# TRANSACTIONS (TM)

The Institute of Marine Engineers

PAPER No.4

# A MARINE FLUIDIZED BED WASTE HEAT BOILER
## DESIGN AND OPERATING EXPERIENCE

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ISSN 0309-3948
Trans 1 Mar E (TM) Vol 91 Part 3
Paper No 4.

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*All papers are indexed in the British Technology Index*

Paper read at the Institute on Tuesday, 10 October, 1978

# A MARINE FLUIDIZED BED WASTE HEAT BOILER
## DESIGN AND OPERATING EXPERIENCE

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SYNOPSIS

The paper describes a project in which a fluidized bed heat exchanger is being used as a marine waste heat boiler for the first time. The shipowner's requirement was to retrofit a waste heat evaporator unit, linked to the existing auxiliary boilers in a 32 000 dwt product tanker to generate saturated steam for cargo heating, tank cleaning, fuel heating and hotel requirements. The waste heat was to be recovered from the exhaust of a 8800 kW (12 000 bhp) slow speed diesel. The paper describes the considerations which led to the choice of the fluidized bed boiler and the problems which were envisaged in transplanting a shore based technology into the marine environment, including the disturbing effect of ship motions on the fluidization of the bed. Laboratory testing on a ship motion simulator enabled the design to be developed to overcome this problem. The paper describes the boiler design and its installation aboard the ship in a bypass to the main engine silencer. Details are given of the boiler's performance over the first year of operation, together with the problems encountered.

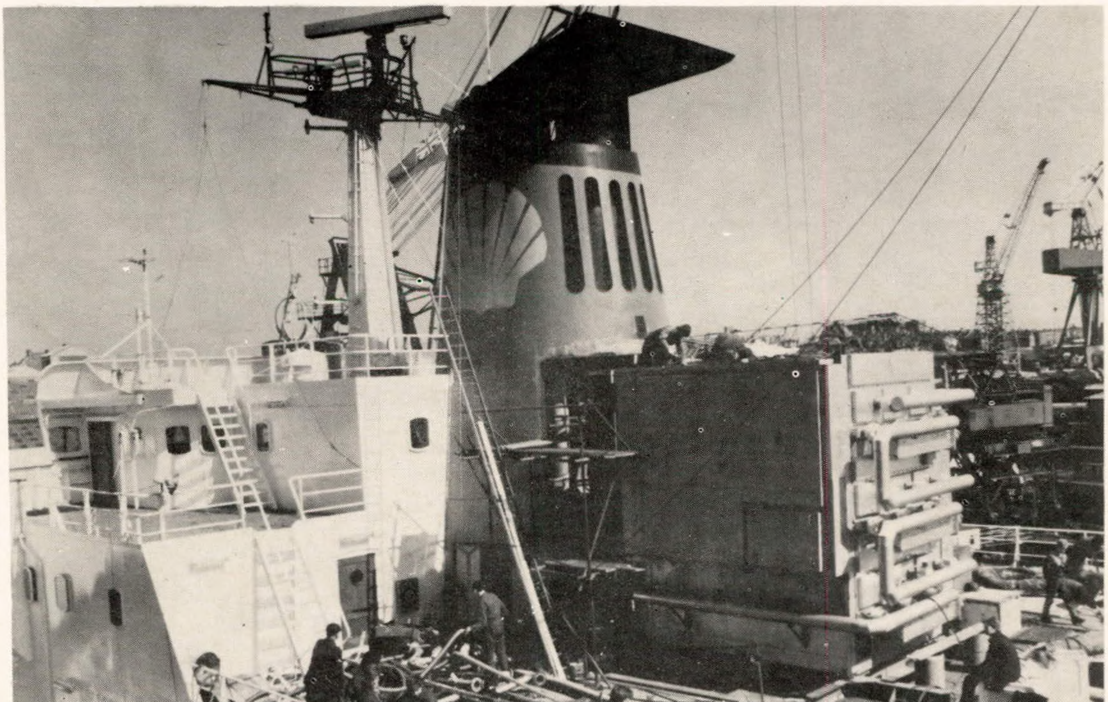

FIG 1 *Installation of fluidized bed waste heat boiler on funnel deck*

* Shell International Marine Ltd

** Stone-Platt Fluidfire Ltd

## REASONS FOR RETROFITTING A WASTE HEAT BOILER

The vessel chosen for the fluidized bed waste heat boiler project was the m.t. *Fjordshell*, a product tanker of 32 465 tonnes deadweight, built in 1974 by Haugesund Mekaniske Verksted A/S in Norway and classified by Det norske Veritas. The vessel is powered by a 6 RND 76 Sulzer engine of 8826 kW (12 000 bhp metric) driving a KaMeWa controllable pitch propeller at 122 rev/min. Three diesel driven electrical generators meet all the electrical power requirements of the vessel. Dry saturated steam is produced by two oil fired Aalborg AQ3 boilers, each rated at 12.0 tonne/h steam at 1.226 MPa (12.5 kg/cm$^2$) pressure. The steam is used for driving cargo pump steam turbines, cargo heating, tank cleaning, fuel heating, engine room and domestic services.

The *Fjordshell* was built for trading around the Scandinavian coast, loading in one of three refineries and discharging at numerous customer and distribution installations. Consequently voyages are short and port time relatively high. The designed trading pattern for the vessel was such that cargoes requiring heating would be few, and the then relatively cheap cost of fuel made the installation of an exhaust gas waste heat recovery boiler uneconomical. During building, however, extra cargo heating coils were installed following a review of the anticipated trading pattern which necessitated additional cargo heating and tank cleaning capacity. The operators of the vessel were thus considering that an exhaust gas waste heat recovery boiler would be beneficial in reducing the additional boiler fuel consumption at sea and studies were made of the economics of retrofitting such a unit.

### *Choice of Fluidized Bed*

Coincident with the above studies, Shell International Marine Ltd were considering with Fluidfire Development Ltd (later Stone-Platt Fluidfire Ltd) the application of the latter's design of shallow fluidized bed heat exchangers as waste heat recovery units for marine use. Such units were already in use as economizer sections of shore based gas fired steam boilers (1).

It was evident from these feasibility studies that the fluidized bed principle offered some advantages when compared with conventional extended surface heat exchangers. These were:

i) The heat transfer rate between the gas, bed particles and tube material is some four to five times that of the conventional gas to extended surface metal heat exchanger. This means that less heat transfer surface is required for the same steam production, which could result in a smaller and cheaper unit.
ii) Because of the shallow bed design and high heat transfer rate, the heat transfer takes place in one row to tubes. This one row can be divided into parallel beds, as described later, and separated by gas spaces allowing easy access for inspection and maintenance.
iii) The continual motion of the bed particles around the finned tubes in the bed prevents the build-up of any fouling on the tubes. Fouling would occur on the underside of the gas distributor plates supporting the bed, which being flat are easier to clean than finned tubes.

The use of fluidized beds for the combustion of coal or oil in marine use is being investigated by several organisations (2), but no fluidized bed had as yet been to sea. It was considered that building and operating a fluidized bed exhaust gas waste heat boiler would demonstrate the advantages, and the disadvantages, of this type of unit in comparison to a conventional waste heat boiler. It would also demonstrate the reaction of a simple fluidized bed to the marine environment before more complex coal or oil fired units were constructed. Ideally, it would have been preferable to build a prototype waste heat unit for use with a land based diesel engine. However, the urgent requirements of the *Fjordshell* for a retrofit exhaust gas boiler offered the opportunity for an excellent floating test bed facility, other considerations being:

a) it was an existing ship without waste heat unit;
b) permanent trading on the Scandinavian coast allowed reasonable access from the UK for inspection purposes and the carrying out of development modifications.

## DESIGN CONSIDERATIONS

The basic principle of a fluidized heat exchanger involves the use of a bed of refractory particles contained in a chamber. The bed is fluidized by passing an upward stream of gas (e.g. engine exhaust gas) through a perforated distribution plate in the base of the chamber and through the bed. In the fluidized state the bed submerges the boiler tubes which are arranged to pass through the chamber. When designing a shallow fluidized bed with extended surface tubing (3), there are a number of basic parameters to be decided upon which were considered in depth by the late Professor Elliott of Aston University (4). Mainly it is a balance between a suitably sized particle to give a reasonably high heat transfer coefficient within the fluidized bed and a particle size and density which will accept a relatively

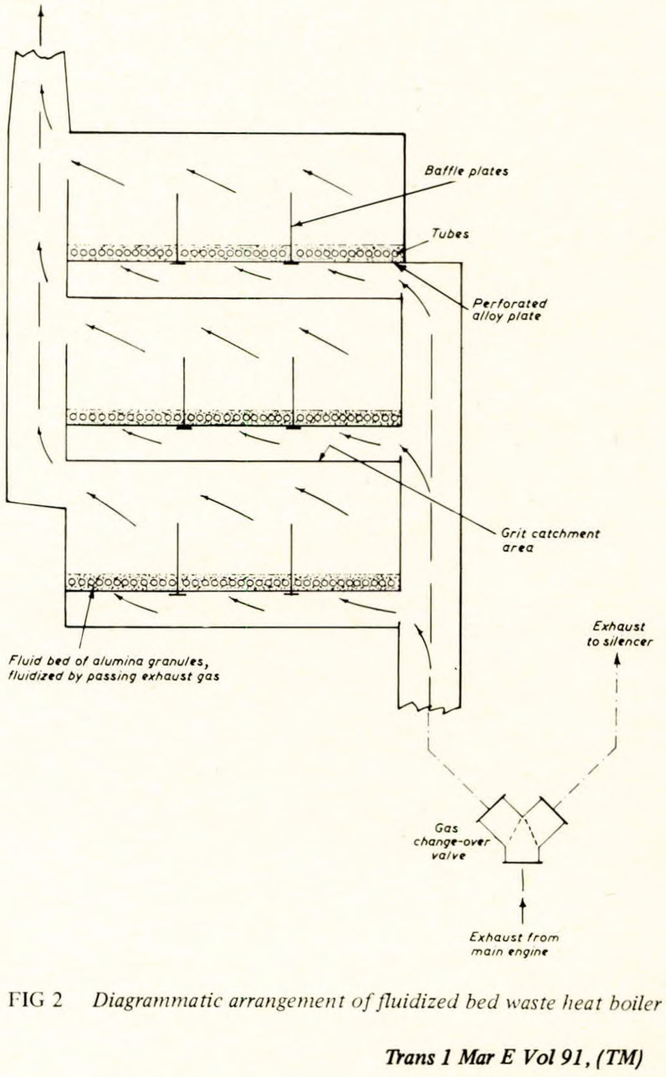


FIG 2 *Diagrammatic arrangement of fluidized bed waste heat boiler*

high gas velocity through the bed without elutriation. Having decided the bed particle material, in this case a high density, high wear resistant and commercially available aluminium oxide, the optimized choice of grit size was to British Standard Sieve No 36, which would normally be operated in the gas velocity range of 1.2 to 1.5 m/s without elutriation. The heat transfer coefficient to the extended surface tubing at these velocities is quite high and the tube fin spacing and thickness can be optimized to achieve maximum steam output for a given area of fluidized bed. It is the gas velocity which fixes the total area of fluidized bed and therefore the total size of the unit. The high gas throughput of this particular boiler made a single fluidized bed an unrealistic size, and it was thus decided to stack the beds as shown in Fig 2, the final choice being for three beds giving a relatively simple design within the space constraints. Each bed comprises a single row of 36 tubes divided into three nests of 12 tubes each. Each nest has its own rectangular water inlet header and can be withdrawn from the boiler individually.

With the size of the fluidized beds decided upon, the next step was to minimize the pressure drop through the bed by choosing a shallow bed depth in which complete immersion of the extended surface tubing could be achieved with the unit running, and selecting distributor plates which would give an adequate flow distribution without excessive pressure drop. This work was carried out in cooperation with Aston University and the experience of a number of industrial installations made the choice somewhat easier. For this application the distributor plates had also to be chosen for their ease of cleaning as it had been decided that the particulate burden of the engine exhaust gas was such that some form of cleaning would be required. A number of possible cleaning methods were considered, including forms of vacuum cleaning, shot blasting, steam cleaning and cleaning with brushes. Eventually, brush cleaning of the distributor plates was chosen because of its simplicity, mechanical reliability, lack of wear and relatively low maintenance prospects. The arrangement for this is shown in Fig 3.

In addition to the foregoing it was necessary to look at the problems associated with dealing with an exhaust gas on board a ship. The exhaust gas flow is pulsating, which combined with the vibration from a slow speed diesel was expected to put considerable stress on the distributor plates, tube supports, cleaning gear and the general boiler structure. Partly for these reasons, it was decided to divide each bed into approximately 810 mm by 900 mm rectangular panels which would each accept a single self-supporting distributor plate. The tubes were also supported every 900 mm. The general structure of the boiler was designed to withstand the pulsating effect of the gases by corrugating the large casing panels to give a reasonable beam strength without unduly heavy structural steelwork.

The other factor which had a bearing on the size of the bed divisions chosen was the effect of ship motion on the behaviour of the bed material. A permanent list or severe rolling in heavy seas could cause the bed particles to flow from one side of the bed to the other, resulting in complete exposure of heat transfer tubing which would be detrimental to the boiler performance. Each complete bed in the actual boiler measures approximately 4.5 m long by 2.75 m wide. This area is divided by baffle plates into the rectangular sections mentioned above. In order to observe the effect of ship motion on the bed material in these sections, a full scale model of one section was built, complete with gas distribution plate and finned tubes. The metal baffles were replaced with glass observation panels and indicator lines were marked on the panels at intervals above the distributor plate. The test section was mounted on a tower assembly and the bed fluidized by air supplied from an electrically driven fan. The entire assembly was then mounted on a Ship Motion Simulator at the Warren Spring Laboratory of the Department of Industry, and subjected to various forms of motion. The behaviour of the bed material was recorded on film. The simulator was able to provide a roll and pitch motion of up to ± 8°, and a total heave of 3 m. These three motions could be controlled independently to give any desired combination, and the time period could be reduced to 5.6 sec. It was thus possible to subject the bed to vertical g forces of 0.385 and horizontal g forces of 0.0198, as well as maintaining a permanent list. The tests showed that the bed material remained very stable, any sloshing being completely retained below 0.2 m of the side baffles. With a permanent list of 8°, some elutriation occurred but a re-design of the distributor plate eliminated this problem and the model has since been tested satisfactorily with a list of 15°, there being no elutriation or

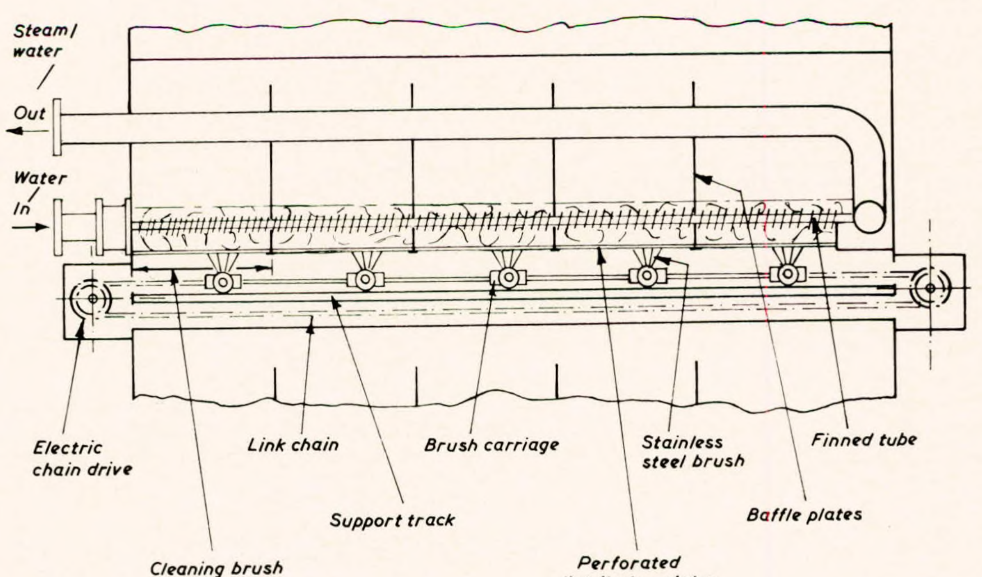


FIG 3 *Arrangement of cleaning system for distributor plates*

serious impediment of heat transfer under these conditions.

With the bed panel size and the tube arrangement selected, it then remained to package the total boiler into the space available on the funnel deck, where constraints were imposed on the height of the boiler house which was not permitted to come above the height of the wheelhouse, and on the width due to the machinery casing bulkheads and ventilation fan ducting. The final boiler arrangement allows the rectangular water inlet headers to be bolted to the boiler structure permitting each of the nine tube nests to be removed from the boiler through the deck house after bulkhead, the latter being provided with removable access plates. Each tube nest has a return header which expands towards the forward casing of the boiler. The return pipes from these headers are brought out through the aft casing of the boiler, thus enabling all external pipe connections to be at one end of the boiler. The floating return header arrangement subsequently provided some expansion problems, although allowances for expansion had been made in the design. The final boiler design is shown in Fig 4.

in engineering. Finally, a plain bearing design was chosen using molybdenum disulphide grease. Most of the structural boiler parts were of mild steel.

The expected performance figures for the boiler as designed are shown in Table I.

TABLE I BOILER SPECIFICATION

| | | | |
|---|---|---|---|
| Engine load | | 85% m.c.r. | |
| Gas weight kg/h | | 74460 | |
| Gas inlet temperature °C | | 320 | |
| Gas outlet temperature °C | | 238 | |
| Steam pressure bar | 8 | | 12 |
| Steam flow kg/h | 2800 | | 2460 |
| Thermal recovery kW | 2073 | | 1829 |

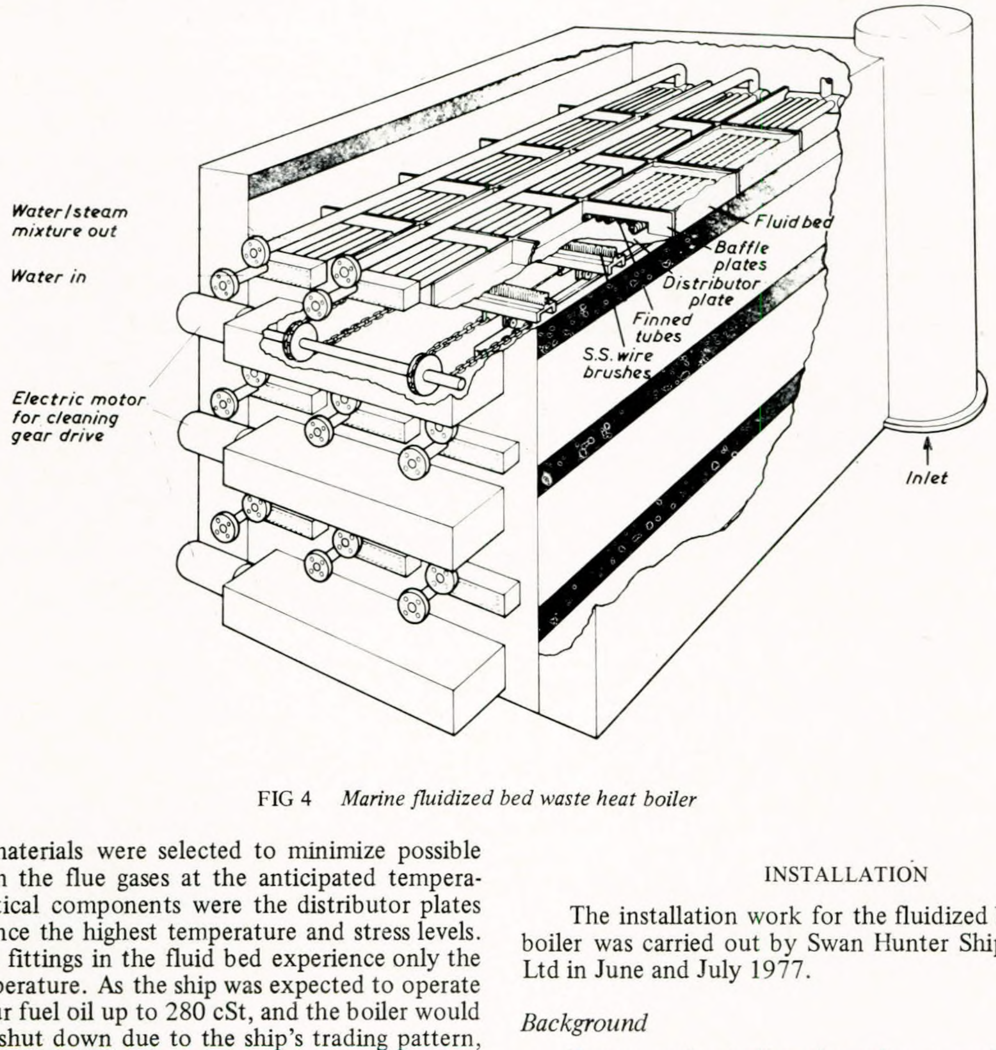


FIG 4 *Marine fluidized bed waste heat boiler*

Finally, materials were selected to minimize possible corrosion from the flue gases at the anticipated temperatures. The critical components were the distributor plates which experience the highest temperature and stress levels. The tubes and fittings in the fluid bed experience only the back end temperature. As the ship was expected to operate on high sulphur fuel oil up to 280 cSt, and the boiler would be frequently shut down due to the ship's trading pattern, it was considered that acid dew point corrosion could be a problem, therefore it was decided to use distributor plates made of a material which could be expected to resist the corrosion that might take place under these operating conditions. As the distributor plate cleaning brushes were also required to operate at high temperatures, stainless steel wire was chosen. Considerable discussion took place in connection with the bearings used on the rollers supporting the wire brush trolleys. Roller kiln truck bearings were considered but were excessively large and caused difficulties

## INSTALLATION

The installation work for the fluidized bed exhaust gas boiler was carried out by Swan Hunter Shiprepairers Tyne Ltd in June and July 1977.

### *Background*

During early studies, the exhaust gas boiler was planned to have five bed sections in a parallel gas path positioned one above the other. It was proposed that the unit should be installed above the main engine maintenance crane at the forward end of the machinery space. A changeover damper was to be provided to direct the main engine exhaust gas to either the boiler or the existing main engine silencer. With this arrangement the boiler would have only been accessible from the aft end, and access to and from the accommodation would have been severely restricted.

A second arrangement was then considered where it was proposed to build the exhaust gas boiler with four bed sections one above the other. At the base of the gas inlet duct to the boiler there would have been a bypass control damper to direct the gases either to the boiler, or through a bypass silencer built on to the side of the boiler casing. This boiler was to replace the existing main engine silencer, and to achieve this within the machinery space would have necessitated tight control on dimensions and the use of a rectangular bypass silencer. Accessibility for maintenance however would have been much better than with the first arrangement. Investigations were made into the design of such a rectangular silencer but predicting the effect on the low frequency noise levels produced by the main engine proved to be very difficult. Supporting a boiler of this shape in the machinery space was also difficult, and access to and from the accommodation and upper parts of the engine room, although improved, was still very restricted. This proposal had therefore many disadvantages, the paramount being that both boiler and bypass silencer would be unproven units, with the operation of the main engine dependent on one or other of the sections.

The third arrangement, the one actually used, was to design the boiler with three bed sections, again one above the other, and to mount the boiler behind the funnel on the funnel deck. The rear of the funnel was to be cut away and a new deckhouse, common with the funnel space built over the boiler. An exhaust gas changeover or diverter valve was to be installed in the existing main engine exhaust below the silencer, directing gas either to the silencer or the new boiler. This arrangement ensured that the new unproven boiler could be bypassed, with minimal alteration to the existing exhaust system. One short length of engine room ladder, which was duplicated elsewhere, had to be removed; otherwise there would be no alteration to engine room accessibility.

*Structural Alterations*

Installation of the boiler behind the funnel meant that the engine room skylight had to be blanked off. This skylight was intended for the removal of machinery parts from the engine room, and a hydraulically operated crane had been provided. Fortunately, this crane also covered a stores hatch on the poop deck, and from this hatch a large passageway led into the engine room. By installing a trackway along the passage and providing a heavy lift chain block, easy removal of parts from the engine room was ensured.

The new deckhouse was prefabricated such that its side bulkheads formed continuations of the engine room casing bulkheads. This meant that the engine room ventilation supply fans had to be moved outboard over the accommodation. In order to avoid intrusion of ducting into the accommodation space and to reduce ventilation ducting bends, the ducts were extended outboard at an angle to the deck and the fans remounted approximately 1 m higher. Fig 5 shows the position of the boiler and the new deckhouse.

*Circulation System*

The exhaust gas boiler is force circulated from the main oil fired boilers, as shown in Fig 6, by sealed, glandless boiler circulating pumps provided with auto start-up facilities for the standby unit, and piped such that each pump can be used with either main boiler. The shells of the main boilers were pierced and pump suction and steam/water return connections fitted.

The ship's original steam and feed water circuit was without a deaerator, there being an atmospheric condenser for the cargo pump turbine and an unlagged hot well. Dissolved oxygen control was achieved by the use of a proprietary brand of hydrazine. Hydrazine injection was, however, difficult to regulate and tests were made on a daily basis. In a ship on coastal trade where steam demand could vary considerably throughout the day, this system was no doubt satisfactory with the type of main boiler fitted, but the waste heat boiler having small bore tubes with forced circulation would probably require a tighter control of dissolved oxygen. To achieve this, an improved hydrazine injection pump was installed. The hot well tank was lagged and fitted with a thermostatically controlled steam injection system to prevent unnecessary undercooling of condensate and additional oxygen absorption, particularly in arctic conditions. Also, a continuous method of monitoring hydrazine reserve did not appear to be suitably available at the time, so a continuous-reading dissolved oxygen meter was installed to sample the discharge from the circulating pumps to the waste heat boiler. It is possible for the watchkeeping engineer to observe any increase in dissolved oxygen level due to changes in steam demand and

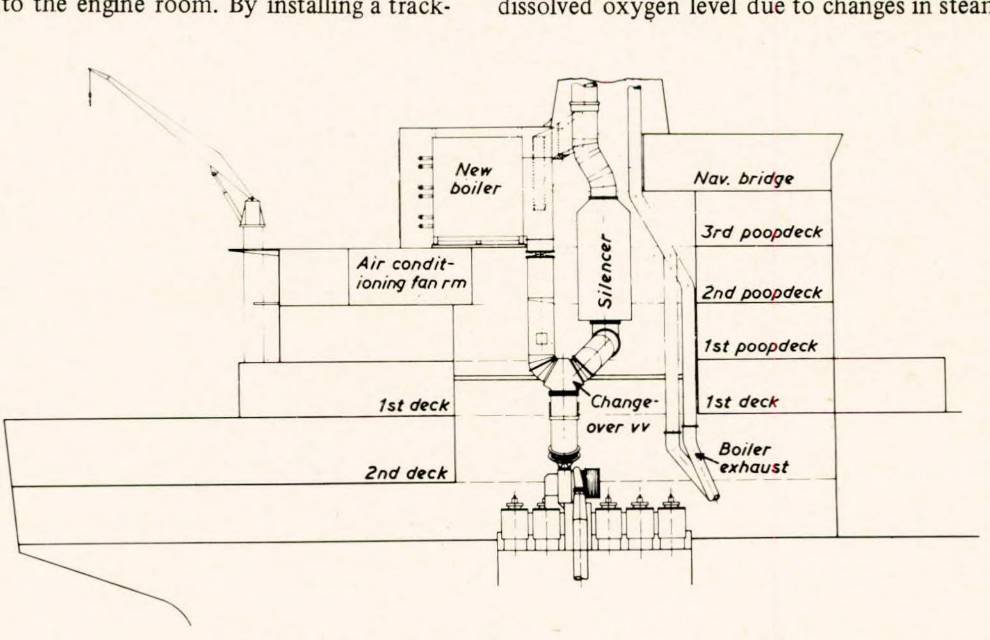


FIG 5 *Machinery arrangement with fluidized bed boiler*

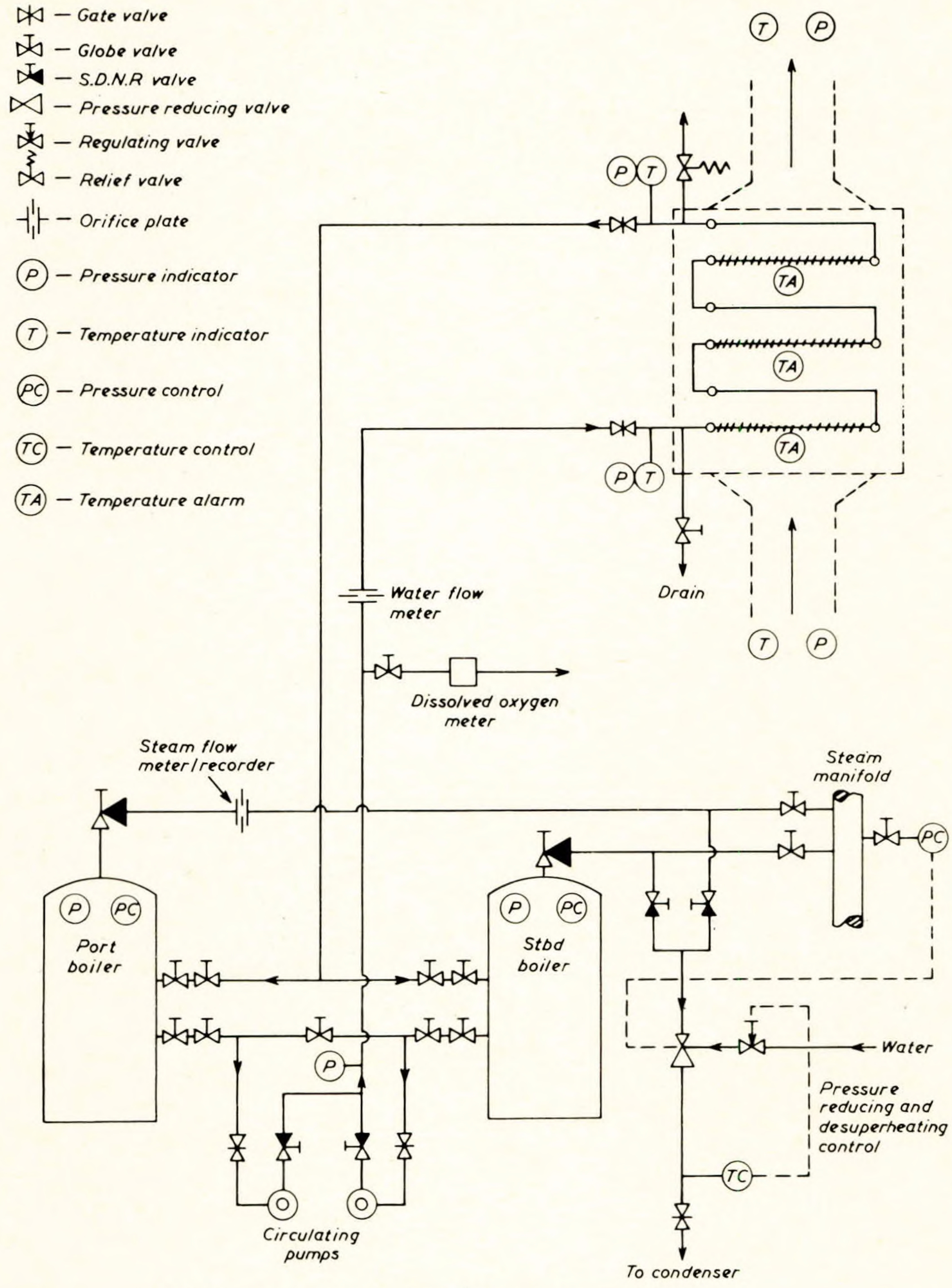


FIG 6 *Waste heat — boiler steam and circulation system*

reduction in hydrazine reserves during the day, and then take the necessary rectification action with the hydrazine injection. This instrument does not, of course, do away with the 24 hourly check of hydrazine reserves.

*Exhaust Gas System*

The exhaust gases from the main engine are ducted to either the original silencer or the exhaust gas boiler via a 1200 mm bore gas diverter valve. In order to limit the bends in the ducting to the silencer, it was necessary to install this valve adjacent to the engine room maintenance crane track. This was achieved by the valve manufacturer making modifications to the operational equipment for the valve before delivery. Nevertheless, installation of this unit proved difficult and time-consuming. The valve is driven by an electric motor operated by the watchkeeping engineer from the engine control room.

Exhaust gas outlet from the waste heat boiler is led back into the main engine uptake above the silencer. No isolating arrangement has been provided here. Expansion bellows are installed in the exhaust duct below the boiler inlet, and in the rectangular outlet gas duct between the boiler and the main engine uptake.

OPERATIONAL TECHNIQUES

The fluidized bed waste heat boiler is integrated into the existing ship's boiler and steam systems as shown in Fig 6. It is normal practice to couple the waste heat boiler to only one main boiler at a time, the shell connection valves on the other boiler being kept closed. The operation of the steam and water circuit is identical to that of a conventional waste heat boiler.

The exhaust gas change-over valve is operated to allow the exhaust gases to bypass the boiler when the engine load is below 60% m.c.r. Above that loading, the valve is opened from the engine control room allowing all of the exhaust gases to pass through the boiler. Steam is generated almost immediately, and the steam pressure control of the main boiler regulates the burners accordingly. Should the steam production be too high with the burners shut down, then the excess steam is dumped to the condenser by the pressure reducing and desuperheating control station, which is provided with a high temperature alarm before the condenser. The exhaust gas change-over valve is not used for steam pressure control as sufficient gas pressure drop must be maintained across the entire bed area to retain even fluidization. If the gas flow were to be used for steam pressure control, then sections of bed would need to be controlled with individual dampers.

## OPERATIONAL EXPERIENCE

The fluidized bed waste heat exhaust gas boiler was started up for the first time on 10th July 1977, and operational checks carried out on the equipment. Unfortunately some of the monitoring equipment had not been received, and accurate performance measurements were not possible at that time. Some adjustments were necessary to the distributor plate cleaning systems, and exhaust gas leakage was occurring through the joints between the rectangular steam header flanges and the boiler casing, causing bed material to be blown into the boiler house. Grit was also found to be dropping through some of the distributor plates into the catchment areas beneath. This was caused by inaccurate machining of the distributor plate resulting in over-sized perforations. Panting of the side inlet duct casing and the bottom of the boiler was also evident. The incorrect distributor plates were replaced and all the tube nests removed and defective jointing material replaced. Some additional stiffening was welded to the inlet duct side casing.

The boiler was put into regular service on 7th August 1977, and up to the end of June 1978 had been operational for 2100 hours. It has been standard practice to use this waste heat boiler whenever the ship is at sea and the engine is operating at more than 60% m.c.r. During this period problems have been experienced in the following areas:-

1) Gas and consequential grit leakage has continued to occur at the joints between the steam headers and boiler casings. The cause of this has been located to a build up of grit behind the tube return headers at the forward end of the boiler, limiting the free expansion of the tubes. This imposes undue strain on the flange joints, causing grit and gas leakage to occur. To cure this, the end structure of a section of the boiler has been modified to provide an angled plate behind the headers so that grit which collects behind the header can be pushed out of the way by the expansion.
2) The cleaning action of the brush gear on the distributor plates has been found to remove burrs left by the punching method, and this has resulted in further grit loss through the plates into the catchment area. These plates have been replaced, and quality control during the plate manufacture has been improved.
3) Some improvements have been required to the drive mechanism of the distributor plate cleaning gear to ensure reliable and even cleaning of the plates. The effectiveness of this equipment is most marked, and only a few strokes are required to clean the plate and thus reduce the engine exhaust pressure before the boiler from 350 mm down to 300 mm $H_2O$. It has been observed that the frequency of use of the cleaning gear has increased over the six month period from November 1977 to April 1978 from an average of 3 hours 20 minutes between operations to 1 hour 40 minutes between operations. This is partly due to a more conscientious effort by ship's staff to operate the gear as soon as the exhaust pressure is at 340 mm $H_2O$ and not to let it increase further. Also, in November 1977 problems were being experienced with the drive mechanism of the cleaning gear which tended to discourage use of the system. With the rectification of these faults, the gear has operated most reliably. The effectiveness of the stainless steel brushes has not been impaired with time in service, as they have consistently brought the exhaust pressure down to approx. 290 to 300 mm $H_2O$ for each cleaning operation since commissioning. As originally installed, the cleaning gear was locally operated from the starters mounted in the boiler house. Reliability has been such that the controls have since been extended to the engine control room. and the system is now operated from there. The advantages of this system over soot blowers are that no warming through is required, there are no steam pipes, steam leaks or water consumption.
4) It has been found that there has been a general loss over a period of time of bed material from the boiler and it is apparent that this material has been blown from the boiler as quantities of bed material conforming in size to 36 grit have been found in the outlet gas ducting.
The measured differential pressure across each bed unit is approximately 22% higher than the design condition, indicating a much higher gas velocity than forecast. This can be attributed to a combination of several factors:

a) The reciprocating motion of the cleaning gear brushes, shown in Fig 3 means that the area of distributor plate under the brush at the end of the stroke is not cleaned. This results in an effective reduction in plate area of about 10%, giving a corresponding increase in gas velocity.
b) The exhaust gas temperature is generally some 34°C to 55°C higher than the design figure. This is partly a result of the increased engine back pressure raising the exhaust temperature by about 15°C. Refer to Fig 7, curves E and F.
An increase in the engine room ambient air temperature of up to 10°C has resulted from the necessity to seal off the engine room skylight when installing the boiler on the funnel deck. Additionally seasonal variations of ambient temperature can also result in increased exhaust temperatures. The cumulative effect of these increases in ambient temperature alone can result in an increase in gas velocity of about 8.5% through the bed.
c) The exhaust gas quantities in practice have been higher than those expected from the engine builder's data sheets. Tests made on the turbo blower performance indicate that there could be a 7% higher gas mass flow and velocity than expected, as shown in Fig 8.
The resultant increased gas velocity through the distributor plates and bed is still well below the velocity at which the bed should be expected to elutriate. In practice the loss of material occurs over a period of several weeks and the main loss is from the section of bed on the gas inlet side of the boiler. It is suspected that the higher gas velocities and possible excessive turbulence at the inlet side have resulted in local velocities being higher still.
Some constructional modifications are now in hand on the boiler to reduce these high velocities and the resultant loss of bed material which

eventually has an appreciable effect on the output of the boiler.

previously it has not as yet been possible to determine any long term effects on the alumina. It is observed that the white "sugary" look of the alumina grit disappears virtually as soon as the boiler is started. Samples of new bed material and material which had been in service for some 600 hours have been compared. It was observed that the used grit had become coated with a black deposit of carbon and a trace of vanadium. Also within the individual size fractions the proportion of darker material increased as these fractions became finer. This could indicate that the smaller grains are more prone to coating with carbon.

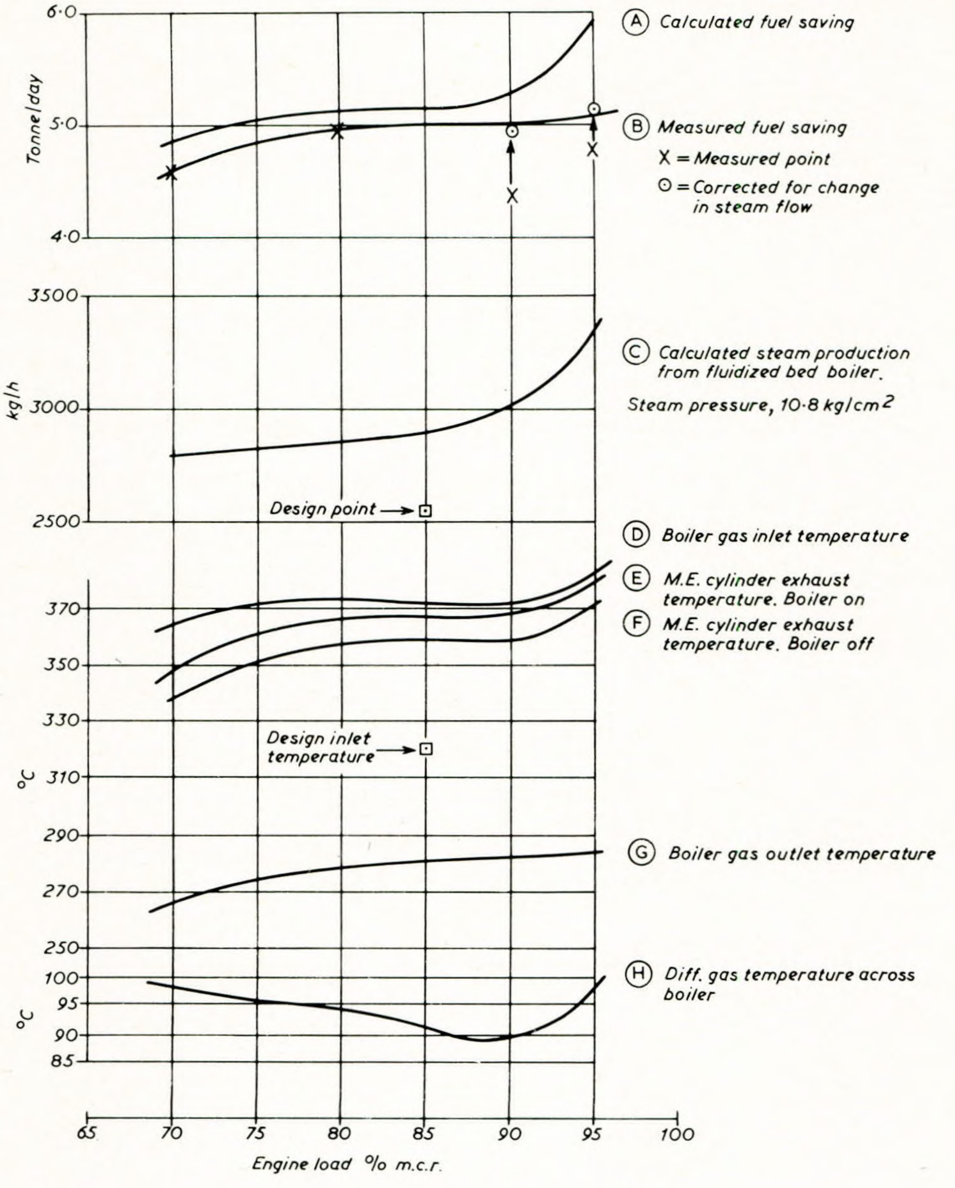


FIG 7 *Waste heat — boiler performance*

*Exhaust Gas Pressure*

At the normal operational condition of the engine at 90% m.c.r., the exhaust gas back pressure at the blower turbine outlet increases from 95 mm to 315 mm $H_2O$ when the boiler is put into operation. When the distributor plates become fouled, this latter figure increases to 365 mm $H_2O$ at which pressure the cleaning system is operated. See Fig 9. Of this total value, at least 50 to 55 mm $H_2O$ is due to the additional exhaust ducting and bends required by positioning the boiler on the funnel deck. The increase in exhaust gas back pressure when the boiler is in use results in a reduction of turbo blower speed of 150 to 200 rev/min. It has also been observed that there is no measurable change in fuel rate to the main engine.

Examination of cylinder liners, exhaust and scavenge ports and general engine condition indicate that there has been no detrimental effect on the engine due to the increased exhaust gas back pressure up to the time of writing.

*Bed Material Life*

Owing to the various losses of bed material described

*Boiler Tube Life*

Interest has often been expressed in the effect on the boiler tubes of the grit motion. Although alumina ($Al_2O_3$) grit is used in the manufacture of grinding wheels, the grit velocity in the fluid bed is only one tenth the normal grinding velocity. A very fine film of carbon builds up on the tubes and the fin surfaces and there has been no indication whatsoever of erosion of the tubes. The arrangement and type of finning of the tubes makes it virtually impossible to micrometer the tube diameters and carbon film.

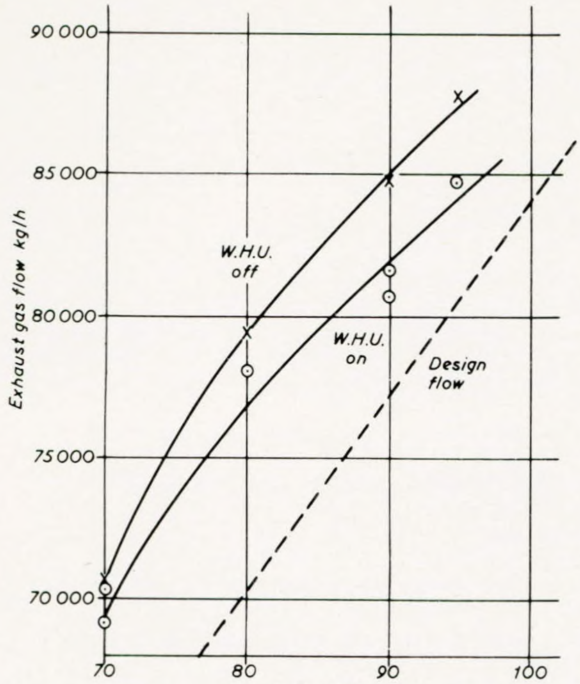


FIG 8 *Exhaust gas flow*

However whenever the tubes are examined after a run they are black in colour and free of fouling. When the boiler has been operating with empty beds, the tubes are usually found to be thick with soot. This soot is soon removed by the bed action once the bed has been refilled with grit.

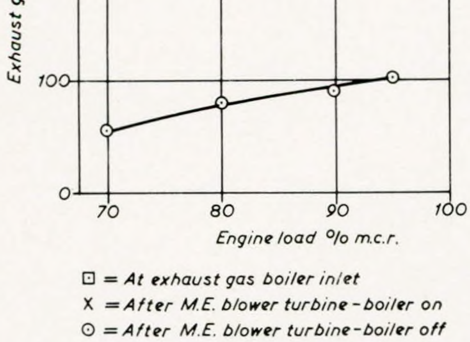


FIG 9 *Exhaust gas pressure*

*Soot Fires*

During the construction stage of the boiler, it had been realized that the action of the cleaning gear brushes might cause a build up of soot in the grit catchment areas below each set of distributor plates. It might be possible for this soot to become a fire hazard and to indicate this, thermocouples were installed in each catchment area. These thermocouples were connected to a panel which would operate an alarm in the engine control room if the temperature in any one of the three catchment areas rose above a pre-set value. The panel would also indicate which of the three areas was affected. To attack any fire in the catchment areas of the boiler, a fixed $CO_2$ fire extinguishing system has been fitted having outlet nozzles at the gas inlets of each of the catchment areas, and supplied by a pair of $CO_2$ cylinders operated manually from the engine room. In practice however there has been no build up of soot in these areas. All surfaces on the inlet side are coated only with a very thin film of ash deposit. The brush action is sufficient to break the deposits up and the exhaust gas velocity great enough to carry the particles through the distributor plate. Soot does tend to build up however on some flat surfaces above the fluidized bed. This soot is very fine and powdery and easily removed by brushing or with the aid of an industrial vacuum cleaner. It is not recommended that water washing should be used as this would soak the bed material. If the bed material does become soaked by water for any reason, it is preferable to dry it thoroughly by circulating the boiler tubes with hot boiler water before allowing the bed to fluidize.

## BOILER PERFORMANCE

Measuring the steam output of a waste heat boiler is very difficult when that unit is forced circulated and delivering a very wet mixture of steam and water to the shell of an oil fired boiler, which also has its own inherent losses to add to those of the waste heat boiler. Although it was possible to measure the water flow rate between the boiler circulating pump and the waste heat boiler inlet, it was not possible to find a practical method of separating the water and steam from the boiler outlet and measuring the individual flows before they entered the main boiler shell. It was thus decided to measure the steam flow in the main steam outlet from the port main boiler, and to carry out all performance tests with the waste heat boiler coupled to the port main boiler.

In practice, these proposals were complicated by various factors:

- the main steam pipe from the port boiler had an insufficient straight length to allow accurate measurement across an orifice plate, although great care was taken in positioning the orifice plate and the pipe tappings. It was also impractical to re-route the steam pipe to achieve a long straight run. This resulted in the steam flow meter reading lower than expected.
- The oil burners fitted to the boiler had a low turn-down ratio, and would cut-out at a steam pressure of 10.30 MPa (10.5 kg/cm$^2$) and cut-in again at 8.83 MPa (9.0 kg/cm$^2$). This arrangement made it very difficult to operate at a steady steam pressure which is essential when trying to correlate steam and fuel flows.
- Although the required rate was that of the waste heat unit, the above methods in fact measured the output of the main boiler, so the losses of that boiler were being added to the losses of the waste heat unit.

In practice, the performance of the unit was assessed in the following manner:

The steam demand of the combined waste heat and oil fired boiler system was adjusted so that the oil burner remained steady at about half its capacity whilst maintaining a constant steam pressure. With the main engine operating under steady conditions, data were recorded from the oil fired boiler, waste heat boiler, main engine and turbo-blower. Data from the turbo-blower and main engine fuel consumption were then used to calculate the exhaust gas quantities through the boiler (Fig 8) which, together with the temperature drops through the waste heat boiler, were used to calculate the heat output of the waste heat unit. From this and the steam pressure, the steam quantity delivered by the waste heat unit to the oil fired boiler was arrived at. (Fig 7 curve C). Assuming a boiler efficiency of 0.8, the fuel required to produce this amount of steam by the oil fired boiler alone was calculated. At the same time the actual boiler fuel consumption was measured, and the steam flow from the boiler recorded. Following a period of steady conditions, the waste heat unit was then shut down and the data from main engine and oil fired boiler recorded again with the steam pressures and flows maintained as before. From this a new boiler fuel rate was achieved, the difference between this and the previous rate being a measure of the fuel saving achieved by the steam production of the waste heat unit. The correlation between these measured values and the calculated fuel savings can be seen in Fig 7 curves A and B. Measurements were also obtained of the port boiler fuel rate at varying indicated steam flows, and then repeated with the waste heat boiler in operation and the main engine at 90% m.c.r. The two curves obtained are shown in Fig 10. The measured fuel

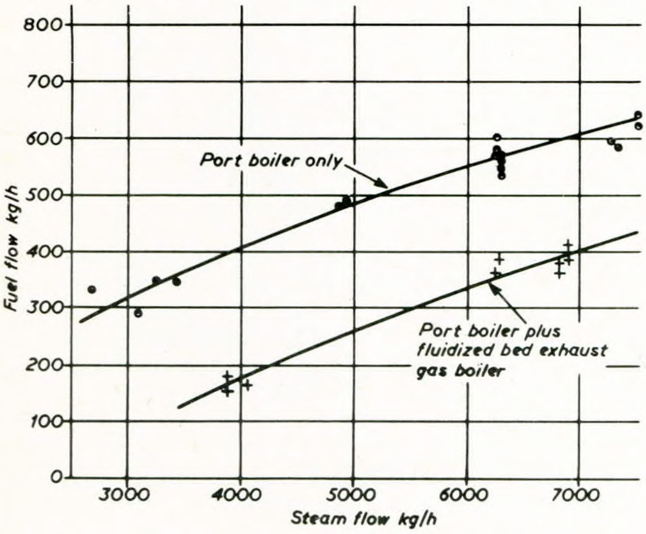


FIG 10 *Port main boiler fuel consumption*

saving of 4.95 tonne/day and the calculated value of 5.25 tonne/day at 90% m.c.r. engine load from Fig 7, can be compared with the savings of between 4.90 and 5.66 tonne/day obtained from Fig 10. Trials were also made with the exhaust gas boiler supplying the full steam demand and the port boiler oil burner shut down. Typical flow conditions are shown in Fig 11.

All the above tests were made after the boiler had been in service for a total of 1700 hours.

*Noise Levels*

Noise levels have been measured on the navigating bridge wings when the main engine has been exhausting through the original silencer, and repeated when exhausting through the fluidized bed unit. The results are reproduced in Fig 12 which indicates a reduction in sound level of 6 dBA when using the fluidized bed boiler.

## DEVELOPMENT WORK

Development work is being carried out to optimize the design, construction and operation of the fluidized bed waste heat boiler, as well as developing a larger unit design to produce superheated steam to drive a turbo-alternator set supplying electrical power for the complete electrical, hotel and refrigeration loads for a liquefied gas carrier.

The main areas of development which are now being pursued are as follows:

1) Extended performance checking of the unit in m.t. *Fjordshell*. The results from the first year have been outlined in this paper.
2) Investigation of a number of improvements with regard to heat transfer surface and distributor plate using a factory-based research and development heat exchanger test rig. There is an opportunity to increase the surface area of boiler tubes by some 12½% in the same plan area, thereby reducing the overall size of the unit.

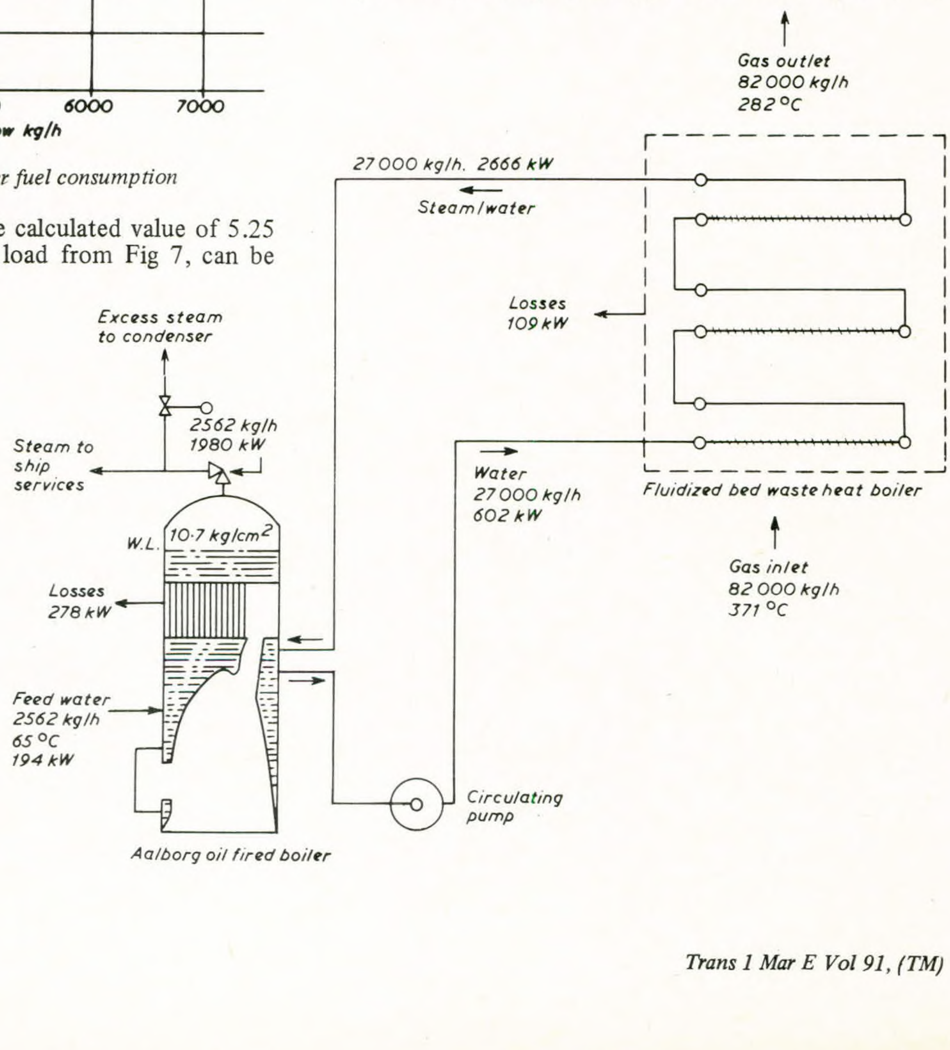


FIG 11

*Fluidized bed waste heat boiler system flow diagram*

3) An outline design of a fluidized bed waste heat exchanger for power generation is being developed. The choice of steam conditions in relation to the diesel exhaust temperature to achieve the maximum electrical output is being investigated with a steam turbine manufacturer.

   Considerable work is now in hand on this aspect and a larger fluidized bed waste heat boiler with a bed plan of 5 m x 5 m, which could handle the exhaust gas from a 15 078 kW (20 500 hp) engine would have the specification detailed in Table II.

   Normal operation of the unit would be in the 10 to 12 bar range at the higher steam temperatures.

4) Further detailed design of the casing has been carried out, eliminating some of the complexity that was inevitably built into the prototype unit including the incorporation of the gas bypass valve inside the heat exchanger.

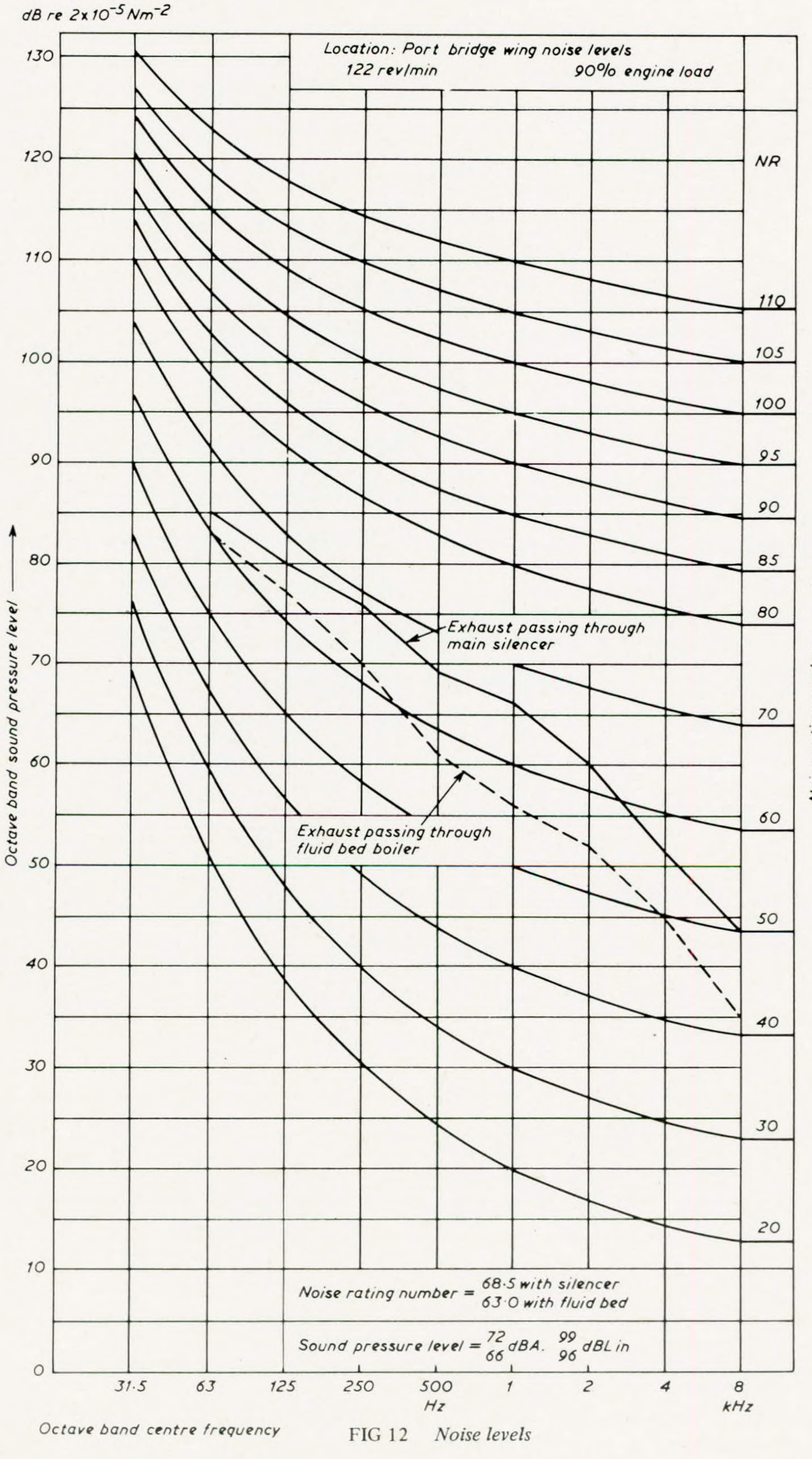


FIG 12 *Noise levels*

TABLE II BOILER SPECIFICATION FOR 15 000 kW ENGINE

| | | | | |
|---|---|---|---|---|
| Gas weight kg/h | | 146 650 | | |
| Gas temperature inlet °C | | 350 | | |
| Gas temperature outlet °C | | 241 | | |
| Steam pressure bar | 12 | 10 | 8 | 6.25 |
| Steam flow kg/h | 6668 | 6900 | 7167 | 7441 |
| Steam temp. °C | 246 | 241 | 235 | 229 |
| Thermal recovery MW | 3.65 | 3.8 | 4.0 | 4.16 |

## CONCLUSIONS

The main advantages of the fluidized bed waste heat boiler in comparison to conventional waste heat boilers which have been used on ships for many years, are as follows:-

1) The steam output is greater than that normally obtainable with a conventional unit with the same heat transfer surface area, because of the high heat transfer rate and the even temperature distribution throughout the bed.
2) Mechanical cleaning of the gas distributor plates eliminates the need for sootblowers, which are seldom 100% effective, and reduces the fire risk from soot formation.
3) There is no build-up of carbon and ash material on the extended surface tubes in the bed as they are kept clean by the continuous motion of the particles, which also maintains the steam output level. A light surface coating of carbon a few microns thick does plate out on the surfaces.
4) The tubes in each panel are straight, and provided with square inlet headers, the covers of which can be removed for inspection and internal cleaning.
5) The individual heat exchanger panels can be easily exchanged in a matter of hours, without the need for cutting and welding.
6) The fluid bed heat exchanger acts as an effective silencer, reducing the sound pressure level by 6 dBA on the navigation bridge wing compared with the normal main engine silencer.

It is recognized that the main disadvantages of this type of boiler are as follows:-

1) The size of the unit may not necessarily be as small as a conventional boiler because the plan area of the fluidized bed is dictated by the fluidization velocity and the volume of gas being passed through the unit. There is certainly some room for optimization of bed arrangements here, and it is quite possible in the future that the size will at least be directly comparable with conventional waste heat boilers.
2) Because the fluidized bed will only satisfactorily work within the fluidization velocity range which at best is about 3 to 1, it is necessary to incorporate a bypass in the system so that when the engine is running at low load the boiler can be bypassed. New arrangements are now being incorporated in the latest design so that this bypass is fitted in the boiler casing, eliminating the necessity for external ducting and silencers.
3) The exhaust gas back pressure on the engine is higher when using a fluidized bed boiler than with a conventional boiler. Normally a pressure drop of 125 to 150 mm $H_2O$ should be experienced across the fluidized bed.
4) The use of the fluidized bed boiler incorporates a third element, the fluidized bed material which in this case is aluminium oxide. This will inevitably result in some grit being lost over an extended period of operation and this will have to be made up. This is relatively easy to do through the manholes provided, but it does introduce an extra maintenance factor, the elimination of which is being actively pursued in future designs.

The fluidized bed waste heat boiler has been operated by the ship's staff, with exhaust gas from residual fuel, for over 2000 hours. The unit is considered by the ships' staff to be reliable and apart from those detailed previously, no special operating procedures are required.

The effect of ship motions on the bed performance, a major uncertainty at the commencement of the project, has been shown to be negligible.


## ACKNOWLEDGEMENTS

The project detailed in this paper has been sponsored by the General Council of British Shipping and the Ship and Marine Technology Requirements Board of the Department of Industry.

The authors wish to express their gratitude to their colleagues for the effort, thought and encouragement they have given during the progress of the project and the preparation of this paper. A special mention should be made of the staff and crew of m.t.*Fjordshell* and the technical shore staff of A/S Norske Shell, without whose enthusiasm and keen cooperation the successful outcome would not have been possible.

The authors also wish to thank the Managements of Shell International Marine Limited and Stone-Platt Fluidfire Limited for permission to publish this paper.


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## Discussion

MR D J GIBBONS, C Eng, F I MarE, expressed his appreciation to the authors for the preparation and presentation of their paper, which he felt was important in the advance of marine engineering because it recorded the first shipboard application of fluidized bed heat exchange, and paved the way for more ambitious projects.

The project was not without its difficulties, which was only to be expected at that stage of development, but Mr Gibbons felt that, with one exception, they had been successfully overcome so that the principles could be said to have been established for future exploitation.

There was also interest in fired fluid beds for advanced boiler and/or superheater designs, and it might only require an upturn in the marine market place to provide the encouragement needed firmly to establish the technology. To enable the technology to be exploited it must be competitive with its alternatives, on both performance and cost grounds. Whilst recognizing the developmental nature of that particular installation Mr Gibbons would be interested to learn how far gas temperatures might be cooled with that arrangement and whether there was the same potential, within gas pressure drop constraints, for heat recovery for power and heating services as with the "conventional" finned tube design of forced circulation heat exchanger. Pressure drop consideration might be a limiting factor for exhaust gas heat recovery and he asked if the authors could give more information on the distribution of that loss between the bed and the distributor plates.

With regard to the silencing properties of the bed, it was not often that the opportunity arose for direct comparison between a conventional silencer and modern waste heat boiler. Fin tube boilers had a silencing effect also, and noise levels measured at the bridge wings of a 9000 SHP installation might be expected of the order 71-73dBA. By contrast, measurements of a 12000 SHP installation with conventional silencer were some 6 dBA higher. Although the figures for the fin tube boiler arrangement were close to the UK guidance levels, they were above the IMCO levels which were closely met by the fluid bed design. Mr Gibbons asked if the authors could give further information on the choice of bed material, and whether there had been any changes in their thinking on the selection of that material for any future installation.

MR D TRUSLER (Marine Technology Support Unit, Harwell) said that his main task was to encourage industry to accept government assistance through the Ship & Marine Technology Requirements Board in the marine research and development field. He pointed out that technical success was not enough: it was necessary to be commercially minded so as to exploit successful developments like the waste heat boiler. Mr Trusler asked the following questions:

1) how did the waste heat boiler compare with a conventional one? Did its through-life cost make it a viable proposition?
2) could the lessons learned be applied to a fluidized bed combustion boiler?
3) could the principles be used for other heat exchangers aboard the ship?

## Author's Reply

Mr Cusdin recalled that one contributor to the discussion had raised some questions on operational and safety procedures of the fluidized bed waste heat exhaust gas boiler.

The first question had concerned cold starting and the time taken to raise steam. The boiler was operated in a similar manner to any other boiler with forced circulation. To raise steam, it was necessary to start the circulating pump and circulate the boiler with hot water from the main oil fired boilers. With the engine operating, the gas changeover valve directed the exhaust gases through the bed and steam was rapidly produced. Calculations showed that it was possible to obtain full steam production in two minutes of starting the boiler. As the unit was installed in a coastal trading vessel, that rapid start up time was very beneficial.

The contributor's second question concerned the effect on the boiler should there be a circulation failure. As temperatures in the bed were relatively low, there was little prospect of overheating the tube surfaces, also the action of the fluidized bed kept the tubes clean so there was little possibility of soot fires.

The third question concerned the effect on the boiler should there be a tube failure. If a tube should fail in service, it was obvious that the bed material in that section would be quickly blown out of the boiler. As each bed was divided into 15 sections by baffle plates, the effect would be limited. The tubes in each bed were divided into nests of 12 tubes each, and each nest could be withdrawn from the boiler individually. As the nests were connected with flanged joints to the main steam/water headers, it was possible to blank off an individual nest containing a leaking tube and to operate with the other eight nests. That would only slightly reduce the output of the boiler. The boiler was also provided with one complete spare tube nest so that at a convenient opportunity the damaged nest could be replaced. Individual spare tubes were also carried.

A further question concerned other fail safe arrangements in the design of the boiler. There was the possibility of the distributor sheets supporting the bed material fa ling, allowing the material to fall through into the inlet side of the boiler. The space under the distributor sheets, as indicated in Fig 2, was designed as a grit catchment area which prevented the bed material falling into the main engine exhaust pipe. When the engine was operating, the gas velocities in the exhaust were far too high to allow bed material to fall down into the turbo-blower. It was normal procedure to keep the exhaust gas changeover valve closed to the boiler when the engine was stopped or below 60% MCR. Thus should any grit fall down the exhaust ducting at such times as the engine was stopped, it would be trapped at the changeover valve. An inspection cover was provided in the ducting adjacent to the valve for such purposes.

The precautions taken against the possibility of soot fires had been described in the paper. Virtually no build up of soot had been observed on the inlet side of the unit and no high temperatures had been indicated by the alarm system. Some fine powdery soot did collect on flat surfaces above the beds, and in the future design, it might be preferable to install the fire detection/fighting equipment in those areas.

In all other aspects of safety, the boiler was like any conventional finned tube waste heat unit.

Another speaker at the discussion asked about the effect of the sulphur levels in the fuel used in that engine. The engine in the *Fjordshell* ran on normal marine fuel oil of 1500 second Redwood No.1, as obtained in Scandinavian oil refineries. That contained some 2½ to 3% sulphur. Using fuels with higher sulphur levels of 4 to 5% could produce some problems with corrosion should the back end temperature be below the gas dewpoint. Measurements made in the vessel indicated the gas dewpoint to be considerably lower than had been previously estimated. The authors were fairly confident that the exhaust gas temperatures could be reduced to the region of 130°C with the fuel used in that engine, without running into any corrosion problems.

The same speaker had commented on the exhaust gas back pressure on the main engine. Exhaust gas pressures observed in the ship had been shown in Fig 9, which compared the back pressure when the exhaust gas boiler was bypassed and when the boiler was in use. It could be seen that at 90% MCR the exhaust pressure on the engine was in the region of 320mm $H_2O$. No detrimental effect on the engine or increase in fuel consumption at that back pressure had been observed and they felt that the engine could be run at some higher back pressure before any detrimental results could be observed. It was obviously beneficial to keep the back pressure as low as possible, but from previous comments they considered that, should that type of boiler be installed as initial equipment with the correctly designed exhaust ducting, then such high pressures would not be experienced.

In answer to Mr Trusler, Mr Virr thanked him and the SMTRB for their support throughout the project, and which was continuing on the development programme described in the paper.

The authors agreed that technical success was not enough, and that the optimization programme was aimed at making the project a commercial success. They answered his questions as follows:

1) So far, apart from one instance (the unit specified in Table II of the paper) their only direct experience of commercial competition had been when selling fluid bed heat exchangers for industrial boilers. Twelve of those units had been sold and, as they competed directly with conventional heat exchangers, the selection had usually been on the basis of price. They found that those fluid bed heat exchangers were most competitive when used as economizer units with steam boilers, which was understandable since the high heat transfer coefficients paid off best at the low temperature end.

2) Lessons could certainly be learnt from the *Fjordshell* exercise when designing fluidized bed combustion boilers for marine use. The authors had recently carried out a design study for consultants on a pair of 60,000 lbs/hr boilers for use with propulsion steam turbines in a 60 000tonne Panamax type vessel, as shown in Fig 13. The study was supported by the SMTRB, and other boiler manufacturers had also submitted fluid bed boiler designs. The marine version of the Fluidfire design submitted was quite small in relation to the space taken up by a slow speed diesel engine. The comparison of the coalfired ship to the diesel version of the same ship on a typical selection of world routes indicated the coalfired system to be economically viable when using the most economic component parts. Marketing information had indicated that on certain routes where coal was carried (e.g. Australia to Japan) the fluidized bed steam powered ship would be the most economic vessel.

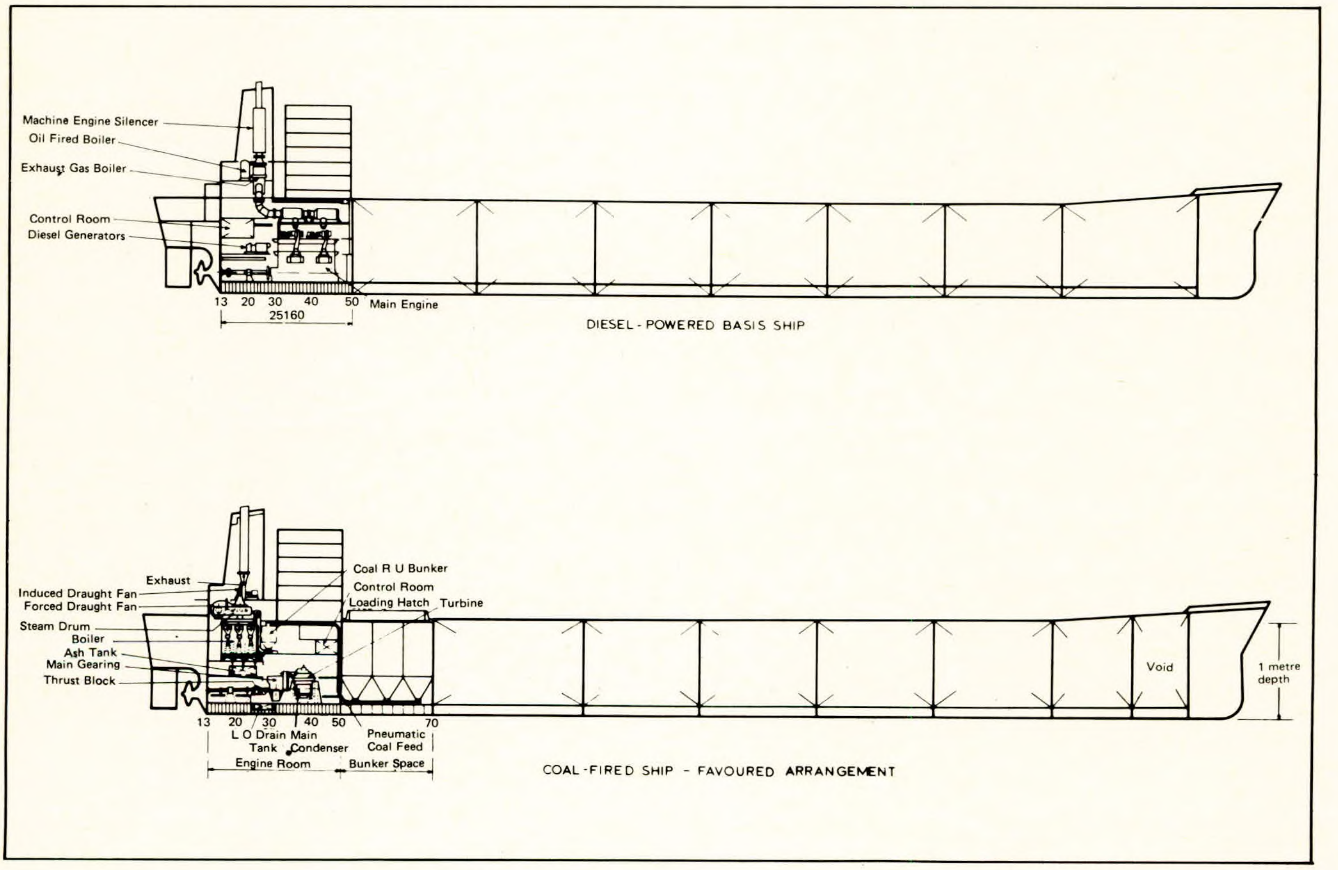


FIG 13 *Coal and diesel powered panamax vessels*

The boiler configuration itself was shown in Fig 14. It could be seen that the fluid bed, shown in cross section, was semi-circular in shape, which caused the bed material to circulate down into the centre and up at the sides. That circulation caused the incoming coal, which was fed by two large diameter tubes to the centre of each bed, to be drawn down under the red hot solids, thus giving long residence time and good combustion efficiency (typically 98.6%). That avoided the need for a carbon burn-up cell, as employed in some other fluid bed boiler designs, to which unburnt carbon was circulated for complete combustion. That boiler design was split into three beds each approximately 12 ft by 7 ft. Each bed was started by lighting up a gas oil burner beneath the bed and coal was admitted when the temperature had reached about 600°C. The bed was controlled at the normal operating temperature of 850°C by regulating the coal feed rate to the tubes by means of variable speed rotary valves. The fluid bed had wing sections into one side of which the main part of the evaporator was fitted, and the other the main part of the superheaters. The boiler pressure, which was 63 bar, was controlled by regulating the fluidization of the evaporator panel. The superheat temperature of 515°C was likewise controlled by regulating the fluidization of the superheater panel. The tubes separating those panels from the main combustion bed were close pitched and acted as baffles both to restrict the detrimental effects of rolling and to isolate the panels from the combustion bed under start up conditions. The panels absorbed about 60% of the heat output from the boiler and were closed down under start up. The boiler had a turn down ratio of about 10:1. The upper part of the boiler was more conventional with a plain tube superheater and a finned economizer section. It was fitted with an air heater so an overall boiler efficiency of 88% was achievable and was of a force circulation type.

That type of boiler design also lent itself to the use of higher superheater temperatures of up to 600°C, because the lower maximum fireside temperature of 850°C, combined with the high heat transfer coefficients available, gave lower metal temperatures with less corrosion.

The experiments and experience of the effects of ship's motion on the fluid bed heat exchanger in *Fjordshell*, led the authors to believe that a boiler of that type would accept ship's rolling of up to 15° without boiler boiler
the feasibility of such a marine fluid bed main propulsion boiler.

3) Mr Trusler had asked if the fluid bed principles can be used for other heat exchangers aboard ship. The authors briefly investigated the feasibility of using them for the brine to air heat exchangers on a refrigerated container ship. The problem here would appear to be the almost certainty of ice forming on the fluid bed on the air side, thus causing freezing up of the bed. That could not be avoided when cooling fruit and other food stuffs containing moisture, so it was judged that that application was not feasible, although in theory very much smaller heat exchangers could be designed. Research in other areas, such as air to exhaust gas heat exchangers was proceeding, and would possibly bear fruit in a few years time.

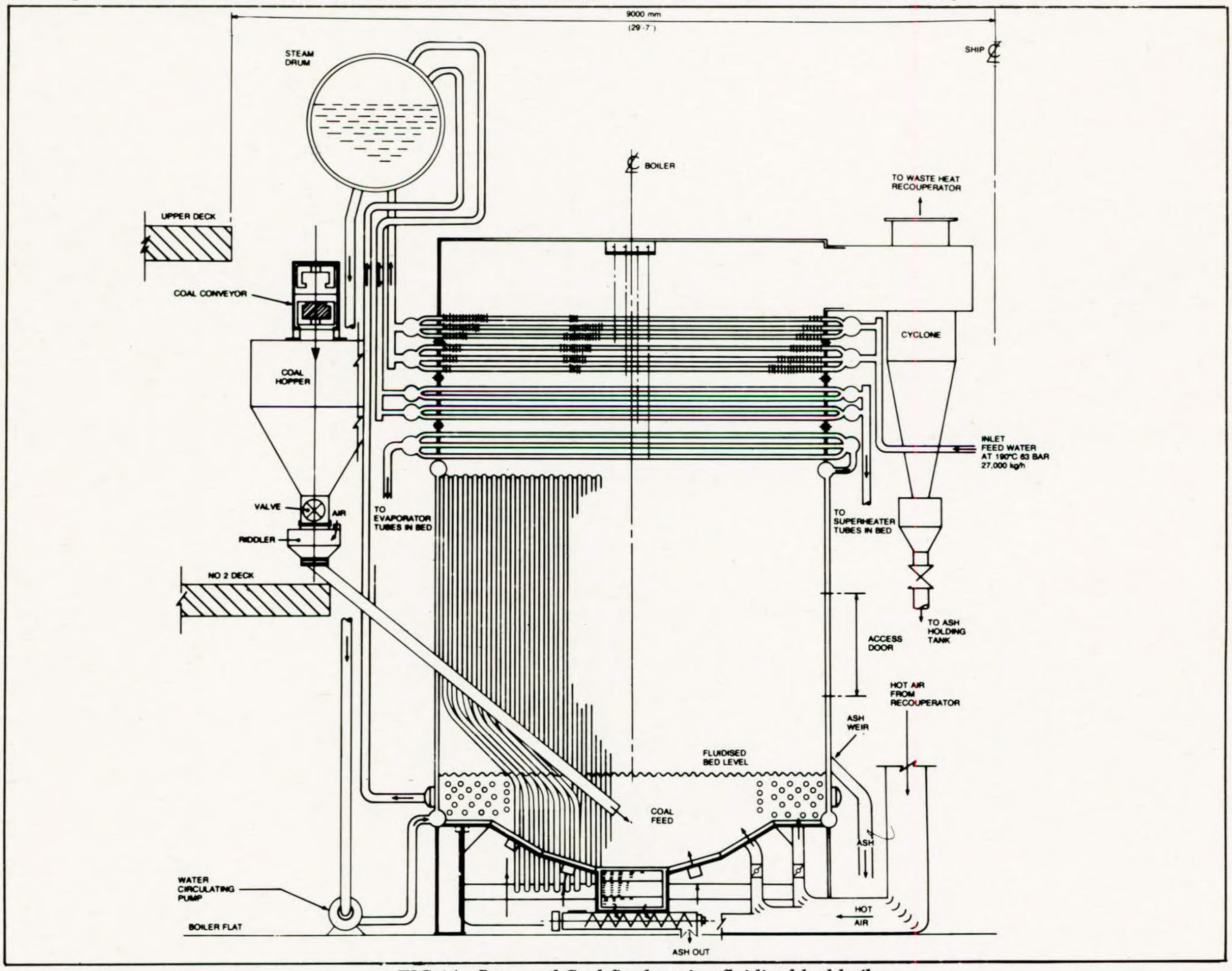


FIG 14 *Proposed Coal fired marine fluidised bed boiler*

Published for THE INSTITUTE OF MARINE ENGINEERS by Marine Management (Holdings) Ltd., (England Reg. No. 1100685)
both of 76 Mark Lane, London EC3R 7JN. Printed by Crown Printers (Bedford) Ltd., Bedford (England Reg. No. 910307).