is now an established practice; unfortunately, this method can have deleterious side-effects on the prime mover. In larger vessels, steam-to-diesel conversions are becoming more numerous.

Current newbuildings have been designed for lower and more economic ship speeds than hitherto. Main propulsion engines have been designed for improved economy. But what of the future?

The ship designers will continue to improve hull and propeller configurations to give higher propulsive efficiency with consequential economies. The leading diesel engine licensors are developing engines with improved thermal efficiencies and lower fuel consumptions, capable of burning 600 cSt residual fuel and of operating at still lower speeds.

While these improvements will have a significant effect on operating costs, there is an area of ship operation which is capable also of producing significant savings—the generation of electrical power.

ALTERNATIVE METHODS OF GENERATING POWER

As this paper concerns economic power generation at sea, perhaps diesel generators should not be mentioned, except to state that it is an alternative to diesel generators that is proposed.

There are several alternatives extant and the author would like to dispose of these first before submitting the Vickers Shipbuilding and Engineering Limited (VSEL) proposal.

Turbo-alternator

Normally this system has been confined to installations with service ratings above 15 000 BHP. The steam-raising plant and ancillary equipment with the turbo-alternators involve a high initial capital cost. The main problem area is the boiler plant where corrosion, due to the cooling exhaust gases, can engender expensive maintenance.

Mr G. G. Pringle CEng, FIMarE, MIMechE served a marine engineering apprenticeship combined with technical studies in Glasgow. From 1954 to 1966 he was chief draughtsman of David Rowan and Fairfield–Rowan. Mr Pringle then joined Barclay Curle as technical manager. In 1978 he was appointed quality assurance manager while retaining all technical responsibilities. Mr Pringle's current priority is the development, for manufacture, of the CSGD concept.

It should be noted that the improvement in thermal efficiency of the slow-speed diesel engine over the past 14 years has resulted in a reduction in exhaust gas outlet temperature of 50°C. The ability to maintain the electrical sea-load requirements by means of a turbo-alternator now requires more sophisticated and, therefore, more expensive heat exchange equipment. The limit on engine power quoted above is possibly too high.

Power take-off systems

Power take-off systems offer the shipowner the opportunity to generate, at sea, all power requirements, utilizing the benefits accruing from the main engine: high thermal efficiency, low specific fuel consumption and, most important of all, the use of the cheapest low-quality fuel for which the main engine has been designed.

Power take-off systems in current use are completely electric in their operation.

The first system is not very common. This is the generator–motor–alternator system, whereby a DC generator is driven by the main engine via a speed-increasing gearbox through a flexible coupling to damp out torsional vibrations, etc. The generator drives a DC motor coupled to an alternator. The excitation field of the DC generator is varied to maintain constant frequency with variable engine speed. The equipment is bulky. Efficiency is around 80%.

The second system is in more common use. This is the shaft alternator, generally incorporated in the intermediate shaft. The latest types use thyristor rectification and a static invertor to obtain a constant-frequency alternating current output. The disadvantages are:

1. An additional machine of similar size is required to generate the necessary reactive power.
2. The alternator is large and heavy.
3. It is near to the tank top and exposed to bilge water, etc.
4. It is susceptible to mechanical damage, particularly when fitted to an intermediate shaft which requires dismantling to permit tailshaft withdrawal. The efficiency is about 81%.

VSEL have approached this problem in a novel way and have developed a power take-off system which is not available currently from any other source. This system is named 'the constant-speed generator drive' (CSGD). It is covered by UK and European Patent Applications.[1]

THE CONSTANT-SPEED GENERATOR DRIVE

The concept

The basic design comprises a speed-increasing gear train in series with a variable-ratio epicyclic train. Usually, a variable-ratio epicyclic will have a constant input speed and a variable output speed.

VSEL have simply reversed this concept. The CSGD operates with a variable input speed and a constant output speed. Figure 1 illustrates the principle.

The system consists of:

1. A train of input speed increasing gears driving one member of the epicyclic gear train (the planet carrier).
2. An output shaft driven by a second member of the epicyclic gear train (the annulus).
3. A fixed-displacement hydraulic pump unit (reaction) connected to the third member of the epicyclic gear train (the sunwheel).
4. A variable-displacement hydraulic pump unit (control) connected to the input shaft.

The sensitivity of control is such that a generator connected to the output shaft can be stabilized at $\pm 1\%$ of rated speed, while the input

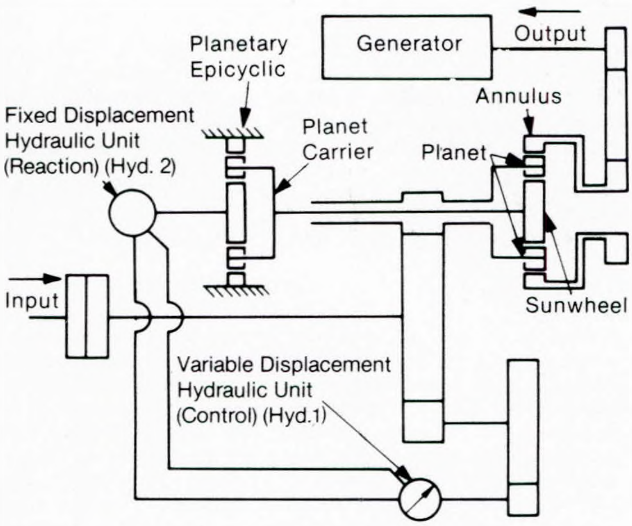

FIG 1 Schematic diagram of the CSGD

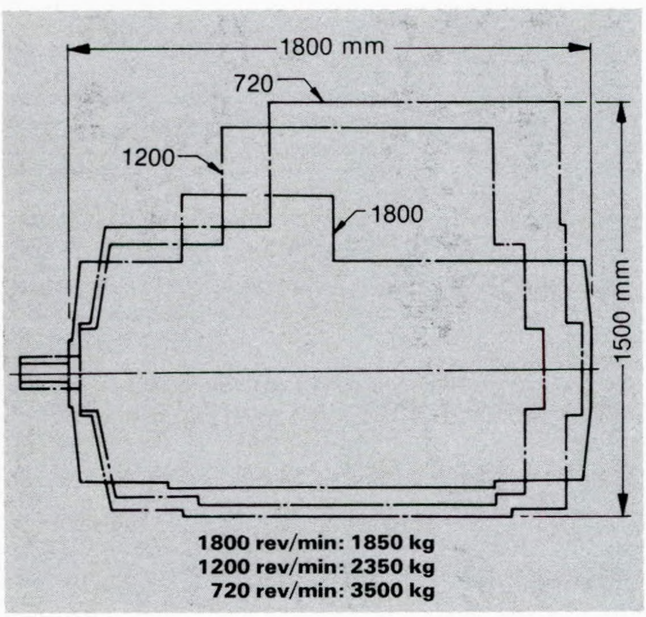

FIG 2 Size and weight variation for typical 700 kW alternators

shaft speed from the prime mover fluctuates over a range of 60 to 100% of design speed. The overall efficiency is above 88% throughout this speed range.

This accurate speed control is achieved by variation of both direction of rotation and rotational speed of the epicyclic sunwheel. This control is achieved by a hydraulic power loop system linking the hydraulic pump units. A speed-sensing device on the output shaft actuates the electronic controls of the hydraulic pumps.

The development

The pre-development stage followed a pattern which can be recommended.

The conceptual design was discussed widely with shipowners and also with the designers of ships and engines in the UK and in Europe. The consensus from the extensive exercise was that the product as proposed was not only viable, it was badly needed by the shipping industry.

Based on these discussions it was decided that initial development would cover a range from 250 kW to 1400 kW. This upper limit was immediately overrun by an enquiry for a design for 1800 kW!

The design would be for a free-standing unit, suitable for connecting to any engine design.

A review of the world shipping market confirmed the view that the slow-speed diesel engine would command the major share of the market in the next decade. Incidentally, the term 'slow speed' appears to be becoming slightly stretched. The latest engine designs emanating from the major European licensors offer an operating range with speeds varying from 70 to 200 rev/min. The decision to base the CSGD design on the slow-speed diesel engine still offers VSEL plenty of scope.

At this point, detailed discussions with Burmeister and Wain Engineering Limited (B & W) set two development requirements, involving close co-operation between the companies.

1. A 700 kW, free-standing unit capable of connection to the current GFCA engine range. This unit would become part of the standard free-standing design range.
2. A 350 kW integral unit for fitting direct on to the forward end of the latest L35GB(E) engine. Since this is the first development of the new engine range from B & W, it will be seen that this application had considerable potential for extension, where practicable, to cover the larger HP members of this engine family.

The author will discuss these two specific applications.

The alternators

A wide choice of alternators is available from various manufacturers. Any of the standard designs can be used in the CSGD concept.

In the opinion of the author, the alternator is an integral part of the total electrical design and should be supplied as part of the electrical outfit. It is the intention that the CSGD will be constructed to incorporate any specified alternator, particularly when a newbuilding contract is involved. The alternator, therefore, does not form the subject of this paper, except in so far as its speed and interface connections are concerned.

With regard to speed, the choice was limited to either 1200 or 1800 rev/min.

1. 1800 rev/min permits the use of a smaller and, most important, lighter alternator. It does require an increase in gearing ratio over the 1200 rev/min unit.
2. At 1200 rev/min the main disadvantage is a larger, and about 30% heavier, alternator. The advantage is a lower gearing ratio than on the 1800 rev/min unit.

On balance, it was decided that the 1800 rev/min alternator offered the best compromise. The standard range is based on that output speed.

As a footnote, it must be mentioned that an interested shipbuilder indicated that he would probably prefer an alternator speed of 720 rev/min. This would give interchangeability with the diesel-driven alternators normally installed. The penalty would be about 90% increase in weight, with a consequential increase in bulk and cost.

There is no limitation on the output revolutions and special non-standard designs can and will be produced, on request.

Figure 2 illustrates the size and weight variation for a 700 kW alternator.

The standard range

It is at present the intention to limit to four the number of frame size variants in the standard range. This, of course, may change as the variants are developed within the proposed range of 250 kW to 1400 kW. However, the fewer variations to be accommodated, the greater the reduction in both development and manufacturing costs.

The overall dimensions of each CSGD unit are controlled by two variables:

1. The input speed range.
2. The output power.

The first-named affects the gearing ratios and controls both width and height of the gearbox. The second affects tooth width, bearing sizes, alternator and hydraulic pump dimensions and controls the fore and aft length of the gearbox.

The 700 kW free-standing CSGD gearbox

The technical description of this unit, as referred to in the introduction, will be sufficient to describe the complete standard range proposed. The specification requirements are as follows:

1. To develop 700 kW output at the alternator with constant speed of 1800 rev/min.
2. To be capable of controlling the speed and, therefore, the frequency of the alternator at that output to $\pm 1\%$ at the rated voltage.
3. To be installed at the forward end of a slow-speed diesel engine.
4. To be aligned coaxially with the forward crankshaft coupling.
5. To be supported by its own seating and holding-down bolts.
6. To be capable of absorbing engine movement, namely: axial and torsional vibrations; thrust clearance, and thermal expansion.
7. To be capable of absorbing engine starting torques and accelerations.

DESIGN CHARACTERISTICS

In the design, the prime mover—a slow-speed diesel engine—will be required to operate over a speed range of 63 to 100 rev/min while the generator operates at its required speed.

The normal space limitations in an engine room, particularly at the forward end, create an obvious need to make the gearbox as compact as practicable. The use of the words 'compact as practicable' is deliberate. The author has suffered, as have the majority of marine engineers, from problem equipment in which the term 'compact' meant inaccessible.

From the start, the accent has been on accessibility. The gearbox is designed with three main features in mind:

1. Ease of manufacture: fabrication, machining and assembly have been considered at every stage of development.
2. Ease of access: this inevitably reduces assembly problems and ensures ease of maintenance.
3. Ease of installation: the gearbox will be installed complete with alternator, pumps, piping and control valves, etc. It will be simple to align to the crankshaft forward coupling and will be secured on chocking material similar to that used for supporting the main engine.

The gearing has two distinct elements. The speed-increasing train is a combination of first-stage parallel shaft gears and second-stage epicyclic gears. The use of epicyclics in the speed-increasing gear system has two main advantages:

1. It allows the gearbox to be more compact and lighter in weight.
2. It reduces the number of take-off pinions and gears for driving the hydraulic units.

The parallel shaft gears have teeth of involute spur design with the journals supported on pre-loaded roller bearings. Gear rims are manufactured in material to BS 826 M31 specification and are induction-hardened and ground in accordance with the highest standards applicable at VSEL. The gear pinions are manufactured in material to BS 655 M13 specification and are gas-carburized and ground to the same standard.

The speed-increasing and control epicyclics incorporate the VSEL patented flexible pin system of achieving uniform load distribution at the multiple meshes of an epicyclic gear train.

This design concept has been described in detail in previous papers presented at this and sister Institutions.[2] It should be sufficient to state that over a period of 19 years VSEL and their associates, Compact Orbital Gears, have produced a large number of epicyclic gearboxes with a minimum of problems.

In fact, the design principle has been applied very successfully to a prototype set of 30 000 shp epicyclic gearing. This unit, after completion of extensive proving trials of 33 000 shp, was eventually run for 500 hours at an output of 48 000 shp with gear elements and bearings showing an excellent condition.[3] An alternative design, more applicable to the upper range of main engine speeds, utilizes a speed-increasing train which is of parallel shaft design with a control epicyclic only.

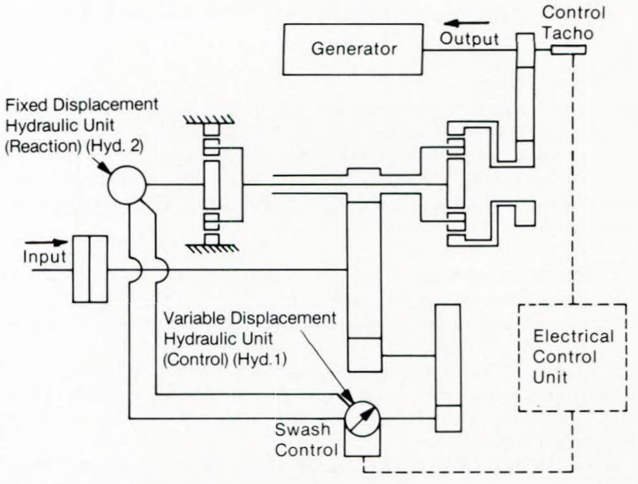

FIG 3 Control system

CONTROL SYSTEM (OPERATION)

The theory of the system has been described earlier. It is of interest to examine how the system actually operates on a specific design.

- Having established the minimum and maximum engine speed parameters, in this instance 63 and 100 rev/min respectively, a reference speed is established at which the epicyclic member (the sunwheel) is stationary. The epicyclic operates at this condition as an inverted (speed-increasing) planetary gear with a fixed reaction member (the sunwheel), to give an annulus speed which will operate the generator at 1800 rev/min.
- Should the sunwheel remain stationary, an increase in engine speed would be accompanied by an increase in annulus and, therefore, generator speed.
- To maintain a generator speed of 1800 rev/min with an engine speed of 100 rev/min, the sunwheel must be allowed to slip. In this condition, hydraulic unit no. 2 is driven by the torque reaction in the epicyclic and power is developed by the sunwheel.
- It would be uneconomic to waste this power. It is therefore introduced via the hydraulic power loop system to hydraulic unit no. 1, which absorbs and reintroduces the power to the input gearing system. Here it augments the power being delivered by the engine through the mechanical transmission.
- Similarly, a decrease in engine speed with the sunwheel stationary would give a reduced generator speed.
- To maintain the generator speed at 1800 rev/min with an engine speed of 63 rev/min, the sunwheel must be driven. In this condition the sunwheel absorbs power from hydraulic unit no. 2 which, in this mode, operates as a motor receiving power through the hydraulic power loop system from hydraulic unit no. 1 operating as a pump. Hence power is delivered from the engine through a hydraulic path and a mechanical path.

The epicyclic adds these powers and delivers the total power to drive the generator at 1800 rev/min.

MAIN ENGINE REACTIONS

One of the problems to be considered when designing a gear installation for direct attachment to a slow-speed diesel engine is how to compensate for the normal engine reactions.

Thermal growth

From cold to normal working temperature, a slow-speed diesel engine will expand lengthwise about 5 mm for a six-cylinder engine. The crankshaft centre line will rise about 0.5 mm with the same temperature rise.

Thrust clearance
The engine crankshaft can move 1 mm when taking up the thrust clearance from ahead to astern.

Axial vibrations
An amplitude of 0.8 mm can be attained in a six-cylinder engine with an acceleration of 0.6 ***g***.

Torsional vibrations
Since the forward coupling is the free end of the crankshaft, the maximum amplitude of torsional vibration occurs here.

The cumulative effect of these engine operating conditions makes the inclusion of a flexible coupling imperative.

ENGINE STARTING TORQUES AND ACCELERATIONS

It is accepted that all slow-speed diesel engines incorporate some form of robust involute spur gears which absorb these forces and operate successfully for long periods. However, it may be considered that a direct-driven gearbox and alternator system should be protected from such loads. This is despite the fact that design calculations show that the actual tooth loading stresses, under engine start-up conditions, are well within the safety margin.

It is possible to incorporate a clutch in the input shaft between the prime mover and the gearing, to isolate the unit completely during starting and manoeuvring periods. The operation of the clutch would be synchronized with engine speed to allow the system to 'clutch in' when the minimum acceptable steady running condition is achieved. Generally this will be at about 40 rev/min.

The inclusion of an isolating clutch would ensure that the gearbox need not run astern and would simplify the hydraulic power control system.

Various alternatives have been considered for separate clutches and flexible couplings, or for combined units. In the 700 kW design it has been decided that the standard arrangement will include a combined clutch/flexible coupling unit. The clutch may be fitted at the shipowner's discretion.

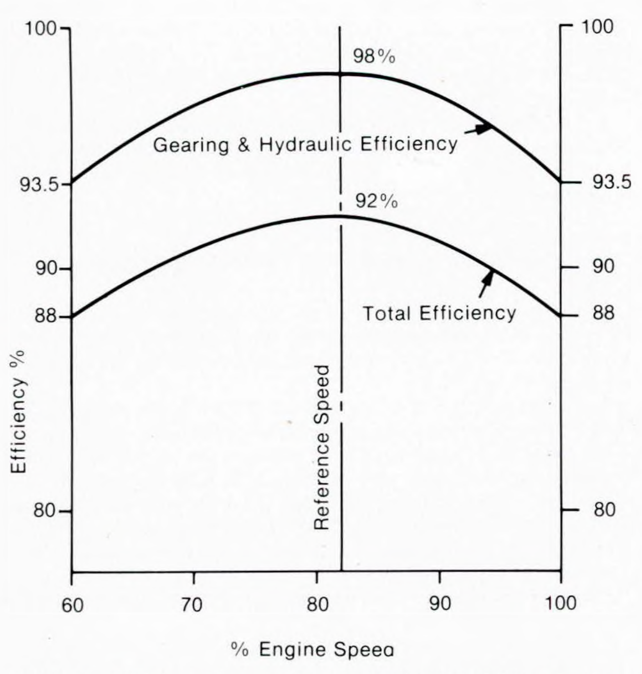

FIG 4 Efficiency curves for the 700 kW CSGD over the main engine operating range

LUBRICATION SYSTEM

An advantage when installing a CSGD unit with a slow-speed diesel engine is that the lubricating oil specification is common to both.

The recommended oil is a rust- and oxidation-inhibited mineral oil of SAE30 viscosity. The specified limit of filtration for the roller bearings is 25 μm.

An integral lubrication system is proposed, consisting of an independent motor-driven starting-up oil pump, a gear-driven oil pump, filters, and cooler. The starting-up pump will be arranged for automatic shut-down when the gear-driven pump reaches the operating pressure.

HYDRAULIC UNITS AND CONTROL SYSTEM

The hydraulic units are of conventional and proven design, incorporating axial pistons and swashplates.

Hydraulic unit no. 1, driven from the input gear train, is a variable-displacement unit, the output flow being controlled by variation of the swashplate angle. A booster pump is close-coupled to this unit for initial pressurizing of the hydraulic system.

Hydraulic unit no. 2, coupled to the epicyclic sunwheel, is a fixed-displacement unit in which the swashplate angle cannot be altered.

The control system, as shown in Fig. 3, consists of an electro-hydraulic servo valve fitted to hydraulic unit no. 1 and controlling the swashplate movement. Electronic equipment incorporated in the units balances the power outflow conditions in the closed hydraulic loop between units 1 and 2 and provides a theoretically perfect regulator against normal variations of engine speed and alternator loading.

The equipment compensates automatically for the inherent slip losses in the hydraulic pump–motor combination and avoids speed variations from this source.

Tacho-generators are fitted in the input and alternator shaft lines to give visual display of speeds on a locally-mounted gaugeboard. Connections for remote monitoring are included as standard.

The alternator tacho-generator connects also to an electronic comparator, adjustable to sense underspeed and overspeed and de-energize protective relays. This interface will be integrated with the overall engine room protective relay system to ensure sequential starting of standby alternators when required.

EFFICIENCY

As previously stated, the overall efficiency of power generation by this method is above 88% at the maximum chosen speed.

Figure 4 shows the efficiency curves for the 700 kW CSGD over the operating range of the main engine. It will be noted that maximum gearing efficiency of 98% is attained at the design reference speed, when the control epicyclic sunwheel is at zero speed and no hydraulic power is being transferred.

The curve of total efficiency is based on an alternator efficiency of 94%.

OVERALL DIMENSIONS

The 700 kW CSGD free-standing unit is illustrated in Fig. 5. The overall dimensions are 2.8 m high × 3.4 m wide × 2.4 m long. The unit weighs about 14 t.

350 kW INTEGRAL CSGD UNIT

This design is shown in Fig. 6. The gear casing has been designed for manufacture in a nodular cast iron. An alternative design is available for fabrication.

The unit is connected to the tuning wheel casing flange with the input shaft bolted to the crankshaft forward coupling. A flexible coupling is included and a clutch can be incorporated.

The main variation from the arrangement described previously is the introduction of final drive bevel gears to permit the use of a vertical spindle alternator, which is supported from the engine column.

A number of manufacturers are prepared to satisfy the requirement for an alternator with a special thrust bearing at the lower end to give support to the rotor.

The overall dimensions of this unit are 2.9 m high × 1.75 m wide × 2.1 m long. The weight is about 9 t.

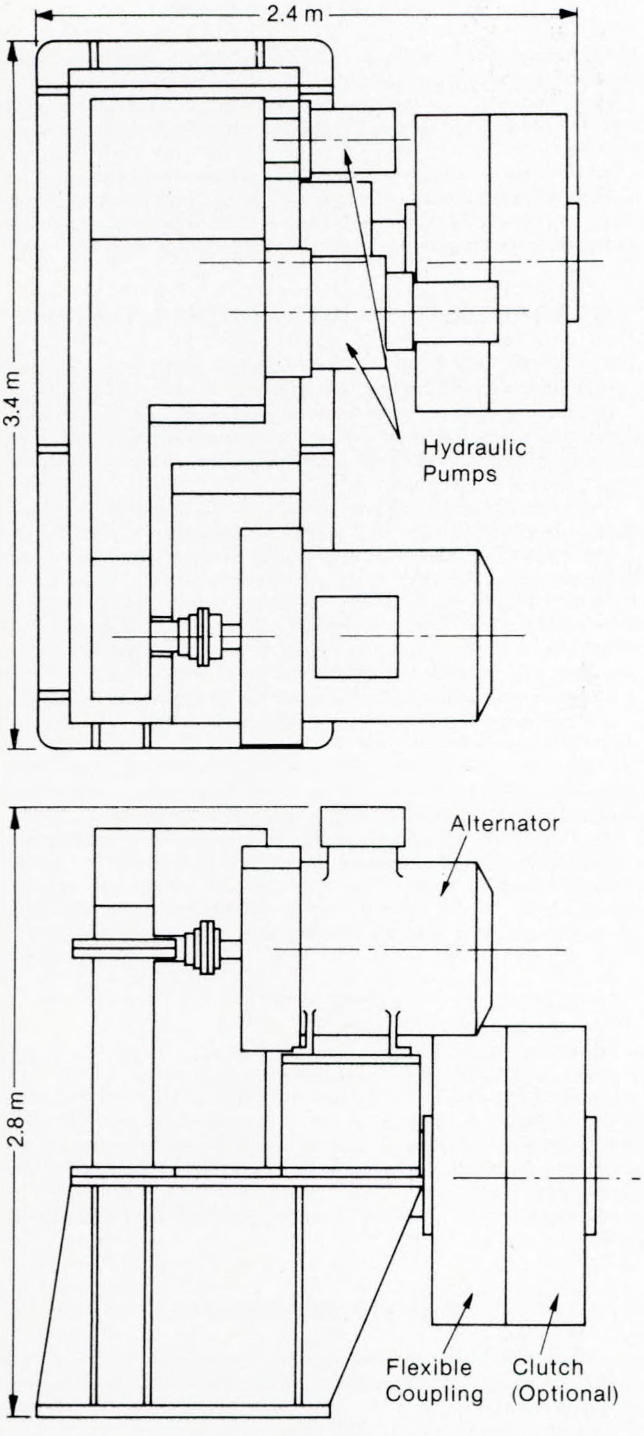

FIG 5 The 700 kW CSGD free-standing unit

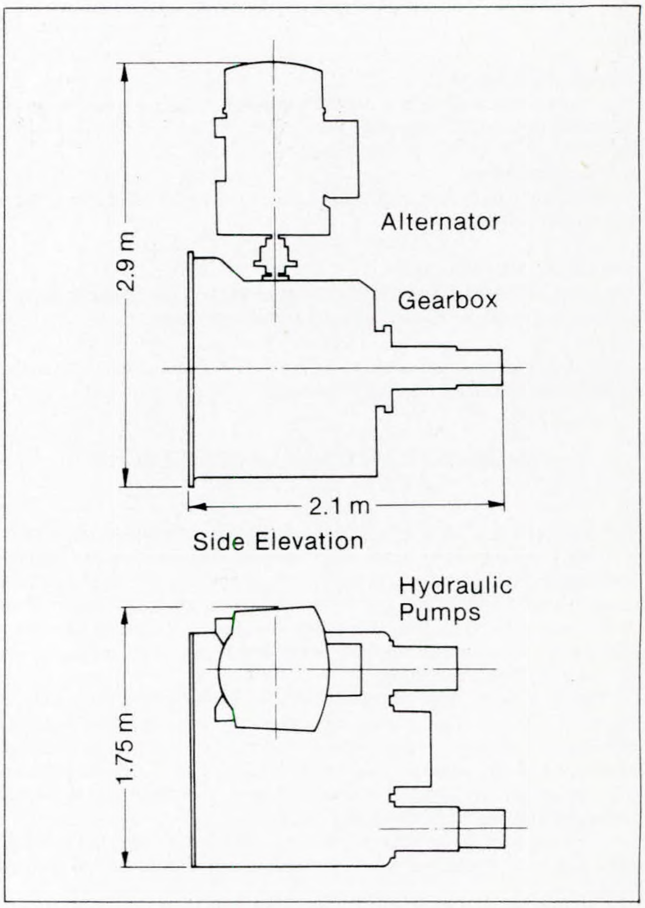

FIG 6 The 350 kW integral CSGD unit design

RETROFITTING

In the existing merchant fleet, many vessels are operating with reduced speeds. This creates a surplus margin of power in the main engine which is available to drive a constant-speed generator.

Normally, the space available in an existing engine room is restricted. However, the possibility of a retrofit CSGD unit merits consideration.

While the basic proposal is direct drive from the forward end of the main engine, a study has been made of the practical application of a flexible connection by means of a multiplex chain drive. This system frees the CSGD unit from the restrictions imposed by the lack of space between engine and forward bulkhead.

For instance, it is possible to consider positioning the unit on either side of the main engine in a parallel alignment. Alternatively, it may be advantageous to consider the introduction of a raised flat, either forward or aft of the main engine, on which the CSGD unit can be mounted.

A number of recent machinery installations have been examined in the context of the installation of a CSGD unit to carry the nominal electrical sea load and it would appear that a retrofit exercise is worthy of consideration in very many of these.

COMPARATIVE COSTS AND SAVINGS

The shipowner can be assured of the positive financial savings which will accrue from the installation of a CSGD unit in place of a diesel-driven generator, savings which more than justify the slightly higher initial capital cost.

The installation cost of a complete CSGD unit is approximately 20% higher than the cost of the equivalent diesel generator.

The remaining diesel generators will operate on a much reduced work-load, covering port, manoeuvring and standby duties.

The cost saving resulting from this for a 700 kW installation has been assessed at approximately £75 000 per annum, made up from:

£65 000 per annum for fuel savings from power generation by 380 cSt fuel against 40 cSt fuel.

£7000 per annum from reduction in cost of maintenance and spares for the diesel generators.

£3000 per annum from the reduction in lubricating oil consumption for the diesel generators.

An additional bonus can be the reduction now possible in diesel fuel storage requirements.

CONCLUSION

The paper has described a novel concept for economic power generation at sea. The individual elements in the design have the dual advantages of simplicity and proven reliability in service. It is the considered view of VSEL that the combination of these elements have retained the same level of simplicity and reliability.

ACKNOWLEDGEMENTS

The author thanks the technical teams at VSEL, Compact Orbital Gear Works and Barclay Curle Limited for their assistance in the preparation of this paper. In particular, thanks must go to J. R. G. Braddyll, and P. W. Murrell of VSEL, but for whom the paper would not have been written.

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Discussion

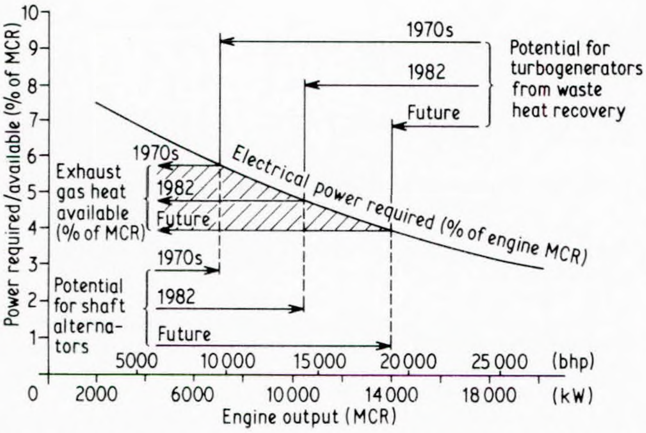


FIG D1 Trends in electric power generation, waste heat recovery and shaft alternators

Table DI: Alternative alternator systems (500 kW)

TYPE	EFFICIENCY (%)	CAPITAL COST (£ × 10^3)	FUEL COST (£ × 10^3)
Diesel alternator (MDO)	94	90	109
Shaft alternator (thyristor-controlled)	83	235	62
Shaft alternator (CSGD)	91	180	56

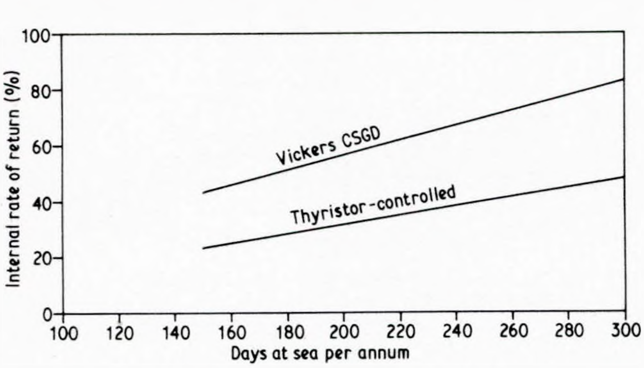


FIG D2 Shaft alternator systems

M. F. CRAIG (British Shipbuilders (Engineering and Technical Services) Limited): The author and Vickers Engineering are to be congratulated on presenting a new and elegant design for providing a constant-speed drive.

I should like to support the author's comments on the choice of alternative methods of generating electricity. There is no doubt that waste heat recovery for electrical generation can offer a most economical form of generation but there is a limitation on the size and type of main diesel engine which can provide sufficient exhaust heat.

Figure D1 highlights how one effect of improved engine and turbocharger efficiency has been to reduce the heat available in the exhaust gas, and the cross-over point at which a waste heat recovery system and turbo-alternator become self-sustaining has significantly increased. If we assume a simple linear relationship between the electrical power required at sea and the engine output, it can be seen that, whilst five years ago one could design a waste-heat turbo-alternator system for 400 kW of auxiliary power at 7000 kW MCR, the current cross-over point is nearer 11 000 kW and moving to about 14 000 kW with the future engine designs.

We have also made a cost comparison of different types of shaft alternator systems (Table DI) compared with the reference case of a 500 kW diesel alternator running on marine diesel oil. The table summarizes the differences between a widely installed type of shaft alternator, mounted integral with the intermediate shaft and thyristor controlled, and the Vickers constant-speed generator drive. The efficiency of the integrally mounted shaft alternator is less and the cost is higher. For the Vickers constant-speed generator drive an extra cost of £90 000 (if we can assume that for a new building one of the original diesel alternators may be deleted) will produce a fuel saving of £53 000 per year for the current differential in fuel oil prices between marine diesel oil and heavy oil fuel.

A fuller economic assessment, taking into account the variation in lube oil consumption, spares and maintenance costs, was considered and a DCF calculation made over a 10-year period for varying numbers of days at sea (Fig. D2). It is clear that, for a reasonable number of days at sea per year, a healthy rate of return is made on the initial investment, equivalent to over 65% for 250 days at sea per annum.

In considering the space requirements at the forward end of the engine, it is clear that the designers have done a good job to shrink the assembly into a small space (Fig. D3) and, by turning the output drive through 180 deg, have made a very compact unit. Superimposing the outline for a 500 kW constant-speed generator design on a typical arrangement would save at least one and a half frame spaces when compared with a conventional in-line generator.

I should like to conclude by asking the author a question concerning the lubrication of the epicyclic gearing. If the design is optimized to place the reference speed with highest efficiency close to the normal running condition, then the sunwheel and drive to the fixed-displacement hydraulic unit will be stationary or rotating at a slow speed when transmitting full torque. Are there any arrangements made to maintain the lubrication?

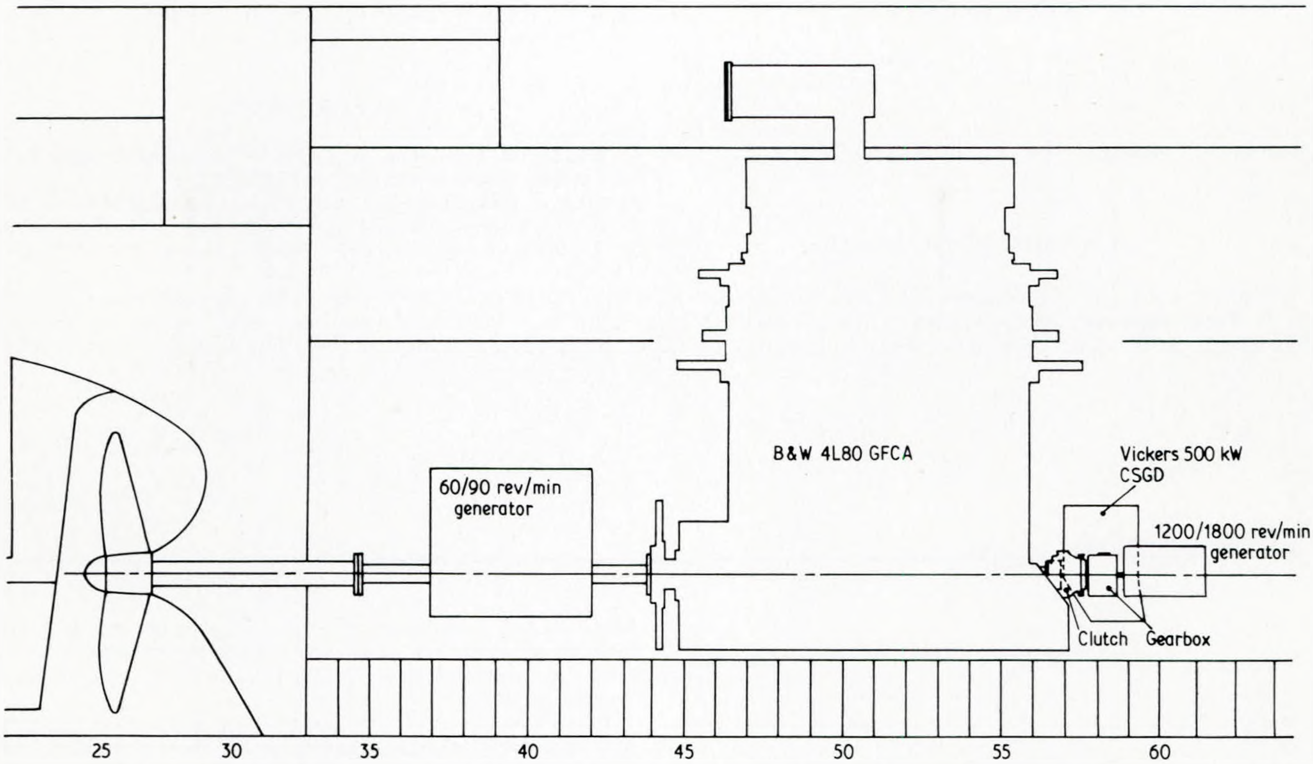


FIG D3 Assembly layout: the CSGD unit is shown superimposed on a typical arrangement

J. B. OLIVER (Marine Manager, Siemens (UK) Limited): We feel it would be of general interest to hear Mr Pringle's reply to the following questions.

First, can this gearbox shaft-driven alternator operate in parallel with auxiliary diesel engines over long periods? What are the droop characteristics?

Second, how does the system react when severe load comes on? That is, could the transient response time cause reverse power on the gearbox, if the control system does not recover as fast as the diesel?

As a comment on the paper, we would point out that interchangeability between 720 rev/min machines is not normally possible, as the gearbox-driven version has a two-bearing foot-mounted arrangement but auxiliary engine sets normally have B16 construction alternators.

P. COTTEE (Positive Infinitely Variable (Transmissions) Limited): Mr Pringle has referred to the fact that Vickers Shipbuilding and Engineering Ltd have developed a power take-off (PTO) system that is not available currently from any other source, and that all other PTO systems in current use are completely electric in their operation. Thus I would like to draw his attention to the PIV RH regulator.

PIV have been engaged in the manufacture of power-transmission equipment and infinitely variable speed gearboxes, with epicyclic combinations, for more than 50 years. The PIV RH regulator controls the variable speed of the main engine and converts it to a constant speed ($\pm 1\%$) for driving an alternator directly off the main engine.

Our present programme includes the production of a range of gearboxes, type RH, which can accept an infinitely variable input speed from a ship's main engine at a ratio of 1.2:1 up to 4:1, with a constant output power capacity range of 5–520 kW. However, we are extending the capacity range up to approximately 700–1000 kW.

The output speed is held constant ($\pm 1\%$) by either a centrifugal governor or an electronic controller via the hydraulic servo system. This low-cost, compact PTO/shaft-driven alternator unit has operated satisfactorily over the last 15 years in a variety of vessels including coasters, ferries, fishing boats, tugs, dredgers, service craft, etc.

Figure D4 illustrates the principle of operation. A rocker pin chain runs between two pairs of conical discs. One of each pair of discs can be moved axially, the other remaining stationary.

An infinitely variable speed change is effected by opposite axial displacement of the movable discs, forcing the chain to a different set of running radii. Power transmission between the spherical faces of the rocker pins and the smooth surface of the conical discs relies on lubricated metal-to-metal traction.

For transmission of power, axial squeeze is produced between the chain and conical discs by a torque-dependent cam arrangement which transforms the torsional moment into axial force. The axial load is carried to the chain via pressure roller and movable disc on one side, and fixed-disc supporting roller and shaft on the other.

By this means the axial squeeze load is supported exclusively inside the rotating shaft disc assembly without loading the housing bearings. When running idle, a compression spring provides adequate contact pressure.

The output speed is controlled by a hydraulic servo system. A pump incorporated into the drive feeds the oil to the hydraulic control device.

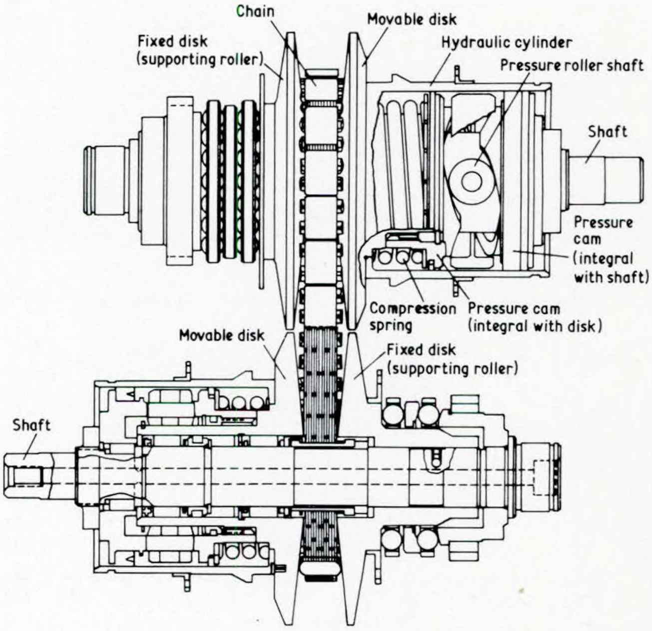


FIG D4 RH regulator: principle of operation

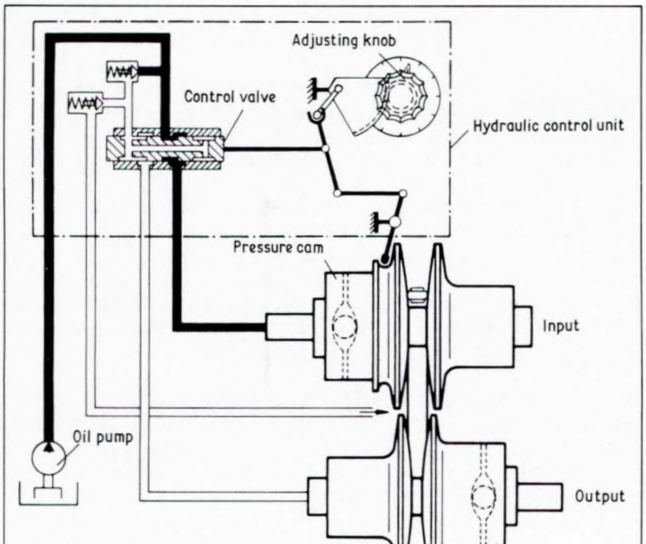
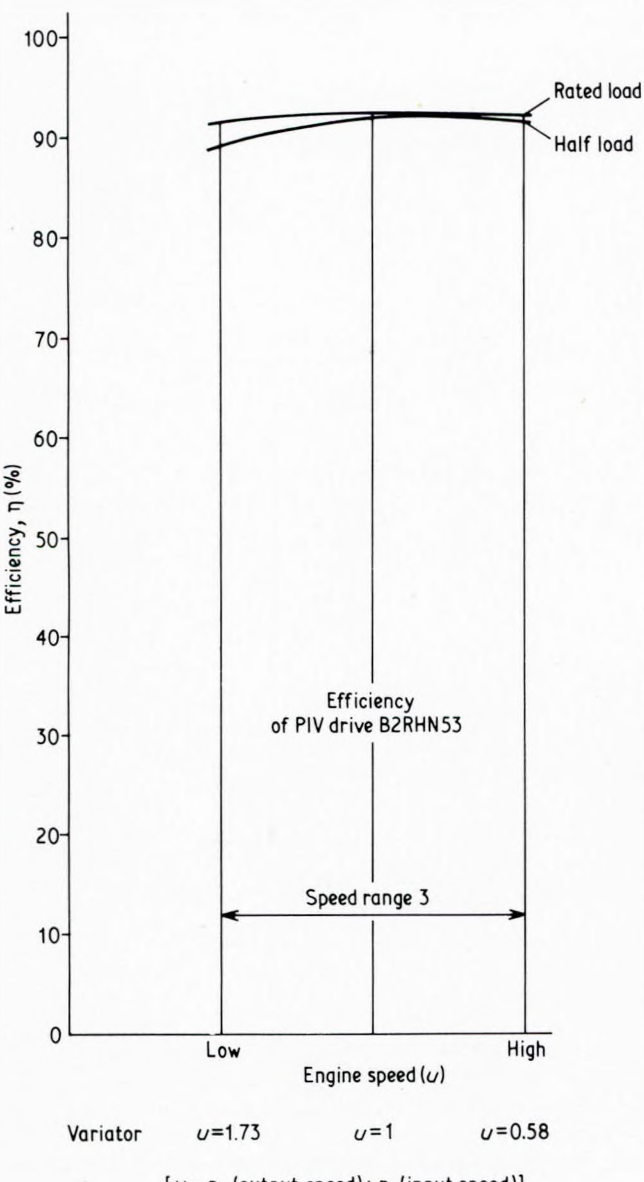


FIG D5 Hydraulic servo system for marine applications

FIG D6 Variation of efficiency with load



The control valve regulates the oil into the rotating cylinders of the shaft/disc assemblies. The hydraulic support pressure provides an additional axial load and controls the axial displacement of the movable discs — i.e. the ratio setting between the drive and the driven shaft/disc assemblies and, hence, the output speed.

Figure D5 illustrates the hydraulic servo system for marine applications. The manual adjusting knob is replaced by either a mechanical governor, driven from the output shaft, or an electronic controller, both of which have mechanical linkage to the control valve, thus giving automatic control of the output speed.

Due to the rather low idling power, the RH regulator operates with a high efficiency of between 88% and 94%, depending upon what auxiliary gearing is necessary to fulfil customer's requirements. This high efficiency declines insignificantly under load conditions (Fig. D6).

Auxiliary gearing contained within the housing of the RH regulator is used to:

(a) Step up the input speed (helical gearing);
(b) Step down the output speed (helical gearing);
(c) As a power bypass for the PIV (epicyclic gearing).

This is currently used when the output power is in excess of 150 kW. However, it is hoped to dispense with the necessity for epicyclic gearing in the power range up to 300/400 kW, with the extension to our product range.

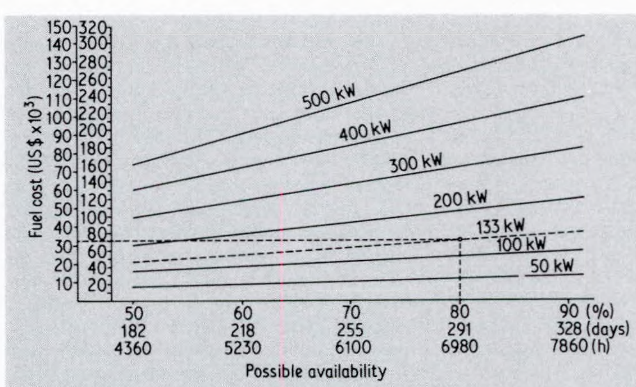
We can supply individual regulator units or an equipment package in which the PIV regulator, alternator, couplings and connections are all mounted on a common base plate.

The major considerations in fitting a shaft-driven alternator system are (Fig. D7):

(a) The potential savings in fuel costs.
(b) The only running costs incurred by a PIV regulator are those of an oil change and clean filters every 2000 running hours. This represents a considerable saving on lube oil costs for an auxiliary diesel.
(c) Very little maintenance has been required on the regulator gears (over 100) sold over the last 15 years to the marine industry. This also applies to the thousands of gears sold to industry for machine drives.
(d) The capital cost of most variable systems appears to be two or three times that of an auxiliary diesel set. However, the cost of the PIV RH regulator system is compatible with that of an auxiliary diesel set and often cheaper.
(e) The payback time for the larger units is approximately one year, depending upon the percentage loading; for small units, it is less than one year.

Some of the typical system tests which we can demonstrate to shipowners/builders are as follows:

1. The PIV regulator driven from a diesel engine at variable speeds and maintaining a constant output speed of ±1%; or driving an alternator and maintaining the frequency at 50/60 Hz ± 1%.
2. Introducing large changes in engine rev/min over varying time periods and maintaining constant output speed ±1%.



	Shaft-driven alternator (A)	Diesel generator (B)
Specific fuel consumption (g/kWh)	198	212
Fuel cost (US$/t)	147	347
Fuel	Heavy (380 cSt)	Light

FIG D7 Fuel cost: comparison between diesel generator and shaft-driven generator

3. Cyclic changes in engine rev/min with the facility to vary the amplitude and time period of the cycle and maintaining a constant output speed ±1%.
4. Fluctuating the load on the output of the PIV and maintaining a constant output speed ±1%.
5. Combination of above efficiency check of the PIV by measuring input and output powers.

Test results can be recorded in print-out form by our electronic monitoring equipment.

H. WOODS (Shell International Marine Limited): Over recent years there has been an inducement, owing to increased fuel prices, to reduce the overall cost of production of auxiliary power on board ship. This initially resulted in the use of exhaust gas waste heat recovery for production of steam with the subsequent use in a turbogenerator: generally a very satisfactory arrangement.

Where sufficient heat was not relatively easily recoverable, or other uses were required for the steam, the shaft generator has become the economic alternative for auxiliary power production. The CSGD system appears as a welcome contender to the thyristor-controlled and eddy-current coupling-driven shaft generator systems, as well as the shaft generator incorporated with a controllable-pitch propeller.

The paper, rather unfairly I think, summarily dismisses the use of turbogenerators. Although the amount of recoverable heat from exhaust gases may be diminishing owing to more efficient engines coming into use, efforts are being made to extract heat for steam cycle purposes from, for example, jacket water and charge air. The capital investment to provide more sophisticated heat-extraction equipment will be quite high but since the heat and the power are obtained at very low cost, it would seem reasonable to use it.

Even with efficient heat-recovery equipment, for many ships there will not at all times be enough power produced from the waste heat system, either owing to the low installed main engine power or the ship's operating pattern, and the auxiliary power requirement will have to be supplemented. I feel a reasonable solution to such a problem is the parallel use of a generator and shaft generator and I should like to know Mr Pringle's opinion on the use of the CSGD system in this mode.

E. J. BANNISTER (Shell International Marine Limited): Ships are now frequently called upon to run at reduced power and on any single voyage they may operate at a fixed speed for many days at a time. Therefore, I should also like to know whether the proposed gear installation is capable of operating for prolonged periods at the reference speed. At this speed, the epicyclic gear connected to hydraulic unit 2 will be stationary but carrying the reaction torque from the main gear. Does the author consider that the gear teeth and bearings in the former epicyclic have adequate design and lubrication margin for the stationary condition, or will there be a barred speed range?

On first glance, the installation appears to have an excessive number of gear trains compared to the basic requirements: for instance, the gears connected to hydraulic units 1 and 2, and that between the main epicyclic and the generator. No doubt there were good reasons for this. Would the author, please, elaborate them?

The larger, free-standing unit has obvious attractions from the point of view of the manufacturer, in that he can produce a complete package for subsequent installation in a ship, with minimum interaction between other designers. The free-standing concept requires the movements relative to the main engine to be taken into account and these appear under the heading of 'Main engine reactions'. However, this assumes that both units are mounted on a base that is completely rigid. Ship's tank-tops are not so strong as we are sometimes led to believe, particularly when it comes to resistance to vibration. I wonder whether the author has taken this into account, either by specifying the rigidity of the tank-top to the shipyard, or some additional flexibility in the coupling system? It may be unwise to assume that the additional requirement can be handled by a margin or safety factor in the design as stated in the paper, without first quantifying the potential problem.

Finally, I should like to ask the author what state the development has reached now. Assuming design work has been completed, what testing programme is envisaged and on what time-scale? Will a prototype be tested onboard ship, and when will a fully developed and guaranteed unit be offered on the market?

B. L. NORMAN (Texaco Limited): I congratulate the author on a very interesting paper, giving us all much food for thought.

Texaco, as a major oil supplier, was requested by Mr Pringle to investigate the possibilities of using the main-engine crankcase lubricating-oil system of large, slow-speed engines such as Sulzer and B & W to lubricate the CSGD gearbox, thus saving in extra equipment costs.

His particular interest and concern was the size of the carbon particles held in suspension in the lubricating oil, as a 25 micron filter is fitted in the gearbox oil system to protect the bearings.

We are still gathering information from used samples of Texaco Doro AR 30 oil, with a TBN of 6 when new, from the above-mentioned makes of engines with reference to the pentane-insoluble level and size of carbon particles. At present, it looks hopeful; with varying levels of pentane insolubles, the majority of carbon particles in suspension are below 5 microns in size. It is possible that a greater hazard would be from water contamination of the oil on the bearings and gearing; but this should be kept to a minimum by good housekeeping and regular purification by centrifuge.

R. G. BODDIE CEng, FIMarE: Designers have had to decrease the mechanical clearances in hydraulic pumps and control systems to cope with the increased hydraulic pressures in use today. Manufacturers seem reluctant to specify the minimum cleanliness standard for the oil to be used in their equipment and rely on the system filter to remove the

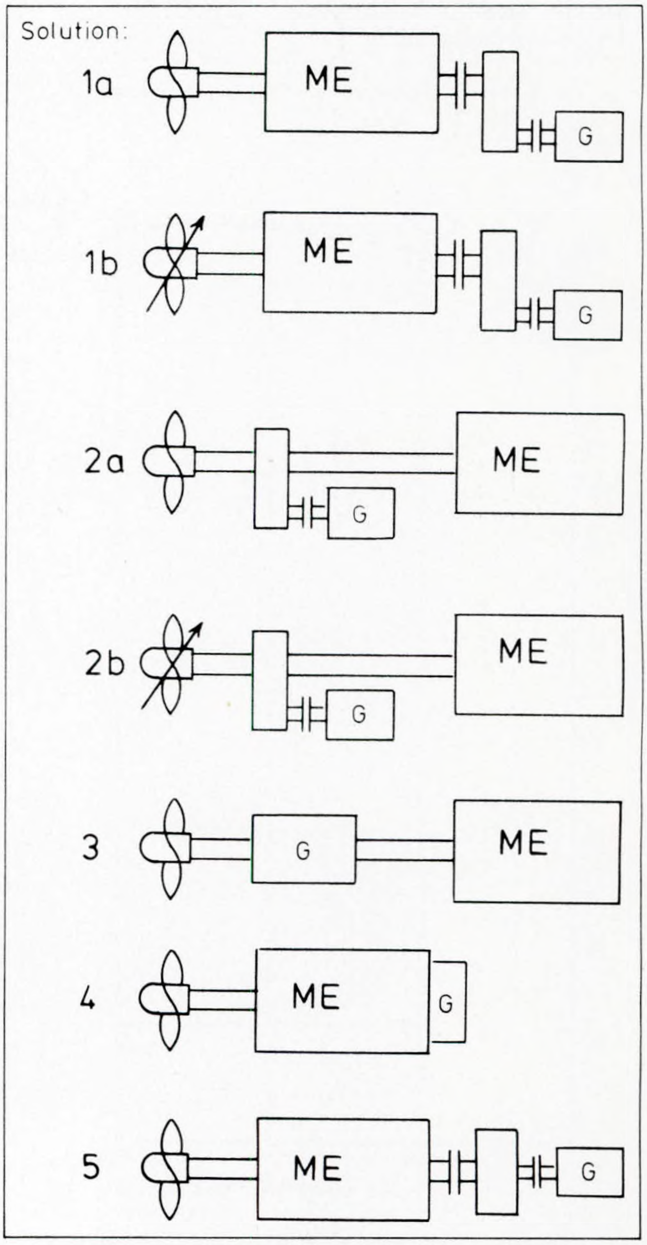


FIG D8 Alternative positionings for main engine driven generators

contamination. However, in a new or overhauled system, the damage may be done before the system oil is filtered down to the required standard of cleanliness. Unless the requirements for flushing and cleanliness are laid down and enforced, hydraulic systems will get a bad name for reliability.

Can the author please provide information on:

1. The cleanliness (in microns) of the oil that the hydraulic pumps and the control valves are designed to operate with, without causing excessive wear.
2. Guidance on flushing the hydraulic system after installation and major repairs, and whether the flushing oil should be renewed or left in the system on completion of flushing.
3. The minimum cleanliness of the oil to be used for filling and topping up the system.
4. The mean running time between overhauls of the hydraulic system.

G. MIKKELSEN and M. BRENDORP (B & W Diesel A/S): During the last 18 months we have conducted an investigation which forms the basis for a survey on the various possibilities of main engine-driven generators, including a solution using epicyclic variable-ratio (EVR) gears. Based on the information received from manufacturers of gears, couplings and generators, etc., we have been able to compare installation length, overall efficiency and investment costs for different generator arrangements, with five plant sizes ranging in electrical output from 300 to 1400 kW.

In our comparison we considered the following alternatives (Fig. D8):

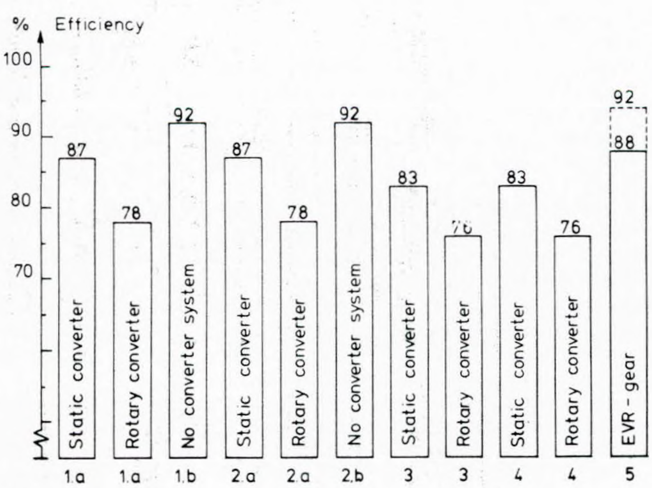
1. Generator and step-up gear in front of the main engine.
2. Generator and step-up gear positioned on the intermediate shaft.
3. Generator with electrical poles mounted directly on the intermediate shaft.
4. Generator directly mounted on the fore-end of the main engine (possible on most B & W two-stroke engines).
5. Generator driven through an EVR gear positioned in front of the main engine.

With the exception of solution (5), we have considered generator arrangements applicable to both static and rotary frequency convertors; as well as a system without a frequency convertor, where the main engine, due to the installation of a CP-propeller, is able to operate at constant revolutions.

Table DII: Comparison of engine room lengths required for different generator arrangements (520 kW)

GENERATOR ARRANGEMENT	ENGINE ROOM LENGTH (m)[a]	DIFFERENCE
Solution 1a: Gear + generator in front of engine	25.2	3.5
Solution 2a: Gear + generator on intermediate shaft	24.5	2.8
Solution 3: Integral generator on intermediate shaft	25.2	3.5
Solution 4: Integral generator on fore end of engine	22.4	0.7
Solution 5: EVR-gear on fore end of engine	23.8	2.1
Propulsion plant without generator	21.7	Basis

[a] Total length of engine, providing this has already been kept to a minimum, i.e. the distance from aft bulkhead to engine should provide space for withdrawal of the propeller shaft.



Note

The efficiencies include losses in step-up gear (if applied), generator and frequency convertor (if applied).

Solutions (1.b) and (2.b) do not apply any frequency convertor, hence the high efficiencies. However, as they do apply a CP-propeller with constant revolutions, the propulsive efficiency is lower than for the other generator plants, which apply fixed-pitch propellers. The efficiency of solution (5) depends on the degree of hydraulic compensation.

FIG D9 Efficiencies of alternative generator systems

Result of comparisons

Table DII gives the total engine room lengths required for different generator arrangements. As can be seen from the figures given for the EVR-gear, this solution offers greater economies on space than more conventional arrangements.

The EVR-gear, in this case, is an extremely compact design from Vickers Shipbuilding and Engineering Ltd (VSEL) (Barclay Curle Ltd). The particular unit under consideration for this comparison is the 520 kW output unit.

Efficiencies

The overall efficiencies for the different generator systems are shown in Fig. D9. It can be seen that the efficiency of the EVR-gear is particularly good when compared with systems using frequency convertors. The figures given are mean values based on the five plant sizes requested.

Investments

After making enquiries at various manufacturers, we were able to make a rough comparison of the price levels for the different generator arrangements.

We have not taken into consideration the installation cost, but only the initial cost of the components. In some cases, however, the installation costs constitute a considerable part of the total price, for which reason they should not be ignored. The prices for EVR-gears are approximately on the same level as for generator plants using static frequency convertors. Table DIII shows the prices for different generator arrangements, as well as for a diesel generator set.

The question of whether investment in a generator plant will be profitable or not depends on several factors: for example, the power requirement, utilization time, price, possible substitution of a diesel generator set, saving in operating costs, etc. The saving in operating costs is perhaps the prime factor in respect of the profitablility of a generator plant.

Comparisons will generally be made with a conventional diesel generator set, for which the running costs are mainly dependent on the type of fuel oil on which the engine is run.

The fuel oil used today on various makes of auxiliary engines will normally be an intermediate fuel, in many cases blended on board. The

Table DIII: Relative cost of components for different generator arrangements[a]

GENERATOR ARRANGEMENT	GENERATOR ELECTRICAL OUTPUT *520 kW*	*700 kW*	*1400 kW*
Solution 1a	169	186	260
Solution 1b	50	64	110
Solution 2a	219	248	306
Solution 2b	100	126	156
Solution 3	191	211	275
Solution 4	180	197	270
Solution 5	195	201	352
Diesel generator set	67	89	165

[a] Installation cost not included.

blending ratio of MDO to MFO varies between owners, and between engine makes.

Viscosity is often used as a measure of the fuel oil's quality, despite the fact that it is usually the chemical constituents which limit the use of a particular quality of fuel in a specific engine installation. Future developments in engine technology and fuel oil treatment, together with the policies followed by shipowners, are considered to be the most important factors in the prevalence of main engine-driven generator systems.

Conclusion

As is apparent from this contribution, and from Mr Pringle's paper, we have been investigating the driving of the generator by means of EVR-gears, in co-operation with, amongst others, VSEL (Barclay Curle).

It is evident from the previous paragraphs that the EVR-gear system (or CSGD, to use Vickers' terminology), is a competitive system for generating electrical power.

We consider the EVR-gear to be a particular good system for generating power and, as mentioned by Mr Pringle, we have, besides the free-standing CSGD unit (Fig. D10), also co-operated with some manufacturers of EVR-gears for an integral system, especially with regard to our smallest two-stroke engines, L35MC/MCE (Fig. D11).

A question often raised is that of results, especially for a new product such as the EVR-gears. To our knowledge, however, several EVR-gears are today running without problems on stationary applications, though with the reverse function, i.e. constant input revolutions, and variable output revolutions.

We are therefore looking forward to the results of the tests Vickers plan to run this autumn on their CSGD.

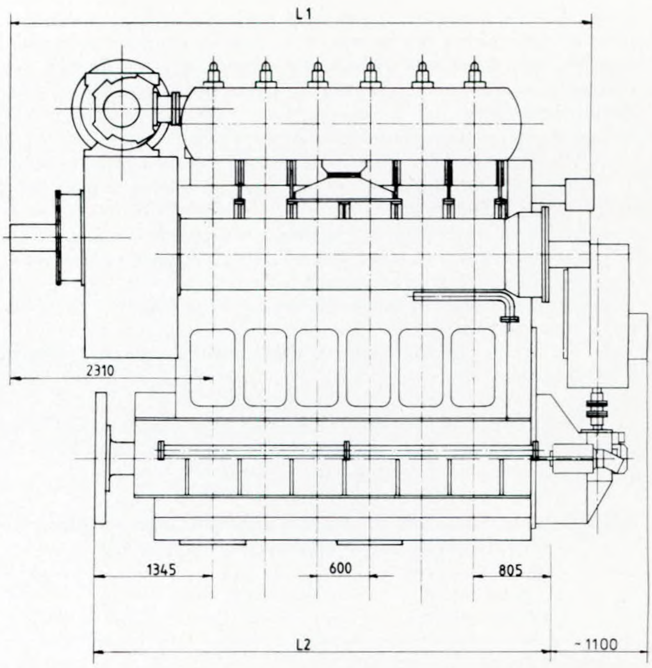


FIG D11 Two-stroke engine (L35MC/MCE) equipped with an EVR-gear and vertically positioned 350 kW generator

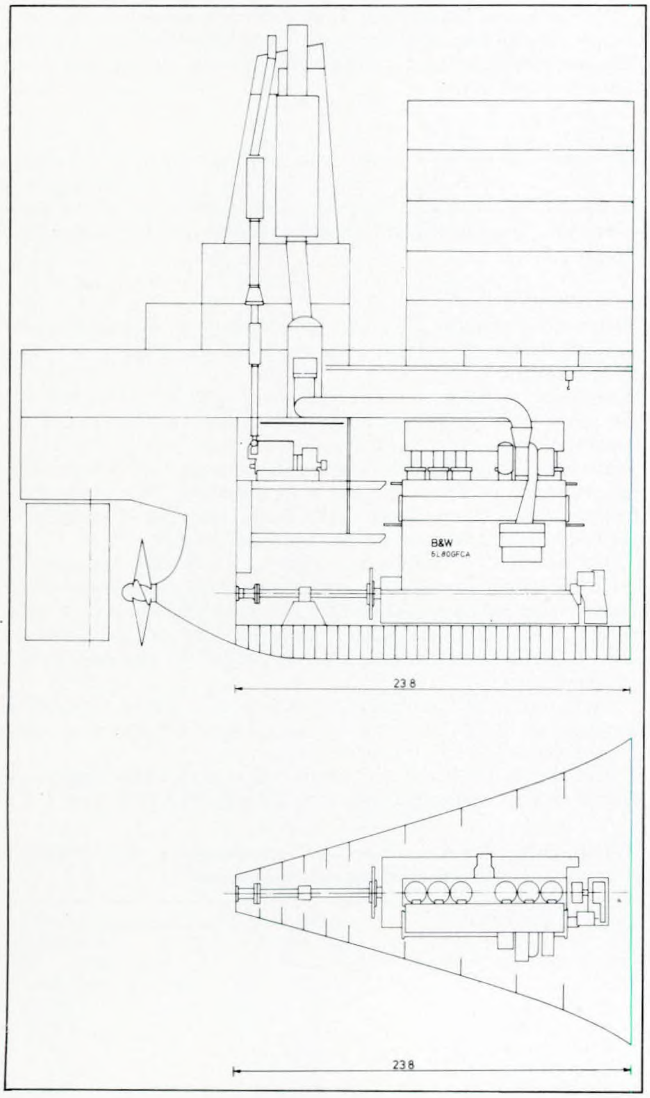


FIG D10 Propulsion plant equipped with a main engine driven generator system as a free-standing unit (consisting of EVR-gear, generator, etc.) located in front of the engine

Author's Reply

Mr Craig's contribution is a valuable addition to the paper. Its comment on the financial advantages of the CSGD concept will be of particular interest to the economy-minded shipowner.

With regard to lubrication at the reference speed, no problem is envisaged. In the control epicyclic, while the sunwheel is stationary, there is still continuous relative rotation between the three elements. Hydrodynamic lubrication therefore results, with lubricant being supplied to the gear meshes and planet pinion bearings. The sunwheel of the control epicyclic is held stationary by the fixed-displacement hydraulic unit, through its associated planetary epicyclic gear unit. This epicyclic unit is therefore totally without relative rotation at reference speed.

Lubrication of the gear teeth is maintained by the forced-feed lubrication sprayers incorporated into the planet carrier; and the planet pinion bearings are also pressure-fed through the flexible-pin assemblies. The torque being applied to this gear train under static conditions is a small fraction of the transmitted torque. The gear tooth and planet pinion loads are consequently small and insufficient to cause a problem. The flexible pins effectively provide flexibility to cushion such loads at the gear mesh and planet pinion bearings.

Such arrangements are already in service and have provided problem-free operating experience to date.

It will be recognized also that, in practice, the prime mover will always operate with a degree of speed variation under its governor control. Some relative movement will be imparted continuously to the epicyclic gear train associated with the fixed-displacement hydraulic unit.

Mr Oliver's questions are very pertinent. The CSGD unit can be operated in parallel with either turbo- or diesel-alternators continuously and the control equipment is designed to facilitate this.

The droop feature will be adjustable to permit the more economic CSGD unit to take the maximum load, while the balance of demand can be taken by the diesel-driven unit.

However, it is our intention to market the CSGD as a unit designed to carry continuously the maximum sea-load and, unless an emergency occurs, there should be no need for continuous parallel operation at sea.

With the CSGD in operation, any change in electrical load is insignificant against the total output of the prime mover. The CSGD alternator will not perceptibly change speed with a suddenly applied load. No problem is anticipated with the effects of transient response time. The control system has been designed to obviate this. Also, as engineers who do not believe everything, we have designed a substantial margin into our rotating element system to cancel out the stress effects of reverse torque.

Mr Cottee's contribution, being not directly concerned with merchant shipping in the range we have been considering, is largely irrelevant to the subject matter of this paper. I would recommend that when his design has been developed to cover the range we have been discussing, he should come back to the Institute with a paper on the subject.

I must apologize to Mr Woods; it was not my intention to dismiss summarily the use of turbogenerators. Rather, my principal aim was to highlight the increasing reduction in available waste heat due to improvements in engine design. Mr Craig's contribution deals also with this point and confirms my views on this subject.

There is no objection to the shipowner installing a combined system of turbogenerator with a constant-speed generator for 'top-up' purposes when increased power is required. It is a question of assessing the economics of such an arrangement. One would feel instinctively that the high capital investment associated with such a combined system might make it unattractive.

Mr Bannister's first point, regarding lubrication at the reference speed, has been answered in the reply to Mr Craig's comments.

With regard to the excessive use of gear elements, the gear trains are necessary to match the prime mover's speed range to the selected alternator speed. The total gear ratios will vary between 18:1 and 30:1 and therefore two trains of gears are required. Gear trains are necessary also to match the operating speeds of the standard hydraulic pump units used in the speed-control system.

It could be argued that the final train of gears to the alternator are surplus, since they are provided as a means of siting the alternator within the length of the gearbox. Without such an arrangement, the length of the unit would be increased by the length of the alternator, with subsequent and unacceptable penalties on engine room length.

It is agreed that the free-standing unit would be influenced by alignment and hull-stiffness problems. For this reason, we have incorporated a torsional flexible coupling which, in addition to vibration isolation, has capacity for accepting axial, angular and parallel misalignments.

I should like to thank Mr Norman for his contribution and for the assistance his company has given.

As mentioned, our early hope was to use the main engine lubrication system to serve the CSGD gearbox. The facts produced by Mr Norman indicated clearly that the only practicable engineering solution was an independent lubrication system. That is what we have standardized on.

Mr Boddie's comments on cleanliness are timely. A significant percentage of machinery problems originate from a lack of thorough cleaning prior to going into service, whether from new or after overhaul. It is agreed that the high pressures and fine clearances in hydraulic systems require a higher degree of attention. Our current production involves my company in the application of CHARN standards for hydraulic oil systems and we shall apply the same level of quality control on all CSGD oil systems. A filtration limit of 10 microns has been set by the designers of the hydrualic pumps and controls to guarantee reliable service.

Guidance on flushing and cleaning of the system will form part of the instruction book provided with the unit. I would not recommend retention of flushing oil in the system. It is a better practice to renew the oil charge and consider replacement of the filter elements after flushing. Any fittings, etc. dismantled during overhaul should have openings capped immediately. It is easier to stop ingress of dirt by this action than to remove it from the system later. In all cases the maintenance of system cleanliness means following strictly a few simple rules.

The recommendations given by Mr M. J. Fisher in his paper entitled 'Good design practice for oil hydraulic systems', read at the Institute earlier this year, deal thoroughly with this subject and should be followed by anyone involved in hydraulic system design or operation.

The comments of Mr Mikkelsen and Mr Brendorp from B & W cover a wide range of alternatives and indicate the depth of the investigations carried out by them into power take-off systems.

While I do not agree with all of their conclusions on comparative costs, it is gratifying to note their confirmation that the CSGD concept is a competitive system for the generation of electrical power at sea.

On reliability of operation, we are pleased to have their concurrence that the epicyclic variable-ratio system, albeit in reverse application, has been operating satisfactorily in land applications for a considerable period.

Conclusion

I should like to finish by thanking the many people and companies who have assisted in this development — some by asking awkward questions, some by offering design comments. As I mentioned in my talk, while the initial development was in co-operation with B & W, discussions with Sulzer Bros. and SEMT–Pielstick have produced valuable proposals which are being developed.

The intention is to have a range of designs and outputs to satisfy most demands.

With regard to a prototype, the company decided that an up-to-date approach was called for and have prepared a mathematical model of the concept. A programme has been prepared, incorporating a range of input and output variables, and is at present being entered into the VSEL computer. The preliminary results validate our calculations.

Mention must also be made of my colleagues throughout British Shipbuilders for the assistance and encouragement they have given during the past 12 months. Their constructive criticism and recommendations have proved invaluable during this development period.