PAPER #204

VIII CONGRESS PAN AMERICAN

INSTITUTE OF NAVAL ENGINEERING

PROPULSION SYSTEMS AND SHIP OPERATING EFFICIENCY

Optimisation of the marine transportation cost tanks to medium speed engines.

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WASHINGTON

SEPTEMBER 1983

The SEMT has introduced in December 1980 two new versions of medium speed engines, the PC 2-6 and the PC 4-2 types, which, it is believed, bring a satisfactory answer to basic requirements of customers : low specific fuel consumption, capability to burn low grade fuels, reliability.

This paper will cover all the aspects dealing with the transportation cost based on two main lines :

- as regards the engines themselves =
  - . most recent developments which permit a further reduction of the specific consumption,
  - . capability to burn low quality fuel oils,
  - . reliability and availability records.

- as regards the engine room =
  - . possibility of using the optimum propeller efficiency,
  - . fuel economy thanks to the high heat recovery capability.

PC 2-6 AND PC 4-2 ENGINES.

1. - Low specific fuel consumption has been obtained, as a result of several improvements.

Marine diesel engines specific fuel consumption has been

significantly reduced, in particular over the last ten years. These savings (figure n° 1) have been obtained for our medium speed engines as the result of a better knowledge of the super - charging, the injection and the combustion processes, and also of technological improvements in the engine design itself.

1.1. - One of the main steps has been the introduction of our so-called MPC supercharging system (figure n° 2), which is basically a constant pressure arrangement, with additional pressure converters located between each cylinder head and the exhaust pipe. This system is, according to our tests, the best possible compromise between both constant pressure and pulse systems, in providing the best efficiency at high loads, and with a still good air excess at lower loads (figure n° 3).

This air excess is, unfortunately, still lower than with the old impulse systém. In order to prevent this inconvenient, a by-pass arrangement may be fitted between compressor air discharge and turbine gas inlet (figure n° 4) to improve the supercharging conditions over the full operating range, in offering three advantages :

- . Increased air excess at low ratings,
- . Possibility to select for the turbo-chargers the adaptation providing the best efficiency at full power without meeting surging,
- . Eventual matching of turbo-blowers for service power, the excess of air at higher ratios being discharged so as to avoid overpassing the design firing pressure.

The MPC arrangement not only permits a reduction of the fuel consumption, but also provide a simpler solution, with evident advantages : lower manufacturing costs, easy access, optimum working conditions for the expansion bellows.

The expansion bellows have been made more reliable owing to an inside guide, and a new concept for assembling exhaust flanges together is systematically used.

1.2. - The use of higher performant turbo-blowers (BROWN BOVERI serie 4), also reduces the specific fuel consumption at high bmep.

The use of such turbo-blowers with an overall efficiency of 0.62 to 0.64 at full load instead of 0.55 to 0.58 with the previous machines, unfortunately led to an, again, reduced air excess at part load making even more necessary to use the already mentioned by-pass system. The saving in specific fuel consumption is of about 2 g/HP.hour at 100 % (figure n° 5), but coming to 0 and even less at reduced load.

Furthermore, non cooled gas casings, available now, will permit to have the exhaust gases at a higher temperature (+ 20 to 25°C), and heat recovery capacity will be still good despite the improved efficiency of the engine.

1.3. - In the PC 4-2 inlet air manifold, air losses are minimized and for both engine types, owing to optimized shapes inside valve cages, transfer losses have been reduced.

1.4. - Injection pumps have been designed with an increased bore/stroke ratio for the plunger (figure n° 6) : the bore of the plunger has been enlarged by 10 %. Then, the injection process is shortened and the injection pressure increased, compared with earlier produced engines. This has been found efficient to reduce specific fuel consumption (about 2 g/HP.hour).

The injection pump driving system has been reinforced, in order to keep the Hertz pressures on the cam, at same safe level as on the already proven engines.

1.5. - Turbulence into the combustion chamber : In view of the first successful attempts made with the early developed

PC 2-6 engines, it was thought that potential possibilities to improve the combustion efficiency still existed. Several solutions aiming to an increased turbulence were therefore tested and it was confirmed that both turbulence and injection spray are interdependent, for the best air/fuel mixing. In the same time, systematic tests were conducted, in view to study how design parameters like :

- engine compression ratio,
- ratio firing pressure/compression pressure,
- air excess,

may influence combustion quality.

Final decision has been a slightly increased compression ratio for both engines, and a new atomizer for PC 2-6 engines using 9 holes instead of 10 previously.

1.6. - Performances of the PC 2-6 and PC 4-2 engines. With the latest improvements, which are progressively introduced on the engine design, following figures are reached :

	Engine type	M.C.R. in HP/cyl. RPM	Specific fuel consumption g/HP.hour	
			at 100 % MCR	at 85 % MCR
Engines ordered from	PC 2-6	750 / 520	133	133
	PC 2-6 E	675 / 470	133	131
	PC 2-6 EE	525 / 420	129	128
	PC 4-2	1,650 / 400	131	129
	PC 4-2 E	1,500 / 380	129	127
	PC 4-2 EE	1,200 / 340	125	124
Engines ordered at the beginning of 1984 (in line engines).	PC 2-6	750 / 520	133	131
	PC 2-6 E	675 / 470	131	129
	PC 2-6 EE	525 / 420	127	126
	PC 4-2	1,650 / 400	129	127
	PC 4-2 E	1,500 / 380	127	125
	PC 4-2 EE	1,200 / 340	123	122

(above figures are valid in ISO conditions, on propeller law, for a low calorific value of 10,200 Kcal/kg).

The first engines PC 2-6 and PC 4-2 manufactured last year, have met the expectations of their builders as regards fuel economy, as it may be seen on figures n° 7 and n° 8, showing test bed measurements.

2. - Improved capability to cope with low grade fuel oils.

2.1. - At normal load.

The diesel engines presently in service on merchant vessels, either of the low or of the medium speed type can, as a rule, burn any type of fuel including the residual ones. It must be however underlined that the quality of the residual fuels is lowering in a considerable way because of the sophistication of the refinery processes which shall allow, during the coming years, to supply the users of petroleum firms with more and more of noble products. This degradation of the quality concerns mainly the following features :

. asphaltene content which comes from 4 to 5 up to 8 to 9 %,
. carbon residue, conradson %, from 10 to 12, up to 18 to 22,
. density : possibily higher than 1,
. content in aluminium and silicium residues reaching up
. sulphur content reaching up to 5 %, to 300 ppm,
. vanadium content reaching up to 600 ppm.

The evolution of the first three characteristics results from the use of the thermal cracking process when the fourth one is proper to the catalytic cracking one. The last two are the natural consequency of a greater concentration of the crude oil impurities ; they are therefore highly variable depending on the nature and origine of the treated crudes.

2.1.1. - Asphaltene content.

Among these characteristics, the main evolution concerns the asphaltene content (or the conradson carbon residue), which, because of a combustion time longer by principle, is assumed to risk to lead to combustion difficulties. These difficulties are said to become more and more severe as the engine rotation speeds increase, since the matter is to obtain a sufficiently long combustion time.

In fact, the tests run have clearly shown that the combustion time, or more precisely, the fuel droplet heating time up to oxidizing temperature, is practically zero as soon as their size is of about 20 microns (figure n° 9).

On the contrary, as soon as the droplet size reaches about 80 microns, this time becomes substantial and even so long that whatever be the engine type, unburnts will be present in the combustion gases. This has been proved by research work done at CHALMERS UNIVERSITY OF GOTEBORG and summarized on the figure n° 10.

Tests achieved thanks to the possibilities offered by the holography, have shown that the main factor to get small droplets size is the injection pressure. This pressure when above 800 bars, allows to obtain easily the 20 microns sizes required when running on diesel oil.

On the contrary, with a 400 bars injection pressure (figure n° 11), the droplet size reaches 40 to 50 microns. This size will still grow with the lower pressures resulting from very low loads. It grows also when using heavy fuel. To prevent this increase, it is necessary to heat the fuel at the proper temperature and to have also a higher injection pressure.

So, the already mentioned increase of the injection pressure has a beneficial effect on atomization quality : the fuel oil droplets get finer in size, which is the key condition to burn correctly a fuel oil having a high asphaltene content, whatever is engine type or speed. The injection pressures are at a leval of 1,200 bars at full load, which is quite sufficient even on heavy fuel as it can be seen on the figure n° 12.

2.1.2. - Density over 1.

A fuel with a density higher than 1, does not allow centrifugal elimination of sea (or fresh) water eventually contained in it.

Several remedies have been investigated, more specially by the use of magnesium salts addition to make the water heavier. As far as the SEMT is concerned, the solution considered consists in leaving the water in the fuel (*) atomizing it in the thus formed mixture. This leads to incorporate an homogenizer in the engine fuel circuit allowing to obtain the required water droplet size of about 5 microns. The mixture is then injected into the engine. Tests are in process to ascertain that the mixture is not injurious to the injection system and the barrel plunger assembly.

2.1.3. - Catalytic residues.

The content of catalytic residues (essentially aluminium and silicium), may reach, in some cases, up to 200 or 300 ppm. These residues have to be eliminated down to 75 or 80 %, because of their abrasiveness.

The mean consists in the use of two centrifuges running in serie (purifier and clarifier), and a 10 micron filter. The figure n° 13 shows the efficiency of this solution.

2.1.4. - Sulphur and vanadium contents.

As far as the sulphur and vanadium contents are concerned,

(*) In so far as the water percent, if sea water is concerned, is limited to a value of 3 % (because of sodium presence).

they are a nuisance because of the corrosions entailed. These corrosions may however be eliminated by technologic means leading to obtain in continuous service, components temperature over the sulphuric acid dew point on the one hand, and below the melting point of the complex salts of vanadium and sodium formed during the combustion in the second hand.

In fact, to prevent exhaust valve burnings, which has been for many years, the main concern of the medium speed users, a new generation of exhaust valves is used (figure n° 14).

These new exhaust valves incorporate the experience gained these last years on heavy oils of different crude origines. Basic design features are :

. efficiently cooled cages, owing to wall thickness decrease and cooling flow increase : less sensibility to high temperature corrosion of exhaust valve seats,

. Nimonc spindles with hardened seats instead of stellite coated steel valves : increased resistance against high temperature seats corrosion, caused by vanadium in the fuel,

. cold surfaces of the cages in contact with exhaust gases are corrosion protected,

. valve guides are remote from valve seats : wear of bushes is considerably reduced, and valve sticking is avoided,

. sealing arrangement of the spindles : less corrosion of the stems and less oil pollution inside the rocker arm casing.

2.2. - At part load.

As already said, at reduced load, the injection pressure and the air excess, are decreasing. Then, to burn completely the heavy fuel, it is necessary to take some more cares :

. due to the high injection pressure at full load, it is already possible to run continuously at load as low as 35 % of MCR,

. to use lower ratings =
  - the by-pass system already mentioned allows to increase the air excess,
  - a two stage air cooler becomes the standard arrangement (figure n° 15). The high temperature side uses the jacket cooling water as cooling media. The low temperature side is cooled by water, with a conventional cooling system, preferably a central cooling system.

As soon as the power rating is below 35 % of MCR, the low temperature air cooler section is not used and the air delivered by the compressor is heated up, by jacket water, to a temperature around 70°C, before entering the engine. Therefore, the fuel oil droplets atomized in the combustion chamber get quicker to the ignition temperature, which means a better capability to burn residual fuel oils at partial loads.

In addition, inespective of the fuel quality, the lower continuous rating performance, which requires a level of 240°C for the exhaust temperatures is improved as a result of the exhaust temperature rise.

As a general conclusion for this paragraph, it has to be said that, in view of these improvements, the engines can be operated with low quality fuel grades having a viscosity of 700 cst at 50°C (7,000 sec. Redwood I at 100°F) and following other characteristics : specific gravity 0.991, sulphur 5 %, conradson carbon 22 %, vanadium 400 ppm, aluminium + silicium 30 ppm, diesel index at least 30.

Of course, the fuel treatment has to be done according to our specifications, which are summarized in the figure n°16.

3. - Reliability.

The increasing share of the marine propulsion market acquired by medium speed engines is the last ten hears is a proof of their reliability. It is true to say that originally, some 25 years ago, they were used particularly for special purpose ships, such as car-ferries, where the space available for the propulsion machinery rules out any other type of engine.

The large number of ships fitted with a single medium speed engine is clear evidence that the necessary level of reliability has been achieved. Of 2,512 medium speed PC engines in service and on order by May 1,983, for 1,601 ships, 920 (37 % of the engines and 57 % of the ships), are installed on single engine ships (figure n° 17).

Anyhow, following comments can be done :

3.1. - Availability records.

They are few records of engine availability, but, thanks to the cooperation of some shipowners, we have been able to get some figures. All these figures are related to engines using 380 cSt heavy fuel and using this fuel from pier to pier, that is to say even for starting, stopping and manoeuvring.

	Number of engines	Total running hours	Number of hours of engine stoppages		Availability of engines	
			Total	for exhaust valve burning.	Total	without exhaust valve burning
PC 2-5	10	99,207	192.5	0	99.81	99.81
PC 4	16	197,373	1,080.5	681.5	99.45	99.80

The above figures are related to the engine stoppages and not to the ship herself, as she is going on particularly in case of a twin en-engine installation.

A precise study of these figures let appear that about 60 % of the time of unavailability of the PC 4 engines is due to exhaust valve burnings. This percentage is zero in case of PC 2-5 engines.This is due to the use since more than three years of the water cooled cages and of the Nimonic spindles mentioned previously. This solution has settled the problem of exhaust valve burnings and all PC 4 engines could now be converted also in the same way.

3.2. - Number of cylinders.

Studies made by LLOYDS REGISTER, have shown that the number of engine breakdowns is dependent on the number of cylinders For medium speed engines, to show up well in a low/medium speed engine comparison, the number of cylinders must be less than three times that of the corresponding low speed engine.

This explains why medium speed engines, for many years restricted to a maximum power rating of 500 HP/cyl., have had to be uprated in terms of power output. The current availability of medium speed engines rated at 1,000 to 1,500 HP/cyl. offers a wider choice and limits the number of cylinders required.

The figure n° 17 of the LLOYDS REGISTER, which summarizes the study of the damages affecting the main engines of ships reported by the end of 1980, enlights that for a SEMT-PIELSTICK engine installation with less than three times the number of cylinders of the corresponding low speed engine installation, the number of incidents affecting PC engines, is less than that in respect of low speed engines.

3.3. - Exhaust valve burnings.

The most serious problem facing designers of four stroke medium speed engines was burning of the exhaust valve seats : this damage is counting for about 60 % of the unavailability time of the PC 4 engines.

As already said, this problem has been overcome by reducing the temperature of the valve seat to a value of around 450°C for all operating conditions and by using exhaust valve spindles in Nimonic 80-A with structurally hardened seats. The service results of the PC 2-5 already modified are giging the proof of this statement.

3.4. - Lubrication of main bearing shells.

Generally speaking, medium speed engines use thin bearing shells with an anti-friction coating based on a copper lead alloy and known as "Kelmet metal". Two stroke low speed engines use a lead tin alloy based anti-friction metal known as "white metal".

Although white metal is only slightly sensitive to the quality of lubrication, Kelmet metal is highly sensitive to all lubrication anomalies, including :

- more than 0.5 % water in the oil,
- excessive oil temperature,
- presence of foreign particles,
- insufficient oil pressure,
- air oil immulsion.

Such lubrication anomalies could damage the bearing shells, leading to immediate immobilisation of the engine, followed by long repairs.

To deal with this problem, it has been decided to fit all new PC engines with thermocouples for measuring the temperature at each bearing, connected to a central alarm unit shown in figure n° 18. Since this measure was adopted, there have been not one single incident requiring maintenance attention to the crankshaft has been mentioned.

ADVANTAGES OF MEDIUM SPEED ENGINES CONTRIBUTING TO THE FUEL ECONOMY.

Medium speed engines have reduced weight, volume and height, conditions favourable to an easy installation of the internal ramp.

In addition, their selection allows to retain several design features contributing to an economical propulsion installation, used also for other types of vessels, as follows :

1. - Free choice of the propeller speed.
Due to the already needed reduction gear, it is easy to foresee a slow speed as low as 50/60 rpm have been already selected for newbuildings. It may result significant saving of propulsive power.

Moreover, it is possible also to have :

2. - Heat recovery capacity.

2.1. - The relatively high temperature level of the exhaust gases, with the four stroke cycle, permits, owing to a steam turbo-alternator, a high degree of power recovery, which means a free of charge production of electricity at sea, for most vessels.

This is now the case on most of the ships propelled by our PC engines. As a matter of fact, as soon as the service power is of 9 to 10,000 HP on even 7,500 to 8,000 HP if the sea water, fresh water and lub.oil pumps are driven by the engine or through a PTO of gear box, the heat recovery is sufficient to produce, at sea, free of charge, the complete electrical power (figure n° 19).

The table n° 1 gives one idea of the economical advantage of this solution : the pay back is obtained in less than 3 years without forgetting the advantage to have no auxiliary engine at sea and very few maintenance work to be done on these diesel auxiliary sets.

2.2. - If the recovered power exceeds the electrical requirements at sea, the power in excess, instead of being wasted, may assist the main engine. In that case, the most efficient recovery system must be considered in order to get full advantage of the arrangement. That may be achieved by using :

. calories of the jacket cooling system, and part of the calories of the supercharging air, in addition to the heat of the exhaust gases,
. a two pressure stage steam cycle.

The figure n° 20 shows one possible cycle, which, by combining these two possibilities allows to the power to reach 12 % of the engine output, instead of 6, in case of a conventional recovery system, using only the exhaust gas heat, and having a one pressure stage.

The surplus of turbine output, not needed to generate electrical power, is fed back to the propulsion system. This can be achieved either by the already mentioned turbo-alternator, or by an electrical motor in both cases connected to the reduction gear (see figures n° 20/a, and 20/b).

In case of the electrical alternative, again two possibilities are offered, depending on propeller type, with fixed or controlable pitch. When a fixed pitch propeller is selected, then, a static frequency converter is to be provided to feed the electric synchronous motor by a current having the frequency matched to the speed.

These electric motors can be acting as alternators and again when a fixed pitch propeller is used, a static frequency converter is provided to ensure stable output voltage and frequency, in a range of propeller rpm, covering the necessary ship speed variations.

As the situation is fully reversible, the diesel auxiliary sets can even supply the electric motor, with variable frequency current - frequency automatically adjusted according to rotation speed - from the 60 Hz current produced by their alternators.

This possibility provides an emergency electrical propulsion at variable speed, ahead and astern, up to a ship forward speed of 7 knots for example (depending on the power of the diesel auxiliary sets).

2.3. - Examples of recent applications.

2.3.1. - Full use of recovered energy through auxiliary electric motor CP propeller at low speed.

This arrangement has been selected for two 140,000 tdw ore/coal carriers and one 80,000 tdw bulk-carrier already delivered by N.K.K. Several interesting features have been retained :

. Due to derating, specific fuel consumption of the engine is 132 g/HP.hour (10,200 Kcal/kg),
. Propeller speed is 62 rpm. It has been estimated that, by reducing the propeller speed from the previously conventional figure of 118 to 62 rpm, the engine power has dropped from 17,100 to 15,000 at engine flange, for a ship speed of 13.8 knots, yielding a fuel saving of 12 %,
. Waste heat boiler and turbo-alternator, recovering calories of main engine exhaust gases and supercharging air. Feed water of the economizer is heated up, through one section of the air intercooler, from 40° to 70°C,
. Accomodation heating is carried out by heat extracted from jacket cooling water. Due to this arrangement, more steam is available for the turbo generator, increasing production by 40 kW,
. Cooling sea water pump is provided with a variable speed device, so that, when engine at 90 % of full power, and with a sea temperature of 25°C, the amount of water is reduced by 50 %, and there is a 30 kW reduction

of the absorbed power.

As a result of these arrangements, the electric power requirement at sea, which is 520 kW, is easily covered by the turbo-alternator.

Total fuel saving amounts to 22 %, compared with an installation having :

. a propeller speed of 118, connected to an engine having the same specific fuel consumption,
. diesel alternators at sea operated on marine diesel oil,
. water heat boiler for only heating purposes.

The heat balance of the whole system is shown by figure n° 21. The efficiency corresponding to mechanical power production is 49.2 % and the overall efficiency including the extra energy recovered for heating purposes, is between 54.5 and 56.5 % depending on wheather conditions.

Records carried out at sea during the first voyage are shown figure n° 22. The turbo-alternator production makes the ship self sufficient at sea for electrical power generation and there is an excess of power of approximately 250 kW, fed back to the line shafting.

2.3.2. - Full use of recovered energy through steam turbine connected to reduction gear.

This is an advanced system developed by I.H.I. in Japan, called "SSG", which has been used for tankers, the first of them delivered in April 1981. A turbo-generator producing the recovered energy is mechanically connected to the reduction gear, in such a way that when turbine output exceeds the electrical energy demand, which is a normal situation at service rating, the power in excess is fed back to the propulsion system (figure n° 20).

When recovered energy is not high enough, then, the turbo-generator is backed up by the main engine to meet the electrical energy requirement.

A record at sea, made on a tanker powered by a 10 PC 4 engine of 15,000 HP, using that system, is shown figure n° 23. In normal service conditions, the power recovered is enough for the electrical energy démand and there is a feed back of approximately 100 kW to the propeller shaft. The system is self sufficient at sea regarding electrical power generation at approximately 60 % of main engine power rating.

An improved version of the system includes (figure n° 24) :

. recovery of calories of the supercharging air in addition to exhaust gas calories,
. steam turbine with a two stage pressure supply.

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Table.1

EXAMPLE of TURBO-GENERATOR SET PAY BACK PERIOD

(Suitable for conventional non reefer ships with a service power at least of : 8,000 HP if shaft driven pumps are used, 10,000 HP if shaft driven pumps are not used)

	Turbo-generator set		Diesel auxiliary sets on :		
			Diesel oil (DO)	Heavy fuel (HF)	
first cost, 500 kW x 750 US.dollars/kW [1]	375,000 [1]		140,000 [1a]	170,000 [1b]	One Diesel auxiliairy set less : 500 kW x 280 US.dollars/kW. (DO) [1a] 500 kW x 340 US.dollars/kW. (HF) [1b]
Installation on board, 500 kW x 75 US.dollars/kW [2]	37,500 [2]		14,000 [2a]	17,000 [2b]	Installation of one Diesel auxiliary set less : 500 kW x 28 US.dollars/kW. (DO) [2a] 500 kW x 32.5US.dollars/kW. (HF) [2b]
Total (US.dollars). [3] = [1] + [2]	412,500 [3]		154,000 [3a]	187,000 [3b]	Total (US.dollars). [3a] = [1a] + [2a] [3b] = [1b] + [2b]
Total excess of first cost (US.dollars).			258,500 [4a]	225,000 [4b]	Over expenditure. [4a] = [3] − [3a] [4b] = [3] − [3b]
Yearly heat recovery system maintenance cost : 500 kW x 10 US.dollars/kW.year [5]	5,000 [5]		166,000 [6a]	95,500 [6b]	Yearly consumption saving : 400 kW x 0.210 kg/kW.hour x 6,000 h. x 330 dollars/ton, [6a] 400 kW x 0.220 kg/kW.hour x 6,000 h. x 190 dollars/ton, [6b]
	DO	HF			
Yearly financial cost, 258,500 x 0.10 [7a] = 0.10 x [4a]	25,850 [7a]	22,500 [7b]	6,850 [8a]	13,700 [8b]	Yearly maintenance cost saving : 10 US.dollars/kW/year x 500 x $\frac{6,000}{8,760}$ [8a] 20 US.dollars/kW/year x 500 x $\frac{6,000}{8,760}$ [8b]
Total proportional cost (US.dollars/year) [9] = [5] + [7]	30,850 [9a]	27,500 [9b]	172,850 [10a]	109,200 [10b]	Yearly saving. [10] = [6] + [8]
			30,850 [9a]	27,500 [9b]	Yearly proportional cost. [9]
			142,000 [11a]	139,450 [11b]	Total yearly saving (US.dollars/year). [11] = [10] − [9]
			$\frac{258,500}{142,000}$ = 1.80 Year	$\frac{225,000}{81,700}$ = 2.75 Year	Pay back time [12] = $\frac{[4]}{[11]}$

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EXAMPLE of TURBO-GENERATOR SET and EXCESS of ELECTRICAL POWER TRANSMITTED to the PROPELLER – PAY BACK PERIOD

(Suitable for bulk-carrier of 130-140,000 tdw with a 12 PC4.2 engine running at 16,000 HP at sea)

	Turbo-generator set		Diesel auxiliary sets on : Diesel oil (DO)	Diesel auxiliary sets on : Heavy fuel (HF)	
first cost, 1,000 kW x 600 US.dollars/kW $\boxed{1}$	600,000 $\boxed{1}$		280,000 $\boxed{1a}$	340,000 $\boxed{1b}$	One Diesel auxiliairy set less : 1,000 kW x 280 US.dollars/kW. (DO) $\boxed{1a}$ 1,000 kW x 340 US.dollars/kW. (HF) $\boxed{1b}$
Installation on board, 1,000 kW x 60 US.dollars/kW $\boxed{2}$	60,000 $\boxed{2}$		28,000 $\boxed{2a}$	34,000 $\boxed{2b}$	Installation of one Diesel auxiliary set less : 1,000 kW x 28 US.dollars/kW. (DO) $\boxed{2a}$ 1,000 kW x 34 US.dollars/kW. (HF) $\boxed{2b}$
Shaft driven generator of 1,000 kW. $\boxed{3}$ Static frequency converter $\boxed{4}$	100,000 $\boxed{3}$ 300,000 $\boxed{4}$				
Total (US.dollars). $\boxed{3} = \boxed{1} + \boxed{2}$ $\boxed{5} = \boxed{1} + \boxed{2} + \boxed{3} + \boxed{4}$	1,060,000 $\boxed{5}$		308,000	374,000	Total (US.dollars). $\boxed{3a} = \boxed{1a} + \boxed{2a}$ $\boxed{3b} = \boxed{1b} + \boxed{2b}$
Total excess of first cost (US.dollars).			752,000 $\boxed{4a}$	686,000 $\boxed{4b}$	Over expenditure. $\boxed{4a} = \boxed{3} - \boxed{3a}$ $\boxed{4b} = \boxed{3} - \boxed{3b}$
Yearly heat recovery system maintenance cost : 1,000 kW x 10 US.dollars/kW.year $\boxed{5}$	10,000		249,000	143,250	Yearly consumption saving : 600 kW x 0.210 kg/kW.hour x 6,000 h. x 330 dollars/ton. $\boxed{6a}$ 600 kW x 0.220 kg/kW.hour x 6,000 h. x 190 dollars/ton. $\boxed{6b}$
			47,400	47,400	Extra saving, thanks to the connection between turbo generator and shaft (efficiency : 0.86), 300 kW x 0.86 x 0.175 kg/kW.hour x 6,000 hours x 180 dollars/ton. $\boxed{7}$
	DO	HF			
Yearly financial cost, 752,000 x 0.10 $\boxed{8a}$ 686,000 x 0.10 $\boxed{8b}$	75,200 $\boxed{8a}$	68,600 $\boxed{8b}$	13,700 $\boxed{9a}$	27,400 $\boxed{9b}$	Yearly maintenance cost saving : 10 US.dollars/kW/year x 1,000 x $\frac{6,000}{8,760}$ $\boxed{9a}$ 20 US.dollars/kW/year x 1,000 x $\frac{6,000}{8,760}$ $\boxed{9b}$
Total proportional cost (US.dollars/year) $\boxed{10} = \boxed{5} + \boxed{8}$	85,200 $\boxed{10a}$	78,600 $\boxed{10b}$	310,100 $\boxed{11a}$ 85,200 $\boxed{10a}$	218,050 $\boxed{11b}$ 78,600 $\boxed{10b}$	Yearly saving $\boxed{11} = \boxed{6} + \boxed{7} + \boxed{8}$ Yearly proportional cost. $\boxed{10}$
			224,900 $\boxed{12a}$	139,450 $\boxed{12b}$	Total yearly saving (US dollars/year). $\boxed{12} = \boxed{11} - \boxed{10}$
			$\frac{752,000}{224,900} = 3.35$ Year	$\frac{686,000}{139,450} = 4.90$ Year	Pay back time $\boxed{13} = \frac{\boxed{4}}{\boxed{12}}$

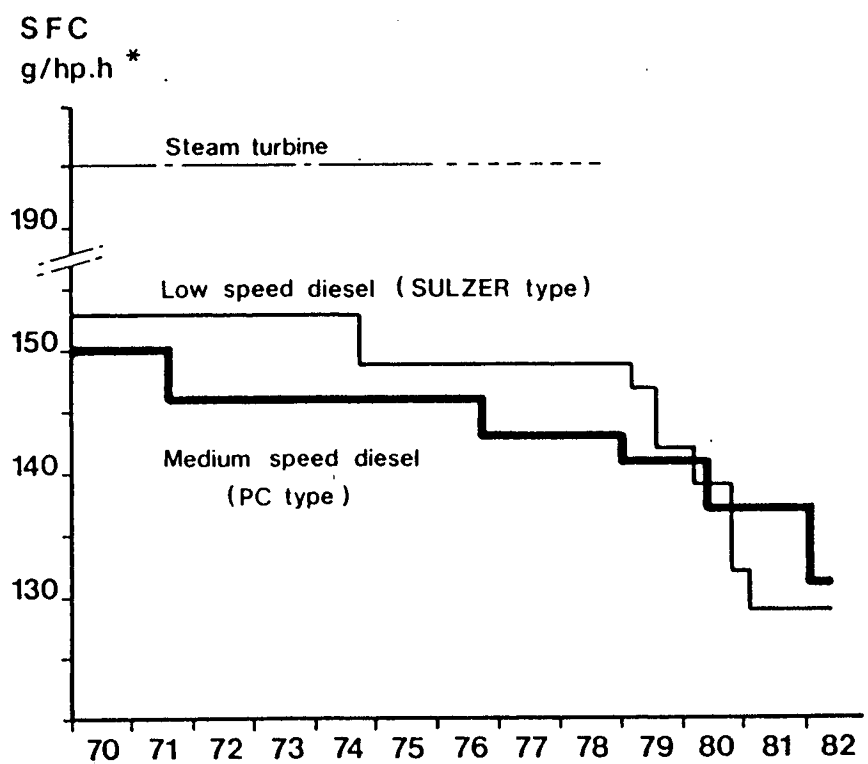
Table.2

S.E.M.T. PIELSTICK

Fig. 1

SPECIFIC FUEL CONSUMPTION OF

PROPULSION MACHINERY



* Based on net calorific value 10200 kcal/kg

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M.P.C. SYSTEM

Fig. 2

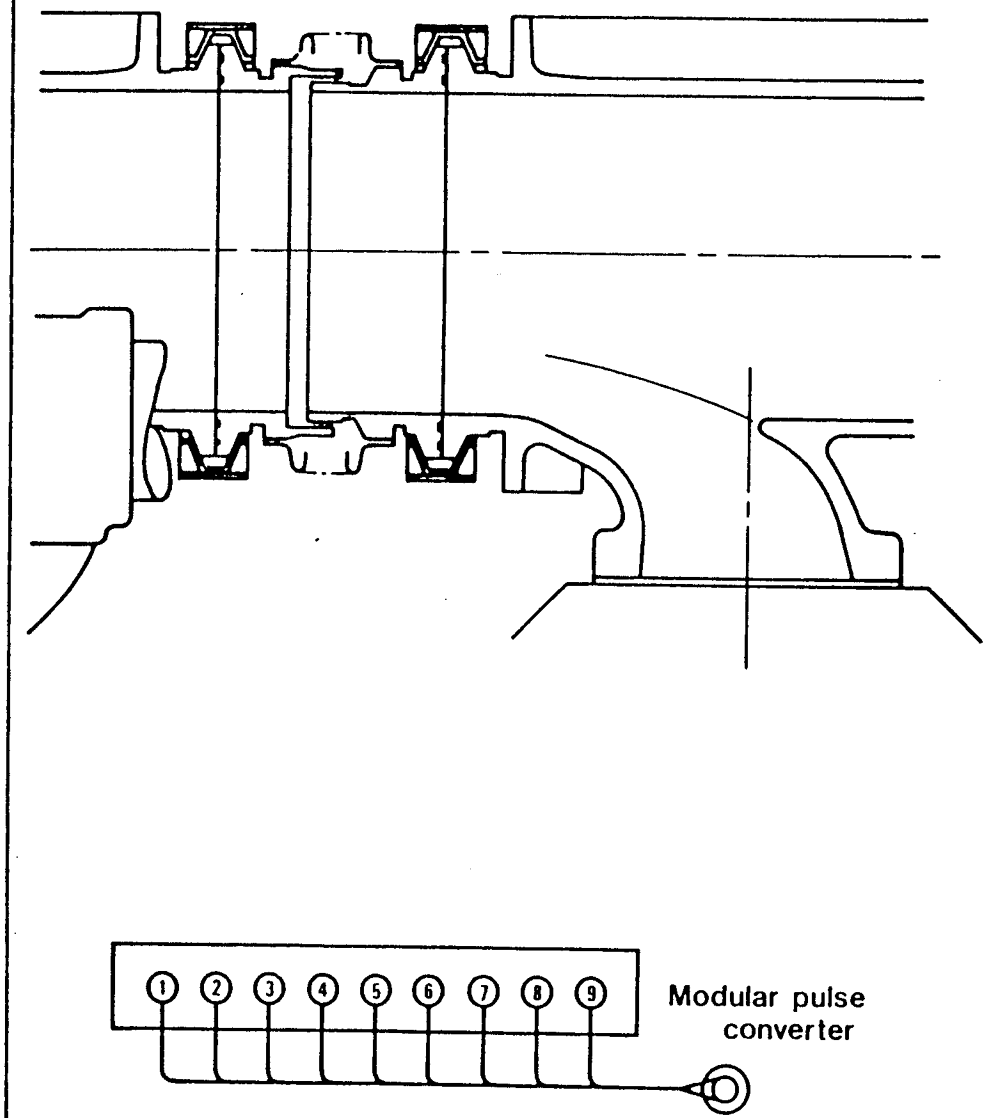

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Fig 3

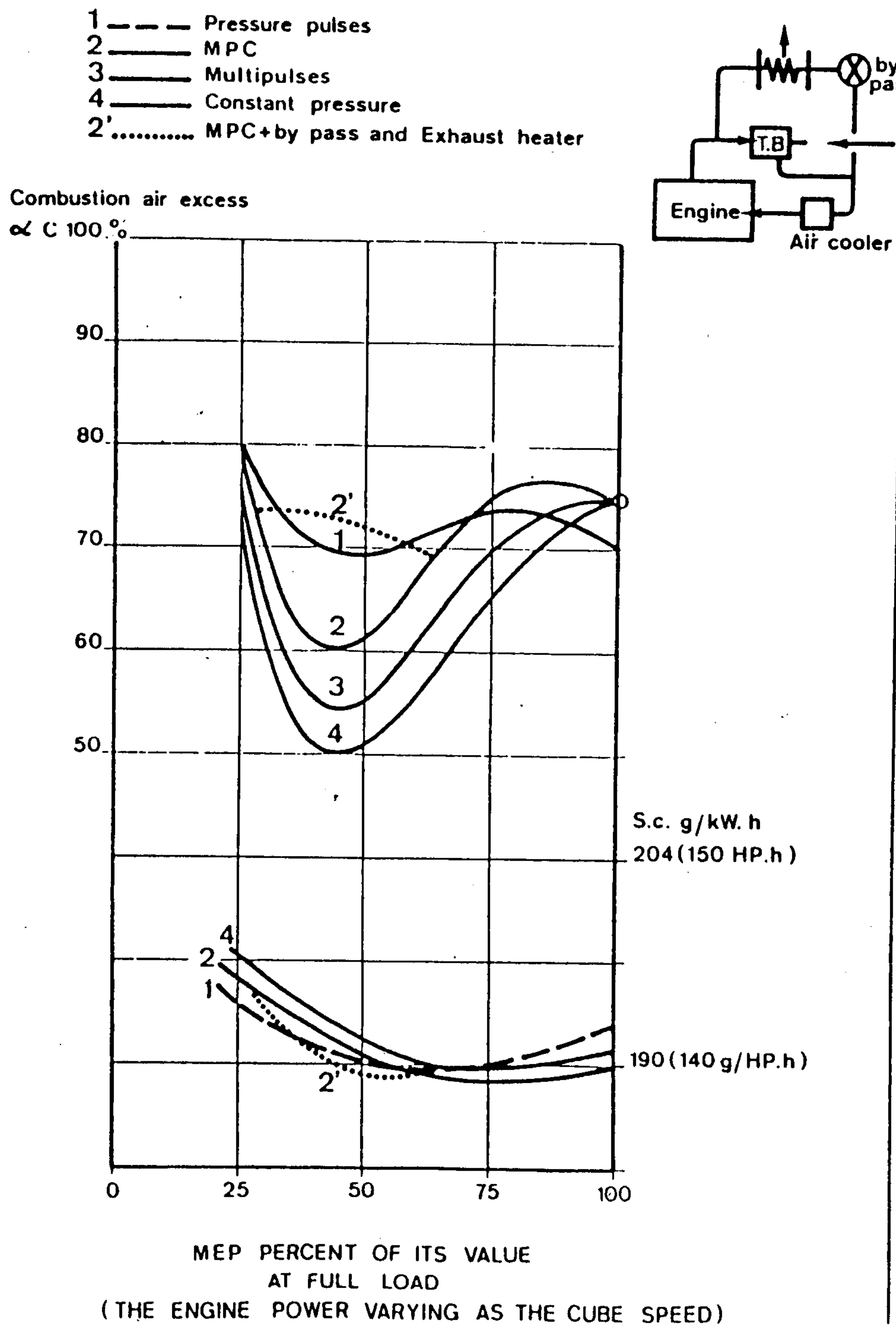

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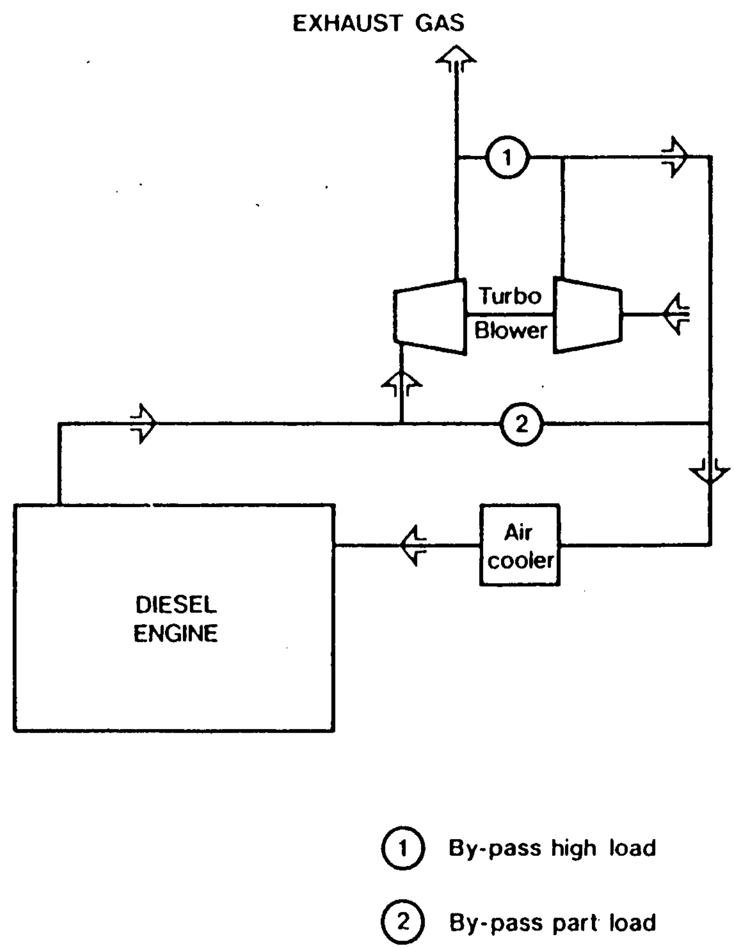
Fig.4

NEW SUPERCHARGING ARRANGEMENT



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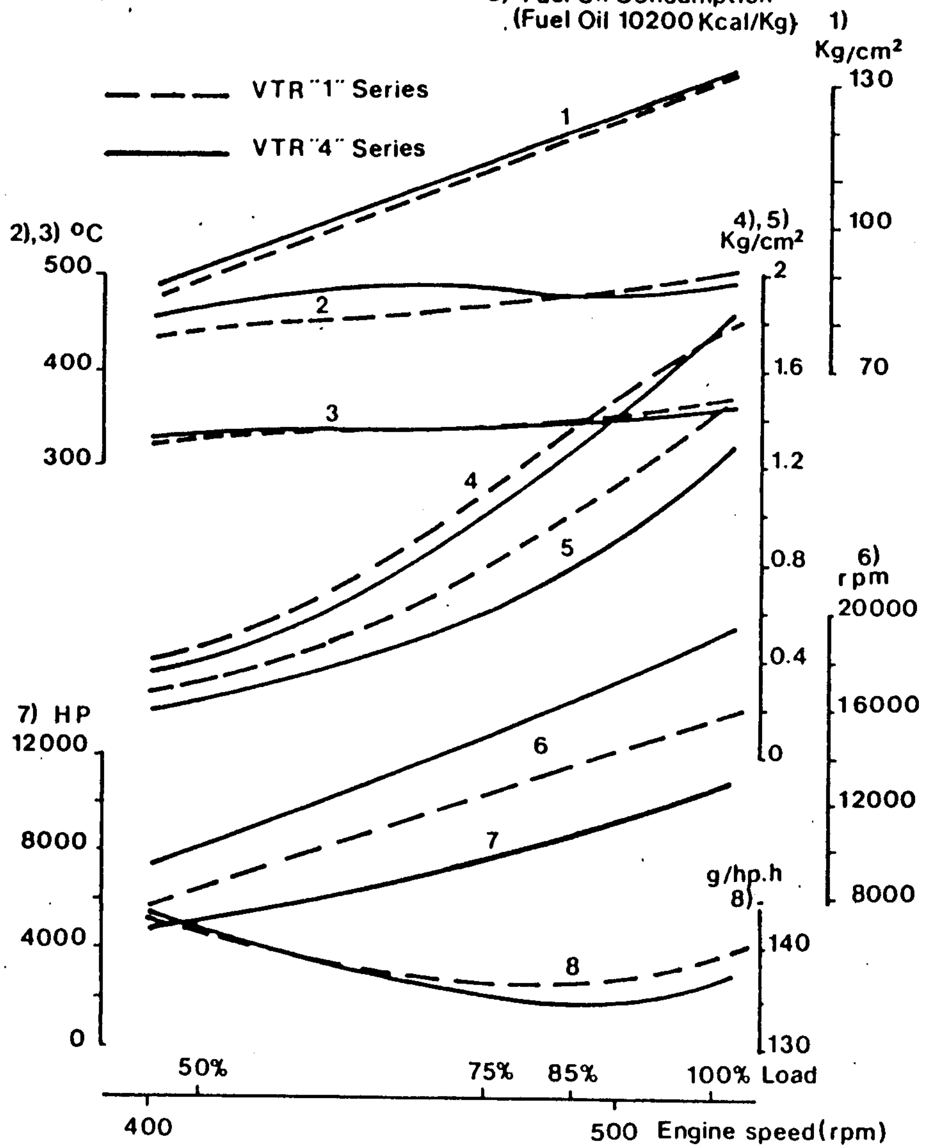
16 PC2-5V ENGINE

Fig.5

PERFORMANCE CURVES

PROPELLER LAW

1) Max. Combustion Press.
2) Exh.Gas Temp. before Turbine
3) Exh.Gas Temp. Cyl. Outlet
4) Charge Air Press.
5) Exh. Gas Press. before Turbine
6) Speed of Turbocharger
7) Brake Horse Power
8) Fuel Oil Consumption (Fuel Oil 10200 Kcal/Kg)



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PC2-6 FUEL PUMP

Fig 6

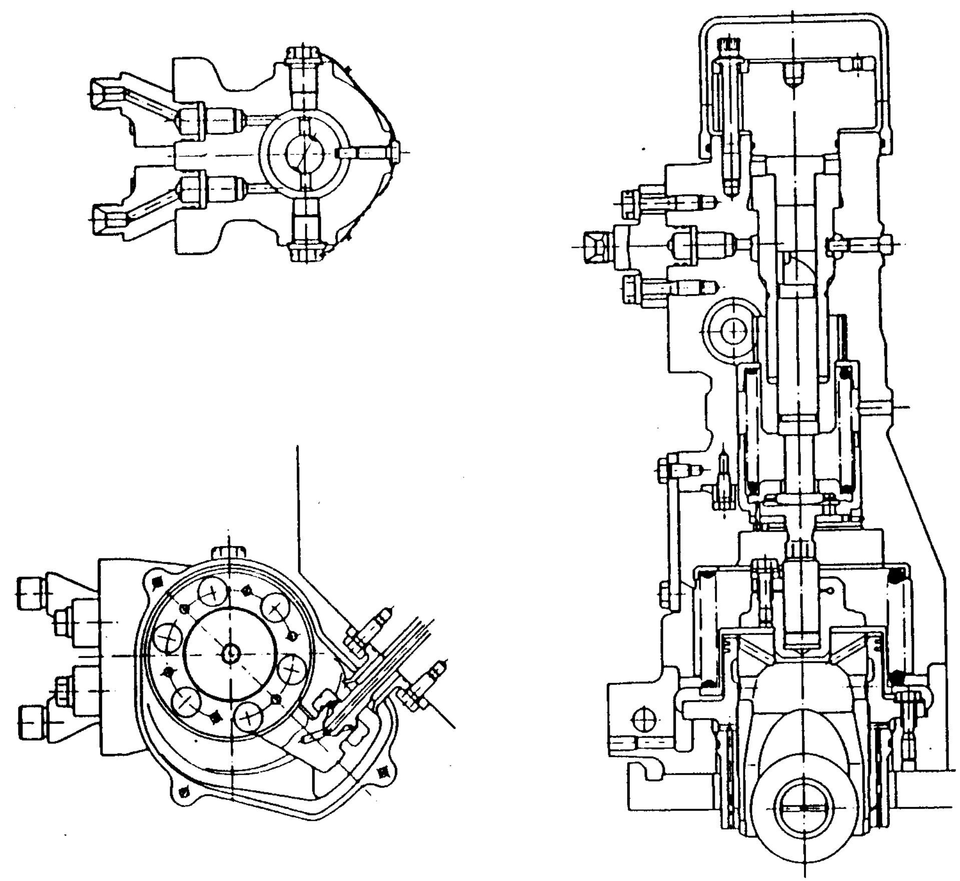

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Fig. 7

SPECIFIC FUEL CONSUMPTION RECORDED ON PC2-6 ENGINES RECENTLY TESTED

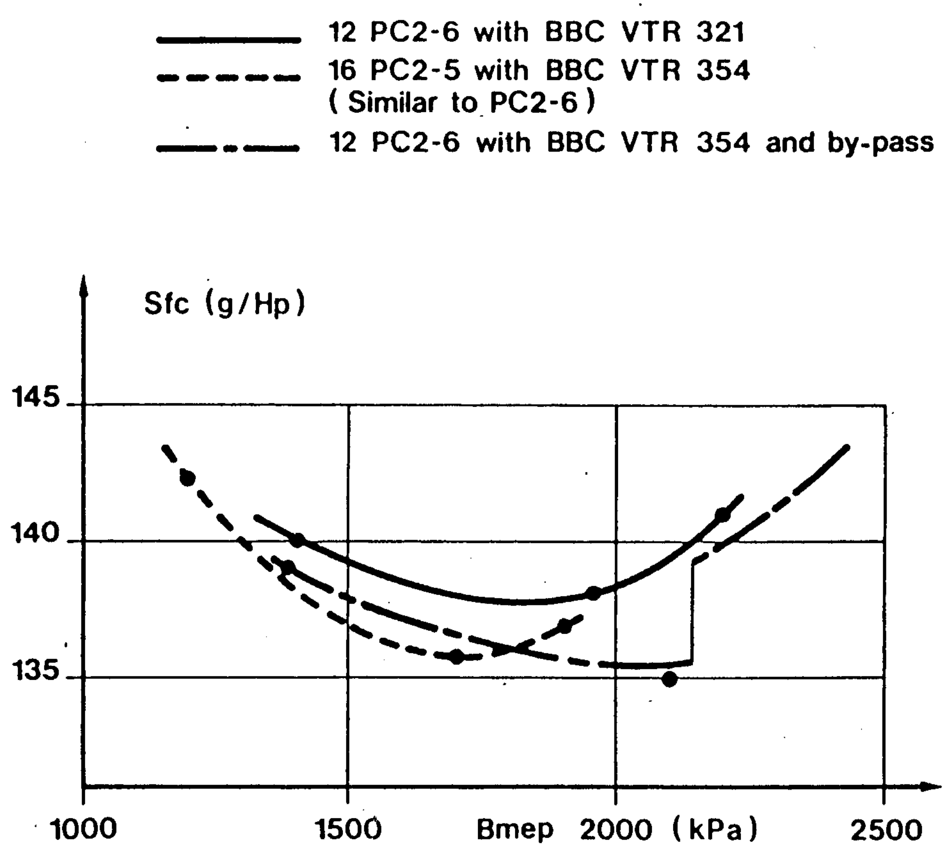

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Fig. 8

SPECIFIC FUEL CONSUMPTION RECORDED ON PC4-2 ENGINES RECENTLY TESTED

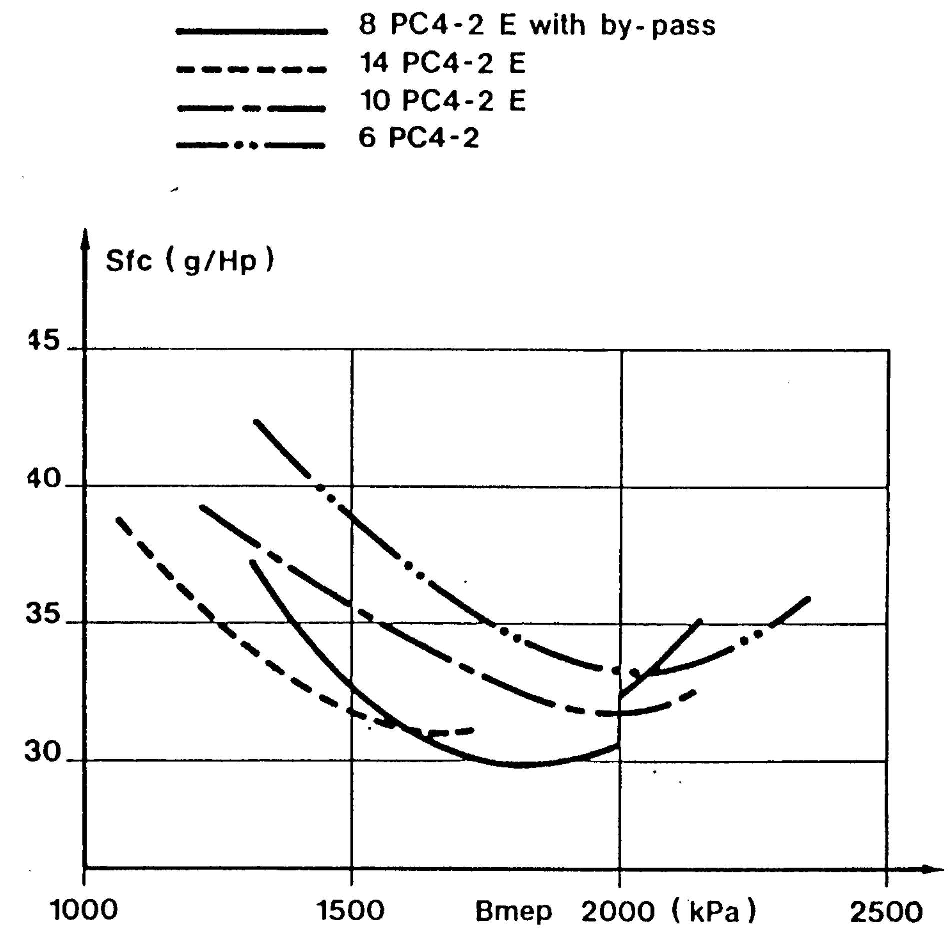

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Fig 9

PRODUCT AVERAGE QUALITY AT DIFFERENT REFERRING STAGES
QUALITE MOYENNE DES PRODUITS EN DIVERSES ETAPES DU RAFFINAGE

	1	2	3	4	5	*Max. Limits* 6 Limites Max
Viscosity / Viscosité cSt 50°C	420	$\simeq$ 500	5-30	1000	600-1000	600
Density Densité	0,96	0,99	1,05	$\simeq$ 1,00	> 1	0,990
Carbon Conradson Carbone Conradson	10	25	*Slight* Faible	22	-	22
Asphaltenes Asphaltènes	5	7	*Slight* Faible	12	-	12
Sulphur Soufre	4 maxi	4 maxi	*Slight* Faible	5	-	5
Silicium Alu. (ppm)	0	0	> 400	-	> 200	≤ 40-50
Cetane Number Indice de Cétane	40-45	35-40	$\simeq$ 0	25-30	15-25	20-40

These values are for information and may change appreciably
Ces valeurs sont indicatives et sont susceptibles de varier assez sensiblement

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Fig.10

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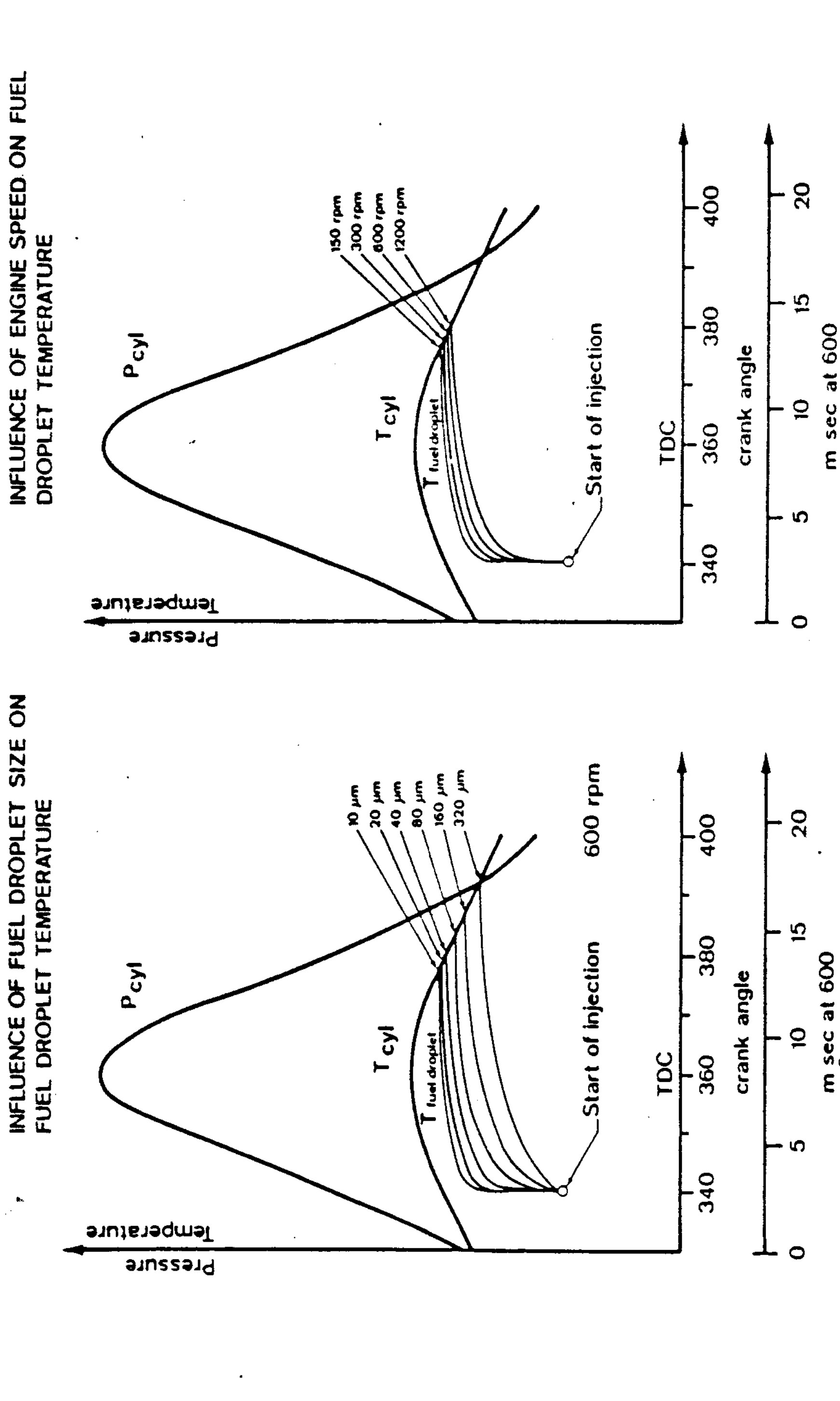

References "CIMAC 79 D7": The future Marine propulsion system, trends, problems and possibilities.

L.T. COLLIN

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S.E.M.T. PIELSTICK	PC2-6 ENGINE	Fig.11

GAS OIL INJECTION TEST BENCH

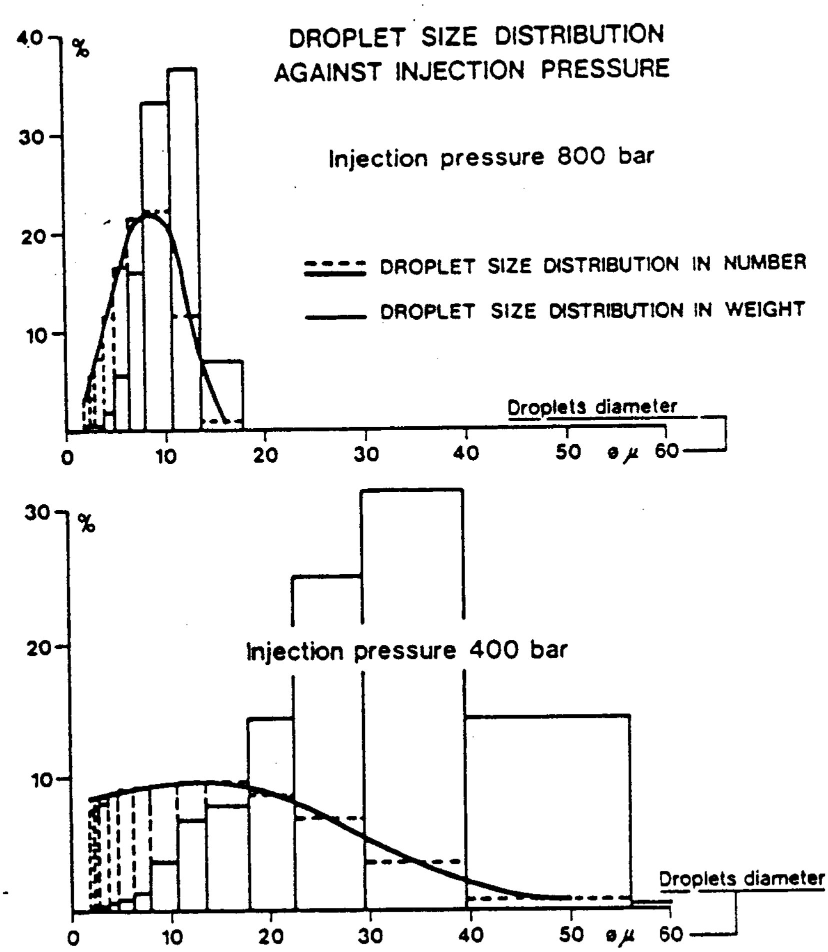

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Fig.12

PC 2-6 Engine
Mean fuel dropplet size in weight

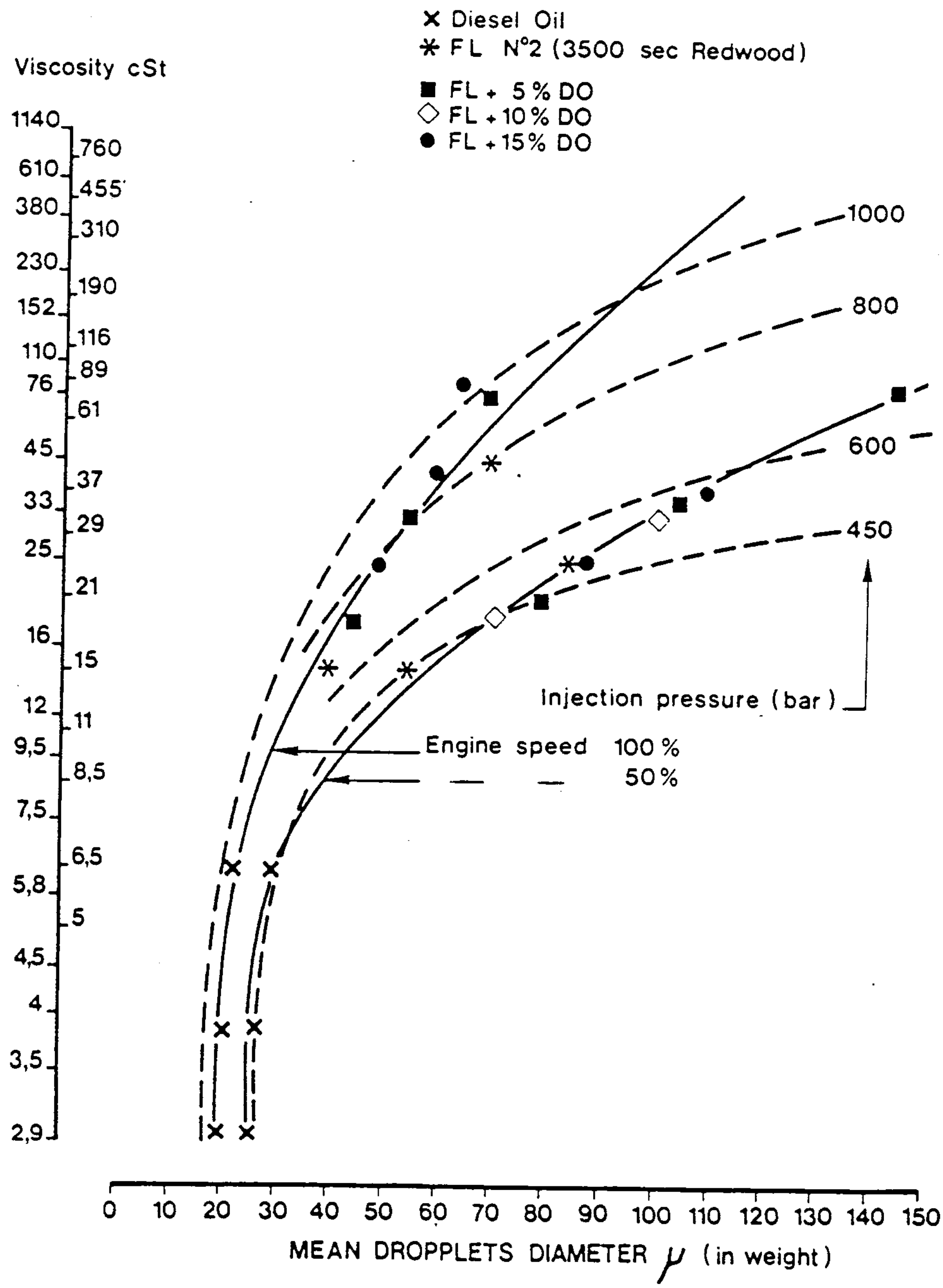

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Fig.13

TREATMENT OF SLURRY WITH CENTRIFUGE AND FILTER

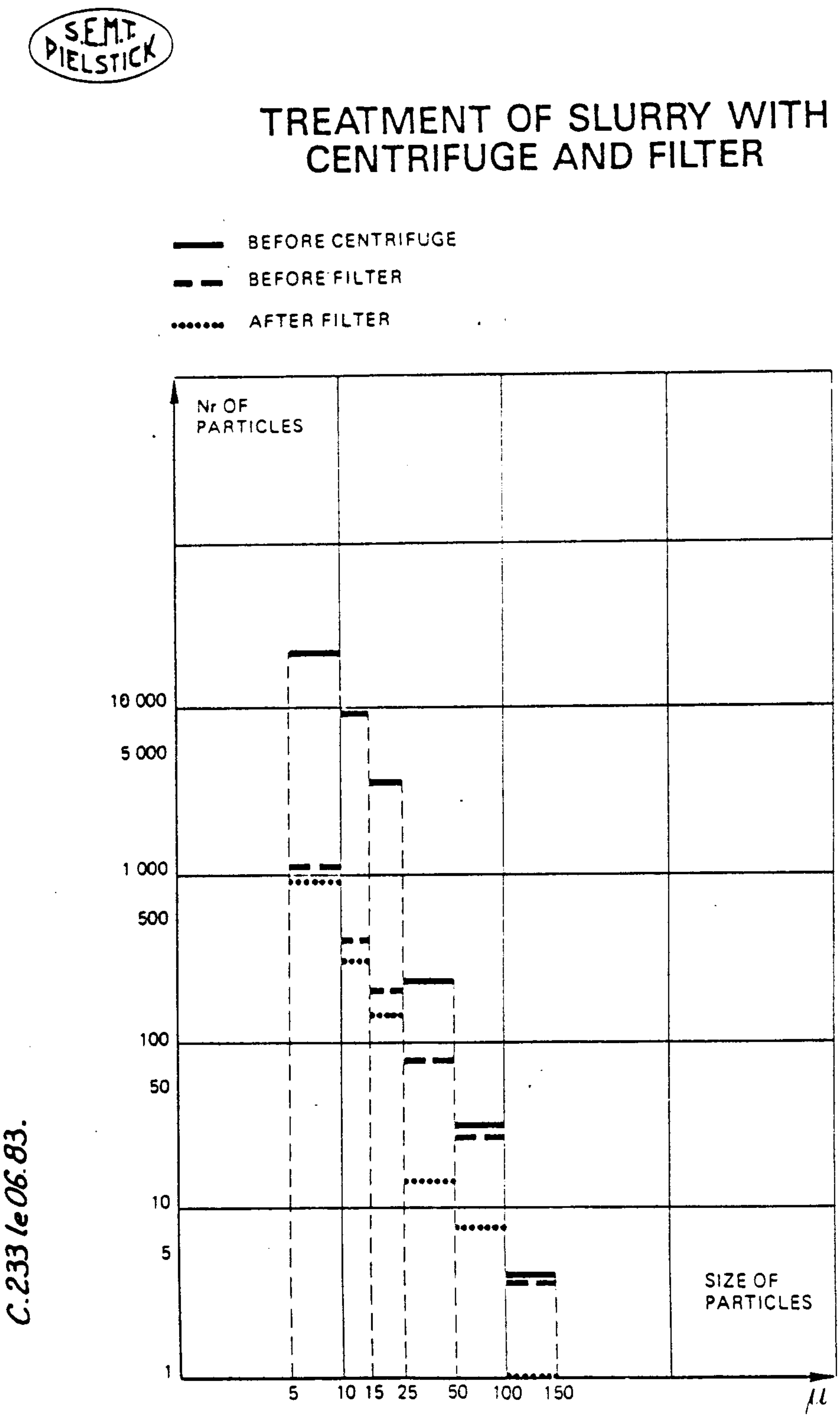

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Fig. 14

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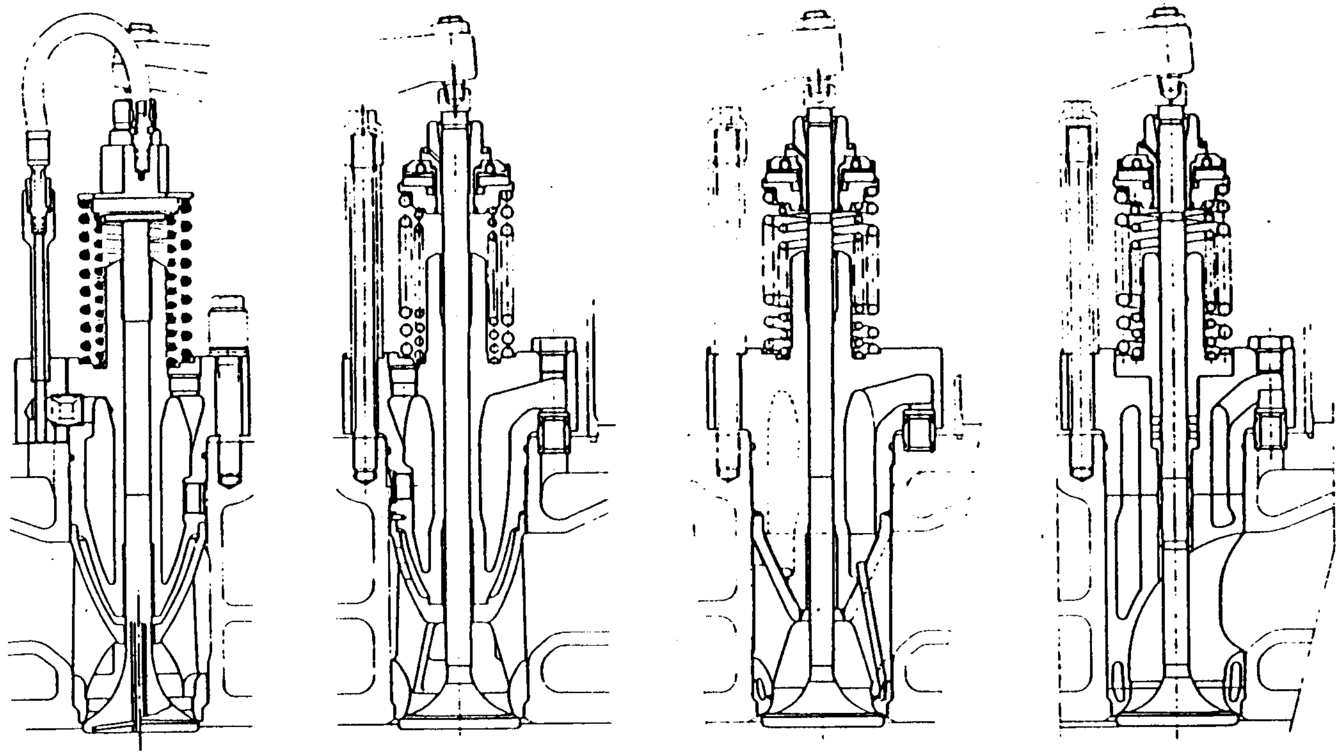
AVERAGE EXHAUST VALVE SEAT TEMPERATURE AT 80 % OF MCR

375 °C   540 °C   475 °C   425 °C



(- 165 °C)   (- 65 °C)   (- 115 °C)

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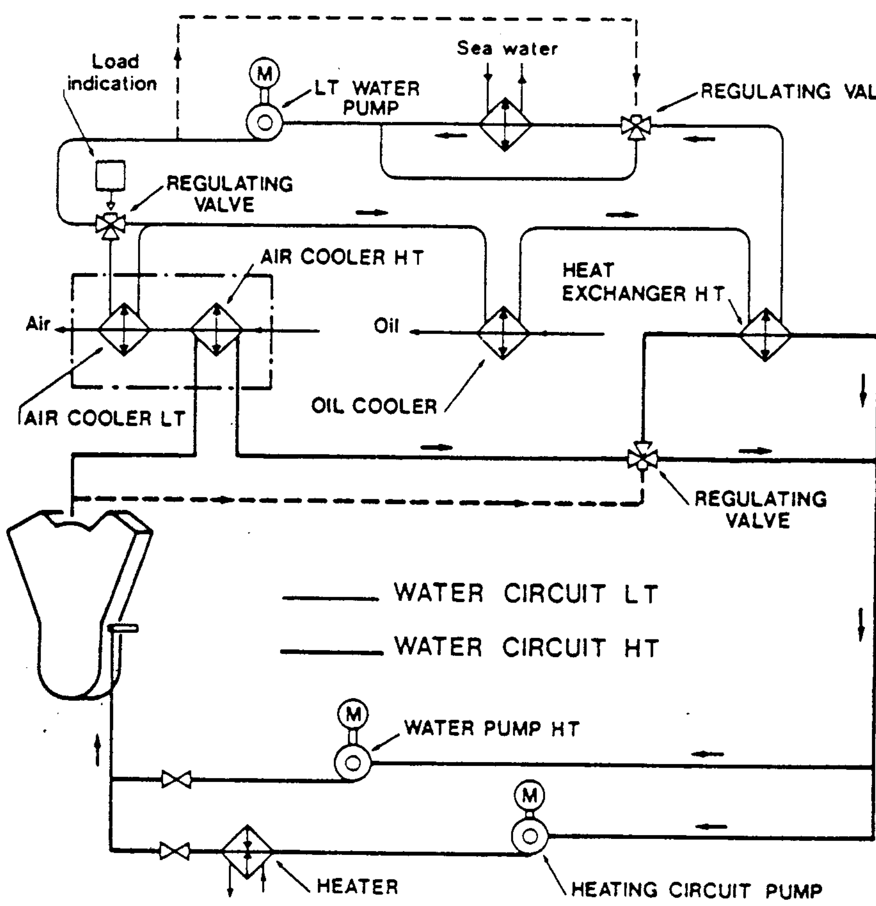
Fig.15



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HEAVY FUEL CIRCUIT

Fig. 16

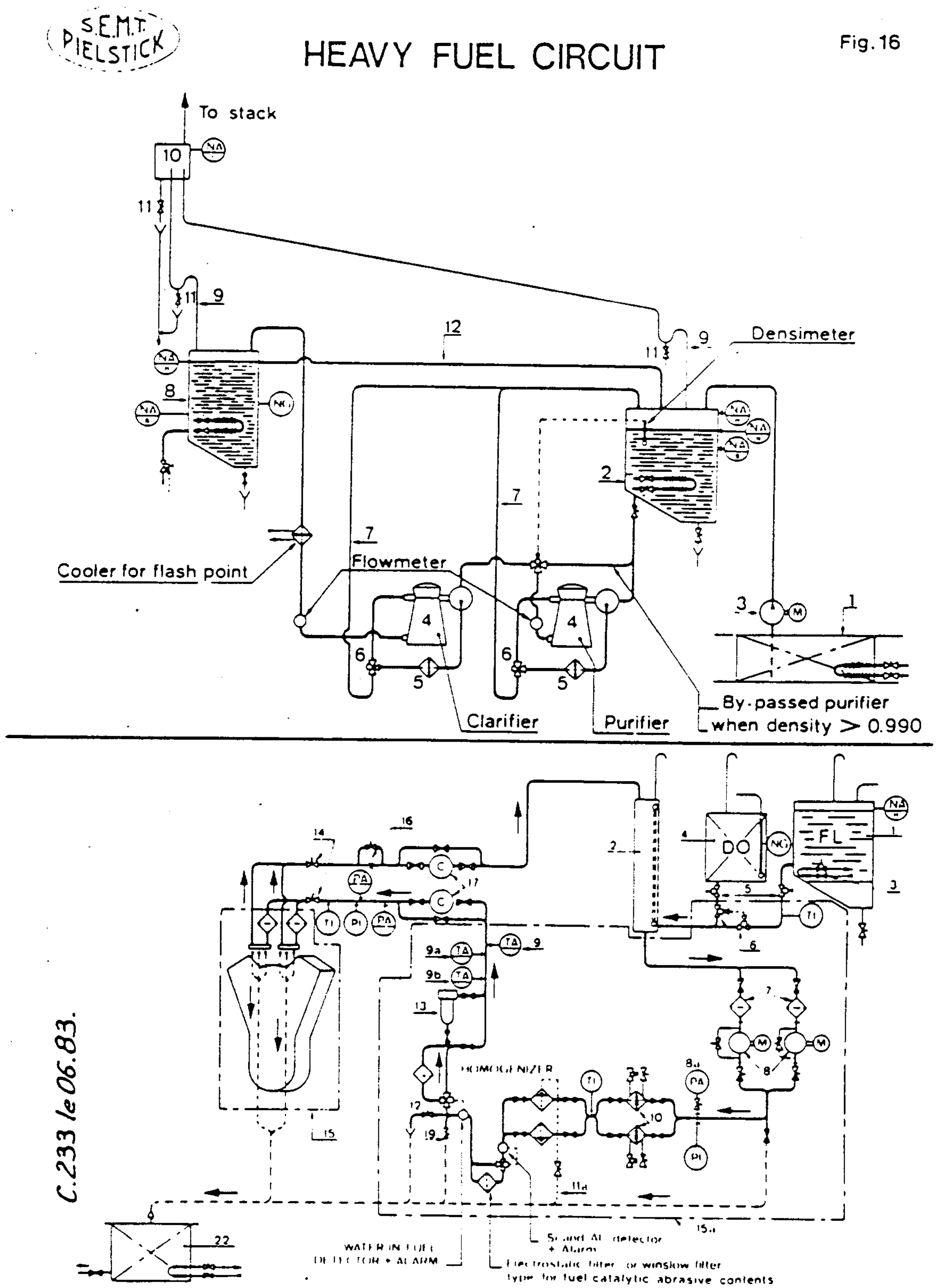

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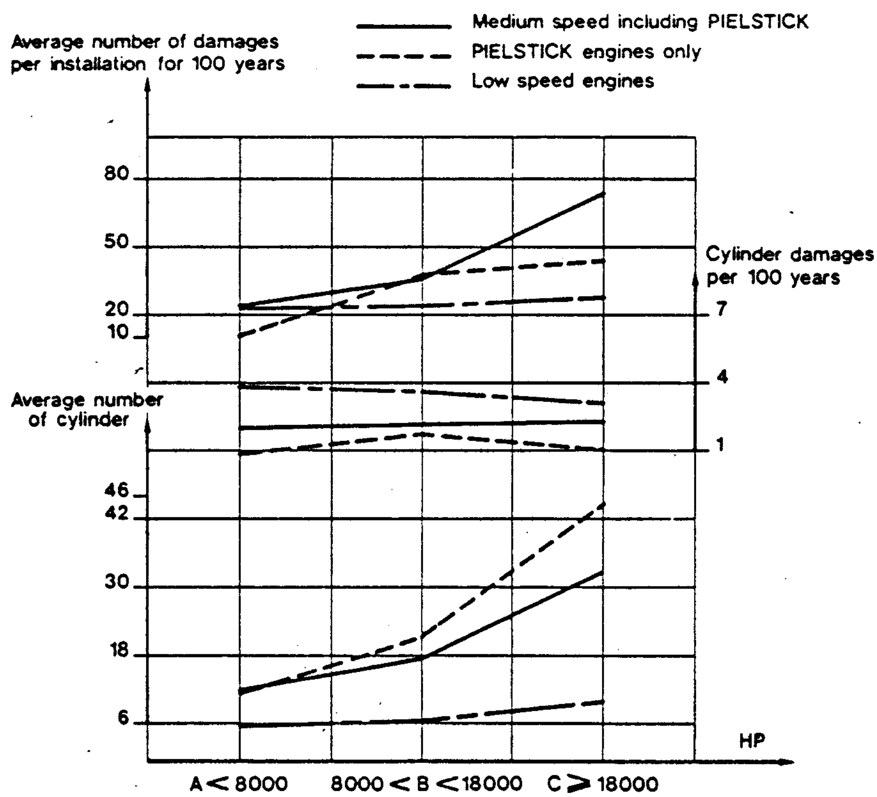
Fig . 17

LLOYD'S REGISTER STATISTICAL



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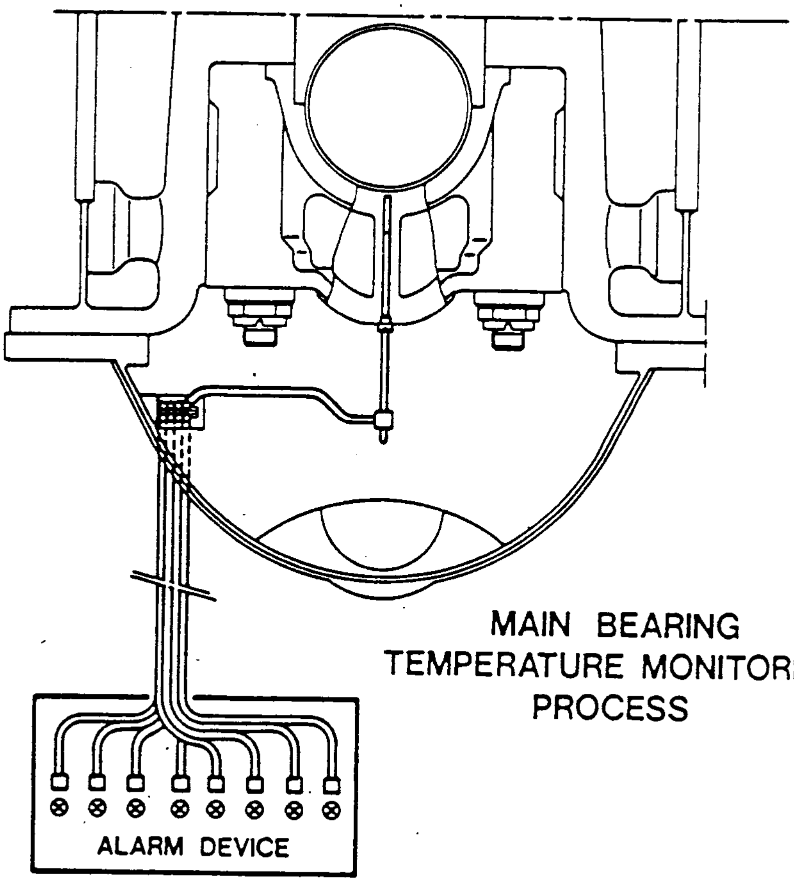

MAIN BEARING
TEMPERATURE MONITORING
PROCESS

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Fig.19

10 PC4 V-ANTE BANINA

EXHAUST GASES WASTE HEAT RECOVERY: CHARACTERITICS AT SEA ACCORDING TO THE POWER OF THE ENGINE.

(MEASURED DURING SEA TRIALS)

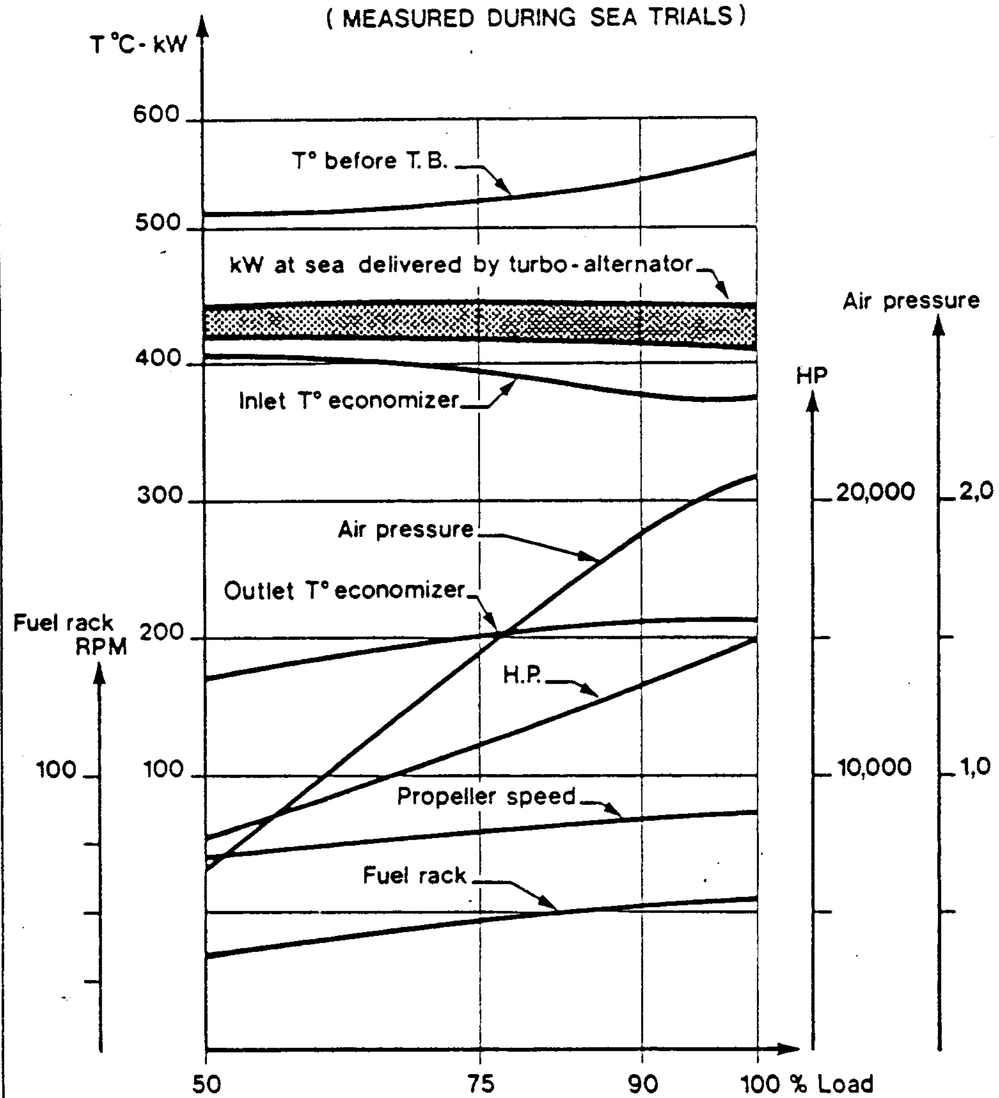

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Fig.20

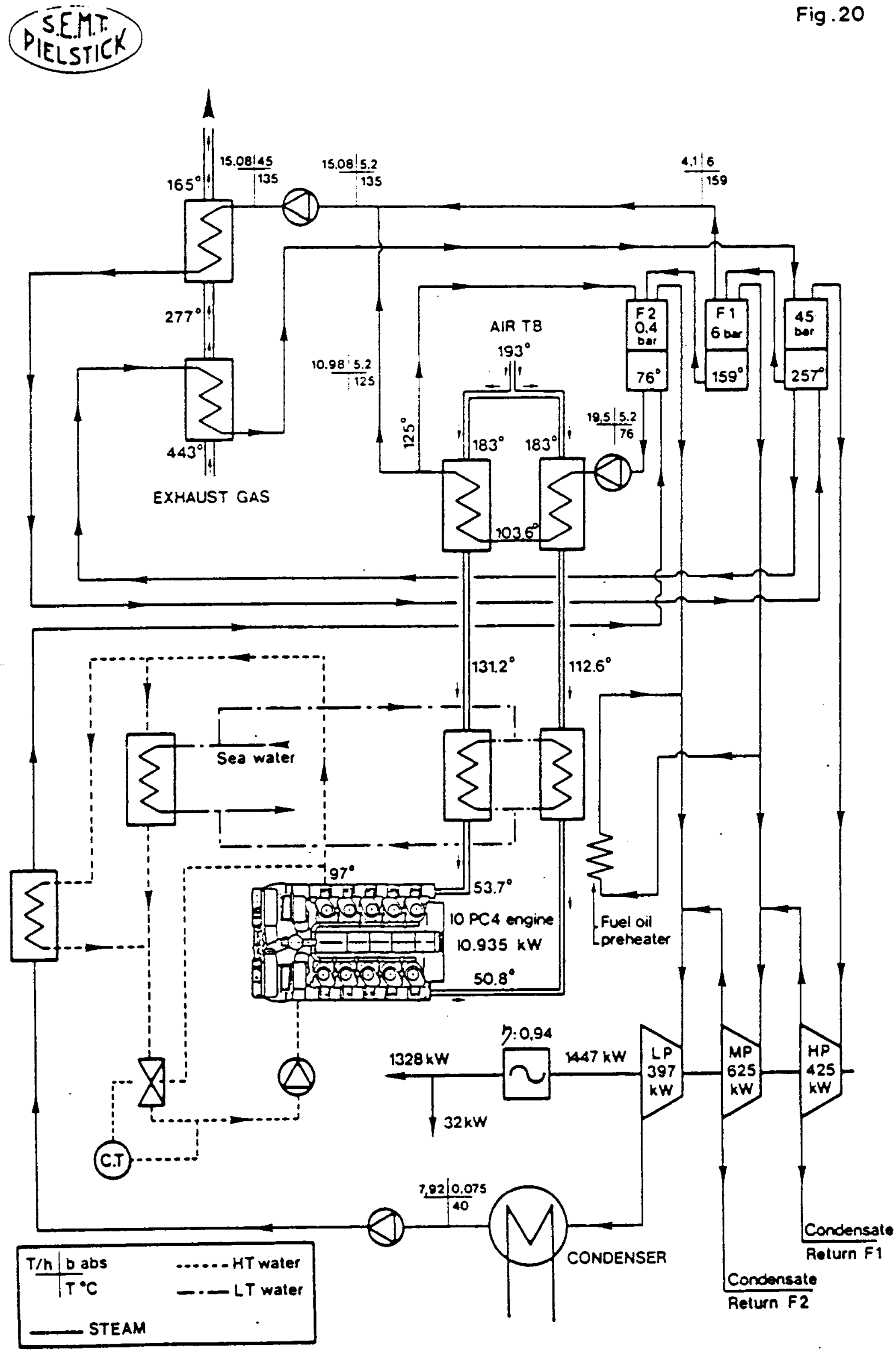

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Fig. 20 a

SCHEMATIC LAYOUT OF S.S.G. SYSTEM

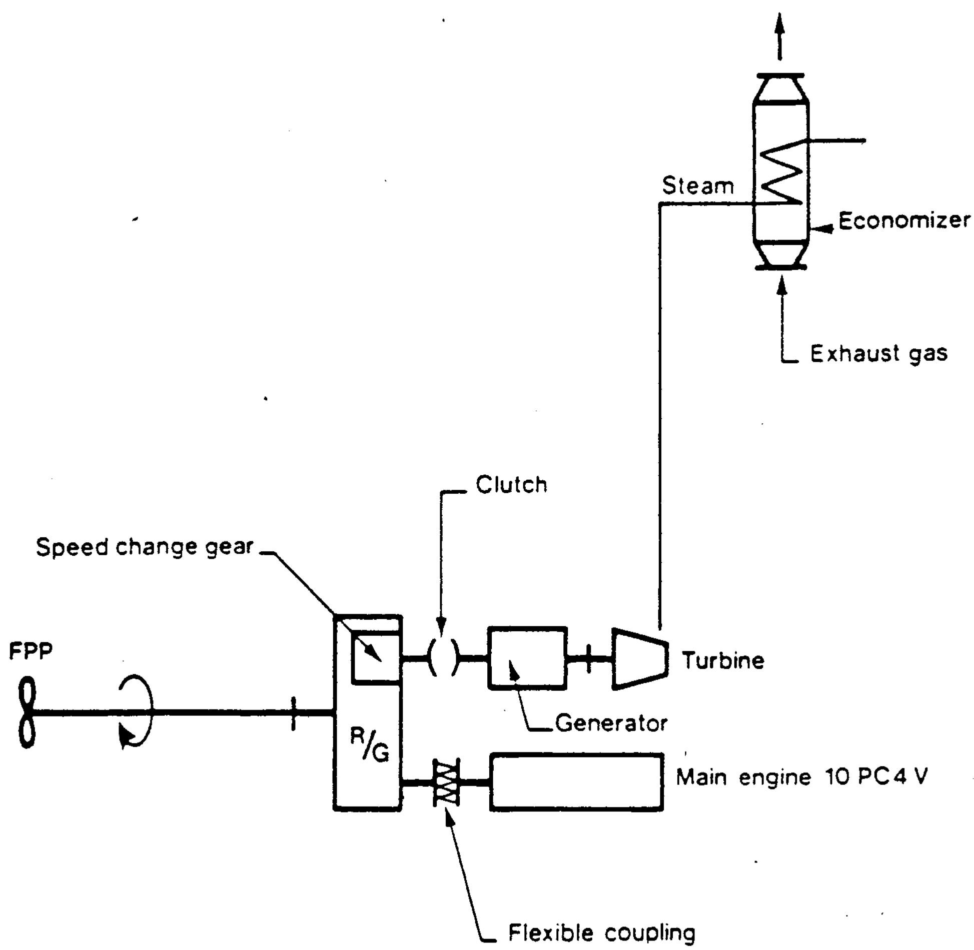

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Fig.20b

SCHEMATIC LAYOUT OF NEW SYSTEM

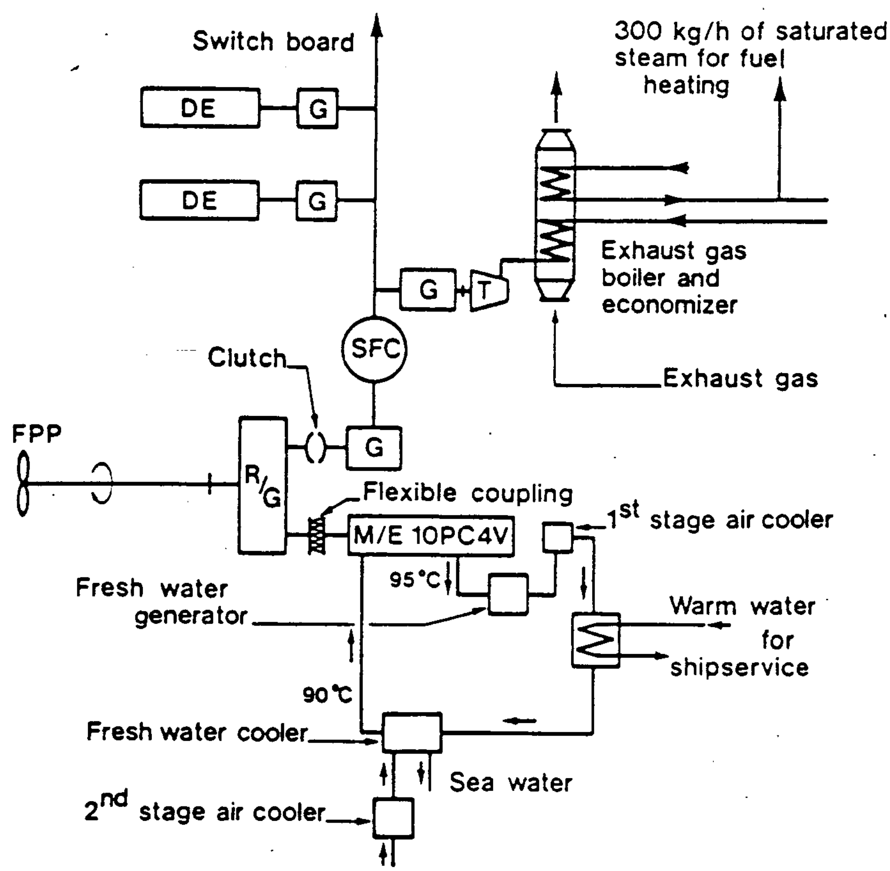

Fig. 21

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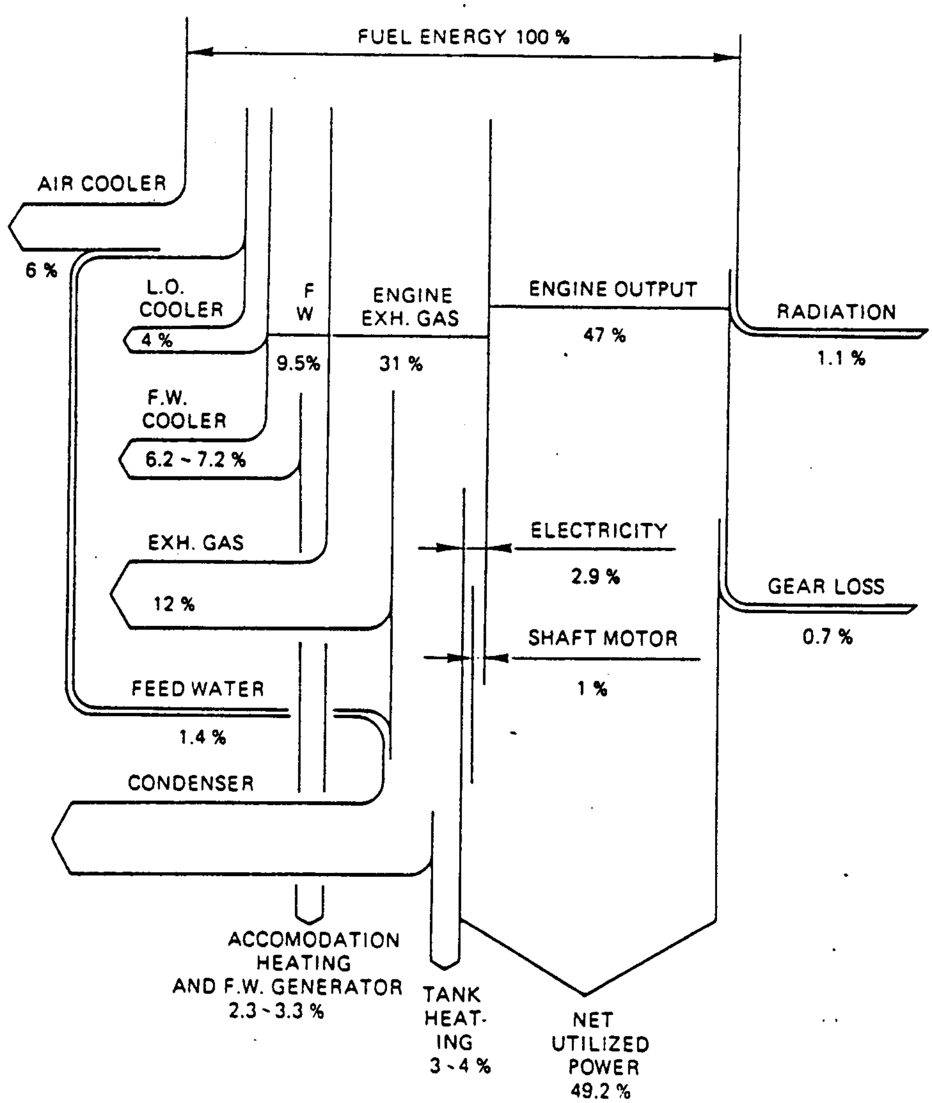
HEAT BALANCE DURING NORMAL SEA GOING CONDITION OF M.S. SHOHO MARU 14 PC4 V



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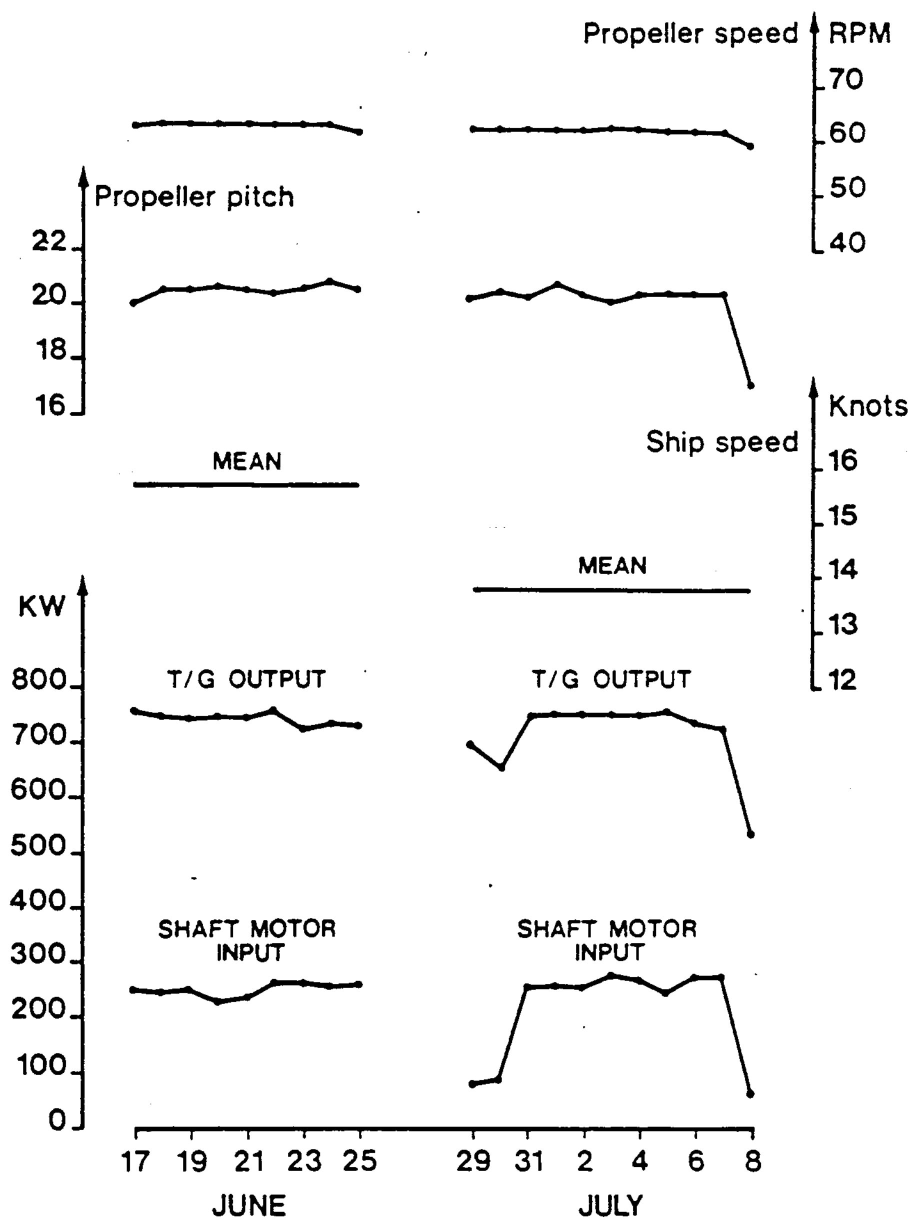
SERVICE RESULTS OF SHOHO MARU FIRST VOYAGE

Fig.22



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S.E.M.T. PIELSTICK		Fig.23

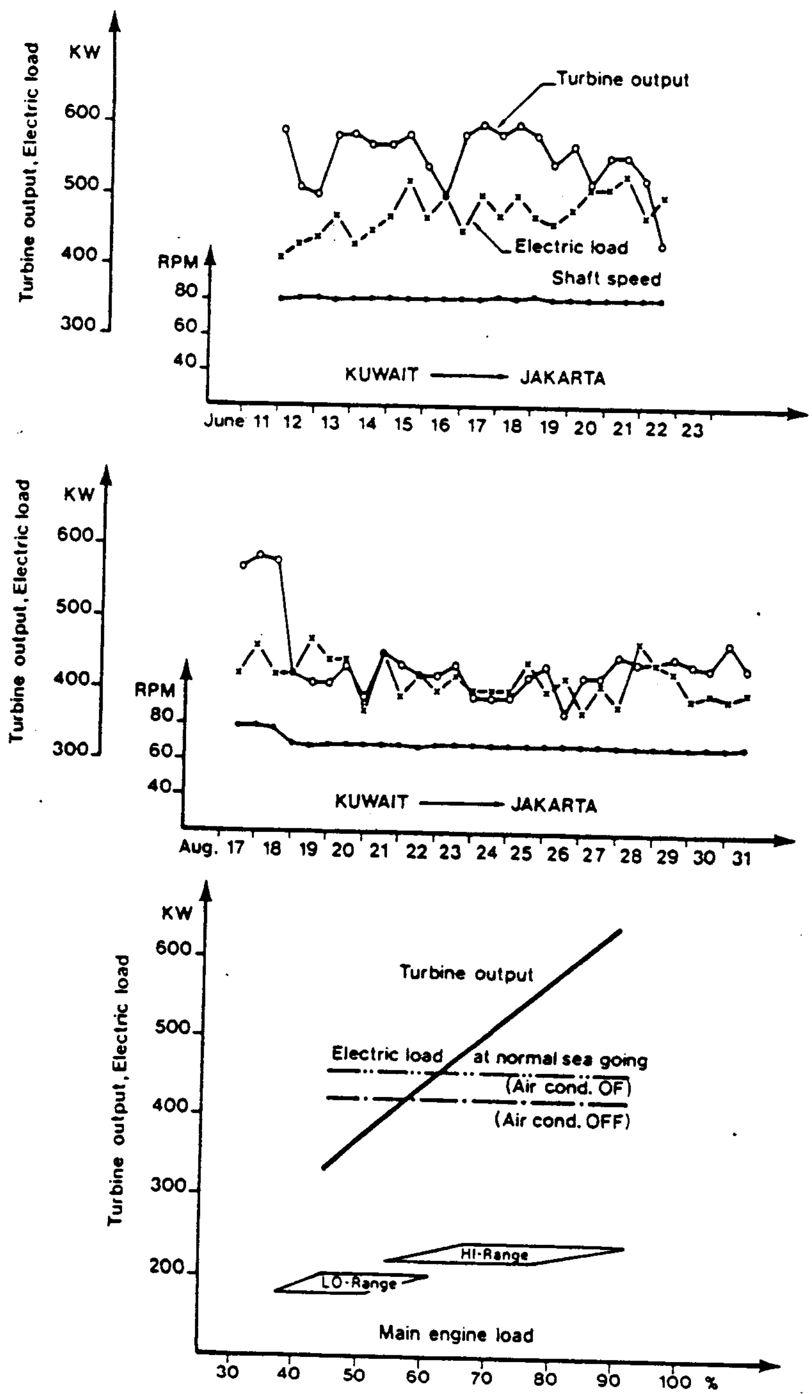

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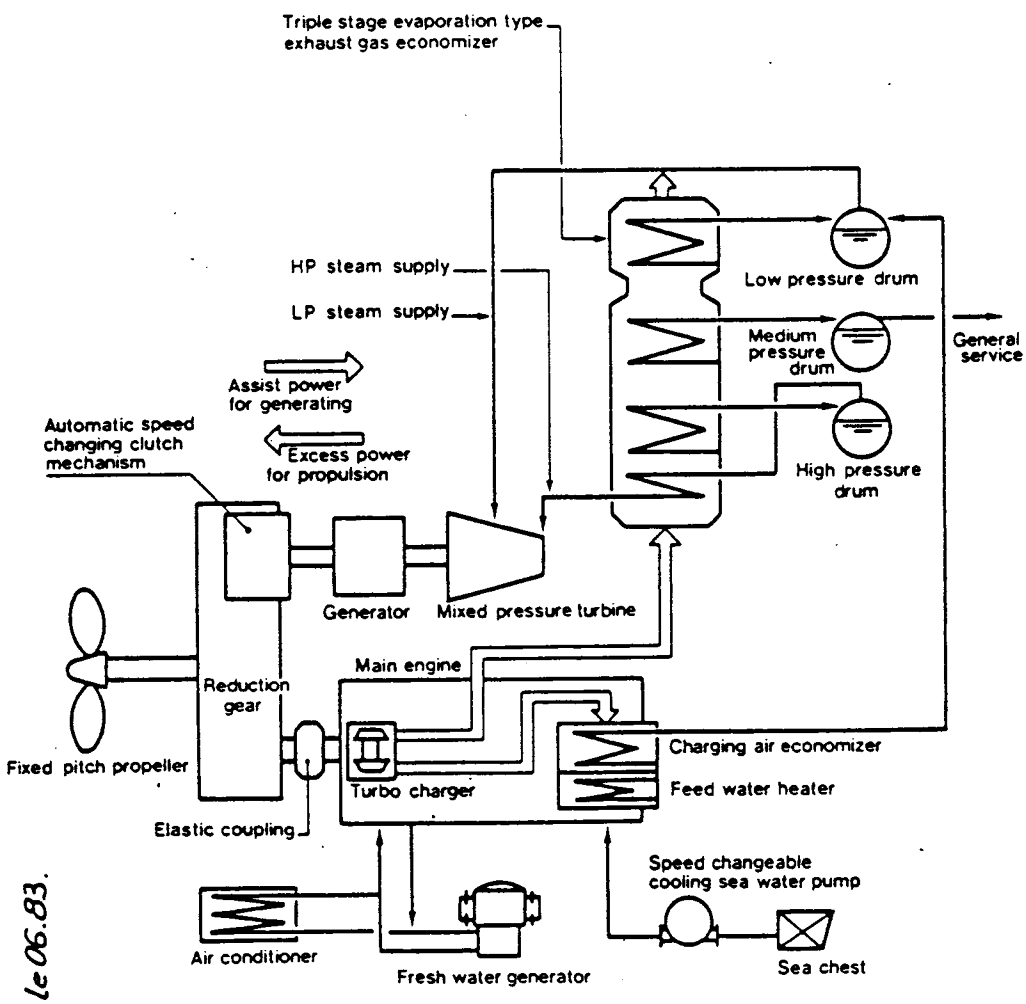
Fig.24

SCHEMATIC DIAGRAM OF THE SSG "MARK-II" SYSTEM



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