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COMPACT COAL-FIRED POWER PLANT FOR MARINE APPLICATION

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Compact Coal-fired Power Plant for Marine Application

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SYNOPSIS

Most recent coal fired ships have used the well established spreader stoker fired water-tube boiler providing steam to a steam turbine. However, there are difficulties that make it unsatisfactory for use in a wide range of short sea vessels, since their architecture excludes the use of the very large water tube boiler and their requirement for good manoeuvring demands much better boiler control than can be achieved with the spreader stoker. These ships need not carry large bunkers and can benefit from using cheap coal as a fuel. The author reviews present coal fired ships and the possible short sea trading routes, establishing the need for compact coal fired power plant. Details are given of the proposed gas producer combustion system and its performance, together with outline designs for compact shell boilers using the system and capable of giving optimal steam conditions suited to the proposed turbo-recompressed medium speed steam engines for main propulsion and auxiliary use. The thermodynamic cycle is detailed, together with a complete thermal balance for the whole plant. It is anticipated that the overall thermal efficiency will approach 30% and that the engine will produce slightly more power for a given size than its turbocharged four-stroke diesel equivalent. The computed dynamic response of the boiler-engine package is discussed. Finally, details will be given of typical machinery installation on board vessels, together with projected economies in their respective trades. The proposed plant offers substantially lower fuel costs whilst the total installed capital cost is in line with that of the heavy fuel burning diesel engine of the same performance.

INTRODUCTION

The energy crisis of 1976 brought the question of the fuel costs of modern ships sharply into focus. As a result, attention turned to the development of more efficient diesel engines burning heavy fuel and to alternative energy sources, especially coal. A number of coal-fired ships were built, some of which have been trading successfully for about 3 years.

The technology of these new ships is based on the use of stoker fired water-tube boilers and steam turbines which, although successful, are large and restrictive in operation. Ways of overcoming these restrictions include the use of fluid bed systems, coal-fired gas turbines and producer gas fuelled diesel engines, as well as my proposal for very compact coal-fired plant using the gas producer combustion system in a highly rated shell boiler and modern reciprocating steam motor with turbo-recompression.

Details of the power plant and the underlying economic and technical reasoning for this design will be given, together with installation details on three vessels ranging from 3000 to 24 000 shp.

Despite the current temporary lowering of the relative price of oil and considerable improvements in the overall efficiency of diesel plant, the use of coal as a marine fuel will be increasingly advantageous, especially for vessels plying the 'short sea' trades, so long as the increase in capital cost of the coal-burning ship can be kept to a minimum.

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He has a special interest in dynamics, systems analysis and control as well as machine design, energy economics and thermodynamics, having obtained a PhD from Cambridge University for his work on optimal design. He has been involved in studies of coal burning power plants for marine, industrial and traction applications since 1976 and has given invited papers on the subject in the UK, USA, China, Europe, Scandinavia and Australia. In 1981 he was awarded a Royal Society Industrial Fellowship for his work in this field.

THE ENERGY CRISIS

As a direct result of the energy crisis, which triggered the fuel price escalation, the modern coal-fired ship evolved from concept into reality and is successfully trading in the Australian bauxite trade.

Before considering in detail the economics and technology of coal-fired ships it is worth looking at the general context of world energy utilization during the next 20–25 years, to examine the underlying reason for turning back to coal as a marine fuel. The many predictions about the demand and supply of liquid fuels and their quality are far too familiar to repeat. In general they all add up to a consensus that there will be insufficient liquid fuel of the correct quality before the end of the century, certainly within the lifetime of any new ship being considered now. Shipping and railways are the only forms of transport that can conveniently burn coal directly as a fuel and it is therefore important that this should be done.

Although considerable interest has been shown in the use of wind, solar and even wave power for ship propulsion, it is unlikely that any real commercial benefit for the normal commercial ship owner will accrue from these. Wind-assisted ships may be introduced during the next few years, especially for coastal and fishing duties, and it is possible that some proposals for sailing ships for particular trades will emerge. The wind-assisted ships that have been built have shown fuel savings of some 10–15%. However, given present building

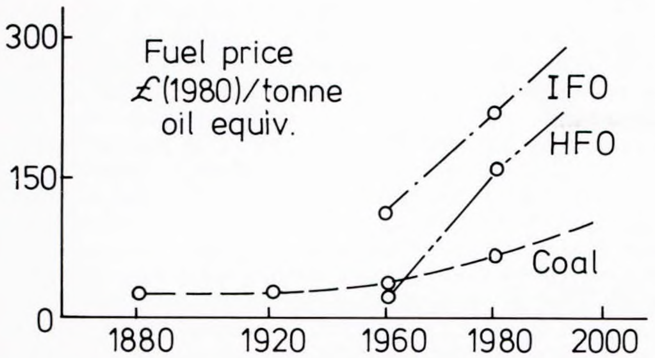
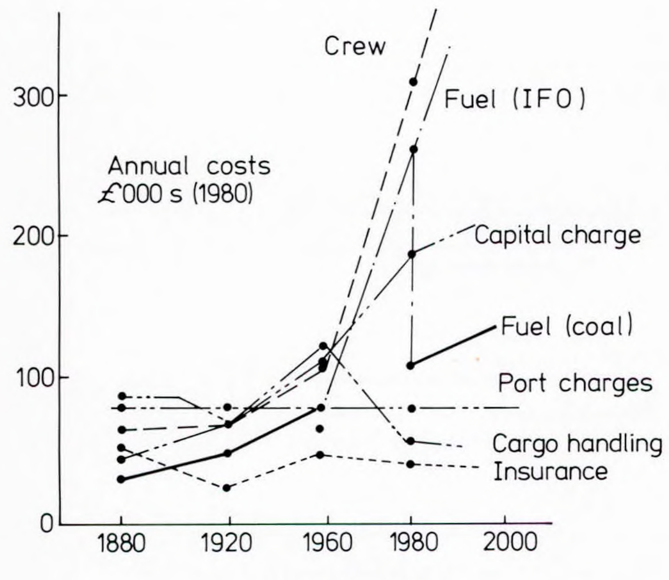
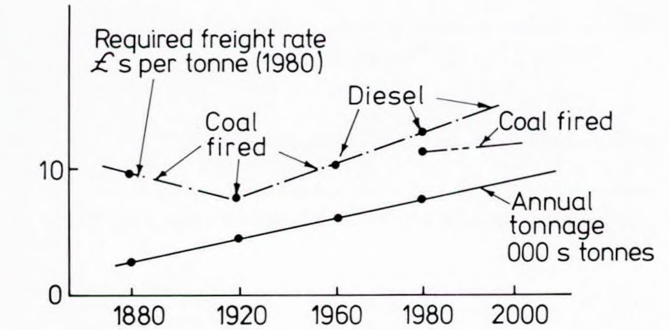


FIG. 1 European fuel prices, 1880–2000, in £(1980) per tonne of oil equivalent for HFO, IFO and coal



(a) Annual costs



(b) Required freight rate

FIG. 2 Economics of 3000 dwt vessel, 1880–2000

costs, it is unlikely that their utilization would be particularly widespread.

More exotic fuels such as hydrogen are too expensive for commercial use, although it is possible that some coastal ships may burn compressed or liquefied gas, where this is available as a by-product of a land-based plant.

This process of elimination of the alternatives to oil as a marine fuel leaves coal, either in a solid form or as a feedstock for liquid fuel derivatives, as the best future fuel source. The development of liquid fuels derived from coal looked an attractive possibility when work began on the development of new manufacturing processes. However, recent realization of the energy costs and the capital costs involved and the impact of the world recession have dramatically reduced development plans, so that it is unlikely that coal-derived fuels will be available to shipowners within the medium time period. Even when it is available the cost, relative to raw coal, will be higher than today's liquid fuels. The use of lump coal appears to be the best practical alternative fuel available, certainly for the next 20–25 years.[1] It is abundant, well distributed around the world and therefore politically less sensitive than oil. Because of the lack of the 'vertical' integration shown by the multinational oil companies, coal is still traded as a commodity in such a way that its price follows that of world inflation. Coal can of course be used directly, with only a minimum of processing.

OPERATIONAL REQUIREMENTS

The attractiveness of coal as a fuel is very dependent on the type of vessel and the trade in question, as well as its availability and relative cost (see Fig. 1). The capital cost of the technology chosen is an important factor in deciding the economics of the vessel, as charter rates are fixed by market forces.

The most important influence on a vessel and its economics is the greater mass and volume required for coal bunkers. Whereas oil may be stored in double bottom tanks, coal must be held in correctly designed hopper bunkers which will allow correct mass flow and avoid any retention of coal in the bunkers.

The transfer of coal automatically to the boilers by dense phase pneumatics, in which a slug of coal particles is pushed by high pressure air along a suitable pipeline,[2] has been very successful, both on land installations and in the new Australian ships.

For the bulk trades in coal, ore, grain etc., where freight rates are low, the longer the voyage between bunkering stations, the larger the coal-fired ship needs to be economic over a diesel ship. For example, on the coal or ore trade between Australia and Japan, a ship of 100 000 dead weight tons (dwt) would be economic, whereas for the similar trade from Australia to Europe the ship should be 175 000 dwt. The present Australian ships operating on a 3000-mile round coastal trade in North Queensland are of 75 000 dwt.

In Europe much of the shipping is in relatively small vessels, or roll-on/roll-off ferries, operating in the short sea or coastal trades. In coastal vessels, the low engine power and short voyage require small bunkers which result in very little, if any, loss of cargo volume. The small engines in these ships and the need to take fuel at small ports mean that these vessels frequently burn higher priced marine diesel fuel. This makes these vessels good candidates for coal firing with a potential reduction in the fuel bill of 60%.

Of course fuel is only one of many costs to the shipowner. It is interesting to note that for the same size ship most costs have stayed fairly constant except that of the crew and fuel. Figure 2 shows the trends over the past 100 years for a ship of 3000 tonnes burden.[3] If such a ship were now to use a compact coal-fired reciprocating power plant, the fuel costs would be cut significantly and, allowing for the marginal increases in capital costs etc., the required freight rate for a typical voyage of 1400 miles would be reduced by some 12%, sufficient to bring the ship into profit.

In the case of the ro-ro ferry, the fuel costs are a much higher proportion of operating costs because of the high powers required to maintain the exacting schedules essential to provide adequate commercial use of the vessel. As with the small coastal vessel, the naval architecture of the ro-ro ship demands a very compact engine room with low headroom. However, the potential saving by burning coal on a typical ferry or bulk carrier of 15–20 000 shp is of the order of $1M per year, based on the current price of coal and heavy fuel oil as reflected in Rotterdam.

Such are the clear economic arguments for coal firing of ships, principally bulk carriers and ships for coastal and 'short sea' trades; but what of the technologies that are being used or considered?

EXISTING COAL BURNING SHIPS

Before considering the new generation of coal-fired ships, it is worth remembering that there are a number of existing ships still operating in various parts of the world, notably China, India and on the Great Lakes in North America, as well as preserved vessels in Scandinavia and elsewhere. All of the vessels are relatively small and use the 'Scotch' boiler. The furnaces generally have manual firing or mechanized firing under manual control. Despite the large crews required, these ships are still economically viable because of the low fuel cost and ease of maintenance.

FIRST GENERATION NEW COAL-FIRED SHIPS

The first generation of newly built coal-fired ships has been steam turbine bulk carriers ranging from 36 000 dwt to 150 000 dwt, designed for long coastal voyages or for the intercontinental coal or ore trade. Figure 3 shows the comparative outlines of the vessels. Despite a number of different approaches for coal firing having been proposed, all the ships use conventional water-tube boilers with the coal being burnt on a travelling grate which is fed from a mechanical spreader stoker.

A diagrammatic cross-section through one of the two boilers of the Japanese-built 75 000 dwt bulk carriers for the Australian National Line (ANL)[4] is shown in Fig. 4. These ships have recently been joined by a further two, built to the same requirements in Italy, for Australian Bulk Ships.[5] These ships have only one boiler, of the same general design but fitted with an oil-burning facility for use in the event of a breakdown in the coal feeding mechanism and have the main coal bunkers just forward of the machinery space in the normal position of the hold. The ANL ships, however, have the bunkers aft, behind the boilers and above the aft peak.

The third ship design to go into operation was built at Quincy in the USA, to carry coal along the north-eastern seaboard between Hampton Roads and Baltimore and power stations around Boston.[6] This vessel, of 36 000 dwt, is the smallest of the first generation ships. It is fitted with two boilers of the same general design as that shown in Figure 4 which provide steam to a single cylinder turbine. Relatively small bunkers are carried forward of the machinery space and aft of the holds. These are in the form of two rows of hoppers which can discharge on to two moving belt conveyers in the space between the hoppers and the double bottom. The coal is conveyed forward to the fore peak from where it is transferred up to a centrally mounted conveyer carried on a boom. With this the coal may be discharged on to the quay or into barges or, if the need arises, may be transferred into the ship's own bunkers. The machinery space is rather cramped on this vessel, especially the twin boiler installation which gives the impression that there is a definite limit to the size of the ship that can be built economically with stoker-fired water-tube boilers.

The Australian and US ships are all designed to burn power station coal, the Australian ships burning the same coal as the aluminium smelter they serve. The US ships use the same coal as they transport to the New England power stations.

The other vessels being built are rather interesting conversions. In Spain there is a project under way to build two steam turbine bulk carriers with oil and coal firing and two diesel oil tankers with high efficiency, slow speed, two-stroke diesel engines, from oil tankers with high efficiency, slow speed, two-stroke diesel engines and two oil-fired steam turbine tankers of about 150 000 dwt.[7] This first involves building two new bulk carriers fitted with slow speed diesel engines and massive coal bunkers amidships and forward of the machinery space. One of the boilers on each of the tankers will be converted to coal firing. After this the ships will be cut in half, the coal-fired steam machinery being attached to the new bulk carrier, whilst the diesel machinery is attached to the old tanker as shown in Fig. 5. These vessels when converted will operate on the Spanish coal trade from Australia, USA and China and will be very flexible in their operation because of their ability to fire both oil and coal. Enough bunkers will be carried for a round trip from Australia to Europe. (Echoes of Brunel's *Great Eastern* which was designed for the same operation – let us hope it does not suffer the same fate!) When not required for fuel the central bunkers may be used as cargo holds.

The other conversion is of the two El Paso liquid natural gas carriers.[8] Like all liquefied gas carriers, these 100 000 dwt vessels were fitted with steam turbine machinery supplied from boilers fired with either fuel oil or 'boil off' from the gas tanks. After many years of litigation over the design and performance of the gas tanks, these ships have now been purchased for conversion to coal-fired bulk carriers. The cryo-tanks have been removed and the boilers converted to coal firing by fitting

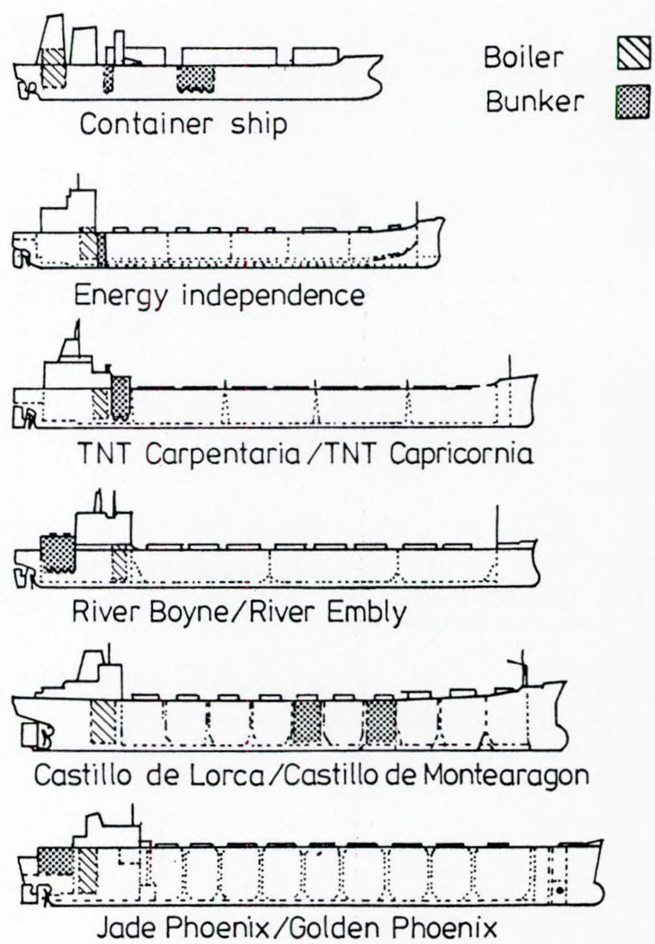


FIG. 3 Comparative profiles of first generation coal-fired ships

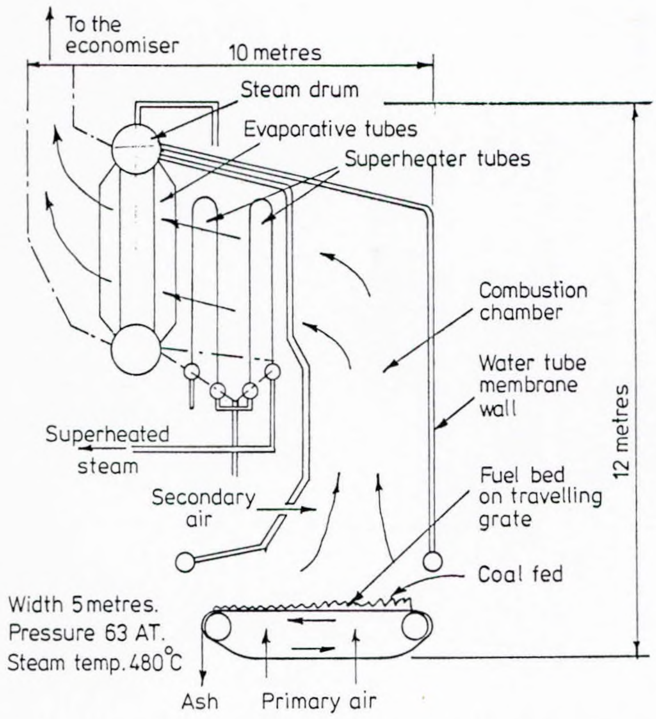


FIG. 4 Diagrammatic section through spreader-stoker fired water-tube boiler for a turbine of 8000 shp

them with travelling grates and spreader stokers. It is expected that these ships will operate on the Pacific coal/ore trade.

The conversion of existing oil-fired boilers to coal firing with travelling grates means a reduction of some 30–40% on boiler output. This is quite acceptable, as the present optimal speeds are lower than before. This lower speed requires much lower power from the turbine. An additional stage of planetary gearing is therefore introduced to reduce output speed, whilst a section of the high pressure nozzle box is blanked off to reduce the mass flow of steam through the turbine. This increases the fuel efficiency and overall economics of the steam plant and has already been applied to a number of oil-fired steam turbine ships.

Shortcomings of first generation coal-fired ships

Although the combination of spreader stoker fired water-tube boiler and turbine has proved satisfactory in service and demonstrated that a fully automatic modern coal-fired ship with unmanned machinery space (UMS) classification can be built and operated, it is limited in its application to reasonably large bulk carriers which can be guaranteed coal of a consistent quality. The experience of the ANL has indicated that coal quality has a marked effect on the performance of the stoker fired grates, especially in relation to 'turndown' (i.e. the ratio of lower to higher firing rate) and its dynamic response.[9] There is considerable room for improvement and the need to develop

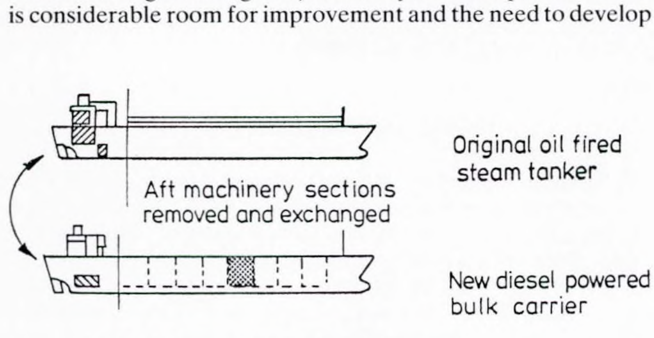


FIG. 5 Conversion of Spanish vessels

compact machinery and boilers better suited to modern naval architecture which has become used to the small machinery space required by modern medium speed diesel engines.

PROPOSED TECHNOLOGIES FOR FUTURE COAL-FIRED SHIPS

An apparently attractive alternative to the stoker fired water-tube boiler is the use of a fluid bed boiler with the existing design of steam turbine. The fluid bed has a number of attractions, particularly in its ability to burn poor quality coal and control pollution. Kawasaki Heavy Industries in Japan are developing a fluid bed boiler[10] which uses four fluid beds separated by membrane walls and containing the 'in-bed' evaporative tubes. The secondary superheater elements pass across three of the fluid beds, whilst the fourth contains the reheat tubes. The products of combustion from the fluid beds first pass through ducts formed by the evaporative tubes of the membrane wall to the primary superheater bank before leaving through the primary economizer. Although the in-bed heat transfer is very good, the low combustion temperature of about 900°C requires extensive convective heat transfer surface which is influenced by the almost total loss of radiant heat transfer. The whole boiler assembly is extremely complex and, like all conventional fluid bed boilers, has very considerable control problems. Part-load performance is difficult because of the limited ratio between the minimum and maximum combustion rates. In the event of a stop in the main steam flow during manoeuvring, an elaborate active control system is required to maintain the correct cooling steam flow through the superheat and reheat elements. It is also necessary to dump substantial amounts of steam. This is a very interesting project but one whose application must be limited by its cost, complexity and control problems.

Another method of using fluid beds has been proposed by GEC Gas Turbines Ltd.[11] This uses a combination of gas and steam turbines. The gas turbine is externally fired with coal using a recirculating fluid bed system, whilst the steam turbine is supplied from a water-tube boiler heated by combustion gases from the fluid bed and exhaust from the gas turbine. The coal is fired into a 'fast' fluid bed operating at a high fluidizing velocity such that particles are lifted from the combustion bed by the combustion gases to drop into a separately fluidized heat transfer bed, in which the air from the gas turbine compressor is heated in tubes before passing into the power turbine. The heat transfer takes the place of the normal combustion and allows the gas turbine to operate with clean gas. Cool bed material is returned from the heat exchange bed to the combustion bed to maintain the correct levels. After carrying the hot bed particles up out of the combustion bed, the combustion gases are separated centrifugally and pass to the water-tube steam boiler. The exhaust from the gas turbines is also used in the boiler and for preheating the combustion air. Although the cycle has the advantage of high overall thermal efficiency at design load, it is very large, complex and necessarily expensive. It can only be considered for installation in the largest ships as shown in Fig. 6, which outlines the relative sizes of different proposals.

An alternative to the use of coal-fired boilers is to gasify the coal. A producer gas plant is installed in the ship, the low calorific value gas being burnt directly in a modified and downrated diesel engine. There are several major problems with this proposal. First is the question of safety, with large volumes of carbon monoxide rich gas in the confined space of the ship. Second is the need to cool and clean the gas before it can be used in the engine, which requires rather bulky equipment and produces quantities of tarry liquids which are difficult to dispose of. Finally, there is the very considerable reduction in performance of the engine burning these gaseous fuels. The producer gas cannot be burnt alone and must either use a spark to assist combustion or a pilot injection of fuel oil. The use of up to 10% of pilot fuel adds complexity to the system and defeats the object of the exercise, to replace oil by coal.

COMPACT COAL-FIRED POWER PLANT

My own research has been concerned with the development of a low cost, high performance, compact steam reciprocating power plant that is as compatible as possible with existing medium speed diesel engines and can fit into existing diesel engine rooms, which will now be described in detail.

Combustion design constraints

If coal-fired power plant is to be economically and operationally competitive, the combustion system must meet a number of design constraints. It must be capable of burning a wide range of coal quality within existing emission standards. The combustor must be compact and have a very high heat release rate. It must be compatible with the lowest boiler costs and provide the basis for the highest heat transfer rates. It must allow for the highest possible 'heat availability'.[12] Finally, it must be capable of automatic control over a wide operating range and have good dynamic response.

The combustion system

Each coal combustion system discussed earlier has problems as well as advantages and not one meets the design constraints.

The fixed grate has the advantage of simplicity and very high heat release but is inherently an unstable process which leads to ash fusion and clinkering, as well as substantial elutriation of the fuel bed and poor emission control. The travelling grate has automated the fixed grate but only with substantial loss in heat release rate, added complexity and a greatly reduced 'turndown' which is determined by the unstable nature of the combustion process and the relationship of the over-feed and grate speed to the coal quality.

The controlled low temperature of the fluid bed (Fig. 7) avoids clinker formation and gaseous pollution but at the price of poor turn-down, grit elutriation from the bed and an almost total loss of radiant heat transfer and consequent need for large convective heat transfer surfaces. High performance fluid beds require very complex control systems and are not self-regulating.

The use of pulverized fuel in land installations is well established but presents many problems when considered for marine use. To obtain satisfactory combustion, the coal particle must be held in suspension in a stream of air and combustion gases whilst the particle burns to ash and slags into a dense particle which will fall out of suspension (see Fig. 8). Combustion therefore depends on the size of the particle, which in turn determines the required terminal gas velocity to support the particle and the burn-out time during which the particle must be suspended. For furnace volumes likely to be acceptable on board ship, the fuel must be ground to a particle size of a few micrometres, at which the particles become chemically very active and easily form explosive mixtures. The fuel must be stored in an inert gas or as a coal/water or coal/oil slurry which is expensive to produce and difficult to maintain.

It is possible to pull together all the advantages of these combustion systems: the simplicity of the fixed grate, the over-feeding of the spreader stoker, the advantage of the deep fuel bed of the fluid bed and the cyclonic combustion of the pulverized fuel burner. This staged combustion of coal was first demonstrated by Porta in Argentina[13] and more recently in South Africa and in the UK. This process is known as the gas producer combustion system (GPCS).

This form of staged combustion (Fig. 9) consists of a deep fuel bed on a fixed grate which acts as a gas producer and supplies carbon monoxide rich off-gas to be burned in an integral cyclonic secondary combustor. The gas producing fuel bed is maintained at a temperature of 900°C like a fluid bed. However, the gas velocities in the bed are much lower and the bed has little inert material.

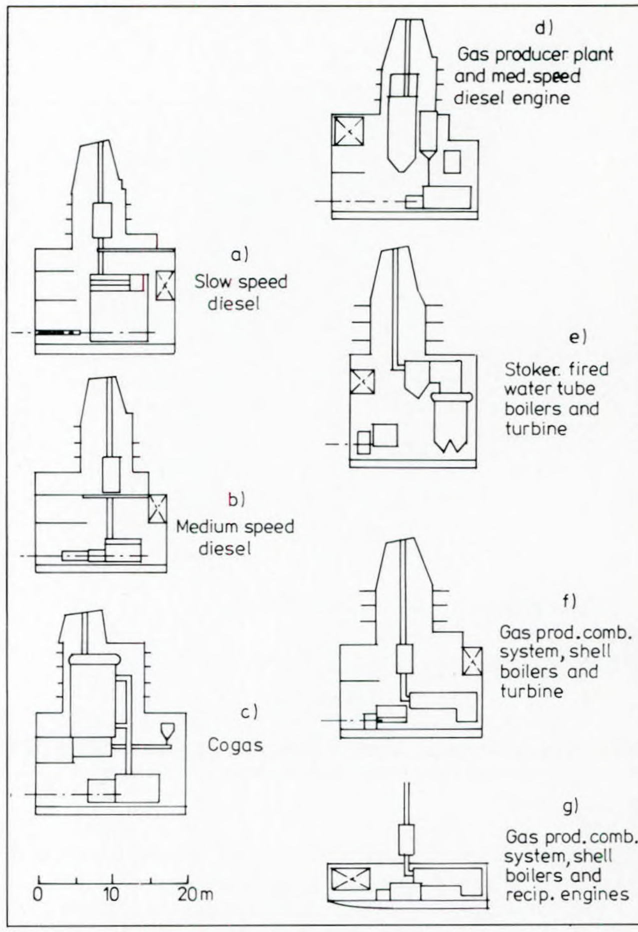


FIG. 6 Comparisons of marine coal burning power plant of approximately 12 000 shp

As shown in Fig. 9, the primary air entering at the bottom of the fuel bed first combusts some of the carbon in a small zone close to the grate to carbon dioxide. This carbon dioxide in passing through the rest of the heated bed is reduced in an endothermic reaction to carbon monoxide, which, together with volatile material driven off the overfed coal and some small particles, emerge from the bed into the cyclonic stream of preheated secondary air where they burn with an intense radiant flame.

During the second or so that the combustion gases spend in the cyclonic secondary combustor, most of the particulate mateial is burnt or removed, producing clean combustion gases. These allow the convective surfaces of the boiler to be designed with very high Nusselt numbers which, together with the high heat release rate of the combustor (5–7 MW/m^2), allow very compact boilers to be designed which will fit into existing engine rooms. The design of a boiler with a heat release rate of 12 MW is shown in Fig. 10.

COMPACT BOILER DESIGN

To achieve the highest thermal efficiency in any steam plant, the highest steam temperature must be used. This temperature is prescribed by material cost considerations, which suggest that 550°C is optimal for the modest pressures of the shell boiler (25 bar). The boiler design is determined by the combustor needs and the superheater temperature. The superheater is placed in the evaporative fire tubes and takes the 'Houlet' form with concentric flow and return tubes. This form of superheater

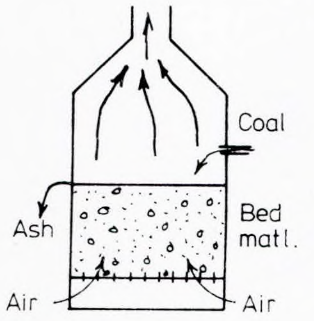


FIG. 7 Fluidized bed combustion

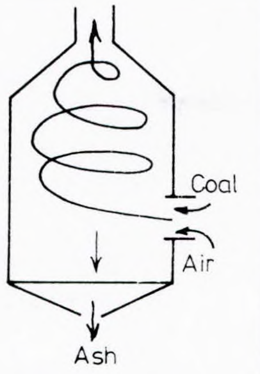


FIG. 8 Pulverized fuel burner

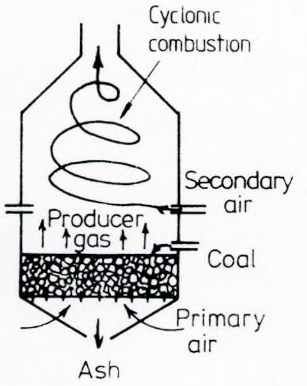


FIG. 9 The gas producer combustion system

has very high heat transfer and is well protected by its radiant proximity to the evaporative tube when the steam flow is reduced during manoeuvring.

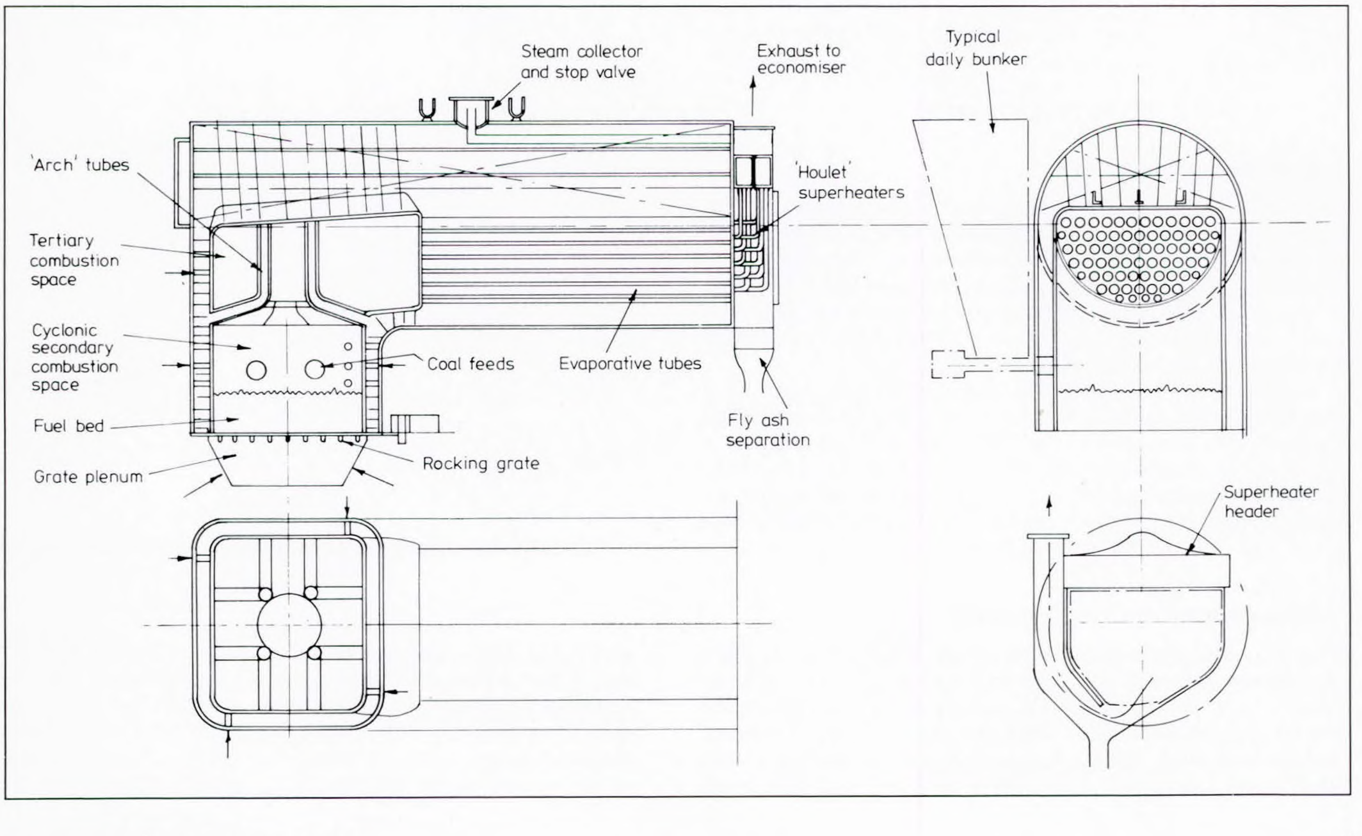
The outlet temperature of the superheater determines the exhaust gas temperature of the evaporative part of the boiler. This is necessarily high at 500°C. The exhaust gases therefore pass to a large economizer in the form of a standard exhaust gas boiler which may be mounted separately from the main boiler on a suitable flat or in the uptake.

The gas flows and inlet temperatures to the economizer are similar to those of an equivalent turbocharged four stroke diesel. The boiler shown in Fig. 10, which is designed to support a 3 MW power plant, is 8.30 m long, 5.30 m high (5.5 m over safety valves) and 3.0 m wide, and weighs approximately 45 tonnes in working order. The heating surfaces are arranged to allow ± 25 deg angle of roll and a ± 15 deg angle of pitch. The 'arch' tubes provide a good thermic syphon to the firebox crown from the generous water space around the combustor. Ready access is provided to the superheater elements and the evaporative tubes from large access doors on the front of the boiler, allowing individual superheater elements and tubes to be removed and blanked off. Internal baffle plates are provided to prevent surging.

Boiler efficiency

The combustion process and the disposition of the boiler heating surfaces are carefully staged to obtain the highest possible overall boiler efficiency within the constraints of the maximum allowable combustion temperature of 1500°C and the back end economizer exhaust temperature of 200°C. This gives a boiler efficiency of 87%. The overall boiler efficiency will depend on ash loss and coal quality.

FIG. 10 Compact shell boiler fitted with the gas producer combustion system
Steam conditions: 25 bar, 550°C
Steam flow: 11 tonnes/h



Boiler control

The control of the boiler water side presents no unusual problems. Since there is a considerable depth of coal in the fuel bed, the precise level of coal is not important and may fluctuate quite widely. For example, the boiler shown in Fig. 10 has a mean depth of 500 mm and can operate between 375 and 525 mm with coal of 10–25 mm. It follows that at full power (5 MW/m^2) the combustor can operate for approximately 30 minutes without coal feed. Coal may be fed by a high speed screw or pneumatically from any of three sides of the combustor. In the example, the coal is fed from a daily bunker mounted beside the boiler by two screws. A rocking grate is provided to remove the ash, which is powdery and easily handled pneumatically.

The required combustion rate and bed temperature are controlled by the ratio of primary air and recirculated exhaust flue gases passing through the fuel bed, while the secondary air is adjusted to maintain the correct exhaust gas composition. The ratio of primary to secondary gas flows may be preset and the boiler output regulated by the induced draught fan.

Coal and ash feeds

The coal and ash feeds to the boiler depend on the design of vessel. However, the use of small coal of 10–25 mm allows the use of established dense phase and screw conveyers and presents no problems. The use of twin feeds from the daily bunker and the ability to operate for long periods without coal feed result in a high level of redundancy.

The controlled combustion temperature of the fuel bed avoids clinkering and produces ash with free-flowing properties which may be readily transferred and disposed of using established methods.

Feed heating and air heating

Improvements in thermal efficiency may be obtained by using a feed heating train from suitable points in the engine thermodynamic cycle. The secondary combustion air may be preheated if required from bled steam or exhaust gases. Whether this is economic depends on the particular circumstances prevailing.

ENGINE DESIGN

Engine design constraints

The decision to use a compact shell boiler imposes severe constraints on the pressure and temperature available to the engine. For reasons of cost and simplicity, these have been chosen at 25 bar, 550°C. For similar reasons of space and cost, it was decided to adopt the high condenser temperature and corresponding pressure of 50°C and 0.1 bar. These constraints are shown on the Mollier diagram of Fig. 11.

In addition to the constraints imposed by the use of shell boilers and the simple condenser, it was desired to match the medium speed four-stroke turbocharged diesel engine in relation to specific power and flexibility of output. Thus the mean effective pressure and the density of steam in the engine are limited, imposing a further constraint below which only a high speed turbine could be sensibly used.

Turbo-recompressed reciprocator

The use of a reciprocating engine with variable cut-off offers the highest potential efficiency so long as it can operate away from saturation. Therefore the arrangement of reciprocating engine and exhaust turbine should be used. The question arises as how best to use the power developed in the turbine. Many ideas have been used. The turbine may be used to provide additional output directly through a gearbox but this does not match the turbine well and the use of hydrodynamic coupling is required. Alternatively the turbine may be free running and used to recompress and reheat the steam between the compounded stages of the reciprocating engine. This concept of turbo-recompression has been successfully applied since the

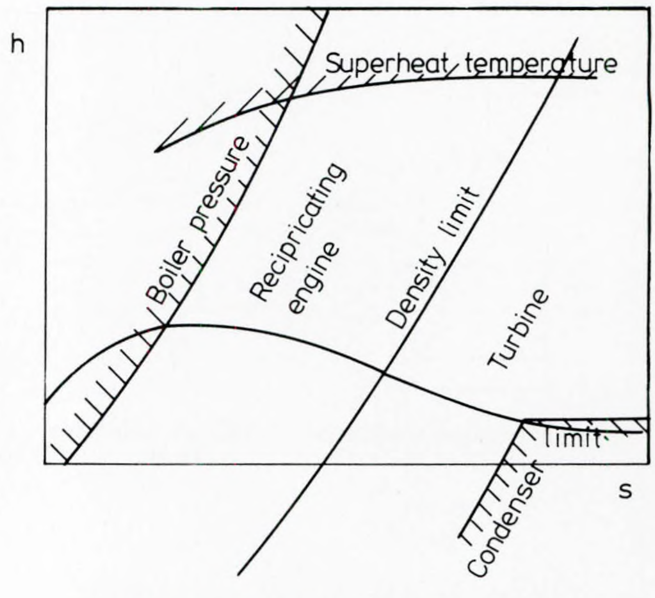


FIG. 11 Mollier diagram showing design constraints

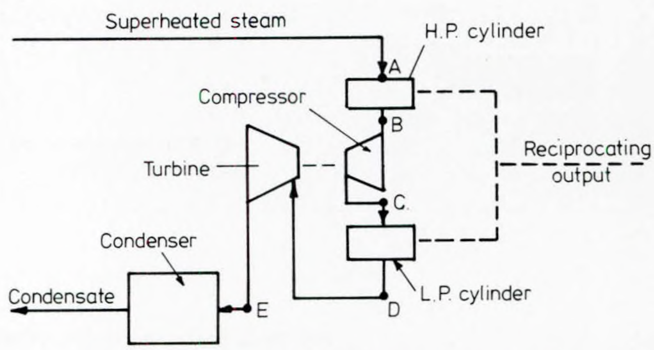


FIG. 12 Diagram of turbo-recompressed steam engine

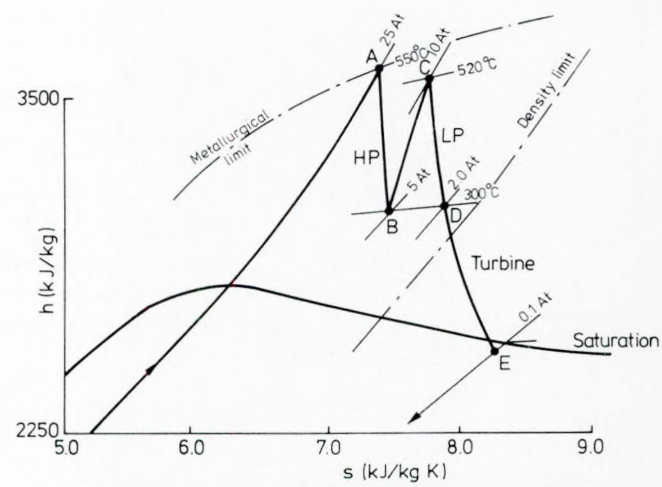


FIG. 13 Mollier diagram of turbo-recompressed steam engine

1930s[14] and is still in use today on the *Borea*, operating between Sweden and Finland. The turbo-recompression avoids the use of complex gearboxes and transmissions and may be designed in a form similar to the turbocharger.

The thermodynamic cycle

Having chosen a turbo-recompression cycle, the steam conditions may be optimized to fit into the constraints shown in Fig. 11.

The cycle consists of a two-stage compound reciprocating engine with an exhaust turbine driving a compressor which recompresses and reheats the exhaust of the high pressure cylinder into the low pressure cylinder. The cycle is arranged to provide equal work in all the elements, with the output being taken from the two reciprocating stages as shown in Fig. 12.

The steam enters the HP cylinder at 25 bar, 550°C and, with a cut-off of 20%, expands down to 4.5 bar, 300°C. This exhaust then passes to the compressor in which the pressure is raised to 10 bar and the temperature increased to 530°C to pass into the low pressure cylinder with the same cut-off, where it again expands to exhaust into the turbine at 1.8 bar, 300°C. The turbine exhausts into the condenser at 0.1 bar, 50°C and 0.998 dryness, as indicated on the Mollier diagram in Fig. 13.

The work produced by each cylinder is 550 kJ/kg of steam, giving a total indicated output of 1100 kJ/kg. This is a cycle efficiency of 33% which is higher than that produced by a multi-stage reaction turbine operating on the same steam conditions.

Unlike the turbine, the recompressed reciprocating engine can maintain high efficiency over a wide operating range, while the free-running turbo-compressor allows the engine to be easily reversed.

Because the reciprocating cylinders operate well away from saturation there is little cylinder heat transfer and thus a cylinder efficiency of 90% may be achieved with suitable inlet and exhaust valves; compared with a turbine isentropic efficiency of 75% under similar conditions.

Turbo-recompression

The use of turbo-recompression has a number of advantages over the use of an exhaust turbine. Although the turbine must be designed for the highest efficiency, the compressor is limited by the achievable pressure ratio and must convert much of its mechanical input to heat with a low isentropic efficiency of about 60%, so that its design is simplified.

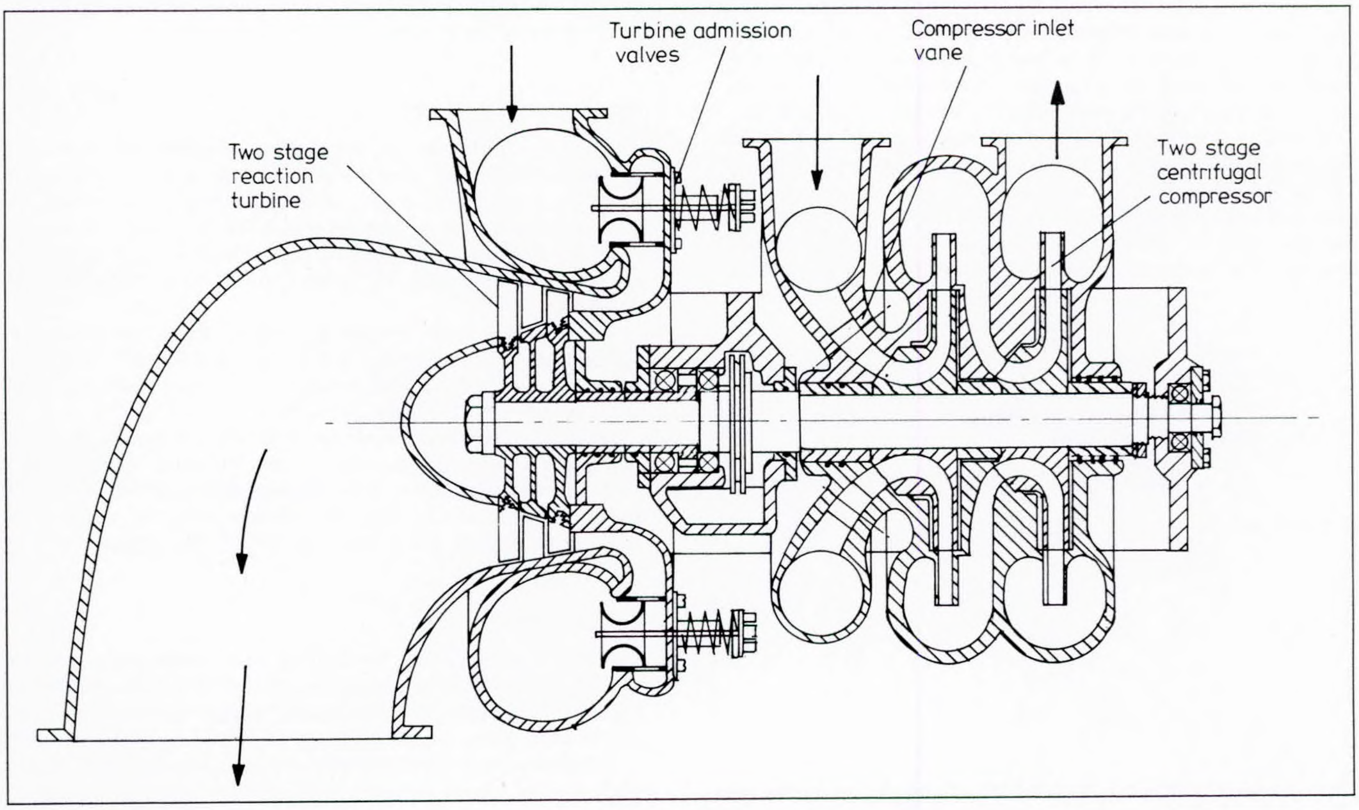
The free-running turbo-compressor may operate at high speed and can operate during manoeuvring. It also avoids the use of an expensive transmission.

Turbo-compressor design

Although the turbo-compressor has many features in common with the turbocharger applied to internal combustion engines, it has a number of important differences. The most significant is the considerable mismatch in the fluid densities and hence specific speeds of the turbine and compressor. A second is the temperature range, the steam compressor being 300–530°C, whilst the steam turbine operates between 300 and 50°C, giving rise to an interesting choice of materials. The radial compressor is of nimonic whilst the axial turbine is of stainless steel. The rotor is designed on a diameter of 450 mm rotating at 40 000 rev/min. The radial compressor has two stages, each with a pressure ratio of 1.9. Since the density of the steam during the compression does not change greatly, the two stages may be the same. The diffusers are not provided with blades so as to give a wide surge limit. This is further enhanced by the use of adjustable inlet guide vanes.

The axial turbine has two reaction stages to give a pressure expansion ratio of 20:1 and exhausts directly into the condenser. To accommodate the wide range of mass flow, three nozzles are provided in the ratios 2:4:4 to give a mass flow over the range 20–100%. This enables the engine to operate efficiently over the same range. The general arrangement of the turbo-compressor is shown in Fig. 14. The rotor is provided with one inboard combined radial and thrust bearing with

FIG. 14 Section through two-stage turbo-compressor



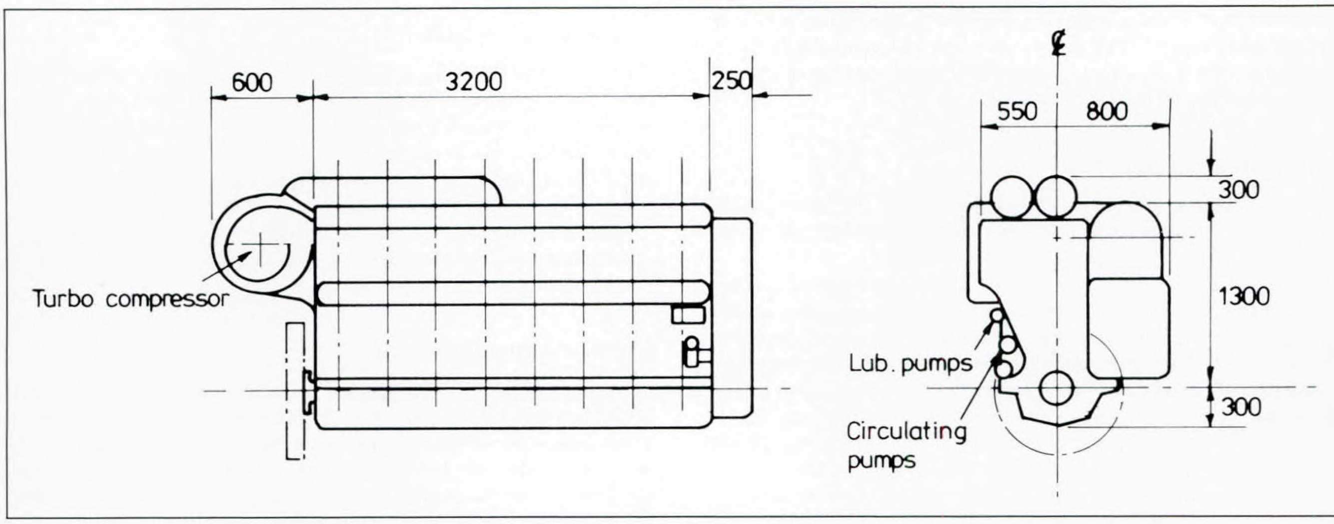


FIG. 15 Outline of eight-cylinder turbo-recompressed steam reciprocating engine of 3000 kW output

self-acting lubrication, mounted between the reaction turbine and the inlet to the compressor. An outboard radial bearing is provided outside the compressor outlet. For the 1500 kW unit shown in the figure, the rotor is approximately 900 mm long and the unit is 850 mm over the casings.

The unit is designed to operate on the constant pressure system, as the exhaust frequency is four times that of the equivalent diesel engine.

Steam reciprocating engine

The steam engine is designed to be as nearly a direct replacement for a medium speed, four-stroke turbocharged diesel engine as possible to enable steam to replace diesel at some future date. It is based on the diesel engine crankcase and designed for a similar output torque and speed (see Fig. 15).

The engine comprises pairs of double acting cylinders (one high pressure, one low pressure) each producing equal power but operating at different mean pressures; although between the same inlet and exhaust temperatures (550°C and 300°C respectively). Because of the use of turbo-recompression between the cylinders, the pressure expansion ratio in each cylinder is controlled by the variable cut-off in the range 5–40%.

Double beat poppet valves are provided for inlet and exhaust on each cylinder. These valves are operated from twin cams which are controlled through epicyclic gears to provide variable inlet cut-off and exhaust cushioning and also to provide for the direct reversing of the engine. A governor operates through the variable cut-off. A section through the cylinder

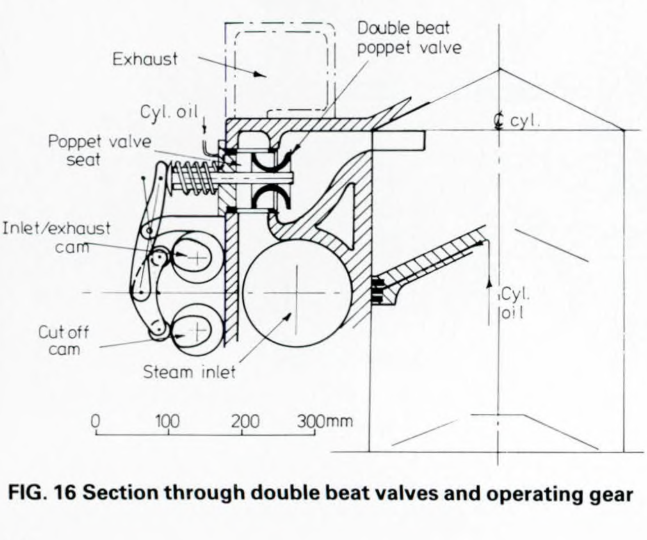


FIG. 16 Section through double beat valves and operating gear

and valve gear is shown in Fig. 16. All the cylinders take the same form and use the same valves and valve gears. The HP cylinder is bored to 230 mm and the LP to 350 mm on a stroke of 370 mm. A diagrammatic cross-section of the engine is shown in Fig. 17. The cylinders are steam jacketed with integral steam inlet receiver passages and external exhaust passages.

All the sliding surfaces of the piston and piston rod and valve stems are fed with metered lubrication, the cylinder oil feed being made through the crosshead and piston rod.

The crankcase is totally enclosed and separately lubricated with integral lubrication pump.

Since the lubricating oil provides no cooling for the engine, no separate oil cooler will be required. Lubrication for the valve cams and mechanism is provided from the main engine lubrication pump.

A gear casing containing the valve drives and auxiliary drives, including power take-off for the condenser circulating pumps, and the liquid ring condensate and air pump is provided at the front of the engine.

Overall engine design

The outline of an in-line eight-cylinder (four HP, four LP) turbo-recompressor steam engine, designed for the steam condensates provided by the shell boiler, is depicted in Fig. 15. The turbo-compressor is mounted at the aft end of the engine over the flywheel, with the condenser mounted alongside the engine. The engine is 3.2 m long and 1.35 m wide over the condenser.

The services to the engine are high pressure steam in and condensate out, condenser cooling water flow and return and lubrication supplies, which may be piped in relatively small pipes which ease the siting of the engine.

Although the engine is shown with the condenser mounted alongside, a separate condenser may be used. This allows a twin engine installation on minimum centre distance (850 mm for the engine in Fig. 15). Ready access to the valve gear, pumps etc. may be made from one side of the engine.

Performance

Computer simulation studies of the turbo-recompressed engine, using filling and emptying models similar to those used for diesel engines, have confirmed the running conditions and anticipated performance of the power plant.

Whilst the performance at the designed MCR is comparable with that of the equivalent turbocharged four-stroke diesel, the steam plant is capable of much greater output torque at lower speed and a considerable short term increase in output power is available by mortgaging the boiler.

The torque–speed characteristic of the complete power plant, including the fine-tube boiler and economizer without bled steam feed heating, is shown in Fig. 18.

The performance of the engine is constrained by the maximum output from the boiler, which for the boiler shown in Fig. 10 is 3.6 kg/s. The maximum allowable speed of the engine is limited by piston speed and particularly the control of the inlet valve events to 800 rev/min for the eight-cylinder in-line engine.

The minimum output power for normal operation with the turbo-recompressor is approximately 500 kW. Below this power the engine must be operated as a straight compound with the turbo-compressor bypassed. This is only required when the engine is driving a fixed pitch propeller. On starting, high pressure steam is reduced in pressure and fed to the lower pressure inlet manifold. The exhaust from the LP cylinders passes to the exhaust turbine and causes the compressor to operate on the exhaust from the HP cylinder to provide steam to the LP receiver, cutting out the starting regulator.

The computed efficiency map shows a large operating area with an overall power plant efficiency of greater than 28%. For steam coal of 30 MJ/kg, this gives specific fuel consumption figures of 320 g/hph or 420 g/kWh compared with a specific fuel consumption of a heavy fuel burning diesel engine, at 48% thermal efficiency, of 178 g/kWh.

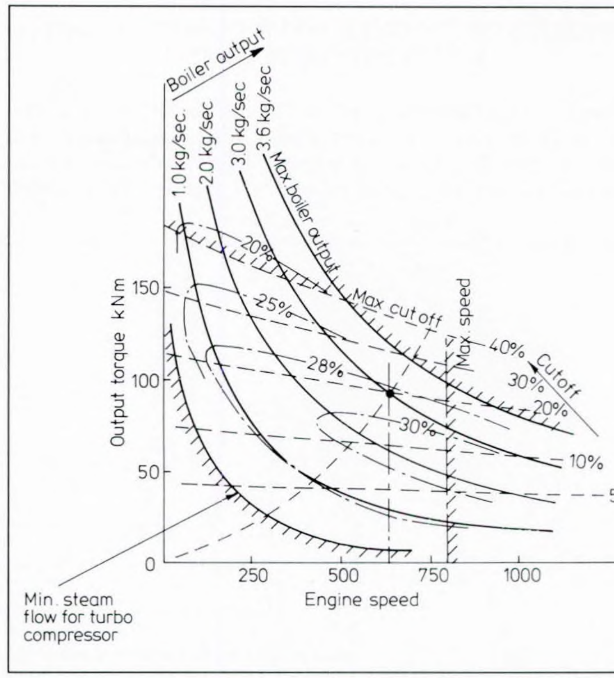


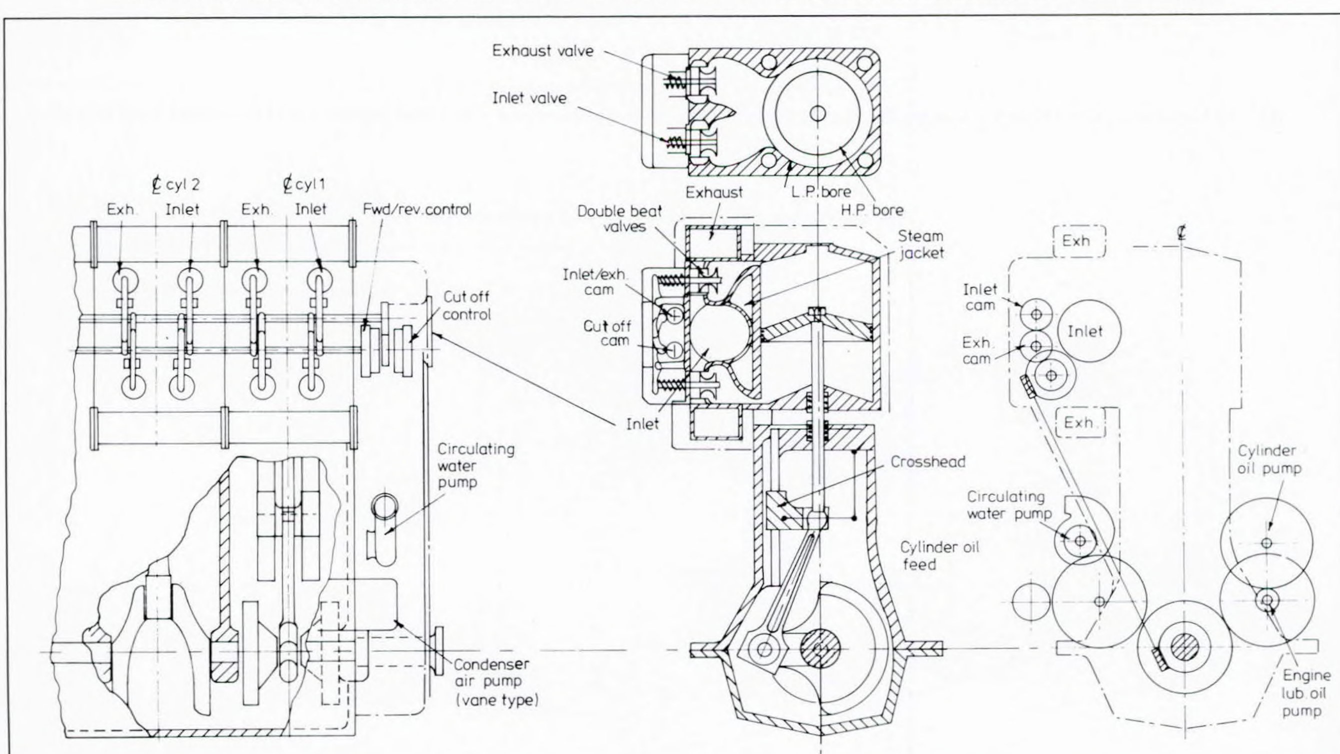
FIG. 18 Performance map of 3000 kW turbo-recompressed medium speed steam engine

POWER PLANT ECONOMICS

With steam coal available fairly widely throughout the world at present (December 1984) at \$40 per tonne, the compact steam plant would cost aproximately \$16.8/1000 kWh. This is half the fuel cost of the equivalent medium speed diesel engine burning heavy fuel oil at the world average price of \$185/tonne. Its fuel cost is \$33.03/1000 kWh, a saving of \$16.23/1000 kWh, which offsets the \$250/kW increase in the capital cost of the coal-fired steam plant within 15 000 hours of operation (i.e. within 2–2.5 years).

Since the compact coal-fired power plant can be readily fitted into existing engine rooms, without large increases in building costs, it makes good sense to use coal in those trades where the bunkers are relatively small and can be readily obtained. This applies to vessels operating in the 'short sea' trades such as small bulk carriers, feeder container vessels and ro-ro ferries, as will be shown in the following examples.

FIG. 17 Cross-sections and part elevation of 370 mm stroke medium speed turbo-recompressed steam engine



EXAMPLES OF VESSELS WITH COMPACT COAL-FIRED POWER PLANTS

In Figs 19–27 details are given of typical machinery installations, each showing a different aspect of the machinery. The machinery installations in the three typical short sea vessels, a small bulk carrier of some 6000 tonnes displacement, a 10 000 tonnes displacement low cost feeder vessel and a 20 MW installation on a ro-ro ferry, are based on the designs presented earlier for a 3 MW engine and boiler. In each case a minimum of two boilers is used with duplicate coal feeds to each boiler, which together with the tolerant nature of the combustor give a great deal of redundancy in the coal feeds and bunkers. The two-stage combustion system is tolerant of coal quality, which

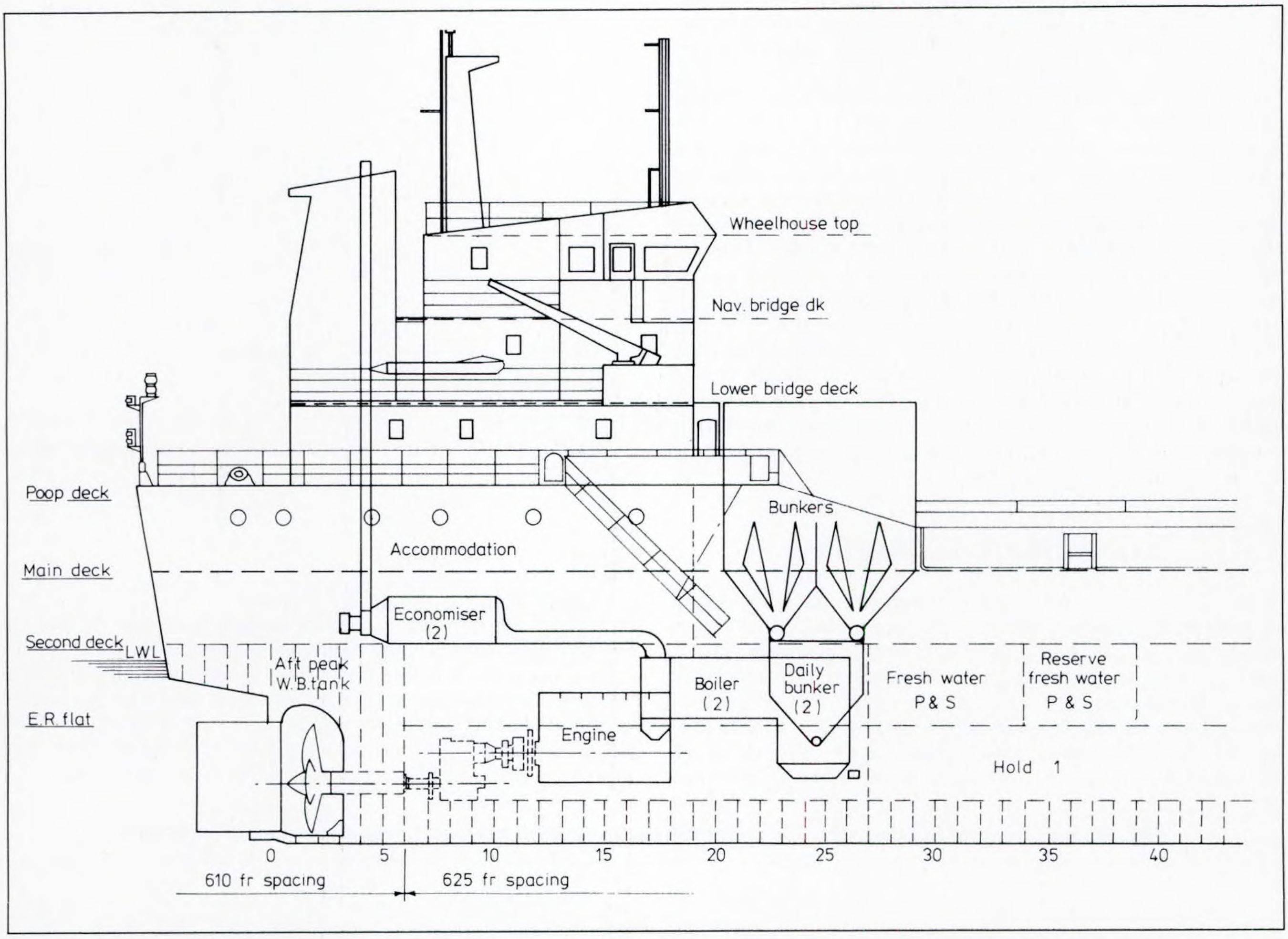
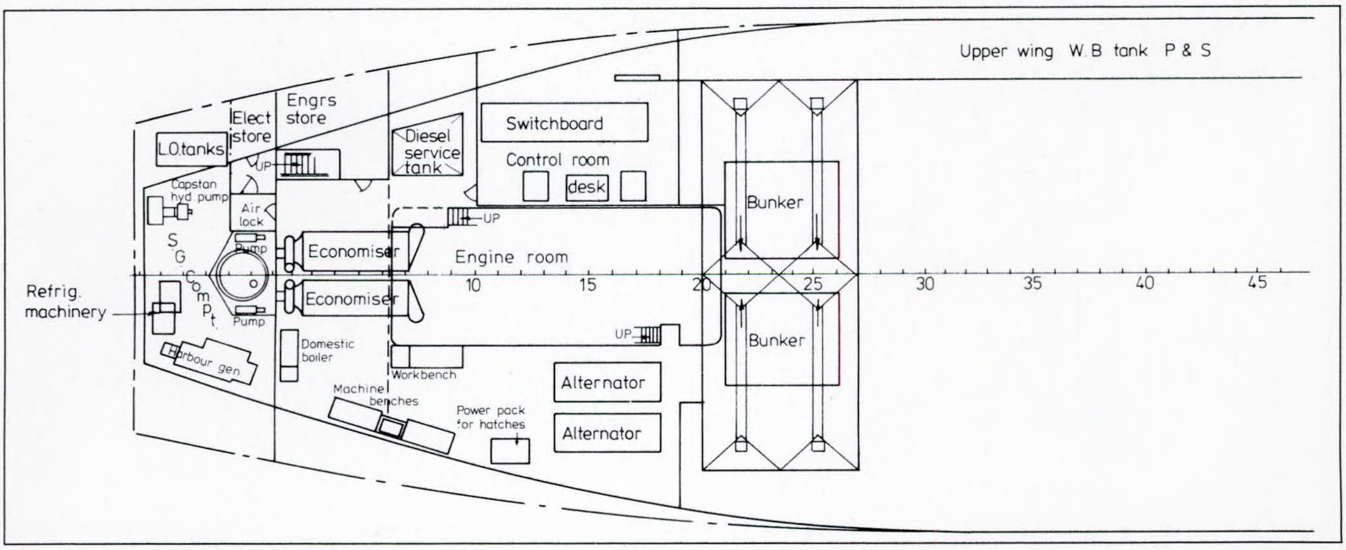


FIG. 19 Elevation of machinery space showing installation of 3000 shp compact coal-fired power plant in a small bulk carrier

FIG. 20 View on second deck of small bulk carrier



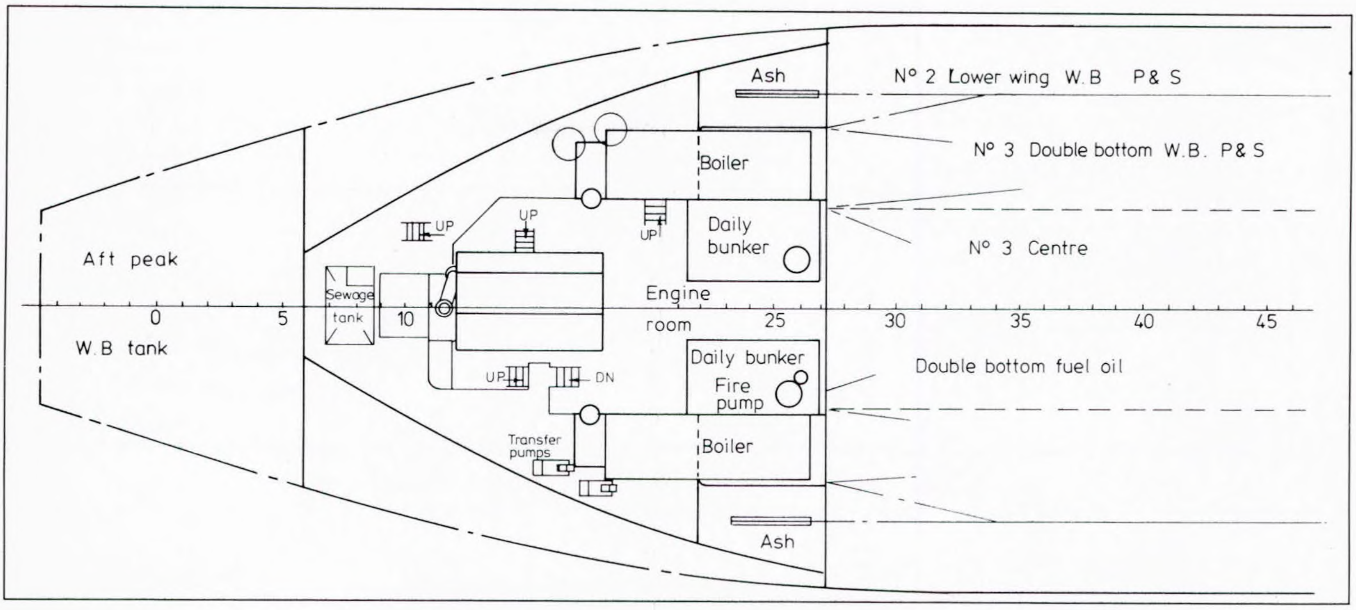


FIG. 21 View on engine room flat of small bulk carrier

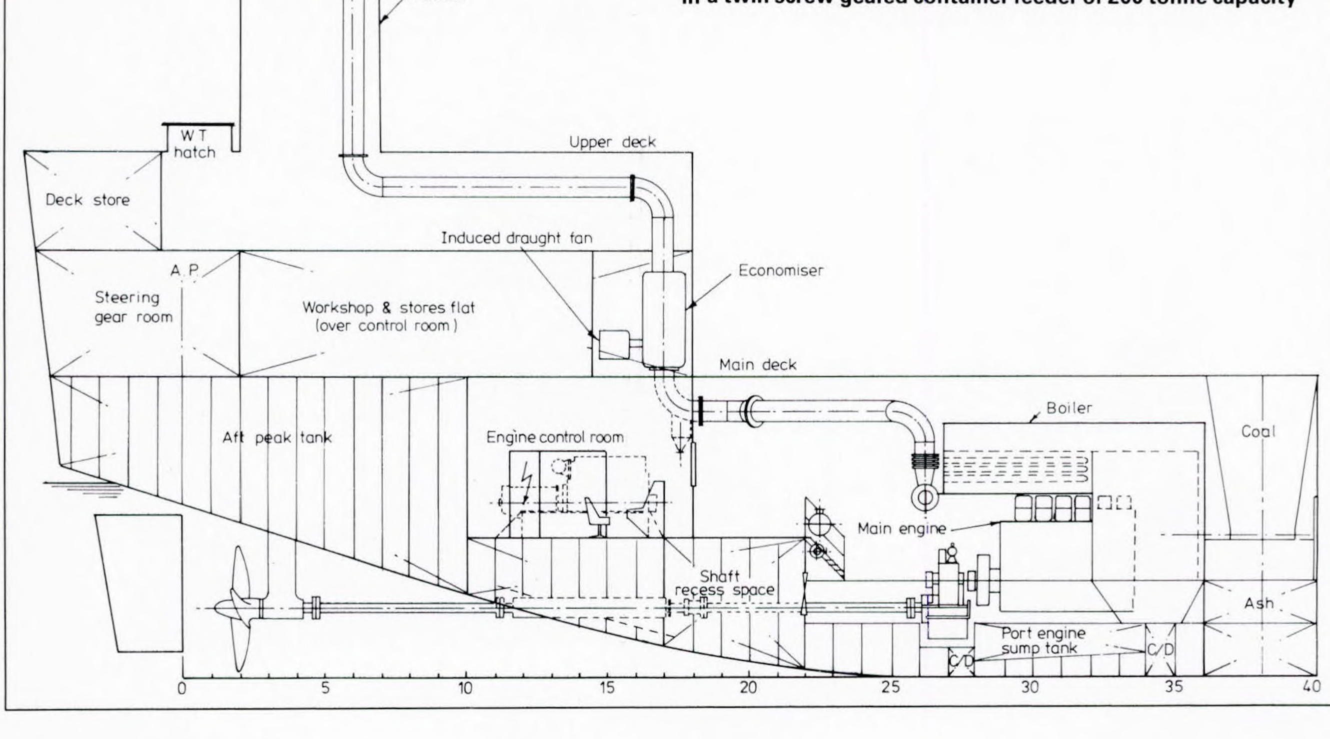
enables the vessels to take bunkers from various sources. Steam alternators using the same machinery design as for the main engine are provided, since these can operate with the same high efficiency as the main engine. Diesel emergency generators and normal engine room equipment are provided.

Figures 19, 20 and 21 show the disposition of the machinery in a small bulk carrier of some 6000 tonnes displacement with an installed power of 3000 shp. The main engine drives a controlled pitch or fixed pitch propeller through an offset reduction gearbox. Steam is provided by two coal-fired boilers mounted forward and outboard of the main engine. The economizers are mounted on a flat at deck 2 level, with the induced draught fans. The engine room has forced ventilation. Each boiler is provided with a daily bunker from which coal is screw-fed into the combustor. The main bunkers are placed forward of the accommodation on the main deck and are divided into four sections each with a screw feed to the daily bunkers below, providing two alternative feeds from the main bunkers to the daily hoppers. The main bunkers accommodate 280 tonnes and provide for a range of some 7000 km. Provision for ash storage is made in the wings outboard of the boilers. Two steam alternators of 100 kW are provided on deck 2 (starboard) with the control room and switchboard on the port side. Access to the coal feeds is also provided forward at this level. The complete machinery installation is contained in an engine room only 14 m long.

Using the December 1984 Rotterdam prices, the fuel saving on this vessel would be $720 per day burning coal instead of heavy fuel oil. There is no loss of cargo space compared with a similar diesel installation.

Figures 22–24 show the machinery installed in a twin screw, geared ro-ro/container ship of 2700 dwt and 200 TEU capacity.

FIG. 22 Elevation showing the twin installation of 3000 and 4000 shp in a twin screw geared container feeder of 200 tonne capacity



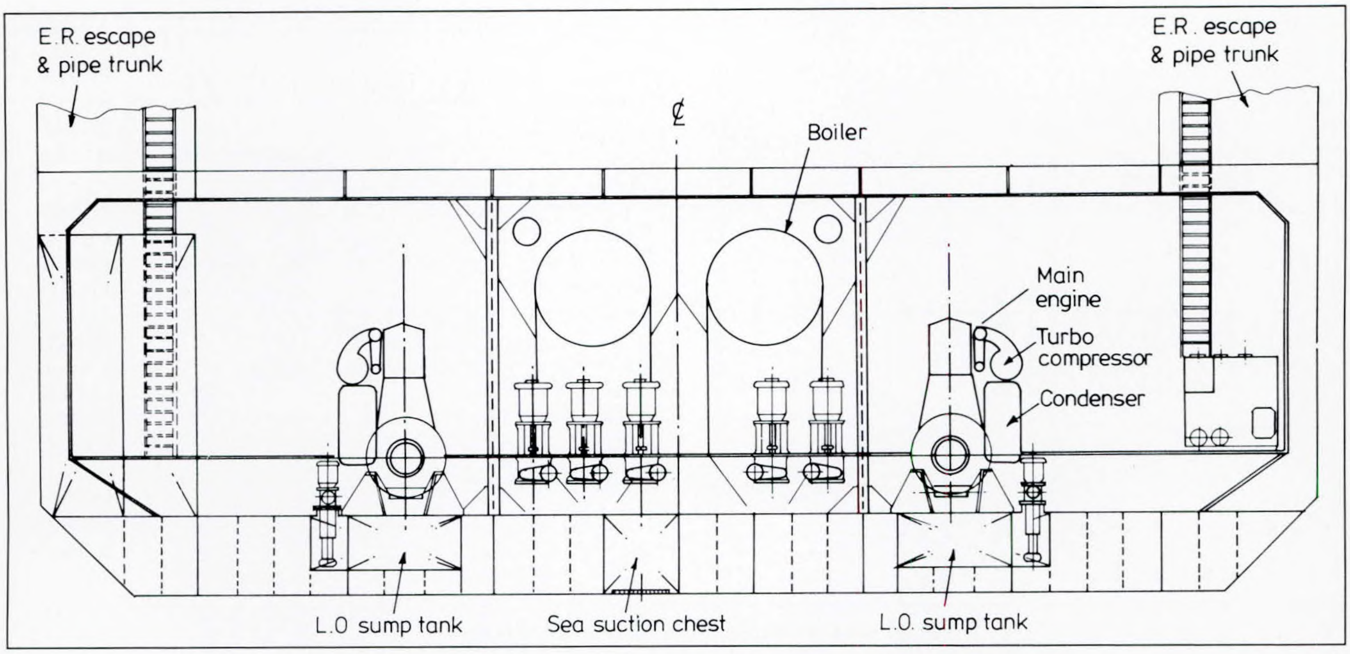
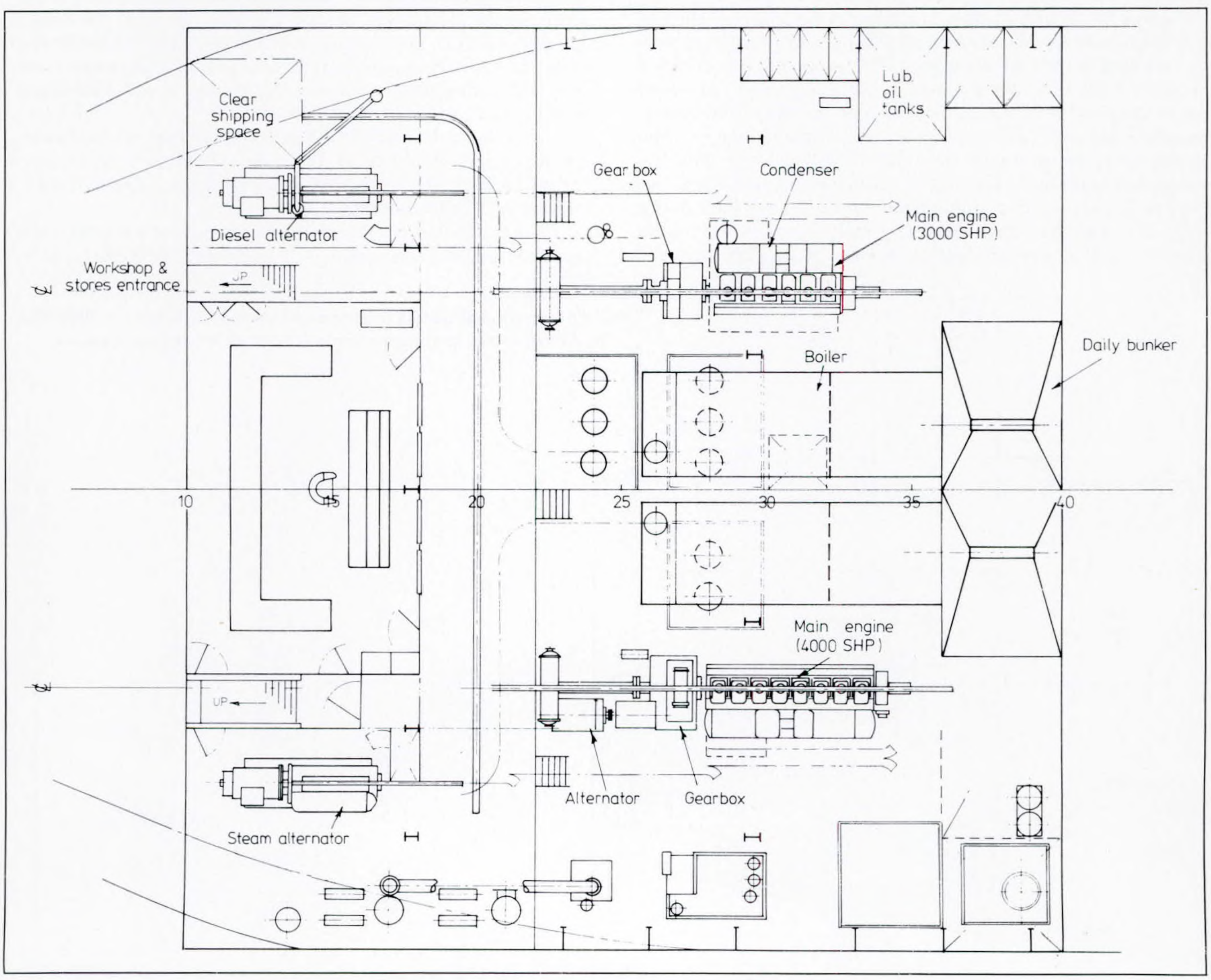


FIG. 23 Section through engine room of 200 tonne container feeder ship

FIG. 24 Plan of engine room showing disposition of machinery and boilers of 200 TEU container feeder ship



This low cost vessel with broad beam has two geared engines of 3000 and 4000 shp driving controlled pitch propellers, with a shaft driven alternator taken off the starboard gearbox to provide for the 40 TEU reefer load when at sea. One steam alternator and one diesel alternator of 1000 kW are provided on the control room flat. Steam is provided from two boilers (to the design shown in Fig. 10) mounted between the main engines with their daily bunkers forward and the ash storage beneath. The economizers for the boilers are mounted in the uptake on a flat above the control room. The vessels are provided with bunkers for 600 tonnes of coal forward of the engine room below the main deck, to provide a range of 8000 km. The bunkers are divided into four, each with a screw feed to a dense phase pneumatic system transferring coal to the daily bunkers. The fuel saving on this vessel is of the order of US$1500/day.

Since the steam auxiliary engines can be as efficient as the main engine, the complexity of the shaft driven alternator and the controlled pitch propellers could be avoided with the installation of a larger (2000 kW) steam alternator, which would reduce the capital cost of the installation.

The third vessel, illustrated in Figs 25–27, is a 19000 dwt ro-ro ferry with 24000 shp on two shafts in a twin skeg afterbody. All machinery and bunkers are below the main vehicle deck with a headroom of 10.5 m. The two 16-cylinder 'V' engines of 12000 shp are each fitted with four turbo-compressors. Steam is provided from four coal-fired boilers aft of the main engines and outboard of the auxiliary machinery. Four economizers are fitted in the main uptake at main deck level, together with the induced draught fans. Two 3000 kW steam alternators are provided, together with one diesel driven alternator of the same output. The main engines are fitted with horizontally offset reduction gearboxes and drive controlled pitch propellers. Each boiler is provided with a daily bunker in front of the boiler, with the ash hopper below.

Cellular bunkers for 1200 tonnes of coal with screw feeds to a dense phase system feeding the daily bunkers are provided amidships, forward of the machinery space and below the main deck. The bunkers provide a range of 6500 km at 20.5 knots.

An alternative to the use of the 'V' engines is to mount two eight-cylinder in-line engines side by side, individually driving into the reduction gearbox. Each engine would be fitted with two turbo-compressors exhausting into either a condenser for each engine or preferably one condenser for each engine group.

The fuel saving on this vessel over diesel engines burning heavy fuel oil would be in the order of US$6500/day.

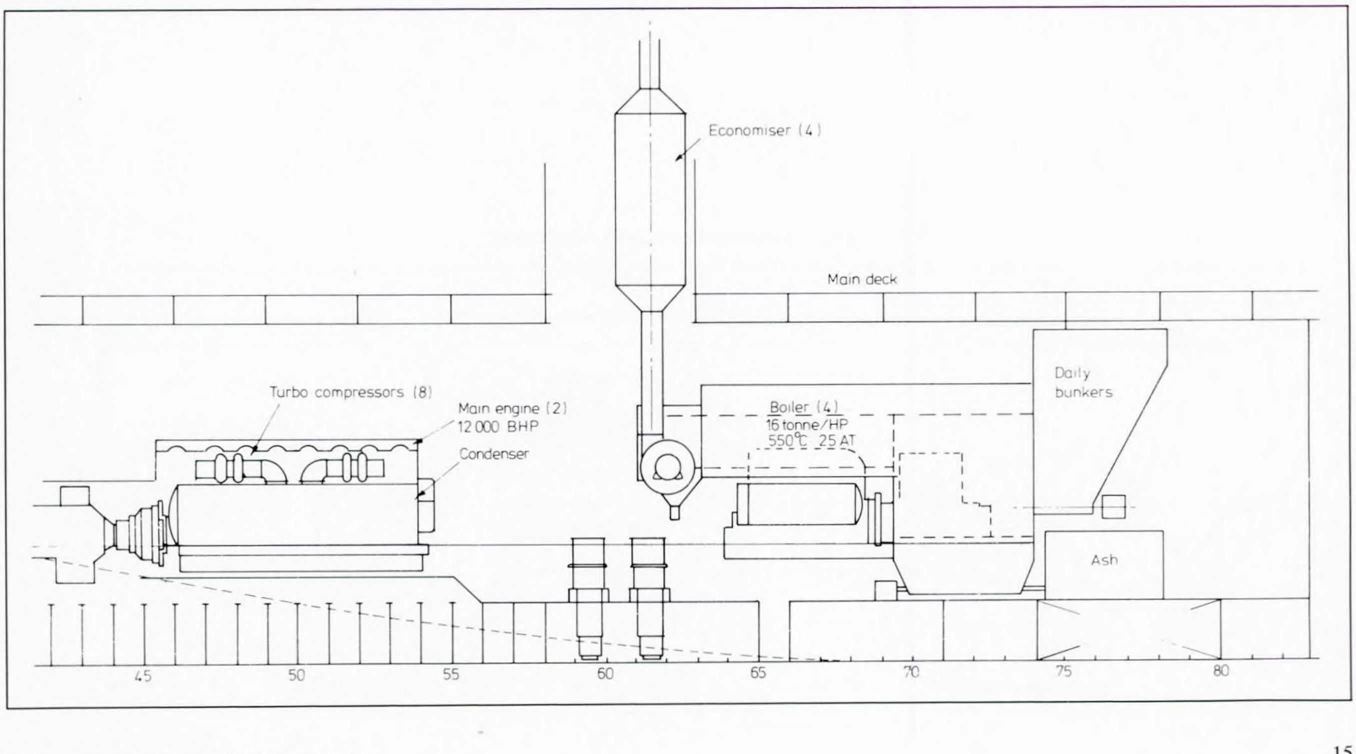
An added attraction of steam machinery for ferry operation is the considerable reduction in noise over a medium speed diesel installation.

In this paper I have not been able to consider all aspects of coal-fired ships, many of which have been dealt with in other papers in recent years; for example, the coal and ash handling and the bunker designs. It is important to note that careful consideration has been given to these aspects. In the case of the small bulk carrier, coal is loaded by grabs into the bunkers and generous mechanical screw feeds are provided. In the case of the ro-ro feeder and the ferry, the coal will be loaded using a dense phase pneumatic system which will ensure that the coal is suitable for transfer by dense phase within the vessel.

THE MULTI-FUEL SHIP

The examples given have illustrated the wide variety of vessels that could benefit from the installation of the compact coal-fired power plant described earlier. Since the steam main machinery is identical in form to that of the medium speed diesel, and the coal-fired boilers can be fitted into the existing engine room space, it is possible to consider a multi-fuel vessel, which may be built with provisions for coal bunkers but initially has diesel machinery which can be replaced by compact coal-fired steam machinery and boilers at some time in the future. This is particularly interesting in the case of ferries which have long hull lives. For any ferry built now there must be considerable doubt over fuel availability within its lifetime.

FIG. 25 Sectional elevation through 24000 shp twin screw ro-ro showing position of main engines, auxiliary machinery and boilers



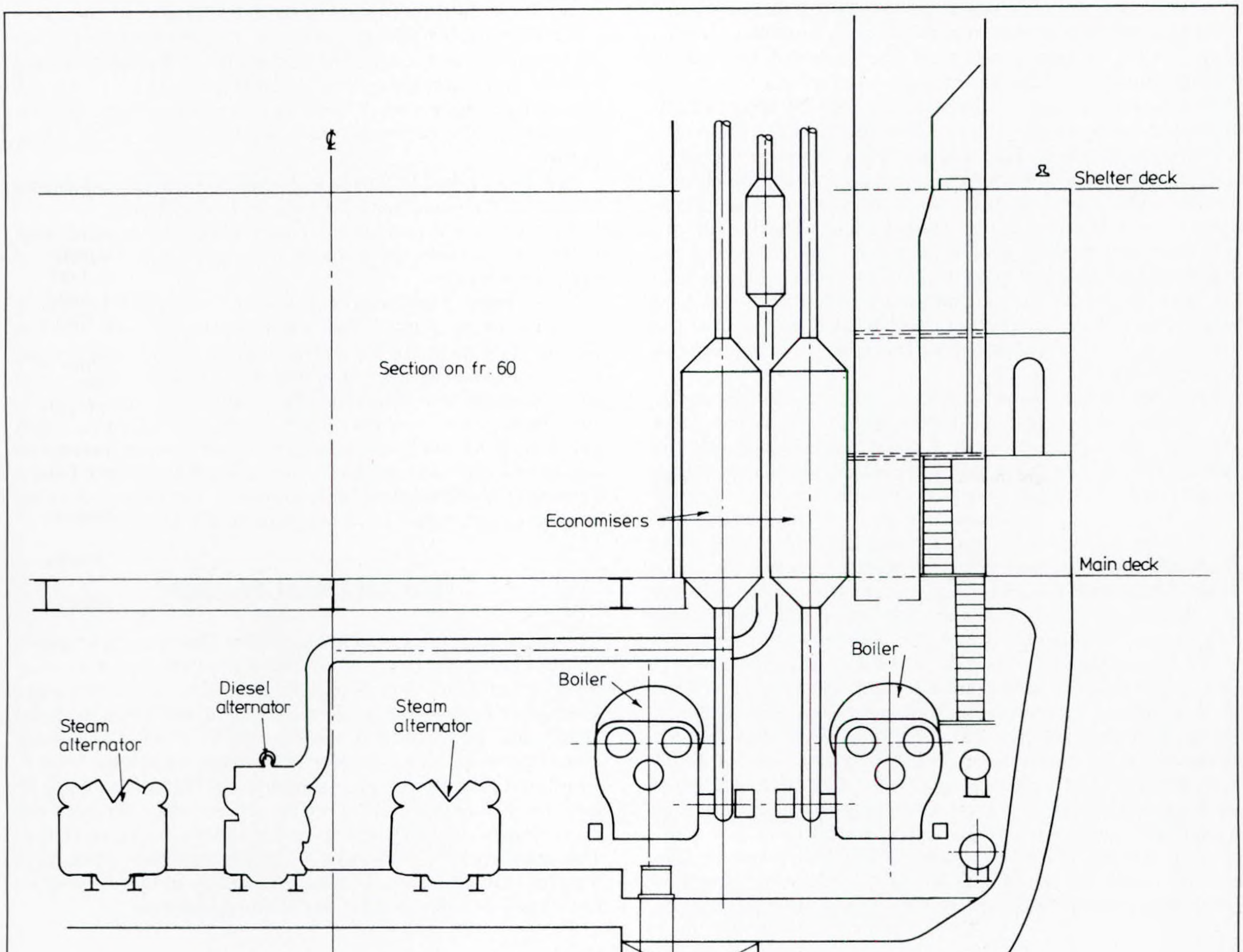
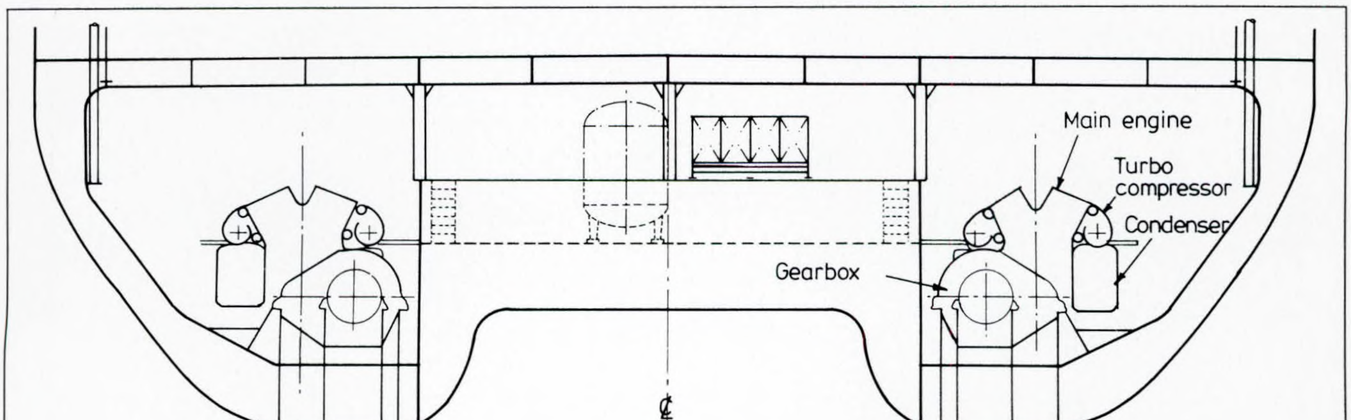


FIG. 27 Section showing main engines and gearboxes in twin skeg aft-body

FIG. 26 Section through boiler room



CONCLUSIONS

In this paper I have outlined the present state of the art for coal-fired ships and have drawn attention to the need for a flexible, compact coal-fired power plant that can fit into existing vessel architecture, particularly for the short sea trades.

Designs for a flexible, compact coal combustion system using staged combustion have been given, together with a low cost, high performance shell boiler design which can provide steam to a turbo-recompressed medium speed steam engine which matches the existing turbocharged four-stroke diesel both in size and output. The thermal efficiency of this plant is such that, compared with a diesel engine burning heavy fuel, fuel costs may be halved (on December 1984 Rotterdam prices). Compared with the existing alternative of spreader stoker fired water-tube boiler and geared steam turbine, the compact shell boiler and medium speed engine represent a considerable reduction in the capital cost of the machinery and avoid any additional capital cost for providing additional machinery space. This indicates a payback time of 2–3 years compared with a similar motor ship on HFO.

Despite the temporary reductions in fuel oil price, I am of the opinion that, particularly with the adoption of the compact coal-fired power plant described in this paper, there can be little doubt that coal will become a major marine fuel in the future, not only for bulk carriers in selected trades but also generally in the short sea trades.

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Discussion

Commander K. I. SHORT (Keith Short Associates): Those who like to consider themselves to be practical engineers are often automatically suspicious of proposals made by pure academics, and it can be difficult for the latter to muster credibility for their ideas. Dr Sharpe is an academic, but nowadays far from a pure one.

To my knowledge he has been peddling the proposals in this paper around the bazaars for over four years. A period long enough, with similar enthusiasm and application, for him to qualify for a degree in another discipline if he had been so minded.

But Dr Sharpe wasn't so minded, and he has subjected himself to the slings, arrows, criticisms, doubts and advice polite and impolite of many shipping managements and their technical staffs in the development of the proposal which he has presented in his paper.

I think therefore that, having voluntarily been infected by such extensive contact with commercial marine people and of course marine engineers, he has earned the right for his proposals to be considered seriously as having been developed beyond the stage of backroom cycle riding.

Dr Sharpe has cleverly avoided the three major logistic difficulties advanced by the oil lobby, in which I include diesel-engine manufacturers. These are:

1. The additional cubic and deadweight needed in a deep-sea vessel for coal bunkers.
2. The current lack of worldwide coal-bunkering facilities.
3. The problems of burning coal of different qualities in the same boilers.

The current yo-yoing of oil prices, the worldwide reduction in the use of oil in response to the massive hike in oil prices in 1972, and the subsequent extraordinary and successful efforts of diesel-engine designers in coping with the low-quality heavy oil fuels, which ship owners are commercially constrained to use in motor vessels, have tended recently to obscure interest in the coal firing of ships.

But as Mr Heath said at the first International Coal Fired Ship Conference in April 1980: '. . . in the next twenty years the world population is going to increase by 50%. The developing world, which has the larger part of the world's population, has 50–60% of people under eighteen. The Catholic population of South America and Central America has the fastest growth. Brazil for example is expected to nearly double in twenty years.

'This population explosion will cause an acute shortage of food in some areas and, in conjunction with industrial explosion in the developing world, will in addition cause acute shortages of raw materials and energy.'

Even the optimists now admit that oil production will become more difficult in the years to come and thus not only more scarce but more expensive.

Thus I believe that in the long run, which may not be too far distant, we shall have no alternative but to turn to coal as fuel for ships, and we ought to be planning for this now. Let us not again be led astray by the blandishments of oil companies and ignore the clear writing on the wall regarding the likely reduction in oil availability in the not too distant future and a concurrent increase in demand for energy.

So I support Dr Sharpe's crusade to get something moving for short-sea-passage vessels as a start. However, no-one wants to risk making a mistake these days in selecting a ship propulsion system.

Not so long ago, when more vessels were being constructed, the odd experiment could be permitted. If it failed it wasn't the end of the world and those responsible for the failure could take satisfaction and credit in having tried something new. Nowadays shipping management and technical staff are not being recruited, and a mistake in the only ship being built over a long period could be the kiss of death for these responsible. So it is not surprising that companies and their advisers will probably 'play it safe' and do what everyone else is doing.

If Dr Sharpe's proposals are to get anywhere they have to fire the imagination of top management, who do not risk the sack if they fail. For this to be accomplished the proposals must be presented in a form which wins immediate credibility from these princes and their advisers. It must be borne in mind that Dr Sharpe is 'preaching to the converted', but it is the unconverted he has to excite and convince (in that order) to the point that they will put up some money.

It may be, and probably is, my ignorance which leads me to criticise and ask for help in understanding certain aspects of this paper. In doing so, however, I stress that I support its aim, and my comments are directed hopefully to improve rather than detract from the thrust of the presentation.

As a matter of tactics I would have preferred to see this proposal presented entirely on its own merits as being superior to the grate-fired boiler or fluidized-bed steam-turbine arrangement for the particular vessels being considered. It does not help the cause of coal to give ammunition to the oil lobby by appearing to 'knock' the deep-sea coal-fired vessel alternatives' operational experience so casually as has been done several times in the paper.

Does the steam engine proposed exist or is it just a gleam in Dr Sharpe's eye? I wonder if he had the advantage of a steam-engine designer running the rule over his proposed design. Recently the Polish Steamship Company was reported as being interested in short-sea Baltic CF bulk carriers and train/lorry/passenger ferries with Skinner engines, working models of which exist.

I admire Dr Sharpe's courage in suggesting a steam engine which will be a direct replacement for a similar output and torque diesel engine, but more information is needed.

Very careful thought needs to be given to the pitching effect on a locomotive boiler. There are recent cases on record of fire tube boilers designed for land use being fitted in ships, for auxiliary purposes, which suffered tube overheating when pitching.

It is perhaps of interest to note that the IOM Steam Packet Company recently investigated prices worldwide for a petroleum coke fire fluidized-bed/steam-turbine installation for a ferry. I understand that it was established that running and maintenance costs for the CF vessels were very much lower than for the diesel alternative.

This was a very serious project and was technically backed by larger boiler and turbine manufacturing companies and coordinated by one of the largest UK consultant firms, which was entirely satisfied as to the viability of the project and the 'numbers' generated in the investigation.

It seems a great pity to many of us that:

1. The Central Electricity Generating Board opted for diesel propulsion in its recently contracted short sea colliers.
2. Dr Sharpe has not so far been able to obtain the necessary backing to authenticate his proposals in practice.

In conclusion, I once again float my own suggestion for the UK to join Spain, Japan, Korea, Italy and USA in the development of a modern coal-fired vessel. I would like to see the Government encourage a shipping company to place an order for a CF (preferably a fluidized-bed boiler) vessel in the UK. The Government would finance the vessel and she would be bare-boated to the owner at the rate applicable to a cheaper diesel-engine installation. This would develop CF expertise in the UK, provide work for UK shipyards, and establish the commercial factors of a design. Also the Government would still, as it wishes, not be ship owners.

Because shipping only uses a small percentage of the world energy resources, the amount of money which it is commercially viable to spend on research on boilers for ships is limited. So I feel some Government help is well warranted.

If I have spoken too long I apologise. My excuse is that John Sharpe did the same thing to me when I allocated him five minutes to speak at a CF Ships' Conference!

P. HOLBROOK (Lloyd's Register of Shipping): Dr Sharpe has pointed to the shortcomings of the first generation of coal-fired ships. I have just come back from Australia and was fortunate to be able to speak with several of the engineers serving on these ships. It is true that inconsistent coal quality has caused one or two problems, mainly to the coal-conveying systems where blockage has been all too frequent, but as far as the boilers, turn-down ratios and all other aspects of the ships are concerned the engineers were very happy. The difficulties with coal quality are almost all attributed to the shore-side coal preparation and loading facilities, which haven't performed very well. I am pleased to be able to report that the most critical aspect of obtaining mass flow from the bunkers has not been a problem.

The paper implies that the current coal-fired ship machinery designs are not sufficiently compact for modern vessels and it is true that the rotary spreader stoker-fired boilers are some 40% larger in furnace volume than an oil-fired boiler of similar output. There is also additional machinery in terms of ID fans, daily coal storage and ash systems to be considered. In general this has resulted in the machinery spaces of the current ship designs being larger by one or two frame spaces than the modern high-efficiency diesel-engined ships.

However, this factor is in some respects insignificant compared with the loss of cargo volume necessary to cater for the coal bunkers, which unlike oil fuel cannot be carried in double-bottom tanks.

I am a little unhappy about ship designs which appear to incorporate coal bunkers in side tanks or other seemingly available space. Achieving mass flow from bulk storage systems is not easy and such designs are likely to require extremely complex conveying machinery which will add to first costs and subsequent maintenance requirements. Rather than stressing the necessity of achieving compact machinery designs in isolation, the whole system from main bunkers right through to ash storage and disposal must be considered overall.

One dimension which is becoming increasingly important as far as ro–ro and thro-deck ships are concerned is the overall height of the machinery, which Dr Sharpe's design appears to have catered for.

Another aspect which appeals to me is the free-running turbo-compressor, which certainly simplifies the gearing arrangement compared with the older pass-out concept. An added advantage would be to reduce the engine revs and dispense with the reduction gearing completely. Since with a free-running turbo-compressor the engine can be reversed, why is it necessary to include controllable-pitch propellers?

From a classification point of view, can the turbo-compressor be by-passed both in terms of engine operation and from the condenser loading for prolonged periods of running and how complicated would such a change over be?

With regard to the boiler combustion system, it would appear that to allow some flexibility in coal quality and size will necessitate quite large alterations in primary air-flow rates, in some cases approaching bed fluidizing velocities. This in turn would have the effect of increasing eludation, higher dust loading on the 'Houlet' superheater section and, perhaps more importantly, increasing unburnt carbon losses. It is noted that there does not appear to be any provision for unburnt carbon grit reinjection, so wouldn't this have an adverse effect on the boilers thermal efficiency?

As well as the present six ships in or about to enter service to Lloyd's Register's Class, there is considerable interest in both small coal-fired bulk carriers and ro–ro ships, which if all goes well should extend the present series of new generation coal-fired ships.

I would like to say that this paper continues the current trend of the excellent work being done on the re-introduction of coal-fired ships, and I read it with considerable interest and enjoyment.

A.F. HODGKIN (Babcock Power Ltd): Dr Sharpe's proposals for a boiler and combustion system capable of extreme compactness are clearly worthy of attention but should not divert effort away from other alternatives. Compactness is something of a necessary evil in many cases since it also implies restricted access for maintenance, which can soon lead to operational difficulties. There will be many coal-fired ships where this degree of compactness is not necessary, allowing alternative methods of burning coal to be employed.

Still in its infancy as far as steam generation is concerned, fluidized-bed combustion can deal adequately with lump coal and it should not be discounted as Dr Sharpe implies in his paper. Some of the disadvantages he suggests can, with development, be overcome or even turned to advantage. Poor turn down and grit elutriation fall into this category. The loss of radiant heat transfer is also compensated by enhanced heat transfer elsewhere.

The locomotive-type boiler is very compact, although the firebox arrangement is considered to be far from ideal. On the locomotive there is perhaps no alternative but for shipboard use it should be possible to substitute a combustion chamber enclosed by panels of water tubes known as membrane construction.

A more detailed description of the 'Houlet' superheater would be welcome, as it appears that 'Monsieur Houlet' had taken a 'holiday' to obtain a final steam temperature of 550 °C with a final gas temperature of 500 °C.

However good Dr Sharpe's proposals may be, what is desperately needed is some practical application, and it is a pity that he could not influence the CEGB during the design stage of their 19000 ton colliers, as this would appear to be a very suitable vehicle for demonstration purposes.

Dr Sharpe has made considerable progress since first floating his ideas and we are all grateful to him for presenting his paper so efficiently on this occasion.

Professor R. O. GOSS (University of Wales Institute of Science and Technology): When considering different power plants for ships it is desirable to consider the ships as systems, ie in their entirety. This is particularly important when, as in the present context, the differences are considerable. Thus, coal has a much lower calorific value than oil, so more must be carried. If the bunkers are to be larger, does Dr Sharpe envisage the ship being the same size and carrying less cargo, or does he envisage the hull being larger and thus more expensive? Has he examined the economic effects of either of these?

Again, there may be effects on the ship's operating pattern. For example, oil fuel may conveniently be taken (perhaps from a fuelling barge) whilst working cargo. Does Dr Sharpe envisage this being done with coal and, if so, how? If not, has he examined the economic effects of the coal-fired ship steaming to a bunkering berth and spending time there?

If economies can be achieved, taking all relevant factors into account, by substituting coal for oil fuel then, inter alia, they may vary directly as the proportion of time spent at sea. This suggests that Dr Sharpe's examples of Australia–Japan and even a 3000 mile round voyage in Australia are appropriate, but his example of cross-Channel ferries seems less so.

Whilst, therefore, Dr Sharp's very interesting paper provides a fascinating technical account of the developments he favours, there seems to be room for further exploration of the overall economic effects of his proposal, eg by simulating a variety of circumstances for the ship as a whole and using profitability as a criterion through the standard discounted cash-flow method. Such variations might include ship size and type, route length and proportion of time spent in port besides the more obvious items such as the relative specific costs of oil and coal. Such comparisons are neither difficult nor time consuming, and it is to be hoped that Dr Sharpe will produce another paper on them in due course.

Dr M. GARRATT (Liverpool University): Dr Sharpe's paper provides a very interesting interface between the technical and economic aspects and opportunities of operating coal-fired ships. My comments are restricted to economic aspects and I take the technical relationships in the paper such as grammes of fuel per horsepower hour as given. I also accept the bunker prices quoted, although their continual fluctuation is a consideration in itself.

The case made for coal-fired ships is summarized by the argument that coal-fired engines can produce one thousand kWh for $16.23 worth of fuel, against $33.03 for oil-fired engines, and that the extra capital cost of coal-fired engines ($250 per kW) can therefore be recovered within 15 000 h at sea. There are three principal complications. First, a vessel may not actually spend sufficient time at sea to reach this figure; secondly, the higher weight of coal bunkers will reduce revenue earning cargo; and thirdly, reaching coal-bunkering facilities may involve extra steaming.

I shall use Dr Sharpe's illustrative vessel of 6000 tonnes displacement to illustrate this, and assume its carrying capacity (deadweight) to be 4500 tonnes. Statistical analysis conducted by the Marine Transport centre suggests that in European conditions such a (general cargo) vessel would make about 60 voyages p.a. of an average 700 miles each, which at 12 knots involves 3500 h at sea. This would require the consumption of 3360 tonnes of coal or 1424 tonnes of fuel oil. If we further assume that the daily charter cost of such a vessel is $2000 per day, and that half of all sailings are in ballast, we can make the estimates of sea freight costs (exluding bunker costs in port) shown in Table DI.

Let us assume that attractive bunker prices can be found at the end of every fourth voyage (every 2800 miles or 9.7 days at sea), and that a 25% 'safety' margin is also carried. This implies carriage of 280 tonnes of coal or 119 tonnes of oil, reducing effective deadweight per laden voyage by some 196 and 83 tonnes, respectively. Annual vessel 'capacity' therefore becomes (30 laden voyages p.a.) 129 120 and 132 510 tonnes, respectively. We can now calculate overall sea freight costs per tonne as:

- Coal fired $8.84 per tonne,
- Oil fired $9.08 per tonne.

In this case, therefore, a coal-fired ship would offer a saving of 2.7% over an oil-fired ship, or some $31 000 p.a. If coal bunker prices rose by $10 per tonne, oil bunkers fell by $22 per tonne or the cost of vessel construction was a further $250 000 (quite likely given that a shipyard would wish to insure against unanticipated difficulties in developing an 'unusual' vessel) then the coal-fired ship loses this advantage. Equally, if the poor availability of coal bunkers required an extra round voyage of 800 miles three times p.a., the advantage would be lost. It may, in consequence, be difficult to persuade an operator to choose a coal-fired ship of this size and type.

It is much more difficult to argue against the concept of a coal-fired ro–ro ferry. Given that such vessels operate on fixed routes, are powerful and in consequence consume large volumes of bunkers, it would appear economic to provide radily available bunkers. Such ships are rarely deadweight constrained and there are large areas of 'wasted' spaces onboard where coal bunkers could be stowed. Providing that the capital costs of providing bunkering facilities were reasonable, and that such bunkering could be achieved rapidly, there would appear to be an interesting case for coal-fired ferries.

The strength of the case is essentially based on utilization. A Dover–Calais ferry would spend 4000 h p.a. at sea and consume over 10 000 tonnes of fuel oil p.a. The case may be strongest for an intensively used freight ferry of say 15 000 shp, where the current ratio of annual bunker costs to capital costs could be 1:8, as compared with 1:20 for a coaster.

Table DI: Sea freight costs

Item	*cost* ($) *coal*	*oil*
Annual charter cost	730 000	730 000
Bunker costs in port	4200	10 000
Bunker costs at sea	134 400	263 440
Additional capital cost discounted over 15 years at 10% p.a.	72 518	—
Port disbursements p.a. (est.)	200 000	200 000
Total	1141 118	1203 440

Given also the level of uncertainty using 'new' technology, it will probably be necessary for coal-fired ships to develop on routes where regular bunkering facilities are guaranteed and where the estimated savings on a given vessel are large enough to exceed the perceived risks of using new designs. These conditions may well be most easily met in the case of such a freight ferry which is at sea for 15 h per day and currently consumes some 11 000 tonnes of fuel oil p.a. Bunker savings for such a vessel could reach $1M p.a., against an additional capital cost of $2.75M, or $0.36M p.a. discounted over 15 years at 10% p.a. Such a case may be the most promising area of opportunity for Dr Sharpe's suggestions.

A. M. DYSON (Senior Green Ltd): Experience has always limited steam temperatures to about 410 °C (770 °F), at which the lubricating oil starts to break down and form varnish deposits on the valves and H.P. piston rings. How has this problem been avoided in the medium-speed design, which Dr. Sharpe suggests has operated successfully at a steam temperature of 550 °C (1050 °F)?

D. M. WILLIS (National Coal Board): Can Dr Sharpe give an indication as to the coal specification required for the shallow fixed-bed gas producer proposed in his paper? Experience suggests that fixed-bed gas producers are selective in terms of coal size, swelling characteristics and ash characteristics. It is also possible that mineral matter in coal could cause deposition problems on the 'Houlet' superheaters which would be difficult to remove.

What does Dr Sharpe think of the potential application of coal liquids, including coal + water mixtures in this field?

Dr A. MOULTON: Dr Sharpe is to be congratulated on the range and depth, both technical and economic, of the study which his paper displays. Indeed one could add how fortunate are the students who have participated in the work, and especially in the design, which shows much innovation. However it all derives from known elements and does not postulate stepping over the existing thresholds of temperature and efficiencies found in other machinery.

To me, the most fascinating aspect of the proposition is the use of a shell boiler with its pressure limitation. I understand that the choice was economic and flows from international standards relating to pressure. Surely these must be archaic and heaven forbid but a rupture of a tube of a once-through boiler at 1000 lb/in^2 must surely be less disastrous than a shell boiler at a third of the pressure? Indeed one might ask if high pressures within the cylinders of a diesel carry some statutory penalty?

The consequence of the low pressure on the engine design is interesting, insofar as the power density of the one-stroke steam engine is apparently similar to that of the four-stroke diesel using a common crankcase and crankshaft. In my own experiments on a diminutive scale, the small size and mass of the steam engine with its two-stroke MEP approaching 300 lb/in^2 is remarkable in comparison with that of any IC engine. Thus in Dr Sharpe's proposals, if higher pressures were able to be used economically from the point of view of the steam generator, what would be the economic benefit of a more compact engine?

These questions apart, the economic attractions of the proposals are such that one can only hope that the NCB would support a trial installation of his steam machinery in a ferry, if only to gain the goodwill of the passengers by being able to claim that the quiet running of the ship was due to the 'burning of British coal'.

Author's reply

Commander Short has raised a number of valuable and interesting points in relation to the successful development of coal-fired ships generally and the use of the compact proposal in particular. He has also asked about the economics of coal-fired vessels.

Commander Short rightly points out that a number of proposals to build coal-fired tonnage have failed because of the very high additional capital costs. The power plant proposal is specifically designed to have to lower capital cost.

Far from avoiding the logistic difficulties of coal-fired ships, I have specifically addressed those classes of vessel that can readily benefit from coal firing and have proposed a combustion system that has demonstrated its ability to burn a wide range of steam coals.

I have nothing but admiration for the first generation of coal-fired bulk carriers, which have shown conclusively that coal firing is an entirely satisfactory modern means of propulsion. My concern is that it is not appropriate for smaller vessels operating in the short sea trades where there is a great potential for coal firing. Whilst unfortunately the proposed engine does not exist, much of the technology has already been successfully used over a number of years. The concept of turbo-compression has been used for many years with more than 100 vessels having the Götaverken and Elsinore system. Much of the understanding that has gone into the design and development of modern diesel turbochargers and more recently exhaust gas turbines is directly relevant to the steam turbo-recompressor and has been used in preparing the design shown.

The turbo-compressor may be readily by-passed if required as is done when operating at very low output. At the designed steam flow the output power will be reduced by some 30% as the engine is operated as a two-stage component.

The flexibility of the steam plant to provide a wide range of torques and speed enables the convenient use of fixed-pitch propellers if desired.

I agree with the need for careful design of the marine boiler to take into account the pitching of the vessel. The boiler shown has well proportioned thermic systems in the firebox to overcome this problem by providing a constant flow to the crown plate and tube bank.

I fully agree with Commander Short's view on the need to provide some initiative to allow Britain to exploit both its substantial coal reserves and its technical ability and expertise.

I agree with Mr Holbrook's comments on the provision of bunkers and in the designs shown I have provided simple and accessible bunkers. The ability of the gas-producing combustion system to operate for up to 30 min without coal feed is an important improvement over the spreader starter. The combustion system is designed on the basis of a good-quality steam coal as this is the most economic, particularly from a bunkering point of view. If a lower quality, more volatile coal is used, both the priming air and the eludation are reduced. However, if an anthracitic fuel is used the priming air will need to be increased but only then to one-third of the total combustion air, still well below that required for fluidization.

There is 'reinjection' of unburnt carbon in the form of the cyclonic secondary combustor. Not only does this provide secondary combustion but also allows any unburnt char lifted from the fuel bed to be returned through sub-vortices in the corners of the combustion space.

I agree with Mr Hodgkin that there is some merit in the use of a membrane wall construction for the combustion chamber. However, I feel that the use of stayed water walls is likely to be cheaper and provide more generous water spaces.

Mr Hodgkin has also asked about the arrangement of the 'Houlet' superheater. It was first designed by Monsieur Houlet in 1890[1] and used extensively, particularly by Monsieur Chapélon, for locomotive boilers. Chapélon made detailed studies of superheaters which showed that the heat-transfer capability of the Houlet type was three to four times better than any other fire-tube superheater.[2]

The 'Houlet' superheater is the nearest possible form to a true counter current heat exchange and for that reason it is possible to have an exhaust gas temperature below that of the stream outlet temperature. The superheater is placed in the boiler fire-tube and takes the form of three concrete tubes with the steam passing counter to the combustion flue gases though the narrow annulus formed between the two outer tubes to return through the central tube. The heat-transfer rates are much greater in the annular counter-current section where the Nussault numbers are highest with the steam temperature rising to about 650 °C, which is maintained for two-thirds of the return path after which it is atempered by the cooler flue gases.

Professor Goss and Dr Garratt have both raised the question of the overall economics of coal-fired vessels, which are demonstrated by increases in the capital cost of the plant and machinery and the need to carry a larger volume of bunkers. The question of the greater value of bunkers largely determines the types of ships that can potentially benefit from coal firing, operating on fixed short sea routes.

However, whilst the spreader starter fired water-tube boiler and steam-turbine combination is well suited to bulk carriers down to 35 000 dwt, it is clear that an alternative low-cost compact plant is required for the numerous small short sea route dry bulk carriers and the freight and vehicle ferries, as suggested in the paper.

Dr Garrett's figures also illustrate the need particularly in European waters for the lowest capital cost.

Mr Dyson has raised the important question of cylinder and valve lubrication at steam temperatures of 550 °C.

Several 'automotive' engines as well as plant used for solar power stations in this country, the USA, Australia and Spain operating at temperatures above 500 °C, and locomotives operating at temperatures well above 470 °C in many parts of the world at different times over the past 30 years, have shown that lubrication is not a major problem, given the correct design of valve and piston rings and the correct, usually synthetic, lubricating oils. The major concern is the correct metering of the lubricant and its subsequent recovery.

Finally, Mr Willis has asked about the coal quality, to which I have referred in my earlier comments, and also raised the question of coal + water mixtures. There are no practical difficulties of firing the compact power plant with a wide range of residual fuels. Whether it is economic is another question.

One of the major attractions of the proposed compact steam plant is that a ship designed to take the plant has the widest possible fuel options. From the views expressed and my own study, I believe it is important that steps are taken to build a prototype power plant at the earliest opportunity with a view to an installation in a ro–ro vessel before the end of the decade.

1. E. Sauvage et A. Chapélon, 'La Machine Locomotive'. *Lib. Polytechnique*, Ch. Beranger, Paris (1933).
2. A. Chapélon, *La Locomotive à Vapeur*. J. B. Bailtiére et Fils, Paris (1938).