Chapter 2 Recent Development in Marine Engines

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2.1 Diesel Engine Emission Technology

2.1.1 Introduction

Everyone is more concerned of these days of environmental issues. Gradually efforts are being made globally to curb pollution in all its forms. Even though the contribution of large diesel engines to global pollution is relatively small, attention has been given to their exhaust emissions since the 1970s. At first, the focus was on stationary applications, and there are now various countries of which have environmental legislation requiring reductions of the exhaust emissions from diesel power plants of varying degrees up to 100 per cent. For example, most stringent emissions regulations apply for the city of Zurich, Switzerland, where the NOx emissions of diesel engines must be reduced by more than 99 per cent [2.1]. Since January 2000, exhaust emissions control has been extended to marine diesel engines worldwide through the regulations of the International Maritime Organization (IMO).

Essentially, diesel engine combustion comprises a series of batch processes. In the batch process, higher initial temperatures and pressures can be used than in the gas turbine, since the exposed components are cooled at the end of each process and between processes. Each piston stroke constitutes a batch process, and the slower it can be while still maintaining its adiabatic thermodynamics, the more efficient it can be. An added advantage of the slow process that takes place in a low speed engine is that the ignition delay may occur, depending on fuel quality and engine geometry, has less impact on a low speed engine. While a low speed engine often gives a longer ignition delay than its medium speed counterpart with the same fuel, the ignition delay is still proportionally shorter in a low speed engine, in terms of degrees crankshaft angle. This is one of the reasons why a low speed engine is considered more forgiving than other types of machinery when low-quality, low-cost fuels are used, as outlined later. Fig. 2.1, the two-stroke diesel engine is unrivalled as the most fuel-efficient prime mover, whether compared with other types of thermal engines. As shown in Fig. 2.2, the efficiency of diesel engines, and especially of two-stroke low speed diesels, is almost independent of load over a wide load range.

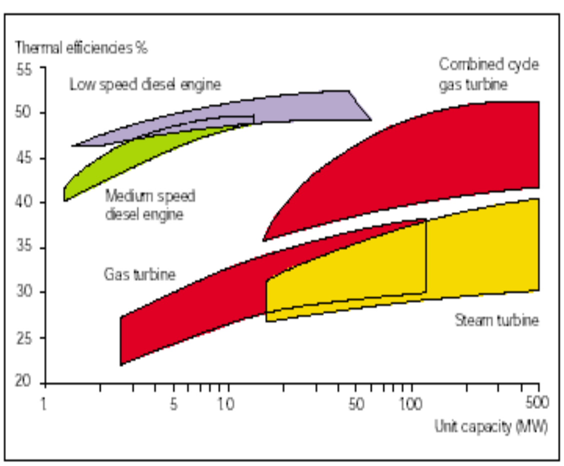


Fig. 2.1: Power efficiency comparison at ISO 3046 [2.2]

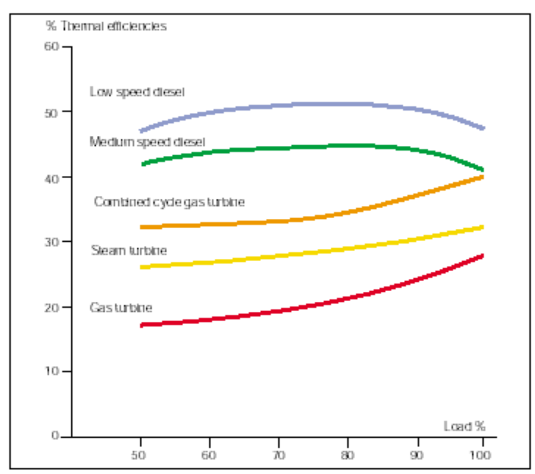


Fig. 2.2: Typical part load efficiencies of prime movers [2.2]

Owing to the high efficiency of diesel engines, the emissions of carbon dioxide, carbon monoxide and hydrocarbons are low in their exhaust gases but high emissions of nitrogen oxides are also inherently characteristic of the diesel cycle. The same high combustion temperatures that give high thermal efficiency in the diesel engine are also the most conducive to NOx formation. By running on low-quality fuels with a low fuel consumption, large diesel engines offer enormous savings in fuel costs compared with those of alternative prime movers. Yet the heavy fuel oil leads to substantial emissions of oxides of sulfur and particulates. They are formed in the combustion process out of the sulfur, ash and asphaltenes that are always present in heavy fuel oil. However, as shown in this paper, there are ways of reducing such emissions, either by combustion control techniques or by exhaust gas after-treatment.

2.2 Diesel Engine Emissions

Exhaust gases emitted from large diesel engines comprise nitrogen, oxygen, carbon dioxide and water vapor with smaller quantities of carbon monoxide, and oxides of sulfur and nitrogen, together with even lesser quantities of partially reacted and unburned hydrocarbons and particulate material, as shown in Figs. 2.4 and 2.5. The composition of major pollutants in the exhaust gas is a result of the engine process.

The composition of major pollutants in the exhaust gas is a result of the engine process, its fuels and the means employed to control the emissions. In the following, each exhaust gas component will be described.

Oxygen (O_2): Low speed, two-stroke, crosshead diesels operate with an air excess ratio of over 3. More than half of the air is available for the combustion process, while the remaining part is scavenged through the cylinder. Hence, the exhaust gas contains some 13-16% oxygen, and this has to be considered when calculating the concentration of various compounds in the gas. Some exhaust gas emission regulations refer to 15% oxygen; so, if the actual content is different, the result will have to be corrected accordingly.

Nitrogen (N_2): Given the above-mentioned air excess ratio, it is clear that nitrogen constitutes the major part of the exhaust gas. As nitrogen is practically inactive, only a small but, as will be seen later, an important part is involved in the chemical reactions in the engine.

Carbon Dioxide (CO_2): Basically, the complete combustion of hydrocarbons will produce carbon dioxide and water vapor, and the relative amounts of these will be a function of the hydrocarbon composition. Carbon dioxide, although not toxic, has recently been given much attention because of the so-called ‘greenhouse effect’. The use of machinery with a high thermal efficiency and of fuels with relatively low carbon content is the only viable means of reducing carbon dioxide emission.

Carbon Monoxide (CO): Carbon monoxide is a toxic gas. The formation of CO is, in principle, a function of the air excess ratio and the combustion temperature. The formation is strongly influenced by the uniformity of the air/fuel mixture in the combustion chamber, and we expect this to be the decisive factor in our engines. Carbon monoxide is a compound that is still burnable, so elimination in the engine process is preferable.

Smoke: A traditional measure of the combustion quality, and a traditional way of qualifying the ‘emission’, is to look at, or to measure, the smoke intensity. The exhaust gas plume, when it leaves the top of the stack, may be visible for various reasons, e.g. its content of particulate matter and nitrogen dioxide, NO_2 (a yellow/brown gas), or of condensing water vapor. Although it may be argued that these components are either subject to separate legislation (NOx, particulate matter) or not harmful (water), it is a fact that smoke and/or opacity limits are enforced in certain countries, e.g. in the USA. Unfortunately, methods of measuring smoke and opacity vary (see the section on measuring methods) and the figures resulting from the different methods are not really comparable. When considering visible emissions, we should bear in mind that the larger the engine, the more likely it is that the exhaust gas plume will be visible. This is because, for a given Bosch Smoke Number (BSN value), the greater the diameter of the plume, the greater the amount of light it will absorb. For instance, a BSN of 1 will mean almost invisible exhaust gas from a truck engine, but visible exhaust gas from a large, low-speed engine. Typical smoke values for the most modern diesel engines are so low that the exhaust plume will be invisible, unless water vapor condenses in the plume, producing a gray or white color. However, the NO_2 may give the plume a yellowish appearance.

Particulate Materials (PM): Particulate materials in the exhaust gas may originate from a number of sources:

- agglomeration of very small particles of partly burned fuel,
- partly burned lube oil,
- ash content of fuel oil and cylinder lube oil,
- sulphates and water.

The contribution from the lube oil consists mainly of calcium compounds viz. sulphates and carbonates, as calcium is the main carrier of alkalinity in lube oil to neutralize sulfuric acid.

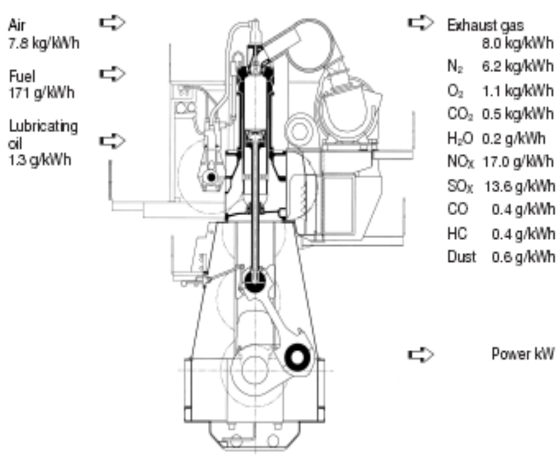


Fig. 2.3: Typical analysis of the exhaust gases from modern two-stroke marine diesel engines[2.2]

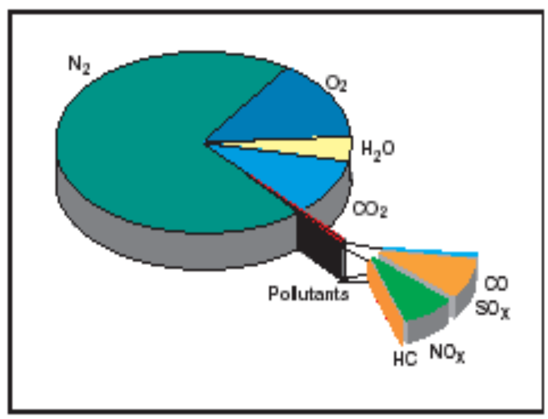


Fig. 2.4: The pollutants in the exhaust gases (% vol) of large diesel engines [2.9]

2.3 Emission Measuring Methods

There are various ways of measuring the components in the exhaust gas. Not only the measuring equipment (the analyzers), but also the measurement procedures and the sampling technique are important in obtaining reliable results. Analyzers and procedures are specified in several standards (see e.g. ISO 8178 [2.10]). In the following, some measuring principles are brief described.

NOx is measured by two methods, the chemiluminescence analyzer (CLA) approved for certification and the electrochemical sensor (ECS) used in almost all portable instruments. The CLA can be heated to avoid NO_2 condensation (together with a heated sampling line). This is a common way of measuring exhaust gas from diesel engines. The CLA measures only NO but, by means of a NO_2-to-NO converter, the NOx is measured as NO. The chemical cells measure each component (NO, NO_2) individually.

CO and CO_2 is usually measured with an infrared (IR) technique. Because this is an optical-light technique, the analyzers are also sensitive to other gas components. Thus, depending on the amount of these components, we must compensate for the cross-sensitivity. Furthermore, because of the wide range of individual components in the exhaust gas, different cells may have to be used to obtain the necessary sensitivity. Similar to measuring NO and NO_2, also for measuring CO, ECS cells may be used for traveling purposes.

SOx: There are several methods of measuring SOx; infrared (IR), ultraviolet (UV) or ECS cells. But the accuracy of all cell types is still being discussed. Therefore, ISO [2.10] calculates SO_2 in the raw exhaust gas from the fuel-sulfur content. If after-treatment is used, or SO_3 is to be measured in the exhaust, different methods must be used.

Particulate emissions are measured with two different principles that may produce very different results. The new ISO standard for diesel engines is a dilution-tunnel method, as opposed to the common power plant raw exhaust methods. The difference is the mixing of air into the exhaust gas to simulate atmospheric condition and the temperature of the filter, where the particulate matter (PM) is collected. Since these conditions are important to the amount of material condensed on the particles, the conditions or the method used must be specified with the PM value.

Dilution-tunnel method: A sample is taken from the exhaust gases, diluted with clean air (to emulate the situation after the funnel), and then passed through a filter maintained at a temperature of 52°C maximum. The mass of the collected material is then determined gravimetrically and referred to as 'particulates'.

Smoke spot methods: A certain volume of exhaust gas is drawn through a filter paper and the reduction in reflectance of the used filter paper caused by the soot captured on it is measured or evaluated. Examples are Bosch Smoke Number, SAE Smoke Number (Scale 0-10), Bacharach (ASTM) Smoke Number (Scale 0-9), etc. These two methods express the degree of blackening of a piece of white filter paper, through which a certain volume of exhaust gas has been drawn.

Opacity methods: The percentage that light is reduced when it is passed through the exhaust gas is determined. Those methods express the percentage of light that disappears when passing through a certain path length of exhaust gas, which means that particulate matters as well as gaseous molecules contribute to the measured value. US legislation prescribes the use of 'a qualified observer', who observes the exhaust gas plume and compares this with a gray-tone scale. This measurement is to some extent subjective and may be considerably influenced by the atmospheric conditions on site.

Three common methods that measure the opacity are Hartridge smoke value (% Hartridge), Ringelmann Number (Scale 0-5), and Opacity in general (various instruments). Those methods express the percentage of light which disappears when passing through a certain path length of exhaust gas, which means that particulate matter as well as gaseous molecules contribute to the measured value.

2.3 Common Rail Fuel Injection Systems

2.3.1 Introduction

Visible smoke has recently become a major issue in the marine market, especially for cruise and passenger vessels. As most harbors in the world are located close to densely populated areas, it can be foreseen that the demand for no visible smoke under any circumstances will become increasingly important in the future. State-of-the-art common rail injection technology now makes it possible to provide smokeless engines. With today's mechanical injection systems the fuel injection pressure is a function of engine speed and engine load. This means that at low load the injection pressure drops, and the result is very large fuel droplets that survive as droplets until they touch the combustion space surfaces. This situation can be changed using the common rail injection system.

2.3.2 Fundamental of Sulzer Common Rail Fuel Injection

The common rail is a manifold running the length of the engine at just below the cylinder cover level illustrated in Fig. 2.5a and 2.5b as an example of the common-rail fuel systems in Sulzer RT-flex engine. It provides a certain storage volume for the fuel oil, and has provision for damping pressure waves. The common rail and other related pipework are neatly arranged beneath the top engine platform and readily accessible from above (Fig. 2.7). The common rail is fed with heated fuel oil at the usual high pressure (nominally 1000 bar) ready for injection. The supply unit has a number of high-pressure pumps running on multi-lobe cams. The pump design is based on the proven injection pumps used in Sulzer four-stroke engines.

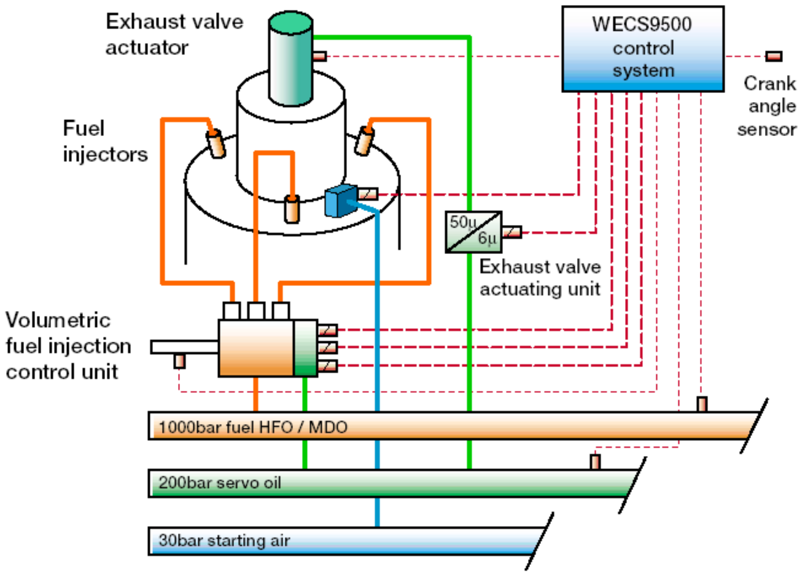


Fig. 2.5a: Schematic of the common-rail systems for fuel injection and exhaust valve actuation in the Sulzer RT-flex engine [2.4]

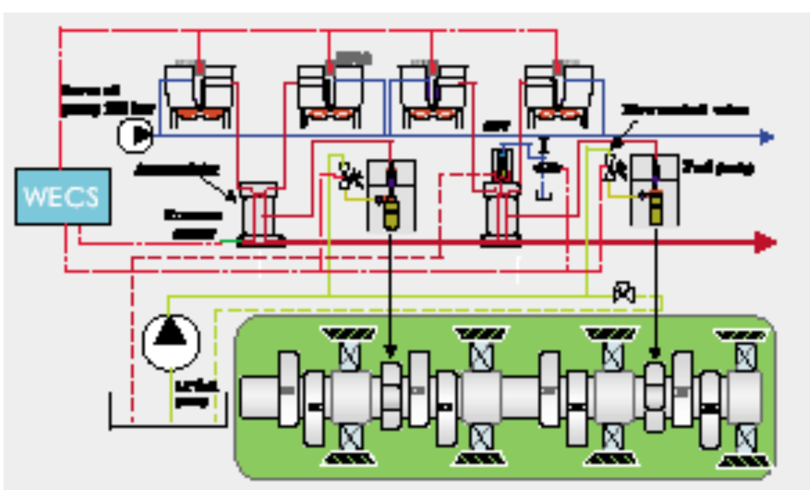


Fig. 2.5b: Schematic of the common-rail systems for fuel injection and exhaust valve actuation in the Sulzer RT-flex engine [2.4]

Fuel is delivered from this common rail through a separate injection control unit for each engine cylinder to the standard fuel injection valves (Fig.2.6) that are hydraulically operated in the usual way by the high-pressure fuel oil. The control units, using quick-acting Sulzer rail valves, regulate the timing of fuel injection, control the volume of fuel injected, and set the shape of the injection pattern. The three fuel injection valves in each cylinder cover are separately controlled so that they may be programmed to operate separately or in unison as necessary. The common-rail system is purpose-built for operation on just the same grades of heavy fuel oil as are already standard for Sulzer RTA-series engines. For this reason, the RT-flex system incorporates certain design features not seen in other common-rail engines using middle-distillate diesel oils. The key point is that, in the RT-flex system, the heated heavy fuel oil is kept away from the precision quick-acting rail valves. The key features of the Sulzer common-rail system are thus:

- Precise volumetric control of fuel injection, with integrated flow-out security
- Variable injection rate shaping and variable injection pressure
- Possibility for independent action and shutting off of individual fuel injection valves
- Ideally suited for heavy fuel oil
- Well-proven standard fuel injection valves
- Proven, high-efficiency common-rail pumps
- Lower levels of vibration and internal forces and moments
- Steady operation at very low running speeds with precise speed regulation
- Smokeless operation at all speeds (Figs. 2.8 and 2.9)

As indicated in Fig. 2.5b the RT-flex system encompasses more than the fuel injection process. It includes exhaust valve actuation and starting air control [2.4]. The exhaust valves are operated in much the same way as in existing Sulzer RTA engines by a hydraulic pushrod but with the actuating energy now coming from a servo oil rail at 200 bar pressure. The servo oil is supplied by high-pressure hydraulic pumps incorporated in the supply unit with the fuel supply pumps. The electronically-controlled actuating unit for each cylinder gives full flexibility for valve opening and closing patterns (Fig. 2.5b). This unit utilizes exactly the same Sulzer rail valves as are used for controlling fuel injection.

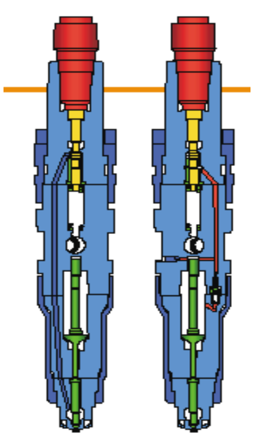


Fig. 2.6: Common rail injector with solenoid valve and hydraulically actuated three-way valve for lowest smoke at low load [2.4]

Fig. 2.7: The supply unit mounted on the side of the engine with the three servo-oil pumps on the near side and four fuel pumps on the upper and outboard sides [2.5]

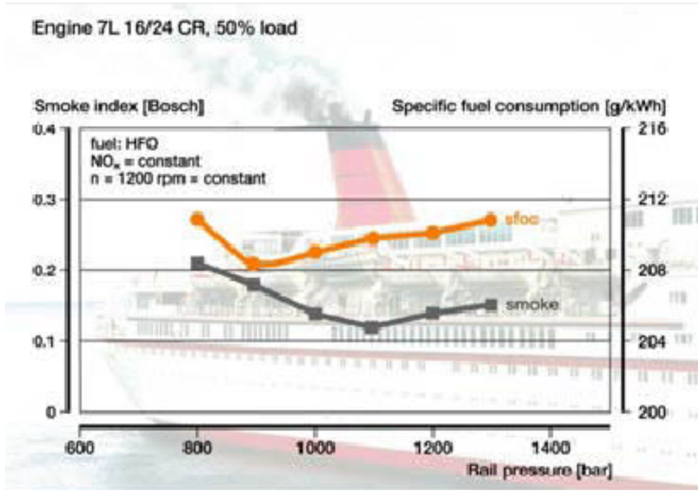


Fig. 2.8: Smoke emission and fuel consumption versus rail pressure on 7L 16/24 CR engine [2.4]

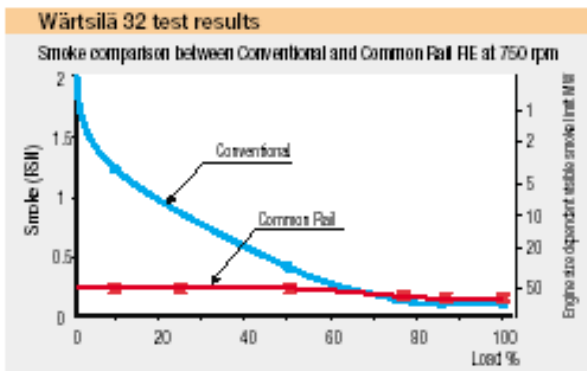


Fig. 2.9 Common rail achieve no smoke at any load tested in the Wärtsilä 32 [2.4].

For upgraded and newly designed engines, either optimized injection equipment or the common rail (CR) technology will be applied to cut emissions. Fig. 2.8 shows some results of tests performed on a test engine 7L 16/24 CR (common rail) in HFO operation for 50 % load at constant speed (1200 rpm): there is a positive influence of the rail pressure on smoke emission indeed, but the increased fuel consumption rate (sfoc) at very high rail pressures has to be taken into account too (Fig. 2.9) [2.4].

All functions in the RT-flex system are controlled and monitored through the integrated Wärtsilä WECS-9500 electronic control system (Fig. 2.5a). This is a modular system with separate microprocessor control units for each cylinder, and overall control and supervision by duplicated microprocessor control units. The latter provide the usual interface for the electronic governor and the shipboard remote control and alarm systems. Sulzer RT-flex engines are designed to be user friendly, without requiring ships' engineers to have any special additional skills. Indeed, the knowledge for operation and maintenance of RT-flex engines can be included in Wärtsilä's usual one-week courses for Sulzer RTA-series engines given to ships' engineers. Training time usually given to the camshaft system, fuel pumps, valve actuating pumps, and reversing servomotors is simply given instead to the RT-flex system.

At its heart, the Sulzer RT-flex engine is the same reliable, basic engine as the existing RTA engine series. The power ranges, speeds, layout fields and full-power fuel consumptions are the same for both engine versions. The Sulzer RT-flex engine offers a number of interesting benefits to ship owners and operators:

- Reduced running costs through lower part-load fuel consumption and eventually longer times between overhauls
- Reduced maintenance requirements, with simpler setting of the engine, for example adjustment of mechanical injection pumps is no longer necessary, and 'as-new' running settings are automatically maintained
- More balanced engine operation. The common-rail system with volumetric control gives excellent balance in engine power developed between cylinders and between cycles, with precise injection timing and equalized thermal loads
- More predictable maintenance costs owing to better balanced engine operation and better retaining of engine settings over many running hours
- Extendable times between overhauls through better engine running conditions, and better prediction of maintenance timing
- Better fuel economy in actual in-service load range
- Flexibility to optimize fuel consumption at selected service loads within compliance with the NOX emission limit in Annex VI of the MARPOL 73/78 convention
- Smokeless operation at all operating speeds.
- The RT-flex system is based on well-proven hardware
- Full electronic common-rail control with integrated monitoring functions
- Lower steady running speeds, in range of 10–12 rev/ min obtained smokelessly through sequential shut-off of injectors in all cylinders. Selective shut off of injectors gives more balanced engine operation than cutting out cylinders
- Built-in overload protection
- Built-in redundancy, as 100 percent power can be developed using three of the four fuel pumps, and with one servo oil pump out of action.

2.3.3 Fundamental of MAN B&W Common Rail Fuel Injection

The general design of the MAN B&W common rail fuel injection system is shown in Fig. 2.10 [2.3]. A common rail servo oil system using pressurized cool, clean lube oil as the working medium drives the fuel injection pump. Each cylinder unit is provided with a servo oil accumulator to ensure sufficiently fast delivery of servo oil in accordance with the requirements of the injection system and in order to avoid heavy pressure oscillations in the associated servo oil pipe system.

The movement of the plunger is controlled by a fast-acting proportional control valve (NC valve). The NC valve is, in turn, controlled by an electric linear motor that gets its control input from the cylinder control unit (Fig. 2.10). This design concept has been chosen in order to maximize reliability and functionality, the fuel injection system is the heart of the engine, and is crucial for fuel economy, emissions and general engine performance. An example of the flexibility of the system will be given below.

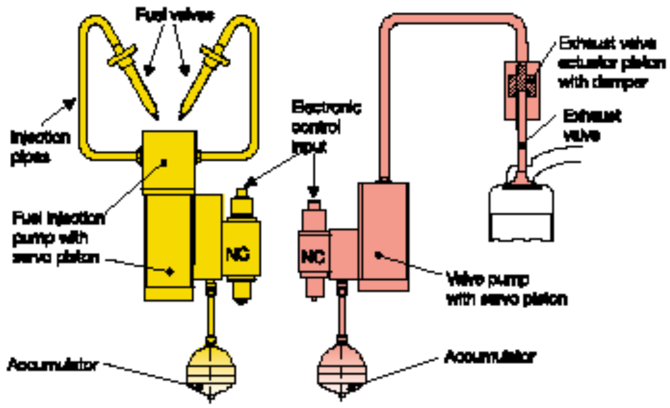


Fig. 2.10: General layout for fuel injection and exhaust valve actuation systems [2.3]

The key components have a proven reliability record: the NC valves have been in serial production for some ten years and are based on high-performance valves for such purposes as machine tools and sheet metal machines in car production-applications where high reliability is crucial. The fuel injection pump features well-proven fuel injection equipment technology, and the fuel valves are of our well-proven and simple standard design.

As can be seen in Fig. 2.11, the 2nd and 3rd generations of pump design are substantially simpler than the 1st generation design, the components are smaller, and they are very easy to manufacture. By mid-2000, the 2nd generation pump had been in operation on the 4T50MX research engine for more than 1400 hours, whereas the 3rd generation is starting service testing on the 6L60MC (see below).

The major new design feature for the 3rd generation pump is its ability to operate on heavy fuel oil. The pump plunger is equipped with a modified umbrella design to prevent heavy fuel oil from entering the lube oil system. The driving piston and the injection plunger are simple and are kept in contact by the fuel pressure acting on the plunger, and the hydraulic oil pressure acting on the driving piston. The beginning and end of the plunger stroke are both controlled solely by the fast acting hydraulic valve (NC valve), which is computer controlled.

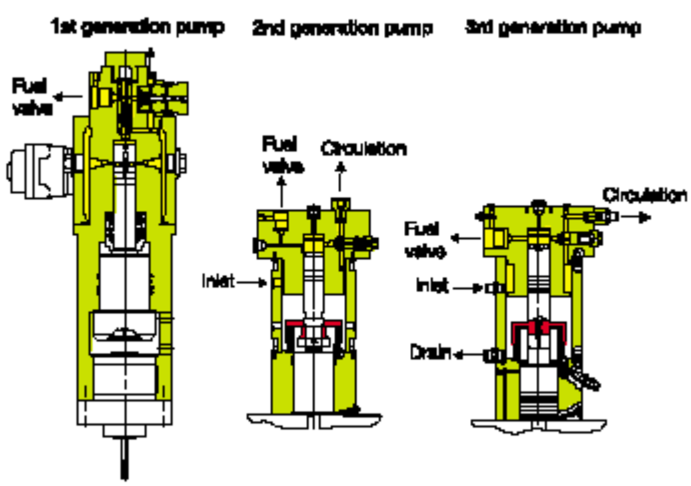


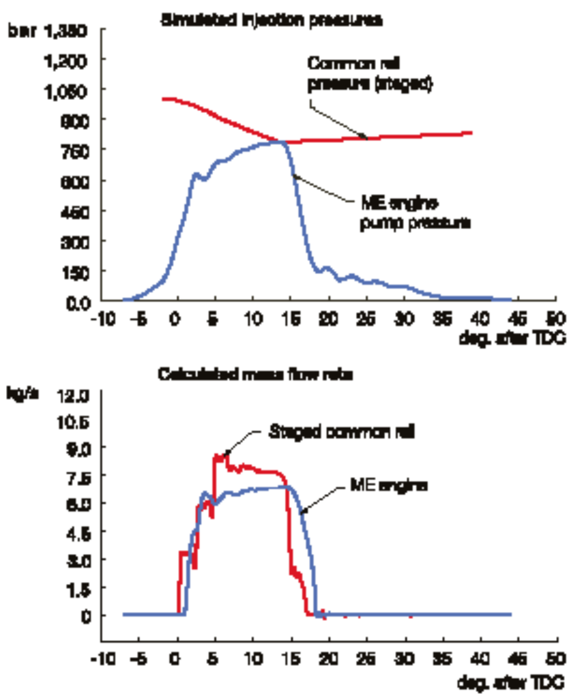
Fig. 2.11: Design development of fuel injection pumps [2.3]

Fuel injection system, rate-shaping capability

The optimum combustion (thus also the optimum thermal efficiency) requires an optimized fuel injection pattern that is generated by the fuel injection cam shape in a conventional engine. Large two-stroke engines are designed for a specified maximum firing pressure, and the fuel injection timing is controlled so as to reach that firing pressure with the given fuel injection system (cams, pumps, injection nozzles, etc.).

For modern engines, the optimum injection duration is around 18-20 degrees crank angle at full load, and the maximum firing pressure is reached in the second half of that period. In order to obtain the best thermal efficiency, fuel to be injected after reaching the maximum firing pressure must be injected (and burnt) as quickly as possible in order to obtain the highest expansion ratio for that part of the heat released.

From this it can be deduced that the optimum "rate shaping" of the fuel injection is one showing increasing injection rate towards the end of injection, thus supplying the remaining fuel as quickly as possible. This has been proven over many years of fuel injection system development for our two-stroke marine diesel engines, and the contemporary camshaft is designed accordingly. The fuel injection system for the Intelligent Engine is designed to do the same but in contrast to the camshaft-based injection system, the IE system can be optimized at a large number of load conditions.



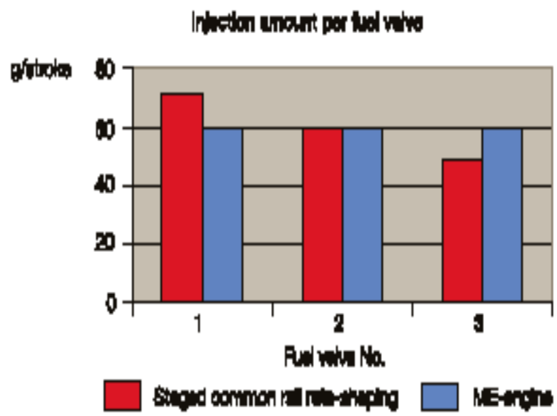


Fig. 2.12: Comparison between the fuel injection characteristics of the ME engine and a Staged Common Rail system in terms of injection pressure, mass flow rate and flow distribution [2.3]

Common Rail injection systems with on/off control valves are becoming standard in many modern diesel engines at present. Such systems are relatively simple and will provide larger flexibility than the contemporary camshaft based injection systems. We do apply such systems for controlling the high-pressure gas-injection in the dual-fuel version of our MC engines, where the (two-circuit) common rail system provides the necessary flexibility to allow for varying HFO/gas-ratios, please refer to [2.3].

However, by nature the common rail system provides another rate shaping than what is optimum for the engine combustion process. The pressure in the rail will be at the set-pressure at the start of injection and will decrease during injection because the flow out of the rail (to the fuel injectors) is much faster than the supply of fuel into the rail (from high-pressure pumps supplying the average fuel flow).

As an example, an 8-cylinder engine will have a total “injection duration” per engine revolution of 160 deg. CA (8 x 20 degrees CA) during which the injectors supply the same mass flow as the high-pressure supply pumps do during 360 deg. CA. Thus, the outflow during injection is some 360/160 = 2.25 times the inflow during the same period of time. Consequently, the rail pressure must drop during injection, which is the opposite of the optimum rate shape. To counteract this, it has been proposed to use “Staged Common Rail” whereby the fuel flow during the initial injection period is reduced by opening the fuel valves one by one.

The Rate Shaping with the IE system (using proportional control valves) and the “Staged Common Rail” are illustrated in Fig. 2.12. This shows the injection pressure, the mass flow and the total mass injected for each fuel valve by the two systems, calculated by means of our advanced dynamic fuel injection simulation computer code for a large bore engine (K98MC) with three fuel valves per cylinder. In the diagram, the IE system is designated ME (this being the engine designation, like 7S60ME-C). As can be seen, the Staged Common Rail system supplies a significantly different injection amount to each of the three fuel valves.

Fig. 2.13: Fuel spray distribution in the combustion chamber (schematically) corresponding to the injection patterns illustrated in Fig. 2.12 [2.3]

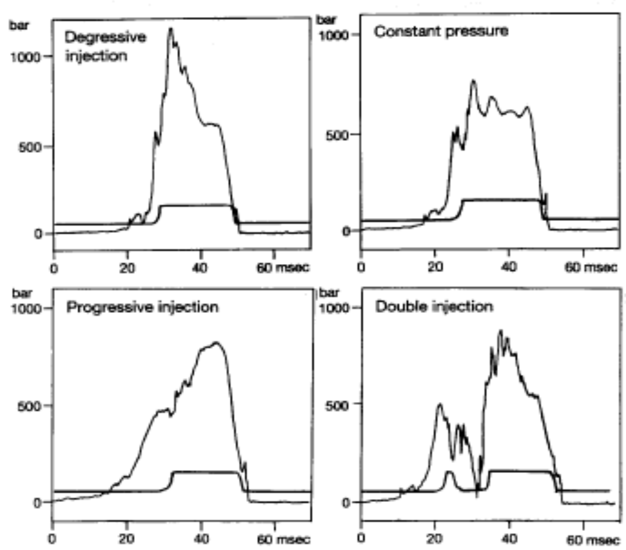


Fig. 2.14: Four examples of fuel injection pressures at the fuel valve, and the corresponding fuel valve spindle lifting curves [2.3]

Though the Staged Common Rail system will provide a fuel injection rate close to the optimum injection rate, combustion will not be optimal because the fuel is very unevenly distributed in the combustion chamber whereas the combustion air is evenly distributed. This is illustrated (somewhat overexaggerated to underline the point) in Fig. 2.13: the valve opening first will inject the largest amount of fuel and this will penetrate too much and reach the next fuel valve nozzle. Experience from older engine types indicates that this may cause a reliability problem with the fuel nozzles (hot corrosion of the nozzle tip).

The uneven fuel injection amount means that there will be insufficient air for the fuel from the first nozzle, the correct amount for the next and too much air for the third fuel valve. The average may be correct but the result cannot be optimal for thermal efficiency and emissions. Uneven heat load on the combustion chamber components can also be foreseen - though changing the task of injecting first among the three valves may ameliorate this.

Thus, the IE injection system is superior to any Common Rail system - be it staged or simple. Extensive testing has fully confirmed that the IE fuel injection system can perform any sensible injection pattern needed for operating the diesel engine. The system can perform as a single-injection system as well as a pre-injection system with a high degree of freedom to modulate the injection in terms of injection rate, timing, duration, pressure, single/double injection, etc.

In practical terms, a number of injection patterns will be stored in the computer and selected by the control system so as to operate the engine with optimal injection characteristics from dead slow to overload, as well as during astern running and crash stop. Change-over from one to another of the stored injection characteristics may be effected from one injection cycle to the next.

Some examples of the capability of the fuel injection system are shown in Fig. 2.14 [2.3]. For each of the four injection patterns, the pressure in the fuel valve and the needle-lifting curve are shown. Tests on the research engine with such patterns (see Fig. 2.15) have confirmed that the "progressive injection" type (which corresponds to the injection pattern with our optimised camshaft driven injection system) is superior in terms of fuel consumption. The "double injection" type gives slightly higher fuel consumption, but some 20% lower NOx emission - with a very attractive trade-off between NOx reduction and SFOC increase.

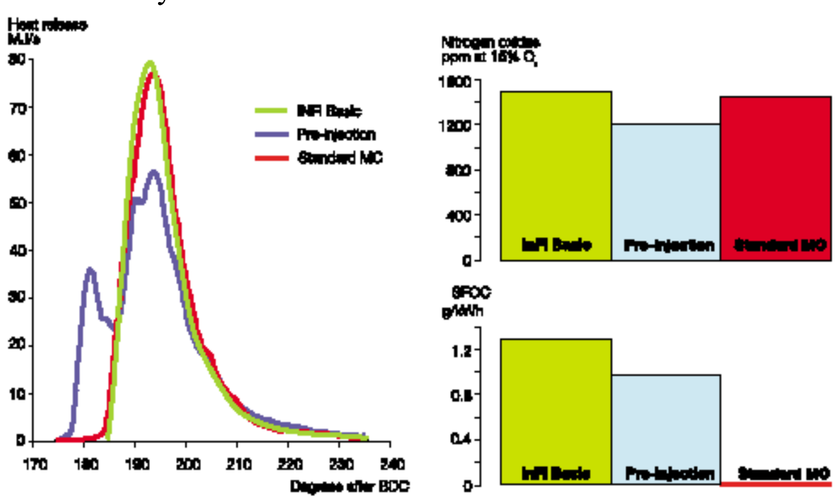


Fig. 2.15: Effect of injection pattern on combustion rate, NOx emission and specific fuel oil consumption (tested on 4T50MX research engine at 75% load) [2.3]

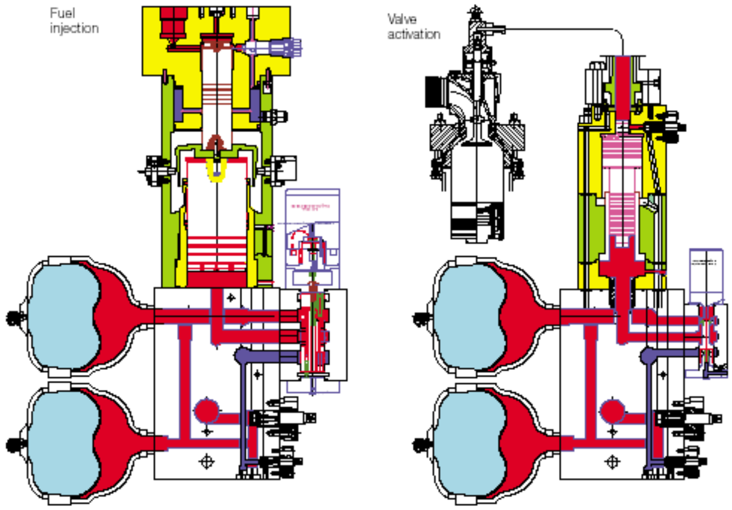


Fig. 2.16: Hydraulic cylinder unit with fuel injection pump and exhaust valve actuator [2.3]

2.3.4 Fundamental of Common Rail Fuel Injection in High-speed Diesel Engine

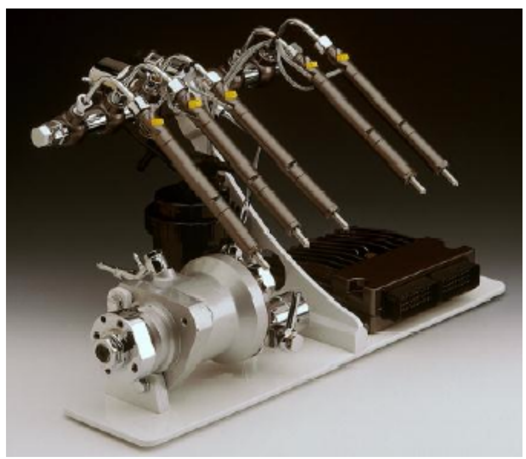


Fig. 2.17: Delphi Diesel Common Rail system (DCR)

The Delphi Common Rail product is a modular system shown in Figs. 2.16 and 2.17, and can therefore be easily adapted for different engines. The main components of the DCR System are:

- Common pressure accumulator (the "Rail")
- High-pressure regulator (option)
- Inlet metered high-pressure supply pump, with integrated lift pump
- Injectors
- Electronic Control Unit
- Filter unit

As illustrated in Fig. 2.17, the Delphi Diesel Common Rail system consists of a common pressure accumulator, called the "Rail," which is mounted along the engine block, and fed by a high-pressure pump. The pressure level of the Rail is electronically regulated by a combination of metering on the supply pump and fuel discharge by a high-pressure regulator (when fitted). The pressure accumulator operates independently of engine speed or load, so that high injection pressure can be produced at low speeds if required. A series of injectors are connected to the Rail, and each is opened and closed by a solenoid, driven by the Electronic Control Unit (ECU).

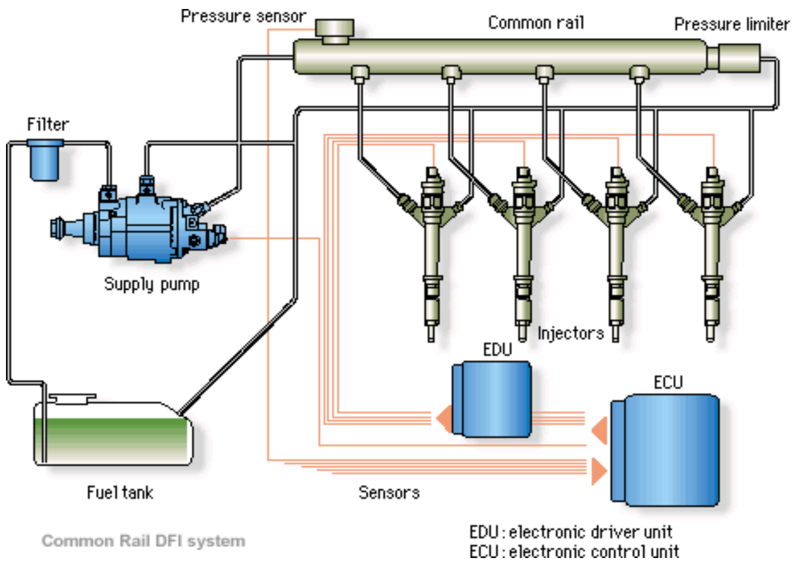


Fig.2.17: Schematic of the Delphi Diesel Common Rail system

DENSO is upgrading its Common Rail technology to comply with Europe's upcoming EURO-4 emission regulations. The improvements include increasing the number of injections per combustion stroke to five, from the present two, and raising the maximum pressure to 180 MPa, from the present 135 MPa. The possibility of adopting piezoelectric actuators for the injectors is also being investigated.

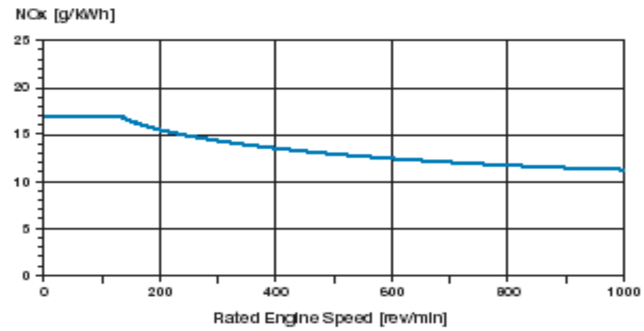
2.4 Emission Control in Marine Diesel Engine

2.4.1 Diesel engine emissions legislation

Growing concerns for the environment have promoted the legislature to install stringent regulations of pollutant emissions. The reduction of NO to N_2 to meet environment regulations has been a major challenge for marine diesel engines. A catalytic approach is the most direct method for reduction of NO to N_2.

In September 1997 the Conference of Parties to the MARPOL 73/78 Convention adopted the new Annex VI. In Regulation 13 of this annex, limiting NOx levels are defined for engines of more than 130 kW output. As shown in Fig. 2.4.1, the limit of NOx emissions is about 17g/kWh for slow speed marine diesels and less for medium speed diesels, depending on rated speed. It is likely that further restrictions will occur. These NOx emissions limits will only apply to engines installed in new ships built on or after 1 January 2000, or to engines in existing ships undergoing a major conversion. The basis of the regulation is that the average NOx emissions of new ships must be some 30 per cent less than the emissions from ships in 1990. In Regulation 14 of Annex VI, the sulfur content in fuels is limited to 4.5 per cent. To protect sensitive areas from sulfur oxides emissions, the concept of SOx special areas has also been introduced.

i) 17 g/kWh when n is less than 130
ii) $45 \cdot n^{-0.2}$ g/kWh when n is 130 rpm or more, but less than 2000 rpm
iii) 9.84 g/kWh when n is 2000 rpm or more
where n = rated engine speed



The three test cycles D2, E2 and E3 apply to three different engine categories namely:
D2: Generating sets with intermittent load, including shipboard generating sets
E2: Constant-speed heavy-duty engines for ship propulsion, including engines with a controllable-pitch propeller
E3: Variable-speed heavy-duty engines for ship propulsion equipped with a fixed-pitch propeller

Table: Weighting factors (f) for the test cycles D2, E2 and E3 in the Technical Code of the IMO regulation for NO_X emissions

% rated speed		100	100	100	100	100	100	91	80	63
% rated load		100	75	50	25	10	100	75	50	25
Weighting factor [f]	cycle D2	0.05	0.25	0.30	0.30	0.10	–	–	–	–
	cycle E2	0.20	0.50	0.15	0.15	–	–	–	–	–
	cycle E3	–	–	–	–	–	0.20	0.50	0.15	0.15

$$NO_x weighted = \frac{\sum_i f_i \times NO_{x_i} \left(\frac{g}{h}\right)}{\sum_i f_i \times P_i \ (kW)}$$

Fig. 2.18: IMO NOx emissions limit

The Baltic Sea represents such an area and others may follow in the future. Norway and Denmark apply an eco tax to shipping, based on emissions. The Clean Design notation of DNV calls for NOx emissions 40% below Marpol. In these areas, fuel with less than 1.5 percent sulfur has to be burned or the exhaust gas has to be treated to reduce the sulfur oxides emissions to a level under 6 g/kWh.

The US Senate has said it will not ratify Marpol. An environmental group called Bluewater Network has successfully sued the US EPA (Environment Protection Agency) to force it to establish emission standards for large ocean-going vessels in US waters. A proposed rule is due in April 2002 and final standards are scheduled for 2003[2.26]. EPA proposed emission standards for new marine diesel engines not covered/regulated by IMO:

- NOx : 9.2 g/kWh
- HC : 1.3 g/kWh
- CO : 11.4 g/kWh
- PM : 0.54 g/kWh
- Smoke : transient cycles (20/50%)

The 1997 Marpol Conference Resolution 3 calls for 5 yearly reviews of the NOx limits. Resolution 8 calls for consideration of CO_2 emissions. This is related to NOx control as there can be a fuel consumption penalty associated with NOx reduction strategies. Annex VI will enter into force 12 months after the date on which at least 15 states, together constituting at least 50% of the gross tonnage of the world's merchant fleet, have ratified it. It will apply to engines installed in new ships constructed from 1/1/2000 and engines in existing ships undergoing a major conversion after 1/1/2000. It is estimated that with a 1.5% yearly fleet replacement rate, NOx controls that reduce emissions by 30% to 50%, would reduce global ship NOx emissions by less than 1% annually [2.14].

When talking about Power Plant stack emissions nitrogen oxides (NOx), sulfur dioxides (SOx) and particles (as dry dust) are in focus. As plotted in Fig. 4.2.21, World Bank Guidelines 1998 for "Thermal Power: Guidelines for New Plants" take into account the air quality in the surrounding of the power plant when defining the applicable stack limits [2.8].

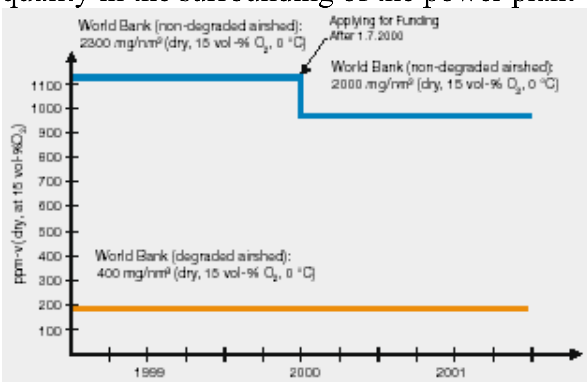


Fig. 2.19: NOx emission limits/World Bank 1998 Guidelines [2.8]

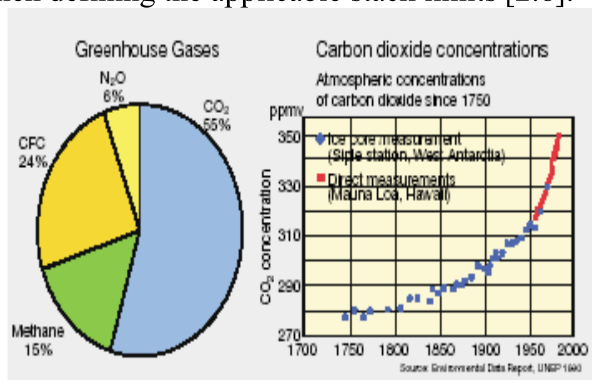


Fig. 2.20: Greenhouse effect and global warming [2.8]

In the future the carbon dioxide (CO_2) emission will be in focus due to its' expected impact on the global warming (Fig. 2.20). An efficient diesel power plant has relatively low CO_2 emissions. The Kyoto Protocol provisions allow for a Clean Development Mechanism (CDM) under which carbon offsets trading between certain countries will become possible in the future, the mechanism is still however under development. Diesel engine emission is only about 60% of coal fired boiler emission as shown in Fig. 2.21 [2.8].

2.4.2 Mechanism of NOx Formation in Diesel Engines

As shown above, the most important gas component that has to be reduced in diesel exhaust emissions is NOx. The strongest motivation is given by the forthcoming IMO regulations. Accordingly, the following sections will predominantly deal with NOx emissions and will give only basic indications of how other mission components might be reduced in the future. Analysis of the combustion process in the cylinder and the reactions which are involved to form NO has identified three main sources of NO [2.15] of which, as mentioned above, some is converted to NO_2 to give the NOx mixture:

- Thermal NO: During combustion, high temperatures are reached. Around 1500 K, and above, enough thermal energy is available to dissociate oxygen and nitrogen and also other molecules formed during the combustion process itself. The recombination of the elements leads to the formation of NO. The reaction processes are quite slow so that most nitrogen oxides are formed during the mixing of the stoichiometric combustion gases with excess air in the cylinder.
- Fuel source: The main molecular components of fuel are carbon and hydrogen. But marginal amounts of nitrogen are also contained. During combustion, 50 to 100 percent of the available nitrogen in the fuel is oxidised to NO owing to very fast reactions.

- Prompt NO: Even if combustion occurs near stoichiometric conditions, nitrogen originating from the air is available near the combustion front and can be combined to NO by the same reactions that lead to NO from the fuel.

In low- and medium-speed diesel engines, by far the most important part of NOx is generated in the thermal NO process. However, fuel-derived NO becomes important when using heavy fuel oil because such fuels contain more organic nitrogen than marine diesel oil and other distillate fuels. Heavy fuel oil typically contains about 0.3% nitrogen, but can contain up to 0.5% nitrogen. Marine diesel oil typically contains 0.1% nitrogen. Most of the nitrogen in the fuel is oxidized to NO in the combustion region. Typically, some mono oxides (3-10 % of the total NO) form from oxidation of the nitrogen in the fuel. In addition, a small amount of the nitrogen in air is converted to NO, called prompt NO, in the combustion region owing to some fast reactions.

Air contains 76.7% nitrogen (N_2) and 23.3% oxygen (O_2) by mass. Combustion of hydrocarbon fuels requires a minimum of about 20 kg of air for every kg of fuel burnt. In diesel engines, air is usually supplied 2 or 3 times in excess of the minimum requirement. Nitrogen is normally an inert gas. At the temperatures in the burning fuel spray, nitrogen is no longer inactive and some will combine with oxygen to form oxides of nitrogen. Initially mostly nitric oxide (NO) is formed. Later, during the expansion process and in the exhaust, some of this NO will convert to nitrogen dioxide (NO_2) and nitrous oxide (N_2O), typically 5% and 1%, respectively, of the original NO. The mix of oxides of nitrogen is called NOx.

A basic requirement for the calculation of NOx is knowledge about the main reactions involved. It was Zeldovich [2.16] who first described with sufficient accuracy the chemical reactions leading to thermal nitric oxide:

$$\begin{aligned} N_2 + O &\leftrightarrow NO + N \\ N + O_2 &\leftrightarrow NO + O \\ N + OH &\leftrightarrow NO + H \end{aligned} \qquad (2.4.1)$$

This is known as the extended Zeldovich mechanism. The rate of reaction is controlled by the concentrations of the species and the temperature. The temperature dependence is of the form $e^{-\frac{A}{T}}$, which is a strong function of temperature. NO formed in this way is called "thermal NO". Figure 2.4.2 shows the temperature dependence of NO formation rate on temperature for burnt gas containing 3% oxygen. As a rule of thumb, it can be said that NO formation rate increases by a factor of 10 for every 100 K temperature rise.

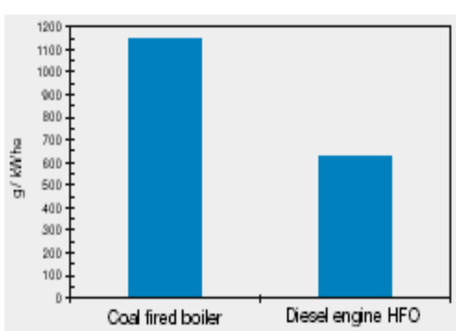


Fig. 2.21: Specific CO_2 emission between coal fired boilers and diesel engines [2.8]

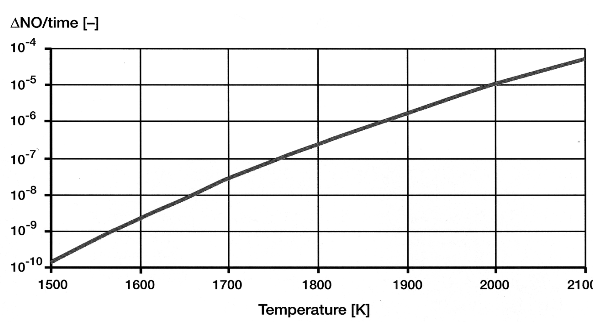


Fig. 2.22: Dependence of NO formation rate on temperature for burnt gas containing 3% oxygen [2.21]

The main dependence of the NOx production rate on the temperature and the nitrogen and oxygen concentrations can be shown in the following two reactions [2.29].

$$\begin{aligned} &N_2 + O + M \leftrightarrow N_2O + M \\ &N_2O + O \leftrightarrow NO + NO \end{aligned} \qquad (2.4.2)$$

By assuming that the concentrations of O_2 and the radicals O, H and OH remain at equilibrium and that the concentration of N remains at steady state, the rate of formation and decomposition of NO can be calculated. The concentration of the O and OH radicals will depend on the concentration of oxygen as well as temperature. Thus, NOx formation depends on the temperature of the burnt gas, the residence time of the burnt gas at high temperature and the amount of oxygen present. Thus, slow speed engines produce more NOx than medium speed engines because the combustion process spans a longer time period so there is more time available for NOx formation.

The burnt gas will be depleted of oxygen due to the combustion process, but there will still be enough oxygen present for NO reactions to proceed. The amount of unconsumed oxygen remaining in the burnt gas will depend on the fuel/air mixture strength in the burning zone. The amount of unconsumed oxygen remaining in the burnt gas will have a significant effect on the rate of NOx formation. Fuel and air can burn over a range of mixture strengths. In diesel engines combustion would occur over a range of mixture strengths, either side of chemically correct [2.33]. Highest NOx concentrations would occur in the burnt gas from combustion at slightly fuel lean mixtures, because there is more oxygen available in the burnt gas. As the burnt gas mixes with surrounding air, the oxygen concentration will increase but the temperature will decrease. NOx reaction rates are only significant at high temperatures. As soon as the temperature of the burnt gas drops significantly, the NOx reactions slow and NOx concentrations can be frozen at the levels reached in the high temperature burnt gas.

Figure 2.23 shows calculated combustion temperatures and NOx concentrations for a medium speed engine at one instant. Figure 2.24 shows calculated combustion temperatures for a medium speed engine at a more advanced stage of combustion. Combustion occurs in a region where the fuel has vaporized and mixed with air over a range of fuel to air mixture strength, around chemically correct. The rate of combustion is mostly controlled by the rate at which fuel vapor and air can mix, which is a function of spray characteristics, air motion and injection rate. Slow speed engines tend to use a high degree of swirl to promote fuel/air mixing. Combustion chamber shape is also important. Slow speed engines tend to have higher stroke to bore ratio than medium speed engines, so the combustion space is of a more favorable shape in slow speed engines. Some oxidation of NO to NO_2 can occur at lower temperatures during the expansion stroke.

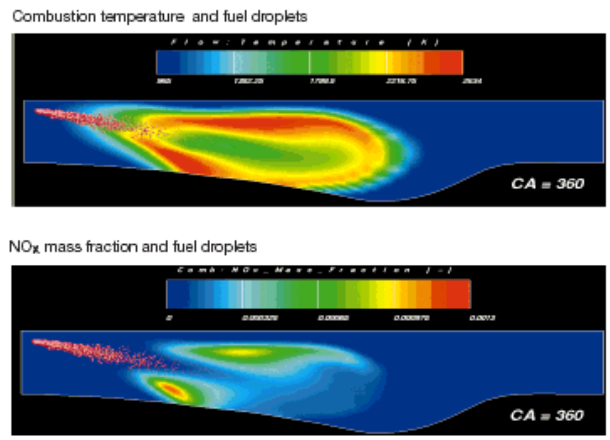


Fig. 2.23: CFD simulation of temperature and NOx in a burning fuel spray for a Wärtsilä medium speed engine [2.32]

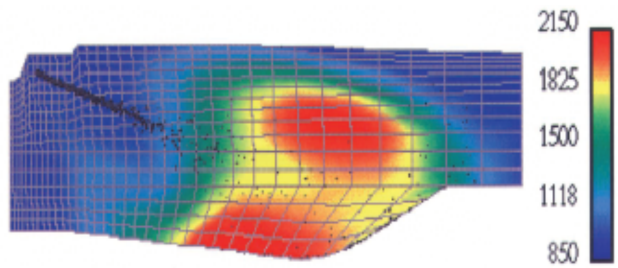


Fig. 2.24: CFD simulation of temperature in a burning fuel spray for a Wärtsilä medium speed engine at 20 degrees after start of injection [2.21]

2.4.3 NOx control measures

To reduce the exhaust emissions of today's diesel engines, both primary and secondary measures may be applied as indicated in Fig. 2.25. Primary measures focus on decreasing the production of emission components during combustion whereas secondary measures deal with the abatement of the emissions in the exhaust gas. The well-known drawbacks in employing catalytic converters in ships, mainly the necessity of a reducing agent together with the additional space required for the catalytic reactor, make them barely acceptable to marine diesel engine users. Consequently, primary reduction techniques are the first choice for to reduce the formation of pollutants on board ships.

As shown above, the main factors influencing NOx formation are the concentrations of oxygen and nitrogen and the local temperatures in the combustion process. Therefore all primary measures (Fig. 2.25), which aim to reduce NOx production, focus on lowering the concentrations and peak temperatures. On the other hand, methods for the reduction of particulates and exhaust gas components, such as hydrocarbons or carbon monoxide, deal with the optimum mixing of fuel and air in the combustion chamber to achieve even more complete combustion of the injected fuel.

Wärtsilä NSD and MAN B&W have done extensive tests over the past years due to NOx control in their Marpol Annex VI compliant slow speed engine specifications. To investigate the effects of different primary measures, extensive tests have been performed over the past years on all types of diesel engines by Wärtsilä NSD and MAN B&W. Tests have involved four-stroke engines from the S20 to the ZA40S and ZA50S, as well as the two-stroke engines available on the test beds in Winterthur, in the Diesel Technology Center in Winterthur, Wärtsilä NSD and in production at the licensees, such as the most powerful engine available today, the Sulzer RTA96C. MAN B&W have also done extensive tests on their MC engines.

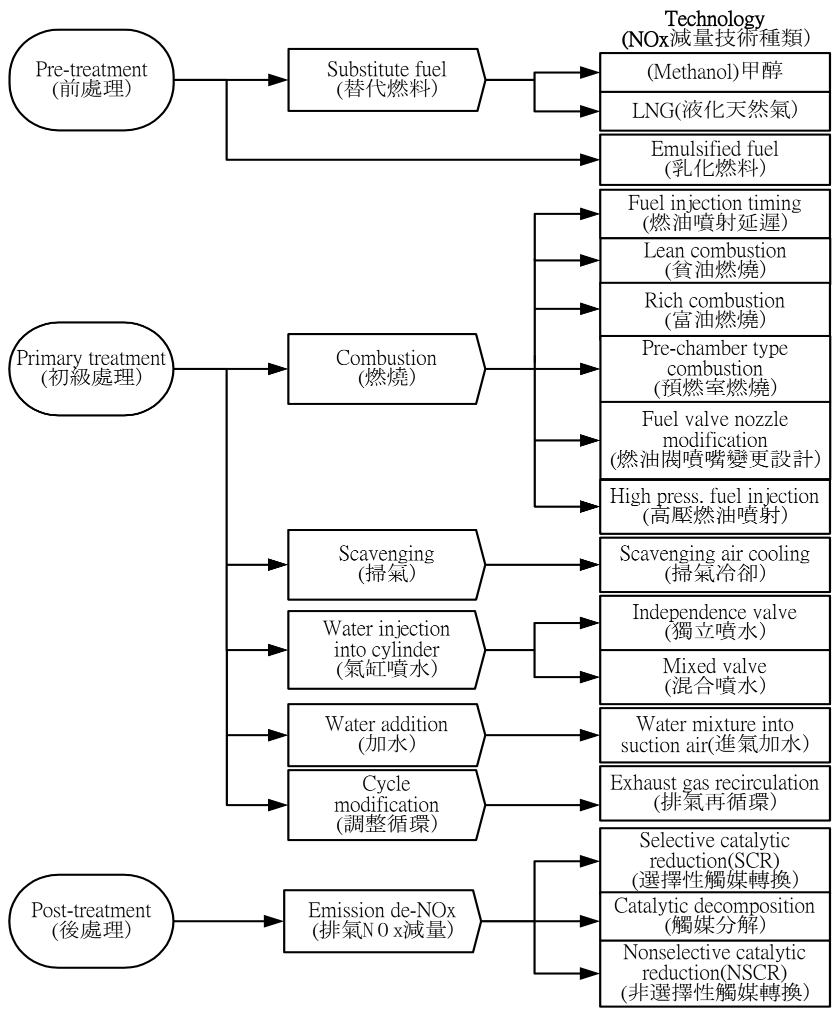


Fig. 2.25: Methods of reducing Nox emissions from marine diesel engines [p]

2.4.3.1 Combustion control techniques

Most engines can meet the Marpol Annex VI limits by tuning the combustion process. Options include modifying the spray pattern, injection timing, intensity of injection and injection rate profile (injection rate shaping), compression ratio, scavenge air pressure and scavenge air cooling. Delayed injection timing is very effective in reducing NOx but increases fuel consumption and smoke. It is usually combined with increased compression pressure and decreased injection duration to minimise or avoid increase in fuel consumption.

2.4.3.1.1 Constant Pressure Combustion

Okada et al [2.28] showed that where maximum cylinder pressure is limited, constant pressure combustion gives the greatest thermal efficiency. The maximum cylinder pressure is limited by considerations of cost and reliability. Combustion approaching constant pressure is achieved by high compression pressure followed by delayed fuel injection and short combustion duration. For constant pressure combustion, the maximum pressure reached during combustion is similar to the compression pressure. This is illustrated in Figure 2.26. The trend in engine design for low NOx without significant loss in fuel consumption is towards the constant pressure combustion situation.

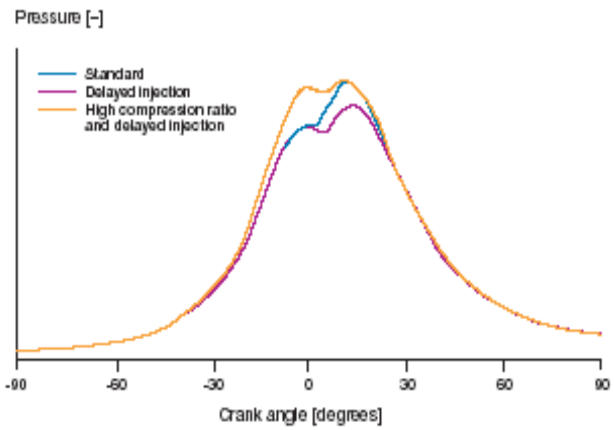


Fig. 2.26: Cylinder pressures for a Sulzer RTA engine with various injection timing and compression ratio [2.21].

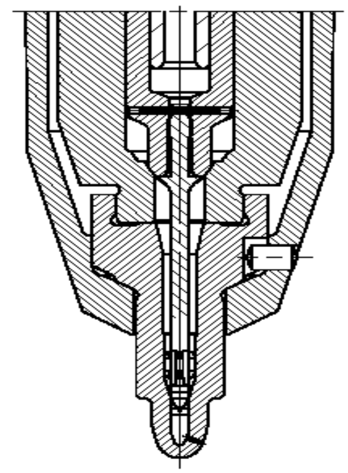


Fig. 2.27: Design of MAN B&W mini-sac type fuel valve [2.2]

2.4.3.1.2 Fuel Injector Valves and Nozzles

Recent developments in low NOx engines involve fine-tuning of the fuel injection process. For instance, advanced three-dimensional computer models of the dynamics of the burning fuel spray in the engine combustion chamber have been used. This is called Computational Fluid Dynamics or Computational Reactive Fluid Dynamics, shown in Figures. 2.23 and 2.24, illustrate the outputs of such techniques. These techniques have been used successfully by engine manufactures to develop low NOx fuel nozzles. Wärtsilä NSD [2.11] describe a trade-off between NOx reduction by optimized spray distribution in the combustion chamber and increased exhaust valve temperature in RTA engines. The relationship is unclear, although it is probably due to impingement of the spray on, or closer proximity of the spray to, combustion chamber surfaces. This would reduce combustion and burnt gas temperature but increase surface temperature. Wärtsilä NSD report that the location of flame zones in relation to metal surfaces was of considerable importance in controlling NOx in their medium speed engines [2.29]. It can be seen in Figures 11 and 12 that the hottest combustion zones are close to the piston and cylinder head. Wärtsilä found that the combustion space that was optimal for NOx was also ideal for low smoke [2.29]. Wärtsilä NSD [2.24] used computational methods to “reduce the space around the burning areas” in the Wärtsilä 64 medium speed 4 stroke diesel. They state that there was too much space around the burning areas available for NOx formation, that there is a layer where NOx build-up is greatest. They also increased the excess air ratio to relocate and shrink the layer where NOx production is greatest. It is not clear what is the significance of the “space available for NOx formation". Presumably, the NOx reduction will be due to decreased burnt gas temperature and decreased residence time at elevated temperature. Rapid mixing of the burnt gas with the bulk gas will be advantageous as well as allowing the burnt gas to be cooled by the combustion chamber surfaces.

An Emission Strategy Group [2.12] found that, for a medium speed engine (RK215), reducing spray core angle from 140 deg to 130 deg reduced NOx by 32% and increased fuel consumption by 6%. The smaller spray angle reduced the air entrainment into the spray resulting in less prepared mixture for the premixed combustion phase. In medium speed engines, the severity of the premixed phase has a significant influence on total NOx production. Combustion rate and thus temperature is high during the premixed phase. A larger mass of fuel burnt by premixed combustion means higher combustion rate and temperature in both the premixed and diffusion controlled phases. The Group found that increasing nozzle tip protrusion from 2mm to 6mm gave 6% less NOx and slightly increased fuel consumption, because the spray was closer to the piston bowl wall giving lower cylinder pressure and temperature. With the new optimized piston shape, there is less distance between the spray and the piston near the centerline of the piston, so there is less air entrainment in the early stages of injection, which reduces the premixed phase. The new design enhances turbulence and mixing for the later stage of the combustion, which improves fuel efficiency and decreases emissions overall.

MAN B&W has introduced the slide-type fuel valve as standard on slow speed engines [26,11]. The slide-type fuel valve, shown in Fig. 2.27, has zero sac volume so the entry of fuel into the combustion chamber after injection ceases is minimized. This leads directly to reduced CO and HC emissions as any fuel leaking into the cylinder after the main combustion process is finished is likely to burn incompletely. The slide-type valve also leads to reduced NOx emissions, although the mechanism is unclear. Probably the fuel nozzle was optimized for NOx simultaneously with the development of the slide valve. Tests on a 12K90MC engine at 90% load show a 23% reduction in NOx emissions for a slide-type valve compared with a standard valve and nozzle, with a 1% fuel consumption increase [2.18]. Tests on a 5S70MC engine show a slight decrease in fuel consumption with the use of a slide valve [2.13]. The K98MC with slide-type fuel valves has NOx over the IMO E3 cycle at 14.3 g/kWh [2.17].

Sulzer report the use of a zero sac volume nozzle in their 4RTX54 research engine resulting in reduced HC and particulate emissions [2.21]. Their tests showed that the main source of smoke and soot deposits was the fuel remaining in the injector sac hole. Their new "mini-sac hole" fuel nozzles for RTA engines are now in field tests. The new nozzles have negligible effect on NOx. It is possible that the reduced smoke by using low sac volume injectors will allow further reduction of NOx. Some NOx control measures, such as EGR and injection timing retard increase smoke. The new nozzles could be used to partially offset the rise in smoke.

Using a low NOx fuel injection valve in its UEC52LSE slow speed engine, Mitsubishi reduces NOx from 18.5 g/kWh to 15 g/kWh, but at 2 percent fuel consumption penalty [2.31]. Kawasaki report increased local heat loads with the slide-type valve [2.25]. This suggests that hot gas cooling by metal surfaces is a factor in the NOx reduction with the slide-type valve. MTU optimized injection on the new Series 8000 to improve mixing and reduce soot generation by optimising number of nozzle holes, hole shape and spray angle. They also increased the rate of opening and closing of the injectors. [2.20] The combustion chamber shape was also optimized. Developments on the Ruston RK215 medium speed engine included smaller nozzle hole diameter, increased number of nozzle holes and initial injection direction closer to the bowl wall [2.30]. Yanmar Diesel [2.28] used an increased number of injection nozzle holes and smaller nozzle holes to give good fuel distribution through a deep bowl combustion chamber.

2.4.3.1.3 Injection Timing Retard

Nitrogen oxide formation depends on temperature as well as residence time. An important factor in NOx production during combustion is the after compression of burnt gases. When fuel and air have burned, high peak temperatures are achieved. If this burnt gases are further compressed, even higher temperatures and pressures will be reached leading to increase NOx emissions. The problem may be overcome by later injection of the fuel. This method may be the best known way to reduce the NOx emissions.

Basically the delayed injection leads to lower peak pressures and therefore to less compression after combustion. Delayed injection leads to lower pressure and temperature throughout most of the combustion as shown in Fig. 2.4.6. Retarding injection timing also decreases the amount of fuel burnt before peak pressure, thus reducing the residence time and degree of after-compression of the first burnt gas. Peak pressure occurs when the rate of pressure fall due to the piston motion outstrips the rate of pressure rise due to combustion. Delayed injection increases fuel consumption due to later burning, as less of the combustion energy release is subject to the full expansion process and gas temperatures remain high later into the expansion stroke resulting in more heat losses to the walls. Smoke also increases due to reduced temperatures in the later part of the combustion process and thus less oxidation of the soot produced earlier in the combustion. An example of the effect of delayed injection on emissions and fuel consumption is illustrated in Fig. 2.28 [2.18]. Injection timing retard of 7 degrees reduced NOx by about 30% and increased BSFC by about 7% from MAN B&W 4T50MX research engine.

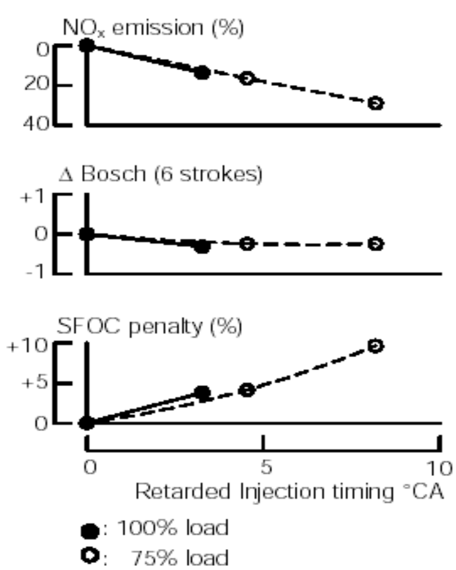


Fig. 2.28 Effect of injection timing on SFOC, Bosch Smoke Number and NOx [2.18]

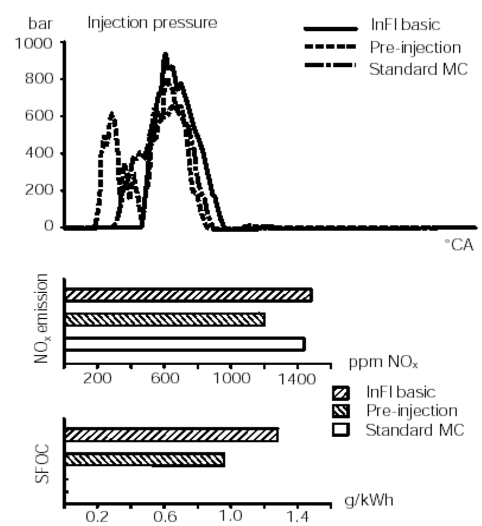


Fig.2.29 Fuel injection patterns,including pre-injection and the effects on SFOC and NOx emissions [2.18]

It has been found that changing injection characteristics of the fuel injection, a NOx reduction of about 20% is possible at a fuel penalty of 3.5%, as indicated in Fig. 2.4.9 [2.23]. Obviously, further development work has improved the fuel consumption. Pre-injection can be used to shorten the delay period in medium speed engines and thus decrease temperature and pressure during the early stages of combustion, resulting in reduced NOx [2.12]. Pre-injection can reduce particulates that are increased by other NOx control measures, thus allowing greater flexibility in NOx control.

With pre-injection, a small part of the fuel charge is injected before the main charge in Sulzer RT-Flex common rail engine [2.19]. With triple injection, the fuel charge is injected in separate, short sprays in succession. With sequential injection, each of the three nozzles in a cylinder is actuated with different timing. The results are shown in Fig. 2.30. For HFO, pulsed injection gave about 20% NOx reduction with about 7% increase in fuel consumption. Sequential and pre injection gave less NOx reduction and less fuel consumption increase. Thus, the NOx/fuel consumption trade-off is still a problem. However, the ability to maintain high injection pressure in combination with early exhaust valve closing at low loads with the common rail engine results in low smoke at low loads without having to compromise NOx at other loads.

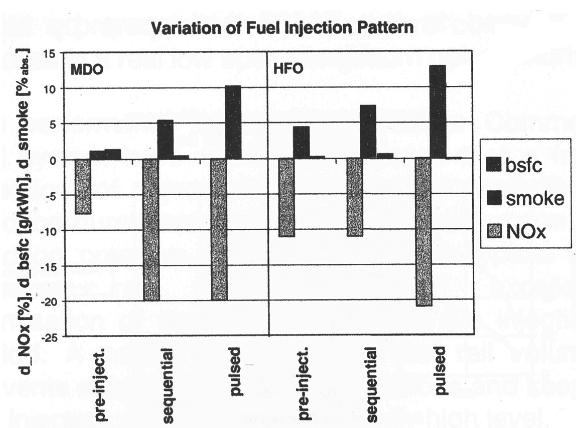


Fig. 2.30: Effects of different injection patterns on Sulzer RT-flex engine [2.19]

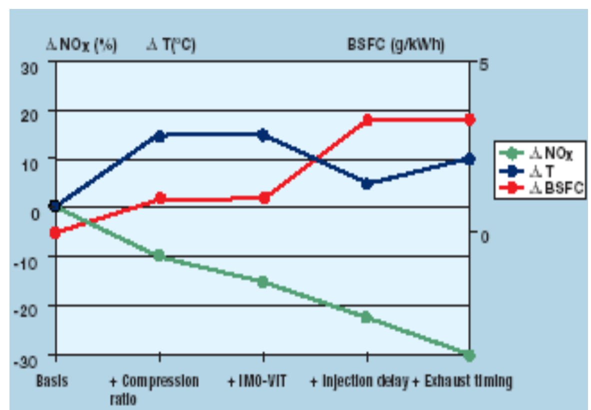


Fig. 2.31a: Effects of combined measures applied to RTA engines [2.22]

The test results of a MAN B&W 6L48/60 engine in February 2000: a NOx cycle value of 7.7 g/kWh and a fuel consumption rate still within tolerance (5%) was measured as shown in Fig. 2.31b. This is 40% below the NOx limit set by the IMO. This result was achieved with only 15% water in the water-fuel emulsion and a slightly retarded injection below 80% engine.

Engine shop test 6L 48/60 with emulsion injection
containing 15 % of water (const.)

Load, %	25	50	75	85	100
Output, kW	1 575	3 150	4 725	5 355	6 300
SFOC, g/kWh	211.7	202.8	191.8	183.9	184.7
NO_x, g/kWh	6.44	7.08	6.97	10.89	9.58

NO_x cycle value (E2): 7.7 g/kWh

Fig. 2.31b: The test results of a MAN B&W 6L48/60 engine [2.7]

2.4.3.2 Scavenge Air Temperature, Miller Supercharging

Both scavenge air-cooling and Miller supercharging aim to reduce the maximum temperatures in the cylinder by lowering the temperature before compression. The straightforward method is the reductions of scavenge air temperature by improving the air cooler efficiency. Tests showed that for every 3℃ reduction, NOx may decrease by about 1% [2.21]. Reduced charge air temperature results in lower overall temperatures and less heat losses, resulting in improved thermal efficiency. Standard air cooling techniques can only achieve scavenge air temperatures of about 50℃, so the potential for this measure is limited. However, on 4-stroke engines, the Miller supercharging concept can be applied to achieve lower scavenge air temperature. Using a higher than normal pressure turbocharger, the inlet valve is closed before the piston reaches bottom dead center on the intake stroke. The charge air then expands inside the engine cylinder as the piston moves towards bottom dead center resulting in a reduced temperature. Even though the pressure supplied by the turbocharger is higher than normal, the mass of charge and thus excess air ratio is similar to normal because the inlet valve closes earlier. In fact, the same charge air mass can be achieved with a lower charge air pressure at the start of compression, because of the increased density. This allows more fuel to be burnt without increasing peak cylinder pressure.

Miller supercharging can reduce NOx by 20% without increasing fuel consumption. Wärtsilä NSD report the implementation of early inlet valve closing on Sulzer ZA40S medium speed engine. Caterpillar introduced the Miller supercharging concept by earlier closing of inlet valves and slightly increased charge pressure [2.30]. However, excessive cooling of the inlet air can lead to increased smoke due to poor oxidation of soot formed during combustion. Caterpillar found that increased smoke at low load limited the applicability of Miller supercharging [2.30].

2.4.3.3 Water Injection, Fuel/water Emulsion, Humidification

It has long been well known that introduction of water into the combustion chamber reduces NOx formation. Engine manufacturers are able to bring NOx levels below IMO levels by water injection, emulsion or humidification. Introduction of water into the combustion chamber reduces combustion temperature due to: (1) the absorption of energy for evaporation, (2) the increase in the specific heat capacity of the cylinder gases (H_2O has higher specific heat capacity than air), and (3) reduced overall oxygen concentration. The reduction in oxygen concentration means an increase in the number of moles of gases that must be raised to combustion temperatures to react a given amount of oxygen with fuel. It also reduces the availability of oxygen for the NOx forming reactions.

Water can be introduced in the charge air (humidification), through direct injection into the cylinder or through water-fuel emulsion. Water-fuel emulsions can reduce smoke, while humidification can increase smoke. Water-fuel emulsions and direct injection of water place the water more directly in the combustion region, where it has maximum effect on NOx production. The influence of water emulsification and direct water injection varies with engine type, but generally, one percent of water reduces NOx by one percent.

2.4.3.3.1 Fuel-water Emulsions

Fuel-water emulsion is a well-known technique for reducing NOx emissions. Running an engine on fuel-water emulsion makes it theoretically possible to reduce NOx emissions by up to 50 percent. It has been found that using emulsified fuel can improve the combustion process and reduce fuel consumption. Under certain conditions, "micro-explosions" take place within fuel droplets when the steam pressure of water overcomes surface tension forces, thus leading to improved atomization of the fuel. Wärtsilä NSD recognizes that the "micro-vaporization" of the fuel drops has a positive effect on the combustion process by producing a more homogeneous air-fuel mixture.

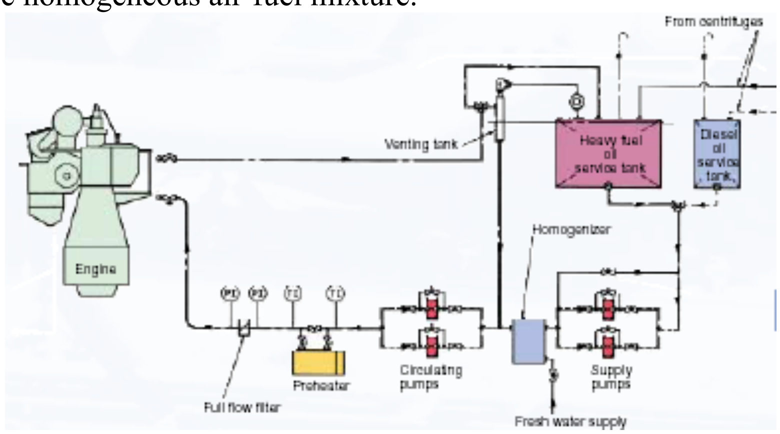


Fig. 2.32: A standard MAN B&W fuel-water emulsion system

MAN B&W prefer fuel-water emulsions as shown in Fig. 2.32. A standard MAN B&W engine design allows about 20% water in emulsion with the fuel at full load, without modifying injectors or fuel injection pumps [2.18]. Figure 2.4.13 shows about 1.5% increase in fuel consumption for 26% water in fuel and about 6% increase in fuel consumption for 48% water in fuel. MAN B&W have described the use of fuel-water emulsions in an "invisible smoke" medium speed diesel (6L48/60). It used a constant proportion of 15% water to fuel over the whole operating range. The fuel water emulsion reduced the opacity of the smoke, particularly at low loads, as well as reducing NOx. There was no fuel consumption increase at high loads. When optimized for low NOx using engine tuning, it yields about 12g/kWh NOx comparing with typical NOx emission of about 16g/kWh. With fuel water emulsions at 15% water to fuel, the NOx output comes down to 7.8 g/kWh shown in Fig. 2.31b [2.7]. MAN B&W claim many years of successful experience with fuel water emulsions in both slow speed and medium speed engines. The emulsion of water with HFO or MDO is stable over a period of several days. A closed pressurized system avoids cavitations and boiling-off. Fuel-water emulsion of 15 to 20% water to fuel results in NOx of about 6.7 g/kWh. It has no effect on specific fuel consumption in the upper load range and reduces smoke [2.18].

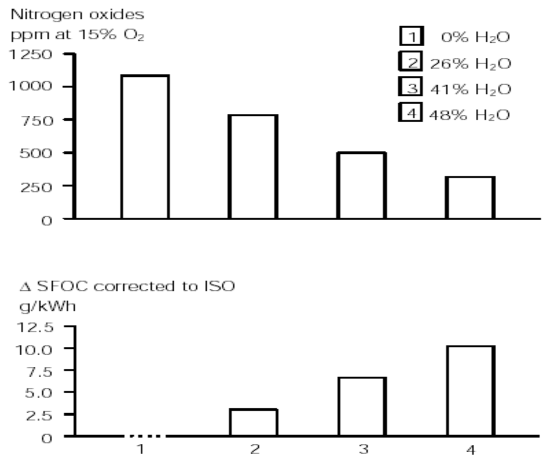


Fig. 2.33: Effect of water emulsification on NOx emission and SFOC from MAN B&W 4T50MX research engine

Wärtsilä NSD, however, highlights drawbacks of fuel-water emulsions. They claim [2.32]: (1) the emulsion is only stable for a few seconds, (2) a significant increase in fuel injection equipment capacity is needed, resulting in increased parasitic load consumption, (3) before every engine stop the emulsion must be flushed from the system to prevent the risk of severe corrosion, and (4) shutting down the water emulsion results in deteriorated combustion performance because the injector nozzles are designed for the volume of fuel and water together. For large proportions of water to fuel in emulsion it is necessary to modify the injection nozzle design to adapt to the increased quantity of liquid injected. This means that fuel consumption and component temperatures may be penalized when operating without water injection [2.21].

2.4.3.3.2 Direct Water Injection

Wärtsilä NSD has generally advocated direct water injection for large NOx reductions. However, they have tested and advocated fuel-water emulsions as an “extended” measure in RTA engines for NOx reductions up to 20% above those achieved with standard low NOx engine tuning measures [16,2]. Wärtsilä state that the additional cost of water injection equipment for direct injection is only justified where larger NOx reductions are required. The mass of water required for 20% NOx reduction is probably about 20% of the mass of fuel injected. Wärtsilä NSD has developed independent water and fuel injection systems for both low speed and medium speed engines, which allow water injection to be turned off without influencing the fuel injection [2.21].

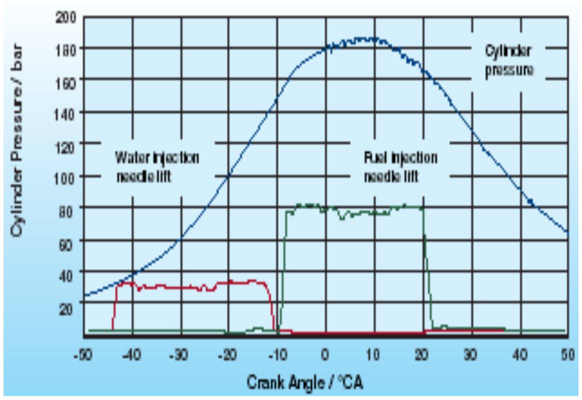


Fig. 2.34: Direct water injection, typical water/fuel timing and duration

Fig. 2.35: The combined nozzle for direct water injection

In the Wärtsilä medium speed engine, water injection takes place before fuel injection begins and ends before fuel injection begins shown in Fig. 2.34. Meanwhile, Fig. 2.35 show the combined water-fuel nozzle developed by Wärtsilä NSD. This early injection of the water can cools the combustion space but prevents the water spray interfering with the ignition and combustion process. Water injection timing and duration is electronically controlled, depending on the engine output. Timing and duration can be optimized for different applications. The typical water to fuel ratio is 40% to 70%, resulting in NOx reductions around 50% to 60% without adversely affecting power output or engine components. NOx when running on MDO is typically 4 to 6 g/kWh and when running on HFO is typically 5 to 7 g/kWh. Wärtsilä claim the system requires minimal space and is suitable for retrofitting.

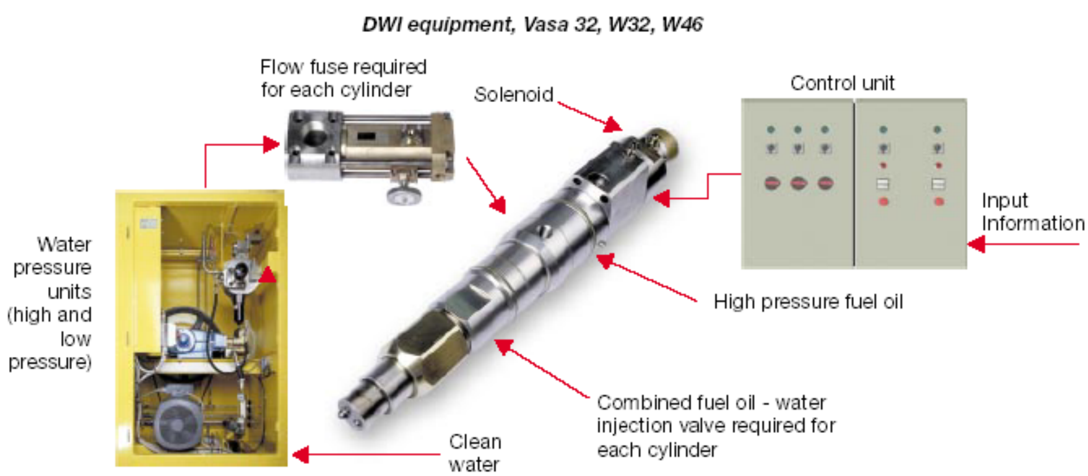


Fig 2.36: The Direct Water Injection package offered by Wärtsilä

Two cruise ferries fitted with Wärtsilä 46 engines with direct water injection have logged many hours of operation as shown in Fig. 25. NOx levels are reduced by about 50% to 6 g/kWh using water to fuel ratio 0.7, with 2% fuel consumption increase [2.32]. Wärtsilä NSD now prefers separate fuel and water injection nozzles in the RTA engines for simplicity and independent of the systems [2.11]. However, direct water injection is not being offered as an option for Sulzer slow speed engines at present. MAN B&W have decided not to follow the direct water injection path [2.18]. The disadvantages cited include increase in fuel consumption, increase in smoke and higher water consumption than for emulsions. MAN B&W also claim that water injection nozzles have a short lifetime [2.18].

2.4.3.3.3 Humidification

The water content of the charge air can be increased by injection of water into the inlet air, or by a method such as that the charge air cooler is replaced by a humidifier to increases the air humidity. A 70% NOx reduction at full load (80% at half load) has been achieved with the Munters HAM (Humid Air Motor) system by introducing typically three times as much water vapor as fuel, by saturating the inlet air with water vapor at higher than normal charge air temperatures. The Munters HAM system, illustrated in Fig. 2.37a, takes its water vapor directly from seawater in a humidification tower, which has a large surface area for contact between the charge air and the seawater [2.36]. The seawater is normally heated by engine cooling water. The inlet air reaches a relative humidity of about 99%. During the compression stroke, the relative humidity decreases due to the rising temperature, so it would be feasible to inject further water directly during compression or combustion. To prevent condensation in the pipes between the humidifier and the engine they are insulated. No deterioration of engine components was detected after 6000 hours of operation on a SEMT Pielstick 12 PC 2.6B 4-stroke medium speed engine (750kW/cyl, 600RPM) on the car ferry Mariella. In fact, carbon deposits were decreased and oil consumption decreased due to reduced component temperatures. Fuel consumption and smoke did not increase significantly. Seawater is the main consumable. Exhaust gas temperatures are reduced by about 20^{o}C, but the mass of exhaust gas is greater, so the energy content of the exhaust gas is similar to the unmodified engine.

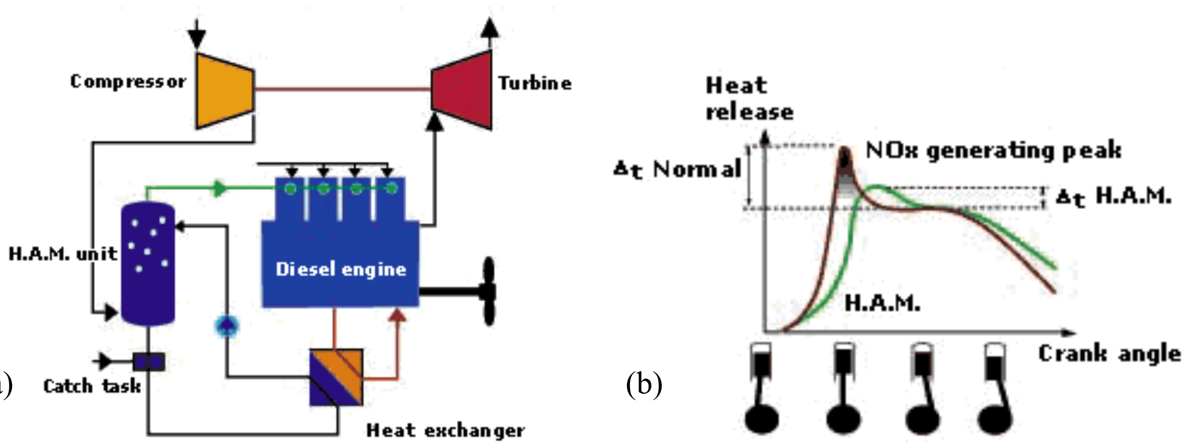
(a)


Fig. 2.37: (a) Schematic of Munters HAM system and (b) Effect of HAM system on mean cylinder temperature [2.36]

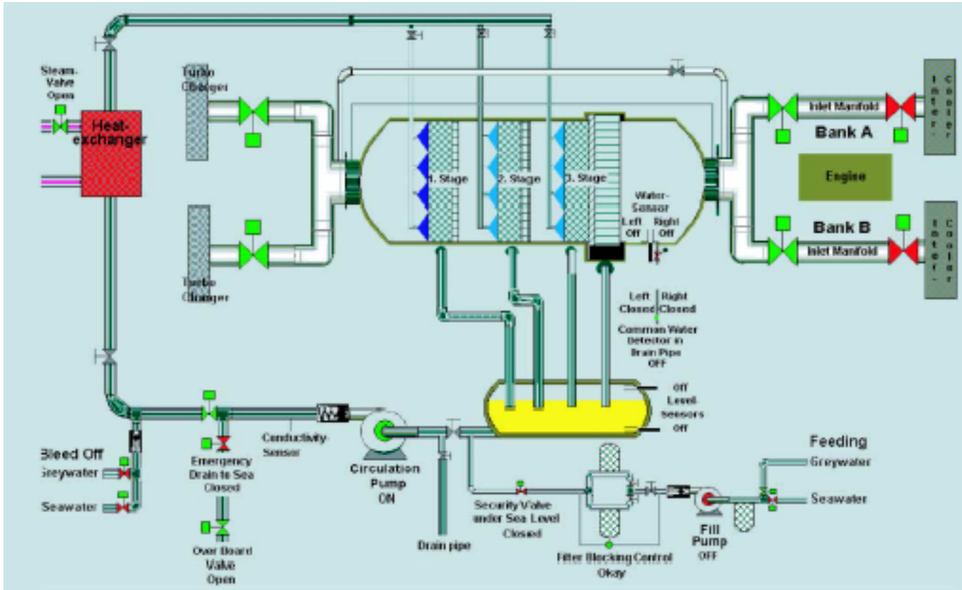


Fig. 2.38: Flow scheme of a Humid Air Motor system (HAM) [2.35]

Table 2.4.1: Total costs of a HAM and SCR system [2.35]

Specific costs in EUR / kW year	HAM	SCR
[illegible]	[illegible]	[illegible]
Specific operating costs at 80% MCR	[illegible]	10.40
[illegible]	[illegible]	[illegible]
[illegible]		
Total costs per year for [illegible] kW	[illegible]	[illegible]

Since fast ferries and similar ships have no use for the exhaust gas heat, the energy is available for evaporating the water. That means the HAM system is absolutely ideal for such kind of ships. Consequently HAM combines many advantages for fast ferries while almost no drawbacks can be seen:

1. HAM provides a very high NOx reduction potential to less than 4.5 g/kWh NOx with hardly any additional energy consumption. With preheated water by adding heat into the water circulation even lower levels of NOx emission can be achieved (less than 3.5 g/kWh).
2. No measurable increase in fuel oil consumption.
3. No detrimental effect on reliability.
4. No operating costs by use of seawater.
5. Easy operation.
   The system responds very satisfactorily to load variations and is self-regulating. In case of failure of the HAM system during operation, enough time is left to switch automatically to low load. In the beginning one of Viking Line's requirements was to be able to switch from standard air system with air cooler to HAM with the engine running. This requirement was met through butterfly valves. Due to the good operating experience, this redundancy is not required any more. In emergency case without HAM and without inlet air cooling it has been checked that the available power is still 50-60%.
6. Low weight of equipment and compact installation. Compared to other NOx reduction measures HAM requires less auxiliary equipment. With the exception of the humidifier and a small catch tank in the water circulation no tanks are necessary.

Contrary to that SCR systems require catalysts in the exhaust gas channels and storage tanks for urea treatment and supply. Also water adding NOx reduction systems like fuel water emulsion (FWE) or direct water injection (DWI) require large intermediate and storage tanks for fresh water. Since operated with seawater no water treatment plants are necessary for HAM systems. As a consequence of this, HAM has a remarkably lower weight than other NOx reduction plants.

HAM system is often claimed to be very expensive. Currently that is true for the installation costs. There are still few plants which have been built, and due to this no series effects on production could lead to cost reduction. However, as happened with SCR plants years ago, also for HAM a cost reduction of 50% can be expected after being introduced as a standard product. When considering the operating costs of HAM the situation is totally different. Since HAM does not cause any operating costs or an increase in fuel oil consumption, it can be stated that HAM is one of the cheapest NOx reduction technologies during operation. The following comparison of the total costs of a SCR and HAM system show that HAM is very attractive also from an economical point of view. Assuming a depreciation period of 10 years with an average interest rate of 5% one can calculate annual capital costs of 12.75% of the initial investment costs. The comparison is based on the same reference vessel as in the previous section with an operating profile of 6,000 h/year running with an average load of 90% MCR and a total installed MCR output of approximately 50 MW NOx reduction is calculated both for HAM and for SCR to be 4.5 g/kWh. The specific investment costs are 50 EUR/kW for HAM and 25 EUR/kW for SCR. The total costs are calculated in Table 2.4.1, which shows that the HAM system is approximately 50% cheaper than a comparable SCR system. When reconsidering the strong requirements for lowest weight and operating costs, it can clearly be concluded that the HAM system will be one of the most attractive NOx reduction technologies on fast ferries.

Wärtsilä is developing a Combustion Air Humidification System [2.29]. The introduction of the anti-polishing ring has allowed higher inlet air humidity without impairing lubrication. They use a water nozzle that produces water droplets of a few microns in size. The water is injected after the turbocharger. Indications are that with twice as much water injected as fuel, NOx can be reduced to 2-3g/kWh for medium speed engines (from about 10 or more g/kWh), apparently without significant fuel consumption increase. This is proposed as an alternative to direct water injection where NOx levels below 5g/kWh are required. With direct water injection, fuel consumption increase becomes significant for NOx levels below about 5g/kWh. Wärtsilä are introducing about 60g of water per kg of air. To get that much water into the air, it is necessary to raise the charge air to 70°C or more. Even though the charge air temperature is higher than normal, the large amount of water vapor results in lower temperatures during combustion and expansion and thus reduced NOx, indicated in Fig. 2.37b. In addition, the increased charge mass results in improved turbocharger performance that increases charge air pressure and thus offsets any increase in fuel consumption caused by the reduced combustion temperature due to the energy absorption by the water. Schematic of Munters HAM system is shown in Fig. 2.37a, while the flow scheme of a HAM system is plotted in Fig. 2.38.

Both the Munters HAM system and the Wärtsilä system result in a charge air temperature greater than 70°C compared with a normal value of 50°C. This is a disadvantage with respect to fuel consumption and NOx. Lower charge air temperature improves thermal efficiency by reducing temperature levels, thus reducing heat transfer. Lower charge air temperature is a means for NOx reduction. The Munters system has been shown to give significant NOx reduction (over 65%) at full load without heating the humidifier water. The reduced charge air temperature compensates for the reduced humidity, as far as NOx reduction is concerned.

Wärtsilä NSD has pointed out that with in-cylinder injection of water the evaporation energy is taken from the cylinder contents and not returned [2.29]. It goes out the exhaust. With the humidification systems, the evaporation energy comes from the engine cooling water and the uncooled charge air. The normal air cooler is eliminated. The energy absorbed by the water vapor due to its high specific heat capacity can be partially recovered and converted into work in the expansion stroke and in the turbochargers. A disadvantage of humidification or injection of the water into the inlet air is that only a small proportion of the injected water will be present in the combustion region. Direct injection or fuel-water emulsions place the water more directly in the region of combustion. However, a larger quantity of water can be introduced by inlet air humidification. Too much water in the inlet air could be damaging to the cylinder condition [2.25]. Increased water concentration in the cylinder air can lead to increased smoke, while fuel-water emulsions can reduce smoke [2.15]. MAN B&W are offering HAM on its 4-stroke propulsion range [2.34]. There is significant potential for reducing the charge air temperature with water evaporation. For instance, if 10g of water per kg of charge air (about 0.3 kg water per kg fuel) were evaporated using the energy available in the charge air, the temperature fall of the air would be about 20^{o}C.

2.4.3.4 Exhaust Gas Recirculation

Exhaust Gas Recirculation (EGR) lowers the combustion temperature, thus lowering NOx. EGR reduces combustion temperatures by increasing the specific heat capacity of the cylinder gases (H_2O and CO_2 have higher specific heat capacity than air) and by reducing the overall oxygen concentration. The reduction in oxygen concentration means an increase in the number of moles of inert gas that must be raised to combustion temperatures to react a given amount of oxygen with fuel. Reduced oxygen concentration also leads to reduce combustion rate, which in turn reduces the combustion temperature and NOx. Reduced oxygen concentration also reduces the amount of oxygen available for the NOx reactions.

EGR can be achieved by recirculating exhaust gases and mixing them with the charge air external to the engine cylinder (external EGR), or by modifying the engine scavenging so that a larger proportion of the combustion products stay in the cylinder (internal EGR). In engines operating on poor quality fuel, external EGR can lead to fouling and corrosion problems. The residue from cooling and cleaning the exhaust gas on ships is difficult to dispose of [2.18].

Kawasaki found 28% EGR yielded 69% reduction in NOx on a MAN B&W 5S70MC engine, with a small rise in smoke and fuel consumption. They increased maximum pressure to compensate for increased smoke and fuel consumption, with a slight loss of NOx reduction [2.25]. Wärtsilä NSD found 6% EGR yielded 22% NOx reduction on the 4RTX54 research slow speed engine, with a rise in thermal load on engine components and a rise in exhaust temperature [2.21]. It has been found that when the exhaust gas was recirculated on the high-pressure side using an additional turbocharger, the deposits in the main charge air compressor and charge air cooler can be avoided. A cooler and water separator was used before the recirculating compressor. Tests on a Wärtsilä 9S20 medium speed engine

(1000RPM) showed that a combination of Miller supercharging and EGR at a rate of 4.5% decreased NOx by about 40% with unfavorable soot increase of a factor of 5 to 7. Increased injection pressure and variable turbine geometry were used to reduce smoke.

However, Wärtsilä have abandoned external EGR due to increased smoke, HC and CO as well as deposition of sulfuric acid and soot if the exhaust gas is cooled before returning it to the combustion chamber [2.29]. For lower rating points, extended measures may be necessary. Internal exhaust gas recirculation (IEGR) is a possibility [2.22]. Wärtsilä NSD is developing internal EGR in two-stroke engines as an extended measure beyond engine tuning techniques [31,36]. By reducing the height of the scavenge ports the scavenge air flow into the cylinder is reduced, so the scavenging process is less effective and more of the burnt gases remain in the cylinder for the next cycle. Lowering the scavenge ports also increases the effective expansion stroke length, so more work is done by the cylinder contents in the expansion stroke, resulting in reduced fuel consumption. To overcome the increased thermal load on the engine with internal EGR, Wärtsilä NSD is developing the "Water Cooled Residual Gas" method that involves injection of water during the compression stroke to bring the temperature in the combustion chamber back to that without internal EGR. The temperature of the combustion chamber is high enough to avoid acid deposits. [2.29] The injected water also reduces NOx. Cooling during the compression stroke could not be achieved with fuel-water emulsions.

2.4.3.5 Selective Catalytic Reactors

MAN B&W believes that Selective Catalytic Reactors (SCR) are the solution when NOx levels around 2 g/kWh are required [2.18]. Wärtsilä has suggested that the trade-off between fuel-optimized and NOx-optimized engine operation could be resolved by the use of exhaust gas after-treatment such as SCR. Using SCR allows the engine to be optimized for minimum fuel consumption [2.18]. The method involves mixing of ammonia with the exhaust gas passing over a catalyst where more than 90% of the NOx can be removed. The reactions are [2.21]:

- Urea decomposition:

$$CO(NH_2)_2 \longrightarrow 2NH_2\cdot + CO \qquad (2.4.3)$$

$$CO(NH_2)_2 + H_2O \longrightarrow 2NH_3 + CO_2 \qquad (2.4.4)$$

- NOx reduction:

$$NH_2\cdot + NO \longrightarrow N_2 + H_2O \qquad (2.4.5)$$

$$4NH_3 + 6NO \longrightarrow 5N_2 + 6H_2O \qquad (2.4.6)$$

$$8NH_3 + 6NO_2 \longrightarrow 7N_2 + 12H_2O \qquad (2.4.7)$$

The ammonia is usually supplied as a solution of urea in water, injected into the exhaust stream upstream of the reactor. Sulfur trioxide in the exhaust gas can react with ammonia to form ammonium sulphate, which is and adhesive and corrosive substance. For this reason, SCR units should not be operated below about 300°C. Some ammonia can pass through the reactor (ammonia slip) and become an exhaust contaminant. There are no changes in engine design necessary and no detrimental effects on engine operation. Disadvantages include: (1) high investment cost, (2) cost of supplying the ammonia, (3) sulfur content of the fuel is limited, (4) will not operate below exhaust gas temperature of about 250 to 300 °C, so may not be useful at low load and has to be fitted before the turbocharger on slow speed engines and (5) for very high reduction rates and for operations dominated by transient load changes, measurement of NOx is necessary to avoid excessive ammonia slip.

Control of the SCR plant is achieved by regulating urea-dosing rate. The degree of NOx removal depends on the amount of ammonia added. The SCR system can replace the exhaust silencer. Numerous SCR systems are in use on ships [2.27]. Figure 2.4.19 shows a slow speed installation. SCR in combination with other measures such as water injection could reduce NOx to 0.5 g/kWh [2.32].

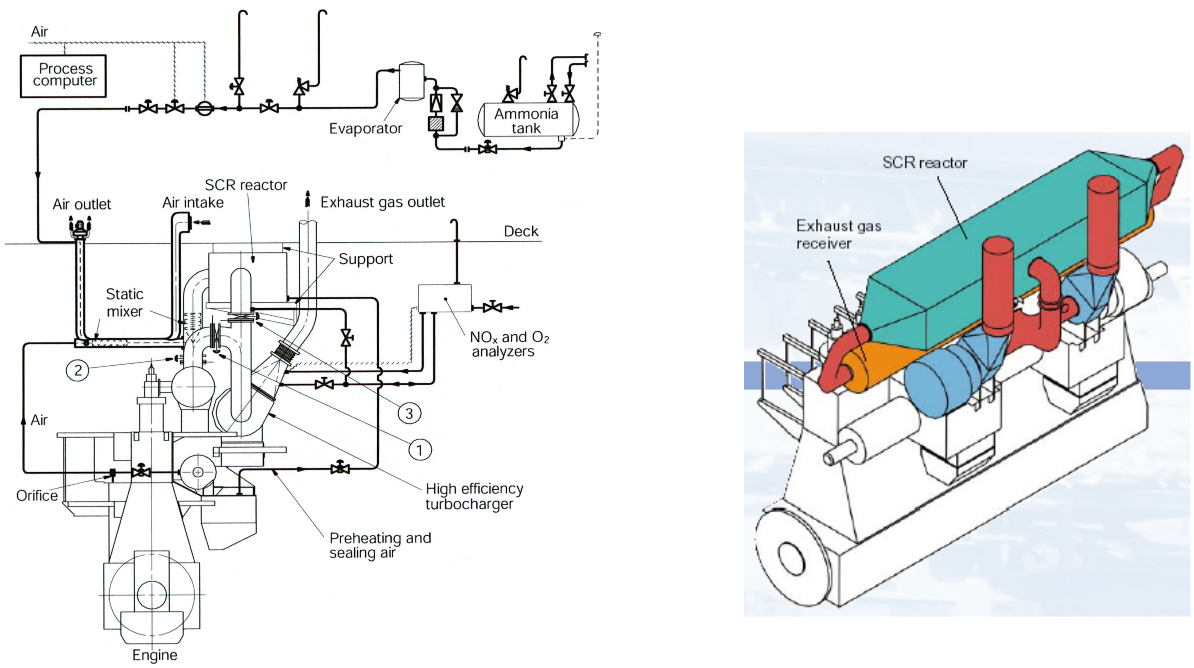


Fig. 2.39: Installation of SCR on a slow speed diesel [2.18]

2.4.4 Future Possibilities

Wärtsilä are developing the Steam Injected Diesel, where steam is injected during the compression stroke, before the cylinder pressure becomes too high [2.29]. The energy from the steam is partially converted to work in the expansion stroke. The large mass of steam would also serve to humidify the combustion air thus reducing NOx. To produce the required steam, the energy of the engine exhaust is increased by reducing the cooling of the combustion space. Presumably the increase in efficiency and reduction of NOx due to steam injection outweigh the tendency for higher NOx due to reduced combustion space cooling. For such large rates of steam injection, it is necessary to inject the steam with a high degree of tangential motion so it remains concentrated in the periphery of the combustion space and thus does not degrade the combustion process. NOx down to 2 to 4 g/kWh is possible, as shown in Fig. 2.40 [2.32].

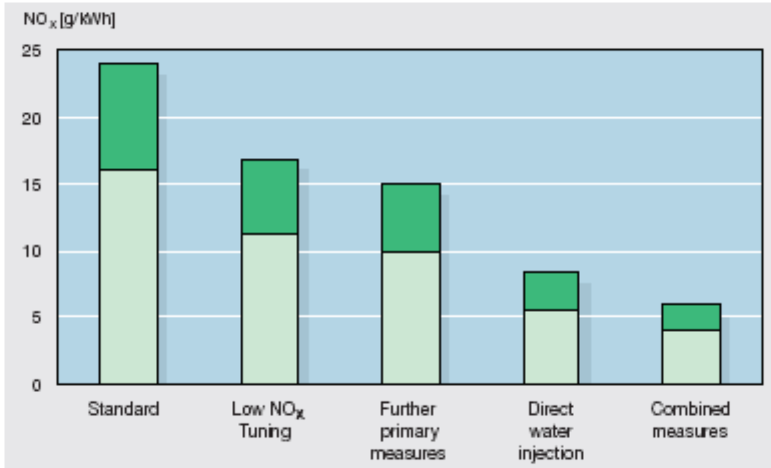


Fig. 2.40: NOx emissions from Sulzer RTA engines using various control techniques [2.32]

The Sulzer common rail system provides the full fuel injection pressure to each injector from a common fuel supply rail. There are no individual fuel pumps for each cylinder. In contrast, the MAN B&W Intelligent Engine (slow speed) uses a common rail servo oil system, where individual injection pumps on each cylinder are driven by a servo piston and controlled electronically. They claim this avoids pressure drops in the fuel supply line that occur in the standard common rail system during injection. Thus, the MAN B&W system gives increased injection rate towards the end of injection. They claim that for modern engines, the optimum injection duration is around 18-20 degrees crank angle at full load, and the peak firing pressure is reached in the second half of that period. In order to obtain the best thermal efficiency, fuel injected after peak pressure must be injected as quickly as possible, to obtain the highest expansion ratio for that part of the combustion energy release. The effects on NOx, fuel consumption and surface temperatures of these measures applied in sequence are illustrated in Fig. 2.31, indicating a no cumulated effect on NOx formation.

2.5 The Intelligent Engine: Development Status and Prospects

2.5.1 MAN B&W Intelligent Engine

This section will discuss MAN B&W's development of computer controlled low speed crosshead engines and the application prospects for such Intelligent Engines. Computerized systems, e.g. for cargo management, satellite navigation and satellite communication, have been used for quite some time in merchant vessels. However, the market has traditionally not favored having electronics integrated as essential parts of the main engine - an exception being the use of electronic governors. It is believed that this situation will change over the next few years, as has happened in the automobile industry over the past 10-15 years. The need for flexibility to cope with diversified emission limits and increasing demands for reliability will undoubtedly lead to comprehensive use of electronic hardware and software in marine engines. Thus, MAN B&W established a separate electronics & software development department some years ago, with hardware and software expertise and the capacity to do professional development work. This will enhance the reliability of conventional as well as Intelligent engines and facilitate new applications - for the former provided, of course, that the owners are prepared to invest in the new systems and that the crews use them.

The basic goal of the development is to reduce the cost of operating the engine and to provide a high degree of flexibility in terms of operating modes. The three major areas of concern in this context are:

(1) Enhanced engine reliability:

- on-line monitoring ensures uniform load distribution among cylinders
- an active on-line overload protection system prevents thermal overload
- early warning of faults under development, triggering countermeasures
- significantly improved low load operation.

(2) Enhanced emission control flexibility:

- emission performance characteristics optimized to meet local demands
- later updating possible.

(3) Reduced fuel and lube oil consumption:

- engine performance fuel-optimized at all load conditions
- as new performance easily maintained over the engine lifetime
- mechatronic cylinder lubricator with advanced dosage control.

2.5.1.2 The Intelligent Engine Concept

To meet the operational flexibility target, it is necessary to have great flexibility in the operation of - at least - the fuel injection and exhaust valve systems. Achieving this objective with cam-driven units would require substantial mechanical complexity that would hardly contribute to engine reliability. To meet the reliability target, it is necessary to have a system that can protect the engine from damage due to overload, lack of maintenance, mal-adjustment, etc. A condition monitoring system must be used to evaluate the general engine condition so as to maintain the engine performance and keep its operating parameters within the prescribed limits and to keep it up to as new standard over the lifetime of the engine. The above indicates that a new type of drive has to be used for the injection pumps and the exhaust valves and that an electronic control and monitoring system will also be called for. The resulting concept is illustrated in Fig.2.5.1.

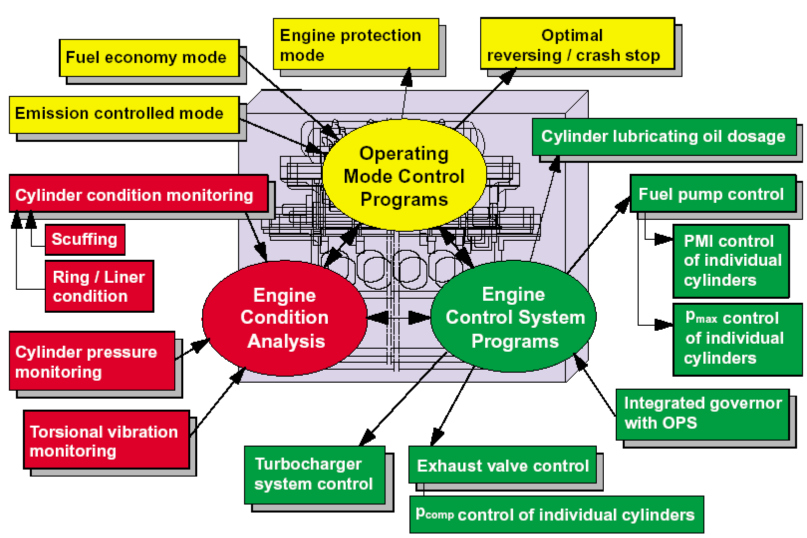


Fig. 2.41: The Intelligent Engine concept [2.3]

The upper part shows the Operating Modes that may be selected from the bridge control system or by the intelligent engine own control system. The control system contains data for optimal operation in these modes, which consist of a number of single modes corresponding, for instance, to different engine loads and different required emission limits. The fuel economy modes and emission controlled modes (some of which may incorporate the use of an SCR catalytic clean-up system) are selected from the bridge. The optimal reversing/crash stop modes are selected by the electronic control system itself when the bridge control system requests the engine to carry out the corresponding operation. The engine protection mode is, in contrast, selected exclusively by the condition monitoring and evaluation system, regardless of the current operating mode. Should this happen in circumstances where, for instance, reduced power is unacceptable for reasons of the safety of the ship, the protection mode can be cancelled from the bridge.

The center of Fig.1 shows the brain of the system: the electronic control system. This analyses the general engine condition and controls the operation of the following engine systems, shown in the lower part of Fig. 2.41, the fuel injection system, the exhaust valves, the cylinder lubrication system and the turbocharging system. Some of the control functions for these units are, as mentioned above, pre-optimized and can be selected from the bridge. Other control functions are selected by the engine condition monitoring system on the basis of an analysis of various input from the units on the left and right sides of Fig. 2.41: general engine performance data, cylinder pressure, cylinder condition monitoring data and output from the Load Control Unit [2.3].

The Condition Monitoring and Evaluation System is an on-line system with automatic sampling of all normal engine performance data, supplemented by cylinder pressure measurements, utilizing the CoCoS-EDS system. When the data-evaluation system indicates normal running conditions, the system will not interfere with the normal pre-determined optimal operating modes. However, if the analysis shows that the engine is in a generally unsatisfactory condition, general countermeasures will be initiated for the engine as a unit. For instance, if the exhaust gas temperature is too high, fuel injection may be retarded and/or the exhaust valves may be opened earlier, giving more energy to the turbocharger, thus increasing the amount of air and reducing the exhaust gas temperature. At all events, the system reports the unsatisfactory condition to the operator together with a fault diagnosis, a specification of the countermeasures used or proposed, and recommendations for the operation of the engine until normal conditions can be re-established or repairs can be carried out.

The 4T50MX research engine in MAN B&W R&D Center in Copenhagen was operated from 1993 to 1997 with the first-generation Intelligent Engine (IE) system. The engine has been running with this system for the IE development as well as for its normal function as a tool for our general engine development. The 1990 running hours logged during that period of time has provided us with significant experience with this system. Being the first generation of IE, the system was somewhat over-engineered and relatively costly compared with the contemporary camshaft system. On the other hand, the system offered much greater flexibility, which has proved its value in the use of the research engine as one of our most important development tools.

In 1997, the engine was fitted with second-generation IE systems, please refer to Fig. 5.42 showing the fuel injection and exhaust valve actuating systems on the engine. The second-generation systems, to be described in more detail in the following, have been developed in order to:

- simplify the systems and tailor them to the requirements of the engine
- facilitate production and reduce the costs of the IE system
- simplify installation and avoid the use of special systems wherever possible.

On the electronic software/hardware side, the original first-generation system was used for a start. Since then, significant development efforts have been invested in transforming the electronic part of the IE system into a modular system, where some of the individual modules can also be used in conventional engines. This means development of a new computer unit and large software packages - both of which have to comply with the demands of the Classification Societies for marine applications.

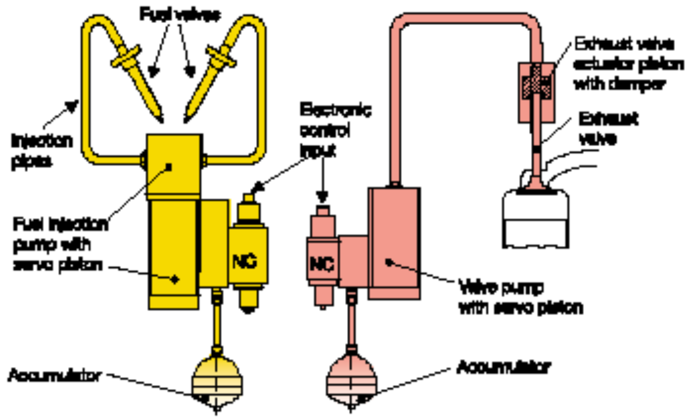


Fig. 5.42: General layout for fuel injection and exhaust valve actuation systems [2.3]

2.5.1.3 Design Features of the Second-Generation IE System

The principle layout of the new system, replacing the camshaft system of the conventional engine, is illustrated in Fig. 2.43. The system comprises an engine driven high-pressure servo oil system, which provides the power for the hydraulically operated fuel injection and exhaust valve actuation units on each cylinder. Before the engine is started, the hydraulic power system (or servo oil system) is pressurized by means of a small electrically driven high-pressure pump. Furthermore, the starting air system and the cylinder lubrication system have been changed compared with the conventional engine series. A redundant computer system controls all these units. The following description will outline the main features of these systems, together with our recent development work and experience.

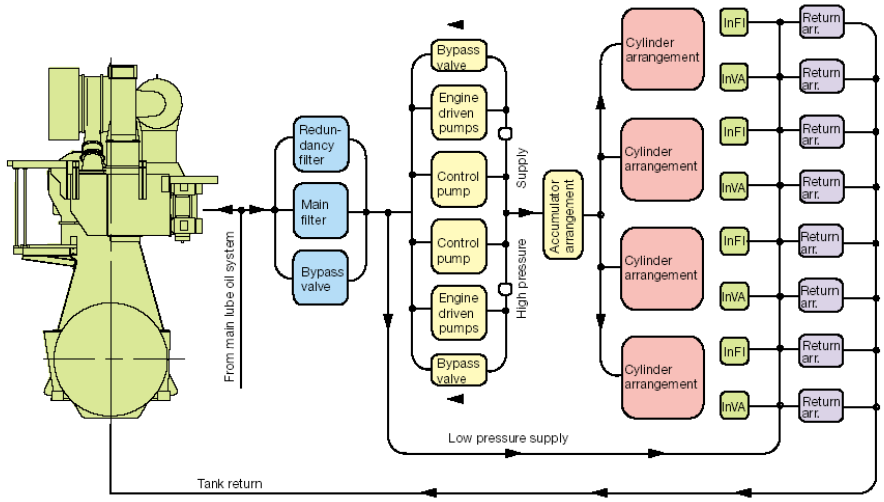


Fig. 2.43: System diagram for the hydraulic flow (servo oil system) [2.3]

Engine-driven multi-piston pumps supply high-pressure lube oil to provide the necessary power for fuel injection and exhaust valve actuation and thus replace the camshaft power-wise. The multi-piston pumps are conventional, mass-produced axial piston pumps with proven reliability. The use of engine system oil as the activating medium means that a separate hydraulic oil system is not needed, thus extra tanks, coolers, supply pumps and a lot of piping etc. can be dispensed with. However, generally the engine system oil is not clean enough for direct use in high-pressure hydraulic systems, and it might be feared that the required 6µm filter would block up quickly. Extensive development work has been undertaken in collaboration with a filter supplier (B&K) in order to ensure the cleanliness required for such systems - the very positive long-term results are described below. Against this background, and based on the fact that the clean lube oil from the engine was at least as suitable for use in the hydraulic system as conventional hydraulic oil, we decided to base our system on fine-filtered system lube oil. This is supplied from the normal system oil pumps, providing a higher inlet pressure to the high-pressure pumps than otherwise - this being yet another benefit.

2.5.1.4 Current Status in Propulsion of Medium Sized Container Ships

As will appear from the above-mentioned engine choice, MAN B&W 60 and 70MC/MC-C engines have an output suitable for medium sized container vessels, maybe with the exception of the upper end of this range, where larger engines are needed. Like the other engines in the MC program, the models in this medium power range have been continuously updated, particularly with a view to enhancing the reliability of the combustion chamber and to controlling (minimizing) the cylinder oil consumption. The combustion chambers of these engines feature a piston crown with "high topland", i.e. the distance between the piston top and the topmost piston ring has been increased, as shown in Fig.2.5.4. The topmost piston ring is a CPR type (Controlled Pressure Relief), whereas the other three rings are with an oblique cut. The topmost piston ring is higher than the lower ones. The piston skirt is of cast iron with copper bands, which has eliminated skirt scuffing. The performance of the piston ring pack has also improved, resulting in higher TBOs (Time Between Overhauls) for the piston. Tests with high-topland pistons have been demonstrated that cylinder oil must be injected into the cylinder at the exact position and time that ensures the optimal use of the lube oil. This is hardly possible with the conventional, mechanical cylinder lubricators for which reason we have developed a computer controlled electronic cylinder lubrication system, the Alpha lubricator.

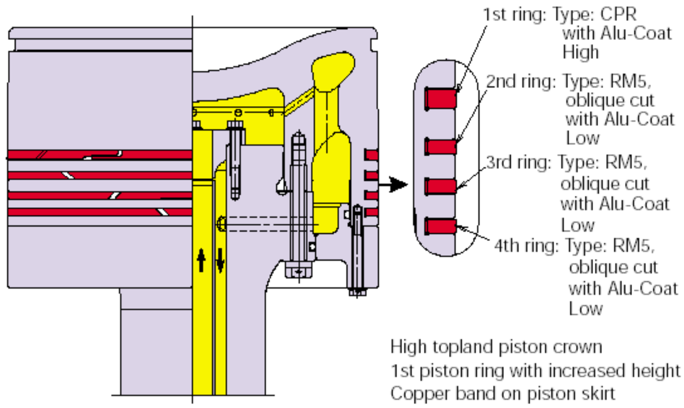


Fig. 2.44: Piston for S-MC-C engines [2.3]

The Alpha lubrication system features a high-pressure pump and an injection arrangement that injects a specific volume of oil into each cylinder for every fourth revolution, as shown in Fig. 2.45. The system is controlled in such a way that the oil can be introduced to the individual cylinder at any piston position but, preferably, when the piston rings are adjacent to the lubricating quills. The system has been fine tuned on MAN B&W's 4T50MX research engine. Good results have been obtained on a 7S35MC engine, and the system has been in service for two years on a K90MC engine, also with good results. The results show high reliability and a very good cylinder condition, giving unchanged wear rates with a lower cylinder oil feed rate than normally recommended. Consequently, the Alpha lubrication system will be installed on a number of MC/MC-C engines that are about to enter service.

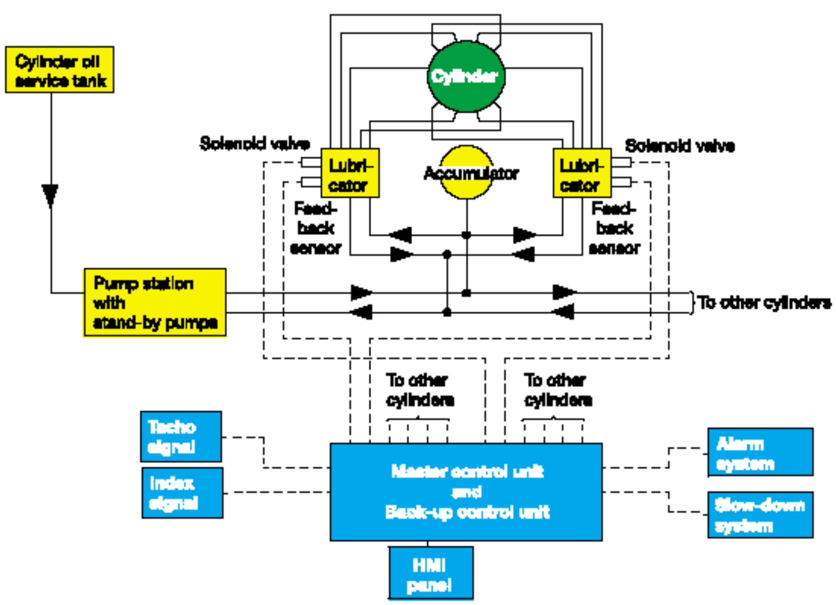


Fig.2.45: Cylinder lubrication with Alpha lubricators [2.3]

2.5.2 Sulzer Intelligent Engine: RT-flex 60C Low-speed Engine

2.5.2.1 Introduction to RT-flex Engines

The RT-flex60C (Fig. 2.46 and 2.5.7) is the first Sulzer low-speed engine to incorporate the RT-flex system with electronic-controlled common-rail technology as standard from the outset (Fig. 2.47) [2.37]. It is not planned to offer the engine in a conventional camshaft version. The decision to follow this path was taken after the very positive reception by ship owners of the Sulzer RT-flex system after it was first applied to a full-scale research engine in June 1998. The RT-flex system offers distinctive operational benefits that are not possible with camshaft engines. The first RT-flex engine went into shipboard service in September 2001, sufficiently before the first RT-flex60C was built in 2002 to incorporate the service experience. The RT-flex60C engine also incorporates TriboPack technology from the outset, as shown in Fig. 2.5.9. TriboPack design measures, today a well accepted standard, provide important improvements in piston-running behavior, for more reliability, longer times between overhauls and lower cylinder oil feed rates.

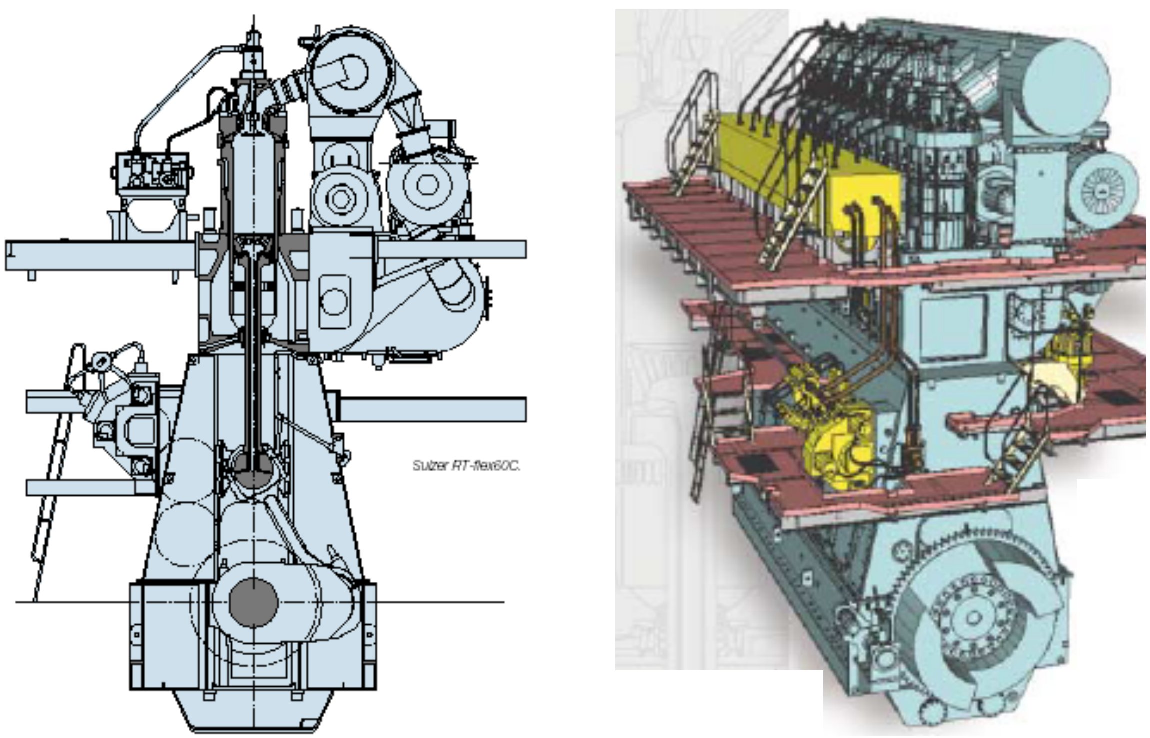


Fig. 2.46: General view of RT-flex60C engine [2.37]

Principal parameters of Sulzer RT-flex60C engines		
Bore	mm	600
Stroke	mm	2250
Output, R1	kW/cyl bhp/cyl	2360 3210
Speed range, R1-R3	rpm	114–91
BMEP at R1	bar	19.5
Pmax	bar	155
Mean piston speed at R1	m/s	8.55
Number of cylinders		5-8
BSFC:		
at full load, R1	g/kWh g/bhph	170 125
at 85% load, R1	g/kWh g/bhph	167 123

Fig. 2.47: Principal parameters of Sulzer RT-flex60C engines [2.37]

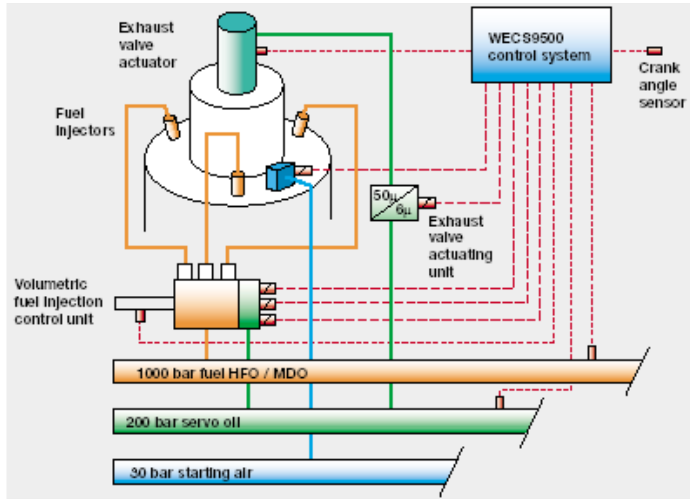


Fig. 2.48: Electronic-controlled common-rail system of RT-flex60C engine [2.37]

2.5.2.2 The RT-flex Engine Concept

The Sulzer RT-flex60C low-speed marine diesel engines are tailor-made for the economic propulsion of container feeder vessels to serve the growing fleets of the major container lines. They are equally suitable for other vessel types such as reefers and car carriers with similar requirements. In these roles, they offer clear and substantial benefits:

- Competitive first cost
- Lowest possible fuel consumption over the whole operating range
- Three years' time between overhauls
- Low maintenance costs owing to reliable and proven design
- Extremely good 'slow steaming' capability through the Sulzer RT-flex technology which offers a high degree of operational flexibility
- Full compliance with the NOx emission regulation of Annexe VI of the MARPOL 1973/78 convention
- Smokeless operation even at lowest loads and speeds owing to the Sulzer RT-flex technology

Development of the Sulzer RT-flex60C engine type was initiated in 1999. It was seen that the changing container ship new building market was changing, not just at the upper end with the container liners but also in the middle range employed as feeder vessels. With container liner new buildings becoming larger, with capacities of 5500TEU or more becoming common, and very many such vessels being contracted, it was clear that more container feeder vessels would be needed, of possibly up to 3000TEU capacity. Development of the Sulzer RT-flex system of electronic-controlled common-rail fuel injection and exhaust valve actuation was timely and has enabled the introduction of marine diesel engines with clear user benefits.

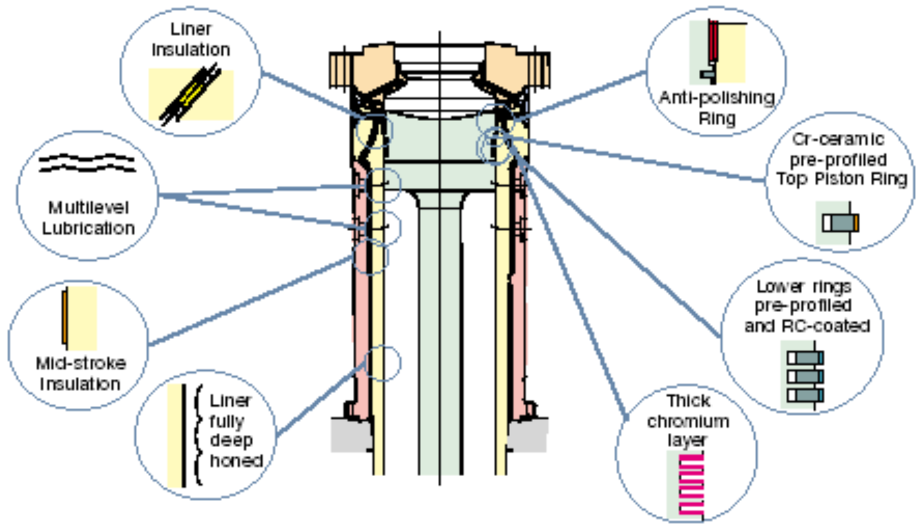


Fig. 2.49: TriboPack design used in Sulzer RT-flex60C engine [2.37]

The common rail for fuel injection is a manifold running the length of the engine at just below the cylinder cover level. The common rail and other related pipe works are neatly arranged on the top engine platform and readily accessible from above (Fig. 2.50). The common rail is fed with heated fuel oil at the usual high pressure (nominally 1000 bar) ready for injection. The supply unit has a number of high-pressure pumps running on multilobe cams. The pump design is based on the proven injection pumps used in Sulzer four-stroke engines. Fuel is delivered from this common rail through a separate injection control unit for each engine cylinder to the standard fuel injection valves that are hydraulically operated in the usual way by the high-pressure fuel oil. The control units, using quick-acting Sulzer rail valves, regulate the timing of fuel injection, control the volume of fuel injected, and set the shape of the injection pattern. The three fuel injection valves in each cylinder cover are separately controlled so that, although they normally act in unison, they can also be programmed to operate separately as necessary.

2.5.2.3 Design Features of the RT-flex Engines

Piston-running behavior

Today the time between overhaul (TBO) of low-speed marine diesel engines is largely determined by the piston-running behavior and its effect on the wear of piston rings and cylinder liners. For this reason, today's Sulzer RTA-series engines incorporate TriboPack technology – a combination of design measures that enable the TBO of the cylinder components, including piston ring renewal to be extended to at least three years. At the same time, TriboPack (Fig. 2.49) offers more safety for piston running under adverse conditions and thus allows standard cylinder lubricating oil feed rates to be as low as 1.0 g/kWh and even less. The design measures incorporated in TriboPack are:

- Multi-level cylinder lubrication
- Liner of the appropriate material, with sufficient hard phase
- Careful turning of the liner running surface and deep-honing of the liner over the full length of the running surface
- Insulating tubes in the cooling bores in the upper part of the liner

- Mid-stroke liner insulation
- Pre-profiled piston rings in all piston grooves
- Chromium-ceramic coating on top piston ring
- RC (Running-in Coating) piston rings in all lower piston grooves
- Anti-Polishing Ring (APR) at the top of the cylinder liner
- Increased thickness of chromium layer in the piston ring grooves.

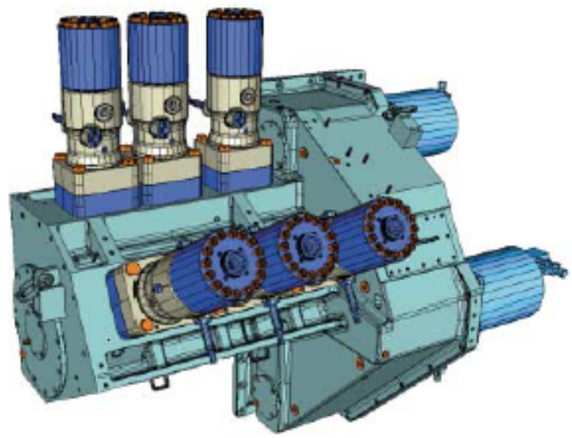


Fig. 2.50: Sup unit with fuel pumps on left and servo oil pumps on right [2.37]

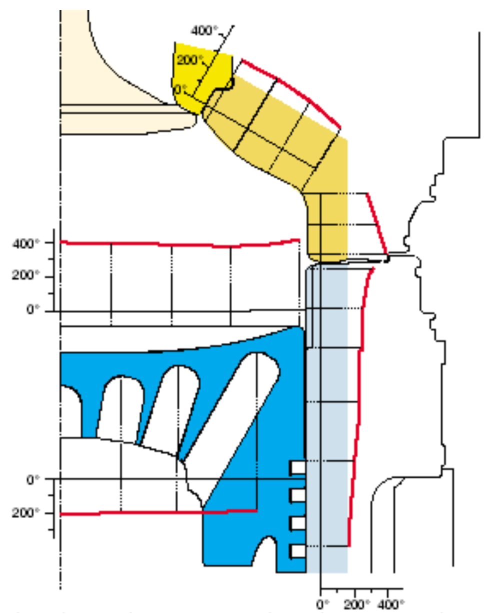


Fig. 2.51: Temperature distribution in combustion chamber of RT-flex60C engine

Most of the design measures in TriboPack have been employed in Sulzer RTA engines for several years. They are now combined systematically as a standard package. Although each individual measure of TriboPack improves piston-running behavior, only the application of the complete package delivers the full benefits to the ship owners.

A key element of TriboPack is the deep-honed liner. Careful machining and deep honing gives the liner an ideal running surface for the piston rings, together with an optimum surface microstructure. The RT-flex60C has four piston rings, all 16 mm thick and the same geometry. The Anti-Polishing Ring prevents the build up of deposits on the top land of the piston that can cause bore polishing on the liner and damage the oil film. Whilst trying to avoid corrosive wear by appropriate optimizing of liner wall temperatures, it is necessary to keep as many water droplets as possible out of engine cylinders. Thus, a newly developed, highly efficient vane-type water separator after the scavenge air cooler and the effective water drainage arrangements are absolutely essential for good piston running.

Load-dependent cylinder lubrication is provided by the well-proven Sulzer multi-level accumulator system. The lubricating pumps are driven by frequency-controlled electric motors. On the cylinder liner, oil distributors bring the oil to the different oil accumulators. For ease of access, the quills are positioned in dry spaces instead of in the way of cooling water spaces.

Another important contribution to fuel economy of the RT-flex60C engines is the capability to adapt easily the injection timing to various fuel properties having a poor combustion behavior. Variable injection timing (VIT) over load has been a traditional feature of Sulzer low-speed engines for many years, using a mechanical arrangement primarily to keep the cylinder pressure high for the upper load range. This is much easier to arrange in an electronic-controlled engine. Yet the settings for compliance with the NOx limit have to be maintained. An important contribution to the overall fuel consumption can come from exhaust heat recovery. The RT-flex60C offers a clear advantage in this respect with an exhaust gas temperature of 285 °C for an unsurpassed high potential for waste heat recovery. High exhaust temperatures are traditional for Sulzer low-speed engines. This can be obtained by efficient scavenging processes.

The bore-cooled steel cylinder cover is secured by eight elastic studs arranged in four pairs. It is equipped with a single, central exhaust valve in Nimonic 80A which is housed in a bolted-on valve cage. There are three fuel injection valves symmetrically distributed in each cylinder cover. This arrangement of injection valves helps to equalize the temperature distribution on the piston crown over the circumference around the liner and in the cylinder cover. Anti-corrosion cladding is applied to the cylinder covers downstream of the injection nozzles to protect the cylinder covers from hot corrosive or erosive attack. The piston comprises a forged steel crown with a very short skirt. The compact piston contains four rings of the same height, as part of the TriboPack.

The cylinder liner is bore cooled which has been proven to meet necessary high demands on temperature distribution. Although insulation tubes are criticized as being complicated, they do allow an invaluable adjustment of the temperature distribution in the liner. Yet only local insulation with physically correct heat conduction parameters, together with a proper geometry can lead to the desired temperatures and, at the same time, limit thermal stress.

Cleaner in the environment

With the current popular concern about the environment, exhaust gas emissions have become an important aspect of marine diesel engines. All Sulzer RTA engines comply with the NOx emissions limit of Annex VI of the MARPOL 73/78 convention as standard. RT-flex engines, however, come comfortably below this NOx limit by virtue of their extremely wide flexibility in optimizing the fuel injection and exhaust valve processes. The most visible benefit of RT-flex engines is, of course, their smokeless operation at all ship speeds, as indicated in Fig. 2.52. This is achieved by the superior combustion gained with the common-rail system. It enables the fuel injection pressure to be maintained at the optimum level irrespective of engine speed. In addition, at very low speeds, individual injectors are selectively shut off and the exhaust valve timing adapted to help to keep smoke emissions below the visible limit. In contrast, engines with the traditional jerk-type injection pumps have increasing smoke emissions as engine speed is reduced because the fuel injection pressure and volume decrease with speed and power, and they have no means of cutting off individual injection valves and changing exhaust valve timing.

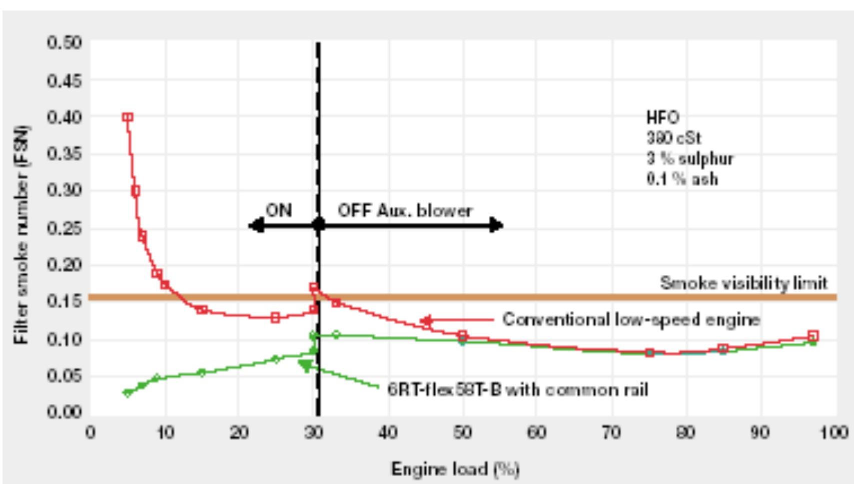


Fig. 2.52: Smokeless operation on the RT-flex engine from the sea trials of the Gypsum Centennial compared with the conventional low-speed marine engine [2.37]

Yet the environmental benefits of RT-flex engines need not be restricted by the current state-of-the-art. As all settings and adjustments within the combustion and scavenging processes are made electronically, future adaptations will be possible simply through changes in software, which could be readily retrofitted to existing RT-flex engines.

For example, one possibility for future development would be to offer different modes for different emissions regimes. In one mode, the engine could be optimized for minimum fuel consumption while complying with the global NOx limit. Then to satisfy local emissions regulation the engine could be switched to an alternative mode for even lower NOx emissions while the fuel consumption may be allowed to increase. As well as investigating the scope of possibilities of the RT-flex system, Wärtsilä is carrying out a long-term research program to develop techniques for further reducing exhaust emissions, including NOx, SOx and CO_2.

Combustion chamber

The combustion chamber is a key issue for an engine's reliability. Careful attention is needed for the piston cooling, as well as for the layout of the fuel injection spray pattern to achieve moderate surface temperatures and to avoid carbon deposits. Low combustion chamber temperatures are not only responsible for long times between overhauls but also desirable to have a higher degree of freedom to reach low NOx emissions (Fig.2.5.11). At Wärtsilä, the optimization of the fuel injection is simulated with modern calculation tools, such as CFD (computerized fluid dynamics) analysis and then is confirmed on the engine. The well-proven bore-cooling is also adopted in all combustion chamber components, together with shaker cooling in the piston, to achieve high heat transfer coefficients over a long time and thus to control temperatures and thermal strains as well as mechanical stresses of the components.

2.5.2.3 Sulzer RT-flex58T-B Engines

The first Sulzer low-speed engine with common-rail fuel injection has successfully completed one-year service. This Sulzer 6RT-flex58T-B engine, of 11275 kW output, is installed in the 47950tdw bulk carrier Gypsum Centennial. It was recently inspected at Tampa, Florida, during the ship guarantee dry-docking after 5295-hour operation. During this first year of operation, the engine ran very successfully. Although the engine has not been without its problems, they were all teething problems and have all been overcome. There have not been any problems with the concept and the very few major faults were largely mechanical with easily defined solutions. This is the world□ first large low-speed engine in service with electronically-controlled common-rail systems for fuel injection and exhaust valve actuation. It must be remembered that the engine was built to operate only using the RT-flex common-rail system with no alternative. It went to sea as a fully industrialized product fully capable of continuous heavy-duty commercial operation. It achieved this performance with very good success. The Gypsum Centennial was built for her owners Gypsum Transportation Ltd (GTL) of Bermuda by Hyundai Mipo Dockyard in Ulsan, Korea. Sea trials were completed in September and she entered service in mid November 2001.

2.6 泵

商船來說，機艙中常用的泵，約有離心式泵、往復式泵、螺旋式泵、噴射式泵、齒輪式泵等。

2.6.1 離心式泵的基本原理

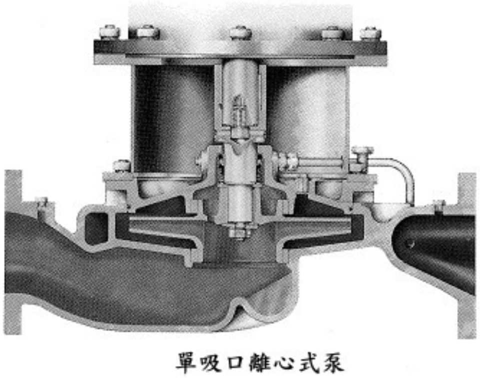
泵殼中當注滿流體時，當葉輪(Impeller)高速旋轉，流體沿著葉輪方向流出，由於葉輪中的流體已被排出，造成葉輪中心壓力下降，於是吸入管內的流體因大氣壓的關係，將流體壓入葉輪中心處，於是完成了離心式泵的抽排作用[1]。

2.6. 2 離心式泵

離心式泵葉片多為單吸口及雙吸口葉片為主，如圖 6.1 及圖 6.2 所示[2]。其中雙吸口葉輪又可用在減輕軸向推力。

另因離心式泵因葉輪有軸向推力的因素，故有推力平衡裝置，下列為各種方式

1. 雙吸口葉輪
2. 平衡口及平衡管
3. 對向葉片
4. 液壓平衡裝置
5. 推力軸承



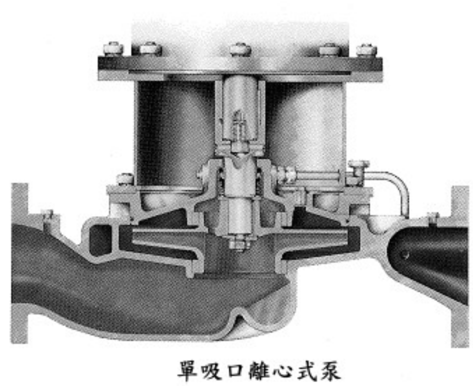


圖 6.1 單吸口離心式泵

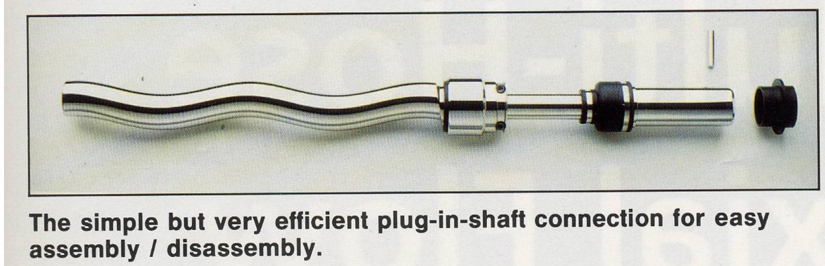
圖 6.2 雙吸口離心式泵

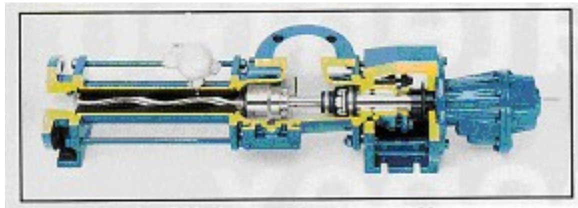
2.6. 3 螺旋泵

於泵殼內有旋轉螺桿，流體在泵殼內，沿著螺桿方向，將流體排出，如液壓油泵、滑油泵、殘渣泵(Sludge pump)等。螺旋泵又可分單螺旋泵，雙螺旋泵及三螺旋泵等[4,5,9]。

2.6.4 單螺旋泵

如下圖 6.3 所示，連接軸易於拆解及組合





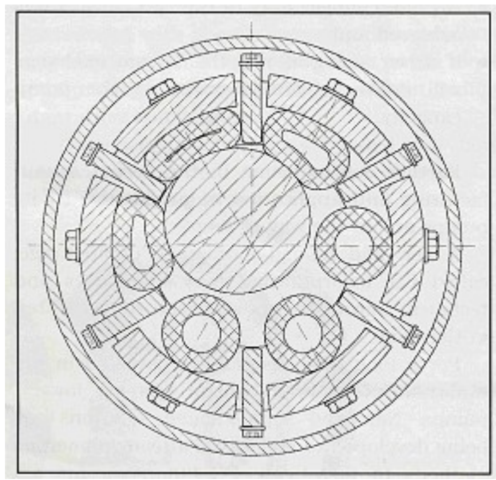


圖 6.3 單螺旋泵

2.6.5 三螺旋泵

三螺旋泵(IMO)，如下圖 6.4 所示：

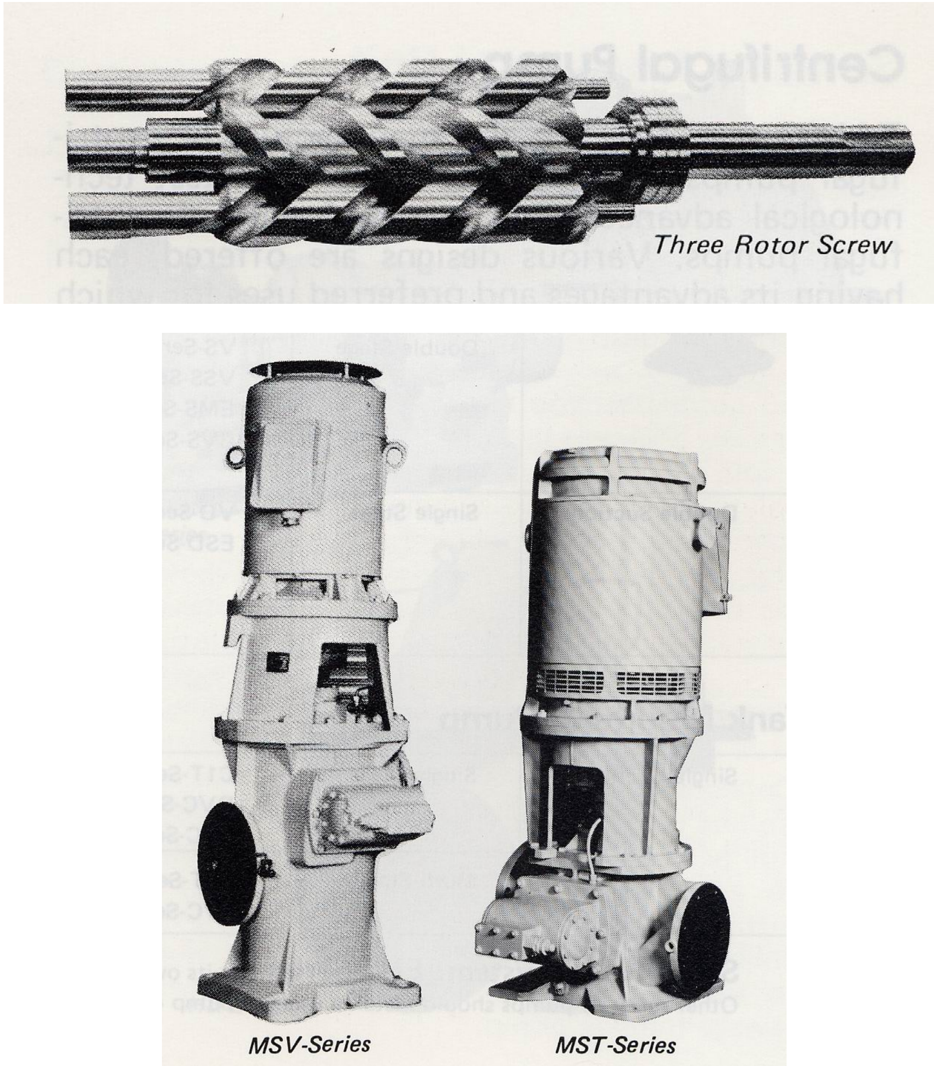


圖 6.4 三螺旋泵

2.7 板式熱交換器

2.7.1 其板子又可分四孔封閉型(Blanked)和非封閉住型(unblanked)

2.7.2 平板材質，因使用之流體不同，而有不同之材質[1]

材質	適合流體
不鏽鋼(SUS304,316)	淨水、河川水、食用油、礦物油

鈦及鈦鈀(Ti，Ti-Pd)	海水、鹽水、鹽化物
20Cr，18Ni，6Mo，(254SMO*1)	稀硫酸、稀硫鹽化物水溶液、無機水溶液
鎳(Ni)	高溫、高溫度苛性鈉
HASTELLOY 合金(C276，D205，B2G)*2	濃硫酸、鹽酸、磷酸
石墨	鹽酸、中濃度硫酸、磷酸、氟酸

2.7.3 平板材質密合墊材質，也因使用之流體不同，而需有不同之材質配合[1]

材質	使用溫度℃	適合流體
NBR	-15～+110℃	水，海水，礦物油，鹽水
HNBR	-15～+140℃	高溫礦物油，高溫水
EPDM	-25～+160℃	熱水，水蒸汽、酸、鹼
Viton	-25～+160℃	酸、鹼、流體

2.8 空壓機

空氣品質是供應空氣中，很重要的一環，將空氣過濾及乾燥，是不可少的步驟。空氣過濾可將空氣中雜質，濕氣及油滴去除。乾燥器可將空氣中水份去除，乾燥方式有下列三種：

1.Desiccant Dryers
2.Refrigerated Dryers
3.Point of Use Dryers

2.8.1 Point of Use Dryers

如下圖 6.5 所示，乃 Point of Use Dryers 之型式

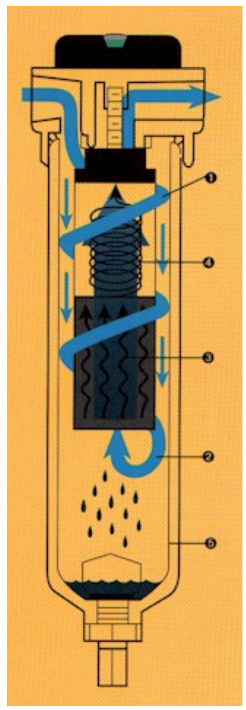


圖 6.5 Point of Use Dryers 型式

2.9 造水機

2.9.1 造水機的原理

造水機的原理，有逆滲透法、蒸發法等方式，逆滲透法因成本高昂，現已不可見。船舶機艙常用之方法，還是以低壓蒸發方式為主，因海水蒸發之飽和溫度和飽和壓力成比例，為了降低蒸發溫度，必須利用噴射泵(Injection pump)來抽真空，利用主機缸套水的溫度，就可使海水蒸發而產生水蒸汽，將水蒸汽凝結後，就可產生淡水。低壓蒸發造水機，以其熱交換器的構造，又可分殼管式及板式造水機。[3,4,5]

2.9.2 板式造水機的優點

1.較少的安裝空間

2.減少更換墊片的開銷

3.減少腐蝕的問題

4.減少熱交換器板子上的鹽垢的產生及阻塞

5.容易維護保養

下圖 6.6 乃板式造水機的分解圖

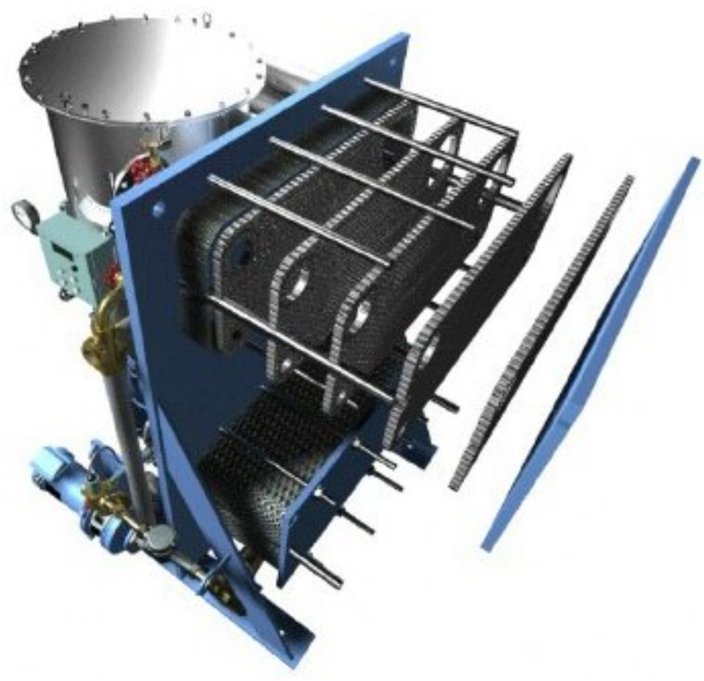


圖 6.6 板式造水機的分解圖

板式造水機的操作原理如下圖 6.7 所示：

1.利用抽射泵(Injector Pump)抽真空

2.抽射泵(Injector Pump)抽未蒸發完的鹽水(Brine water)

3.再利用缸套水(內燃機之船舶)或蒸汽(蒸汽機之船舶) 來加熱海水

4.產生之水蒸汽經冷凝器冷卻後之淡水，被淡水泵(Distillate pump)抽送至水櫃

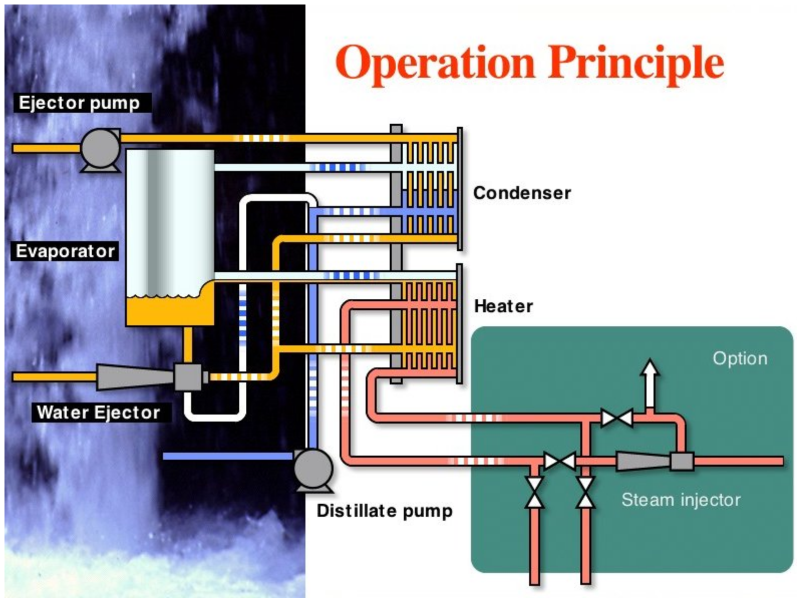


圖 6.7 造水機的系統

下圖 6.8 為 NIREX 板式造水機的剖面圖，及各部元件名稱：[1]

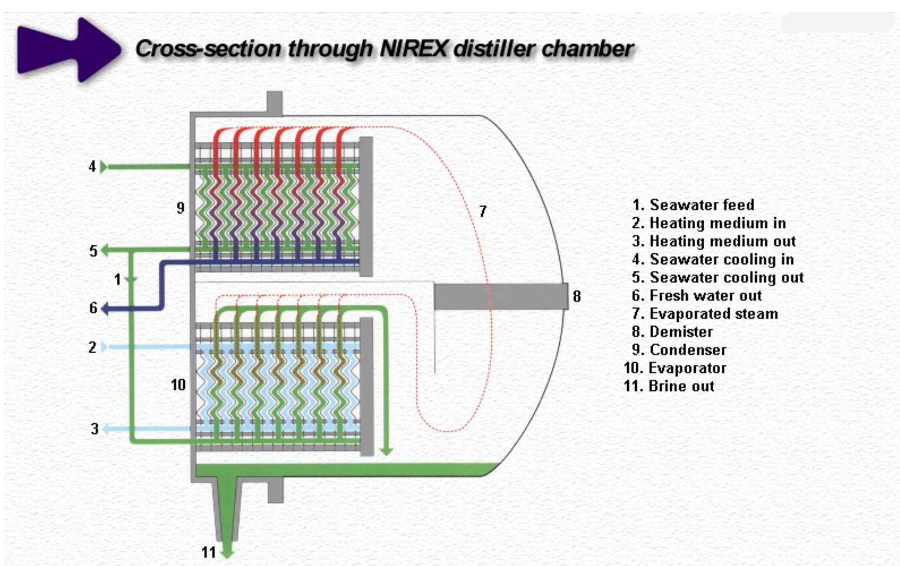


圖 6.8 NIREX 板式造水機

下圖 6.9 是蒸發室及冷凝室的剖面示意圖：[1]

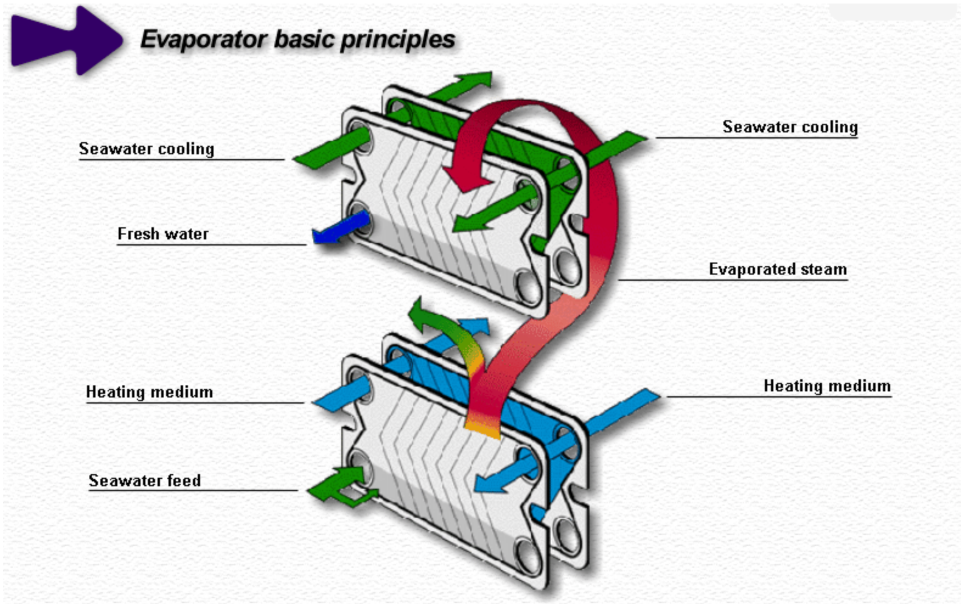


圖 6.9 蒸發室及冷凝室的剖面圖

下圖 6.10 是實際板式造水機之剖面圖，可清楚見到蒸發室(Evaporator)、冷凝室(Condenser)、除霧器(Demister)。[1]

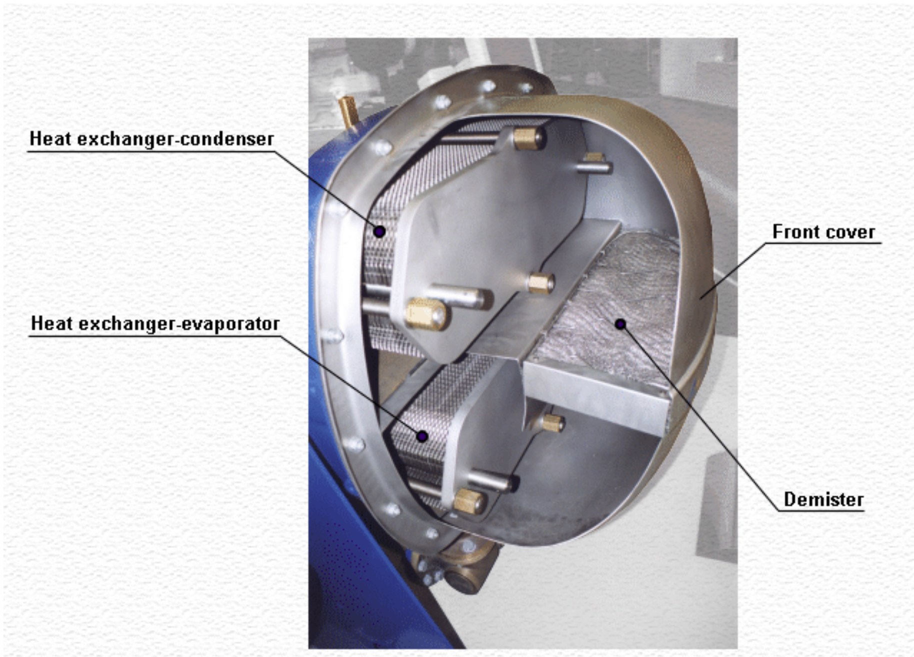


圖 6.10 實際板式造水機之剖面圖

下圖 6.11 是板式造水機之系統圖，缸套水已加熱海水，並產生蒸汽，蒸汽由海水冷卻為淡水，造出之淡水由檢鹽計(Salinometer)監測中，現淡水中鹽份含量為 0.4PPM。[10]

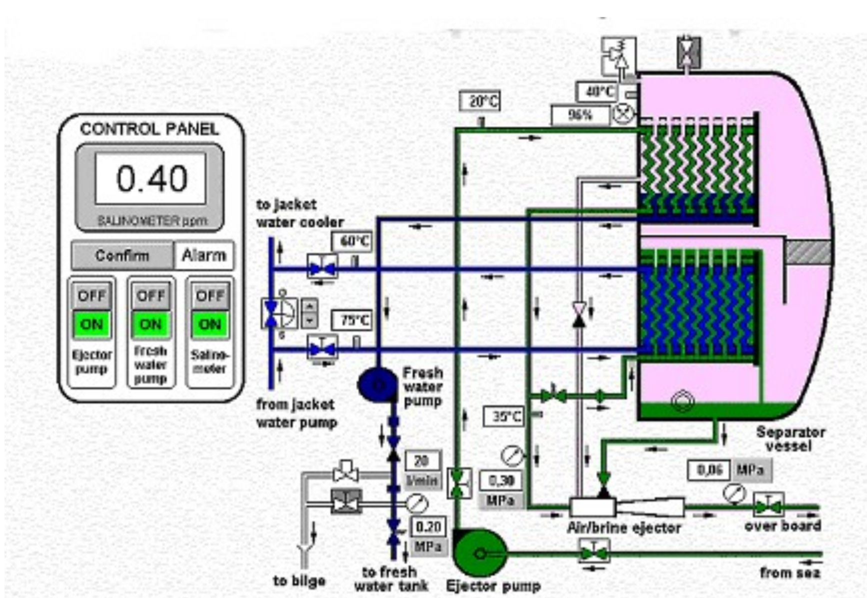


圖 6.11 是板式造水機之系統圖

2.10 淨油機

2.10.1 淨油機的基本原理

由下圖 6.12 所示，若假設淨油機反轉 90°，此時燃油僅受重力的影響，就如同下圖之沈澱櫃一般。[1]

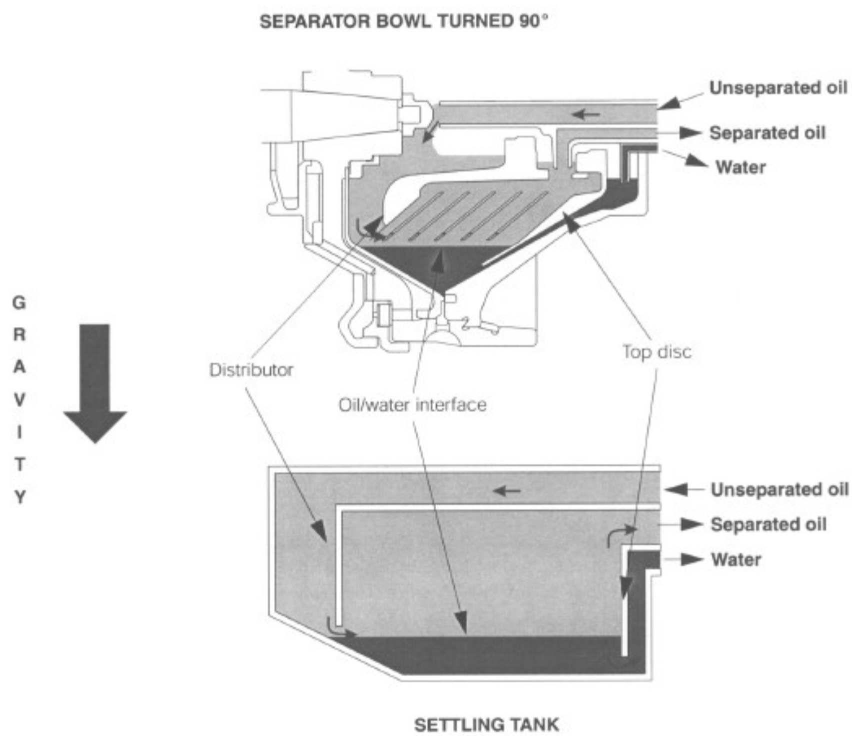


圖 6.12 淨油機反轉 90°

未清潔的燃油經由分配器(Distributor)進入盤堆(Disk stack)下方，再經由孔道進入各盤堆之間。因水份及雜質較重，可因比重不同，產生的重力不同而將水及雜質分離至最下側。

但現在的燃油愈來愈差，若只是靠重力的分離力量來分離水份及雜與，時間必須

長久且效果不佳，故必須提高分離之力量，唯一途徑就是增加淨油機的轉速，才可增加離心力。

2.10.2 Alfa-Laval SU400 型淨油機[1]

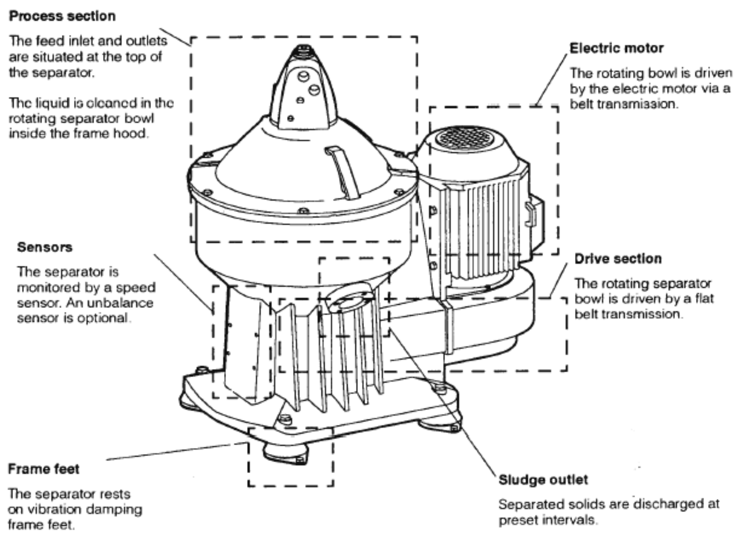


圖 6.13 Alfa-Laval SU400 型淨油機

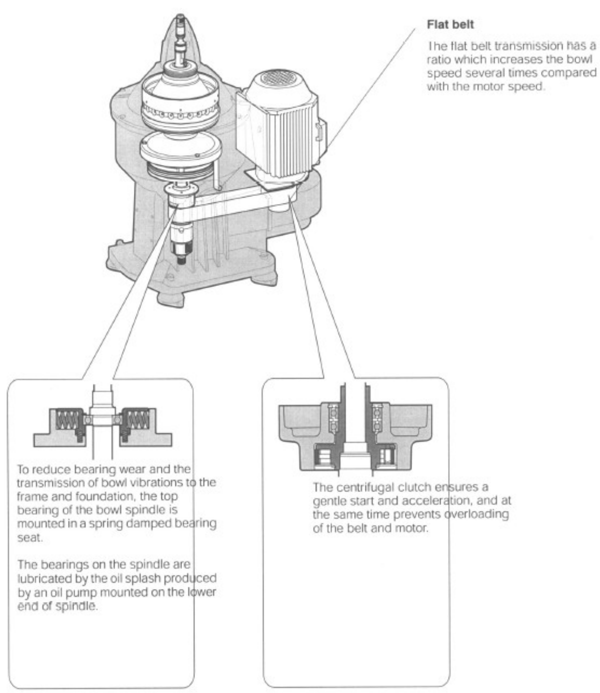


圖 6.14 Alfa-Laval SU400 型淨油機

上圖 6.13 及 6.14 乃 Alfa-Laval SU400 型淨油機，其改進之情形，說明如下：

1. 馬達帶動扁平皮帶，皮帶再帶動淨油機，淨油機起動後的轉度，可由速度感測器來監測。
2. 淨油機起動後，因不平衡所造成的振動，可由振動感測器來加以監測。
3. 淨油機經由馬達帶動的皮帶帶動，因為比例的不同，使得淨油機的轉速是馬達轉速的數倍。
4. 為減少淨油機的軸承磨損及振動，立軸上端的軸承裝在彈簧軸承座中。
5. 離心式離合器，可確保淨油機起動和加速過程較平緩，同時又可防止皮帶及馬達過負載。

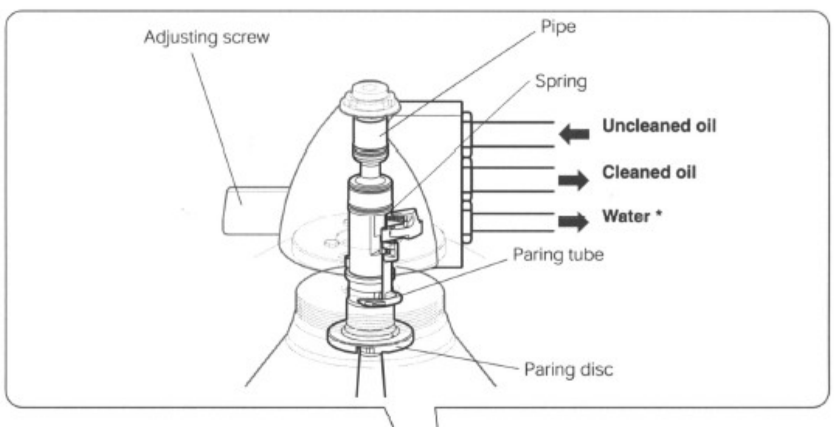


圖 6.15　淨油機刮削環(Paring disk)和刮削管的構造

圖 6.15　淨油機刮削環(Paring disk)和刮削管的構造說明如下：

1.刮削環(Paring disk)和刮削管(Paring tube)位於淨油機的頂部，將乾淨的油及水打出淨油機外。

2.刮削管(Paring tube)可以徑向移動，它靠彈簧來平衡，如上圖所示；在某些操作

狀況，刮削管的徑向位置可被兩調整螺絲給固定住。

3.高度調整環(Height adjusting rings)可調整刮削環(Paring disk)和刮削管(Paring tube)對缽(Bowl)的相對高度。

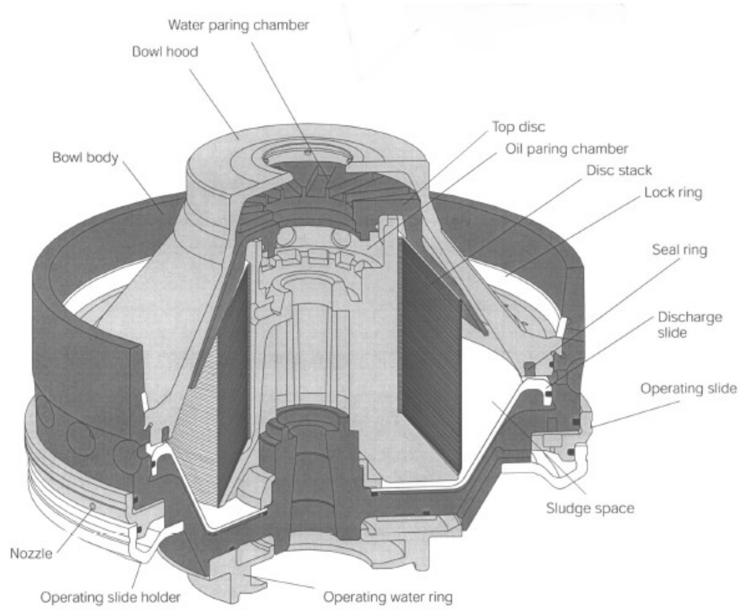


圖 6.16 淨油機缽蓋(Bowl hood) 和頂盤構造圖

4.圖 6.16 淨油機缽蓋(Bowl hood) 和頂盤(Top disk)之間，形成水的空間，由刮削環(Paring disk)和刮削管(Paring tube)將水打出淨油機外。

5.操作水環(Operating water ring)控制操作水的進入，操作水再來控制操作滑閥(Operating slide)和排渣滑閥(Sludge slide)的升降，如此就可控制淨油機的排渣(Sludge discharge)。可由下方圖 6.17 淨油機操作水作用情形看出。

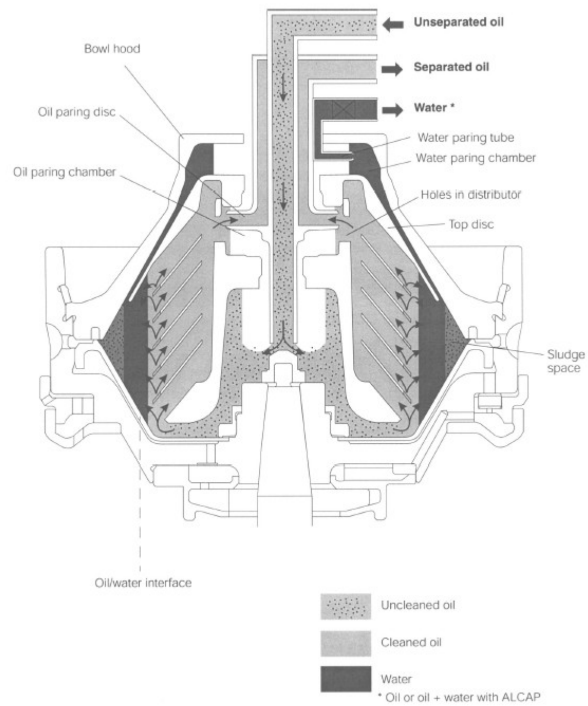


圖 6.17 淨油機操作水作用圖

2.10.3 淨油機的最佳化

由分離效果來說，較小的的缽及較快轉速的淨油機，效果是一樣的，較小的的缽容易製造及成本較低廉，減少維修次數，增加軸承運轉時間及軸承負荷愈小愈好。

2.10.3 淨油機的燃油系統

淨油機在使用時，有一台單機操作，兩台雙機操作之情形。雙機操作情形，又可分兩台淨油機為並聯或串聯狀態。並聯使用時，多考慮要加大淨油量之情形；現今燃油品質低下之情形，就採雙機操作模式，可一台為淨油機(Purifier)而一台為潔油機(Clarifier)，或者兩台都為淨油機的情形。

1. 單機操作:

下圖 6.18 乃一台淨油機使用時之燃油系統圖：[10]

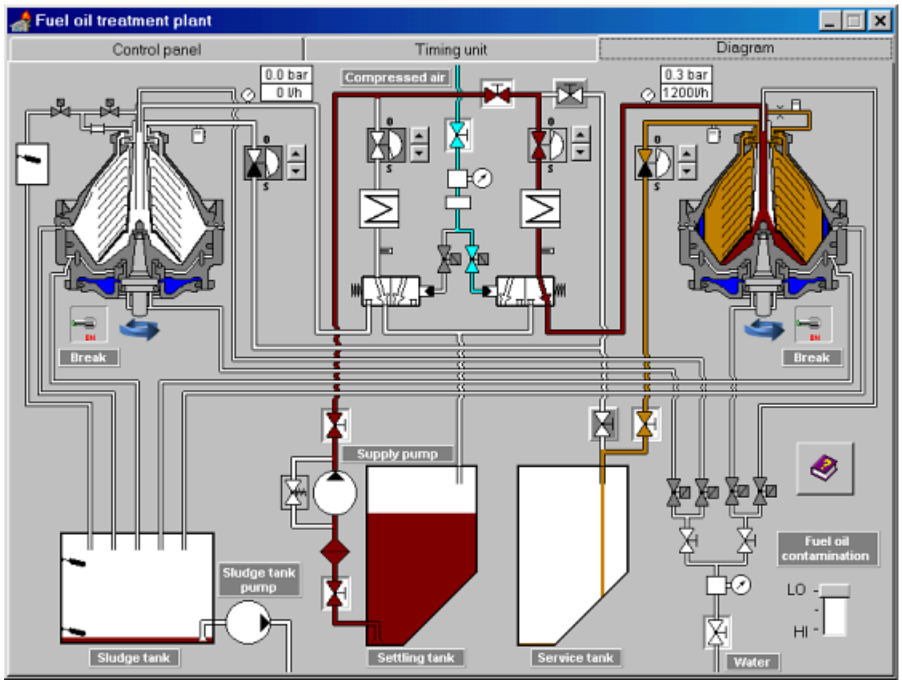


圖 6.18 一台淨油機操作情形

2. 兩台淨油機串聯使用[10]

下圖 6.19 乃兩台淨油機串聯使用時之燃油系統圖，可改善燃油淨油之品質：

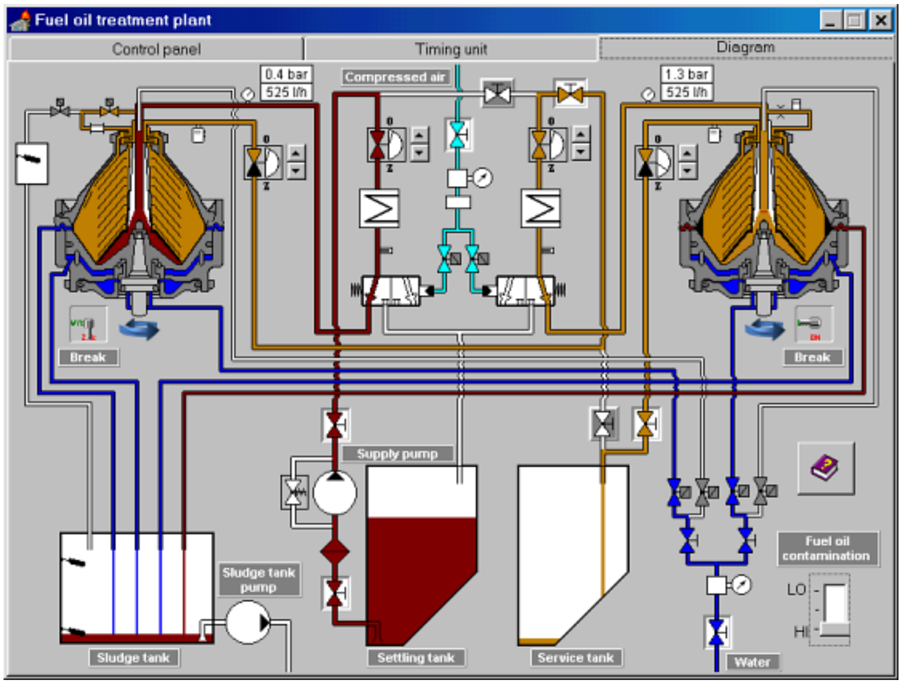


圖 6.19 兩台淨油機串聯使用

2.10.4 淨油機的排渣構造[10]

淨油機將燃油分離後之殘渣排出淨油機外，如此可延長淨油機的淨油時間，淨油機基本殘渣排放機構如下圖6.20淨油機的排渣構造

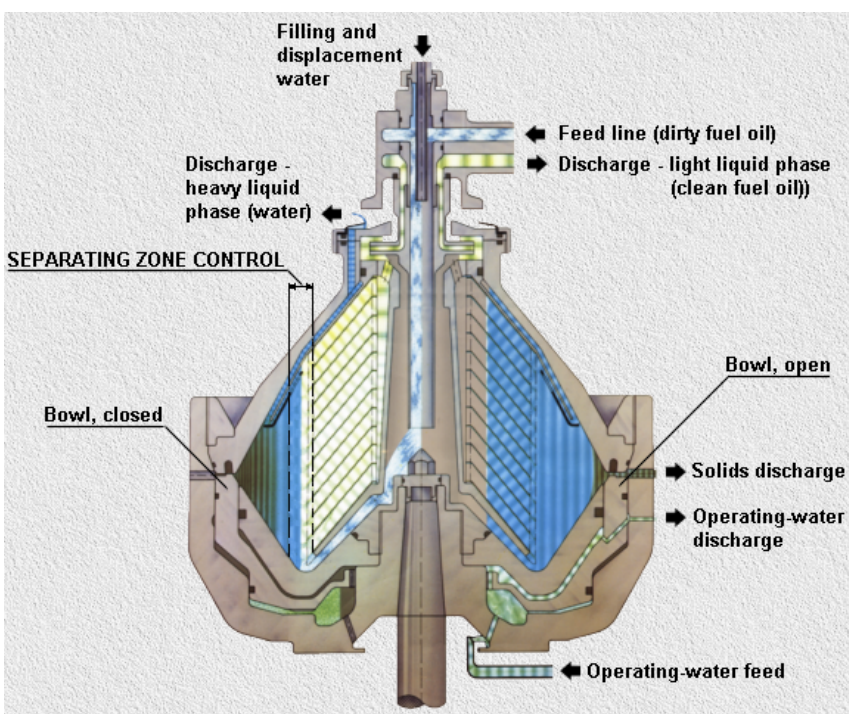


圖 6.20 淨油機的排渣構造

1. 操作水將缽封起，淨油機淨油時之情形：[10]

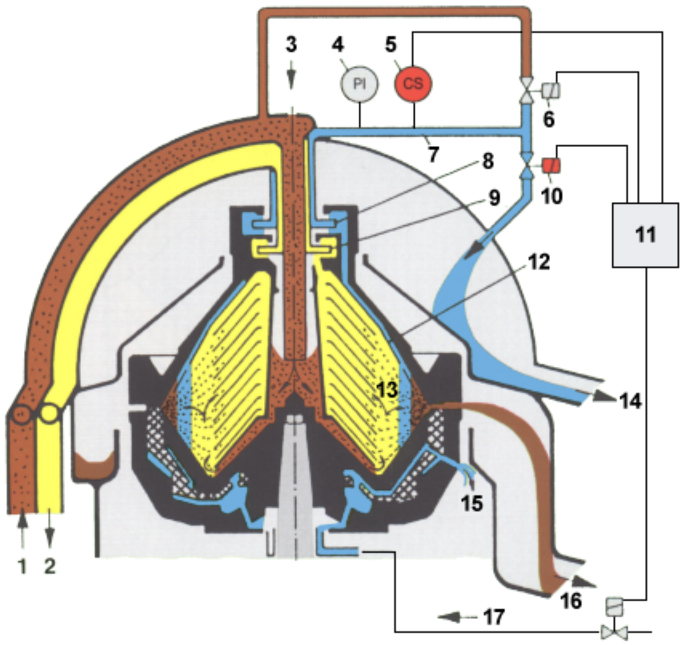


圖 6.21 淨油機淨油情形

2.10.5 淨油機新型排渣機構如下[1]

下圖 6.22 乃新型排渣機構，排渣滑閥(Discharge slide)，固定於較小半徑處，其餘部分可彈性的活動。如此設計可減少滑動缽(Sliding bowl)底部的問題。

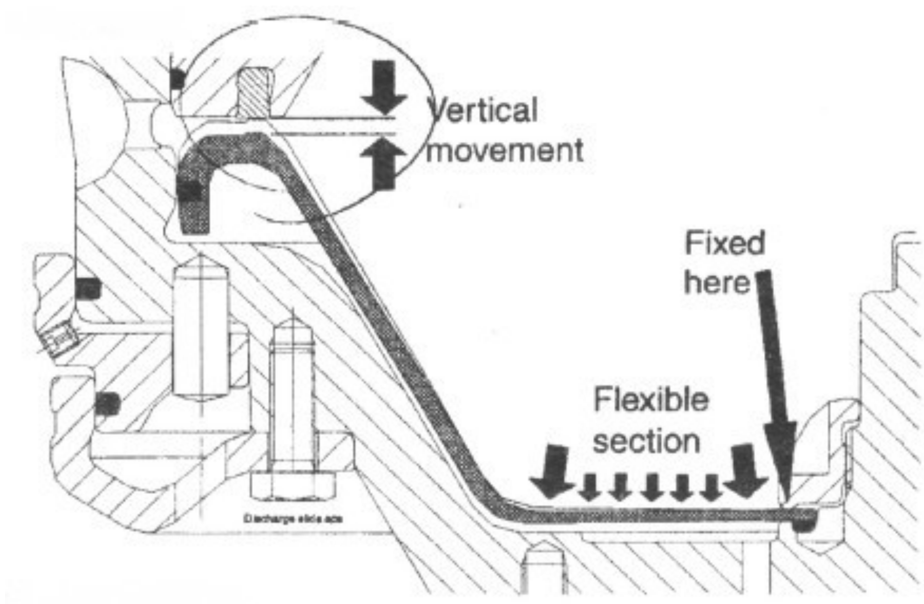


圖 6.22 新型排渣機構

2.10.6 離心式潔油機(Decanter centrifugal)[2]

船用燃油品質愈來愈差，為了加強潔油效果，除了淨油機外，又有如下圖 6.23 之離心式潔油機，圖 6.24 為內部作動分解圖。

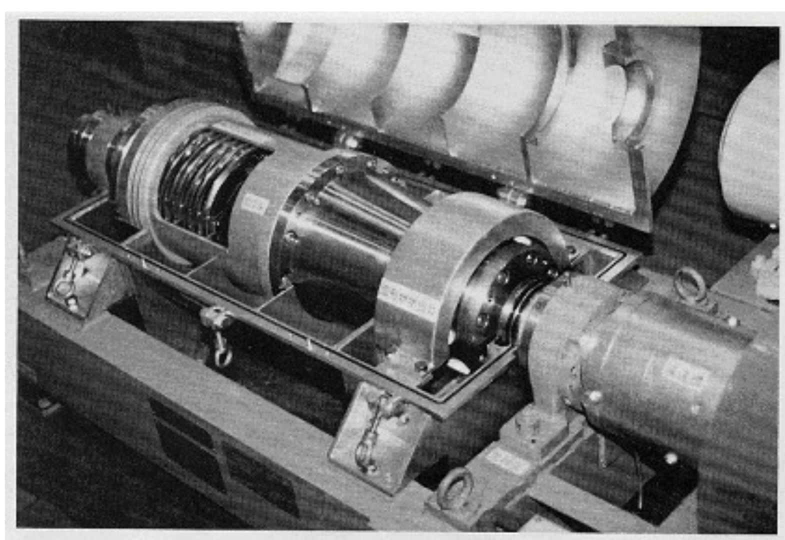
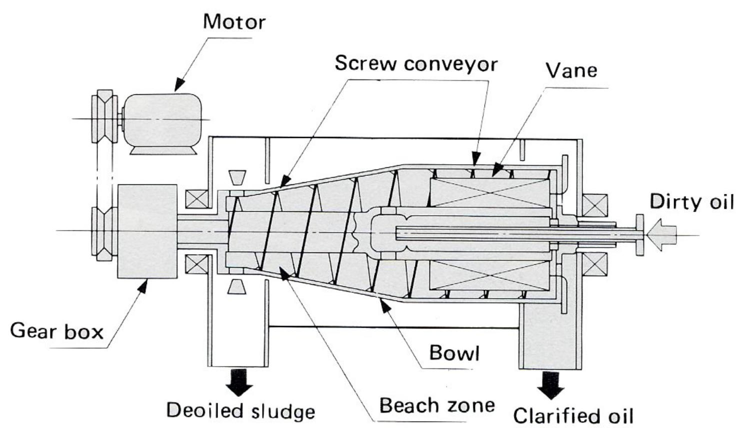
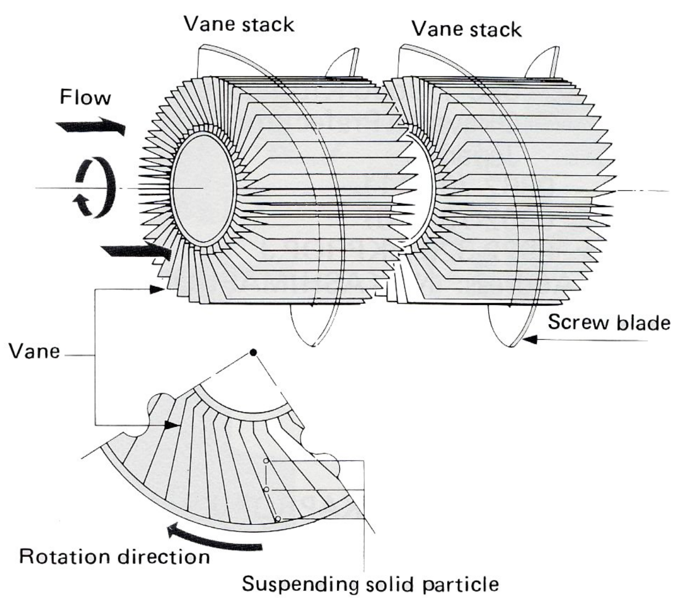


圖 6.23 離心式潔油機





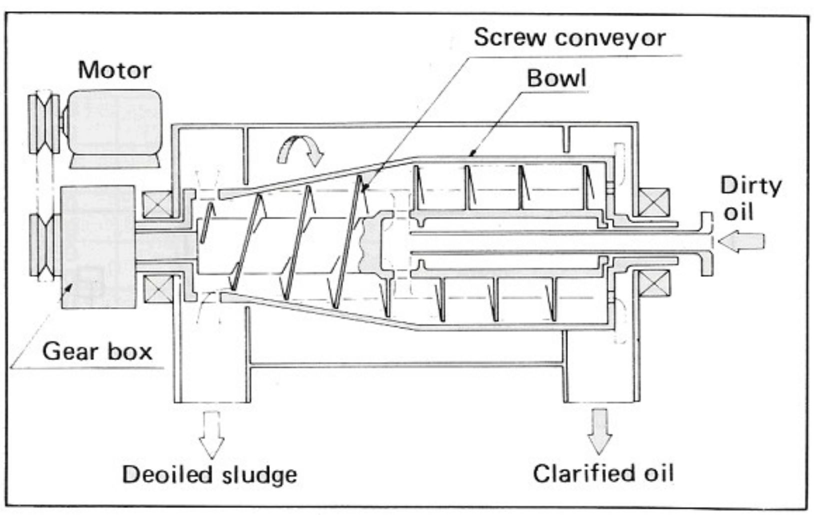


圖 6.24 離心式潔油機內部結構

2.11 油水分離器

2.11.1 油水的基本原理

油水分離器，基本是利用油及水不能混合的特性，目前船用油水分離器，所應用的原理，可分：氣泡分離法、加熱分離法、超音波分離法、過濾分離法、比重差分法、電氣分離法及生物氧化法等。因考慮以不消耗能源為主，故船上之油水分離器多以比重差分法為主。[3,6,7]

2.11.2 起動油水分離器前

圖 6.25 乃污水櫃(Bilge water tank)的污水漸多，準備起動油水分離器，起動油水分離器前，先將油水分離器加入海水，上部探測器被海水包圍。[10]

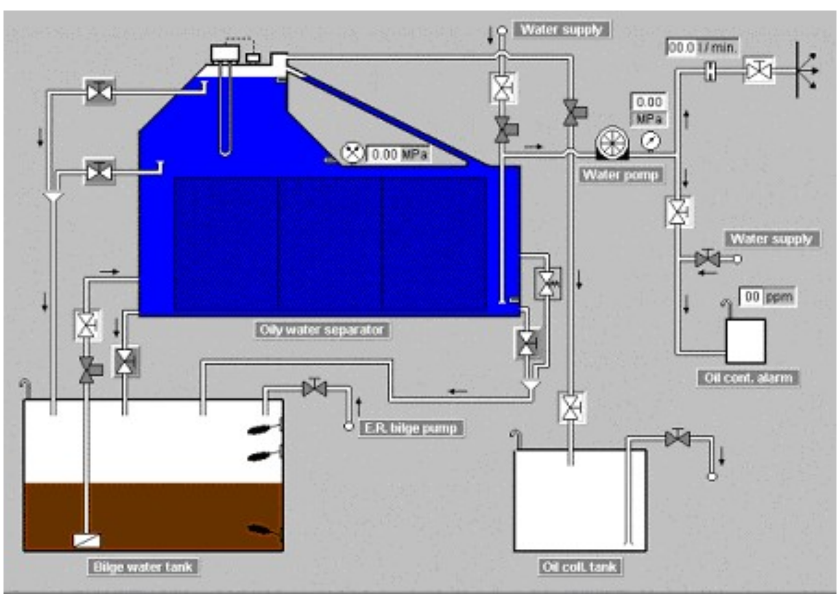
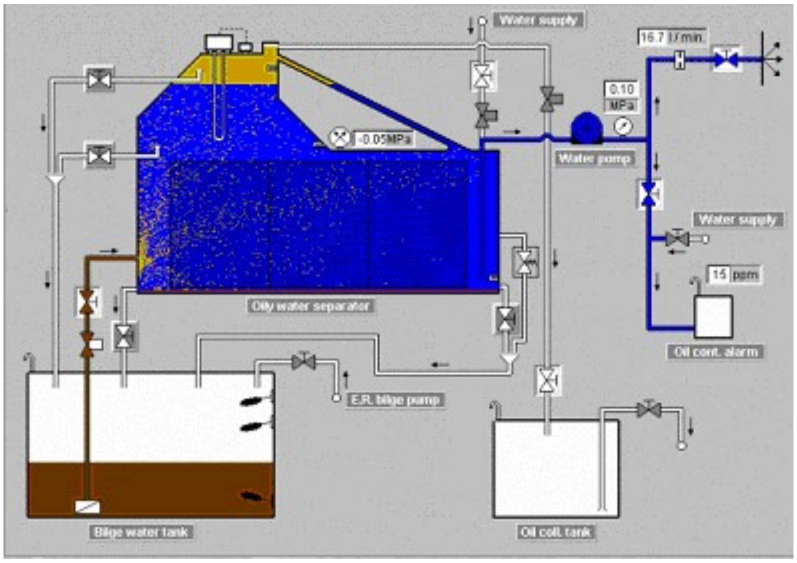


圖 6.25 油水分離器注入海水

2.11.3 油水分離器開始分離油及水

圖 6.26 油水分離器已分離出污油和污水



2.11.4 油水分離器排出污油

圖 6.27 分離器下部應測器已感測到污油，電磁閥開啟，將污油送入污油櫃中(Oil coll. Tank)

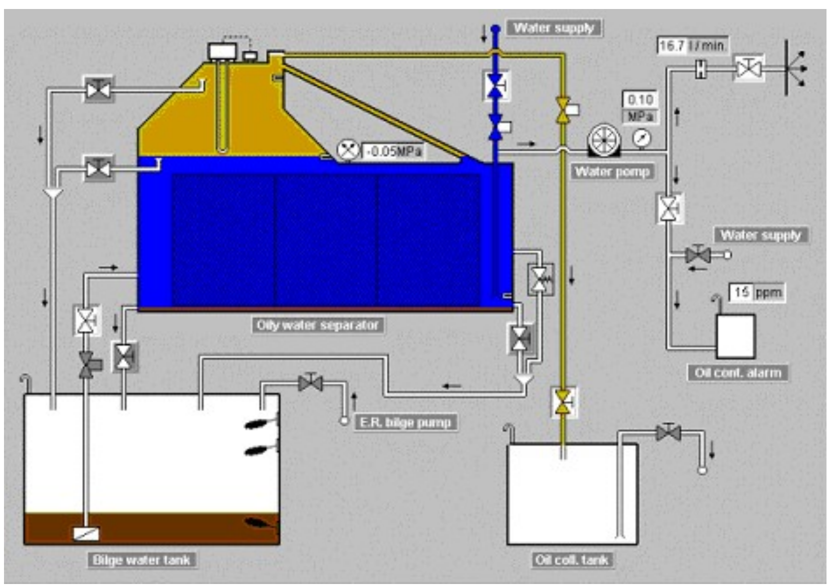


圖 6.27 油水分離器排出污油

2.11.5 油水分離器結束排出污油

圖 6.28 當海水水位上昇，上部探測器被海水包圍，停止污油泵。

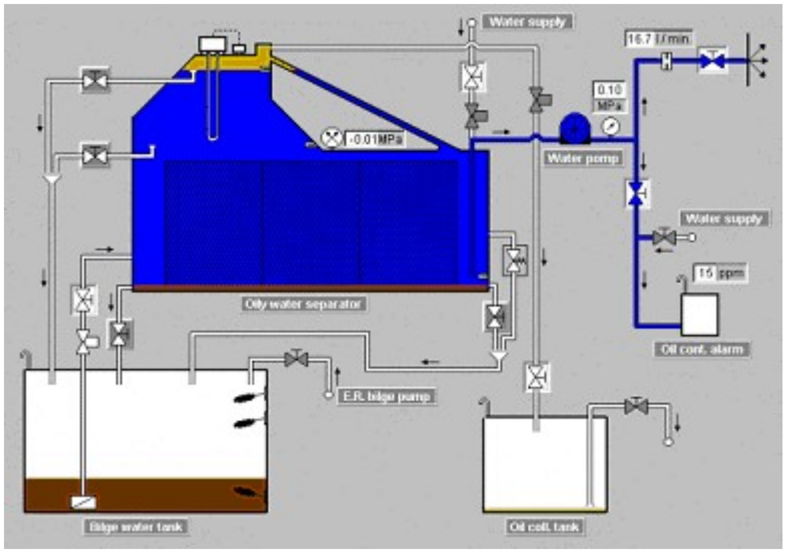


圖 6.28 油水分離器結束排出污油

2.12 污物處理器(Sewage treatment)

2.12.1 污物處理器的基本原理[3,5,8,9]

污物處理器的基本原理，是將空氣打入處理櫃中，利用空氣中微生物，將污物分解，要排出去的流體，再經殺菌室處理，且需符合下列規定排洩物的大腸桿菌標準

1.懸浮固體標準

2.五日生化氧氣需要標準

2.12.2 污物處理器的系統

1.圖 6.29 污物處理器起動前，先將各處理櫃加水[10]

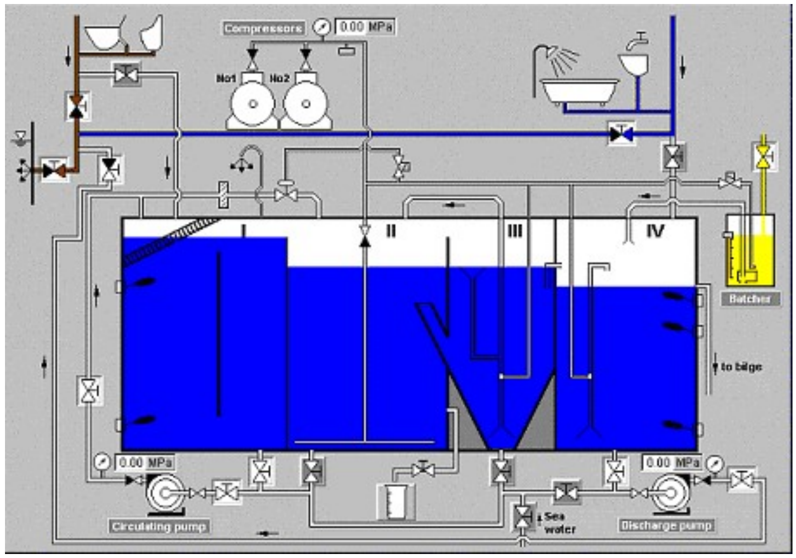


圖 6.29 污物處理器，先將各處理櫃加水

2.圖 6.30 自動模式下，初級處理器中的污水循環中，也將污物打入第二級分離器

中，此時空氣打入第二級處理器中，加速污物分解。

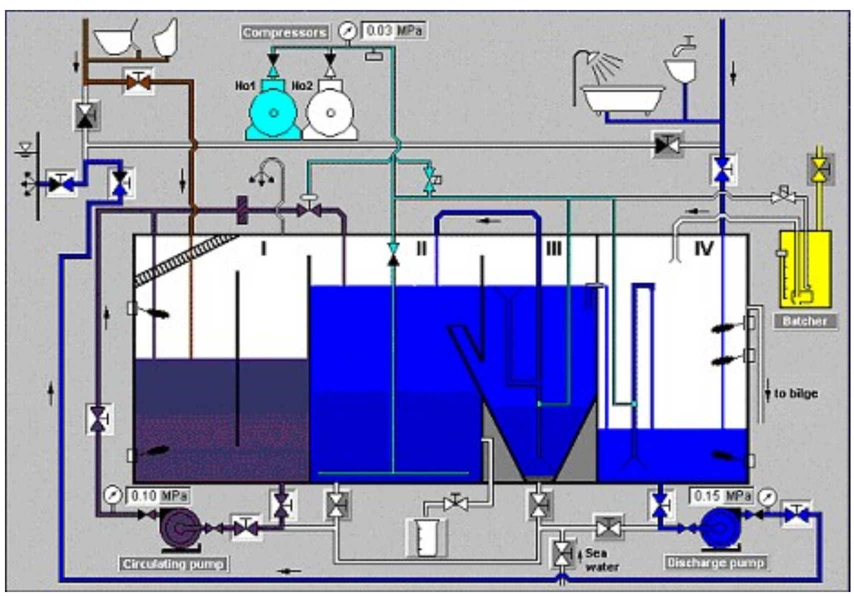


圖 6.30 自動模式下

3. 圖 6.31 經過分解後之污物，進入殺菌室

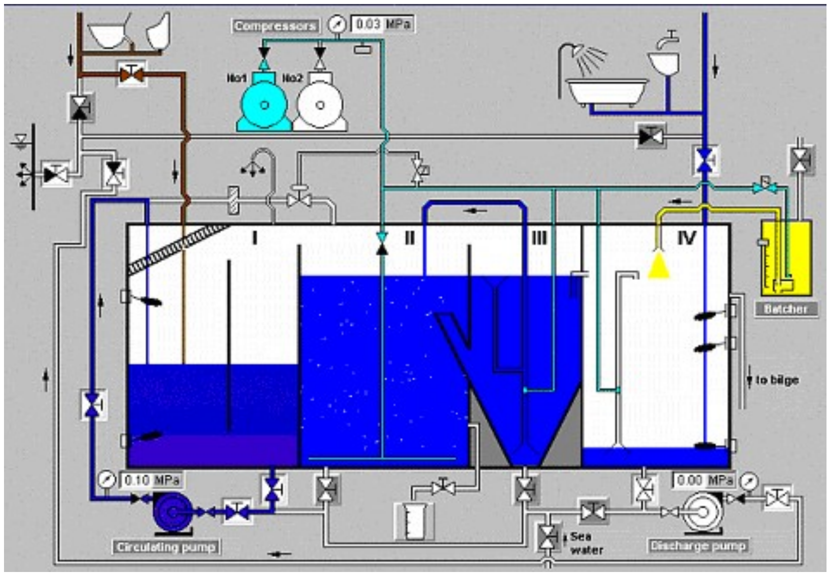


圖 6.31 經過分解後之污物，進入殺菌室

2.13 舵機(Steer Gear)

2.13.1 舵機基本原理

舵機之種類繁多，常用者多為電動油壓舵機(Electro-hydraulic steering)，現又以一柱兩缸或兩柱四缸為主。[10]

電動油壓舵機有兩種，常用者又以雷勃遜滑動型(Rapson-slide type)，及旋轉葉輪式(Rotary vane type)。利用油壓來推動油壓缸或葉輪，使得舵板轉動，船舶就可完成轉向動作。其中以雷勃遜滑動型的轉矩大，轉角大為其優點。大型船舶以兩柱四缸為主，圖 6.32 為兩柱四缸舵機。

圖 6.32 兩柱四缸舵機

2.13.2 舵機油壓系統

1. 兩柱四缸之油壓系統，如圖 6.33 所示[10]

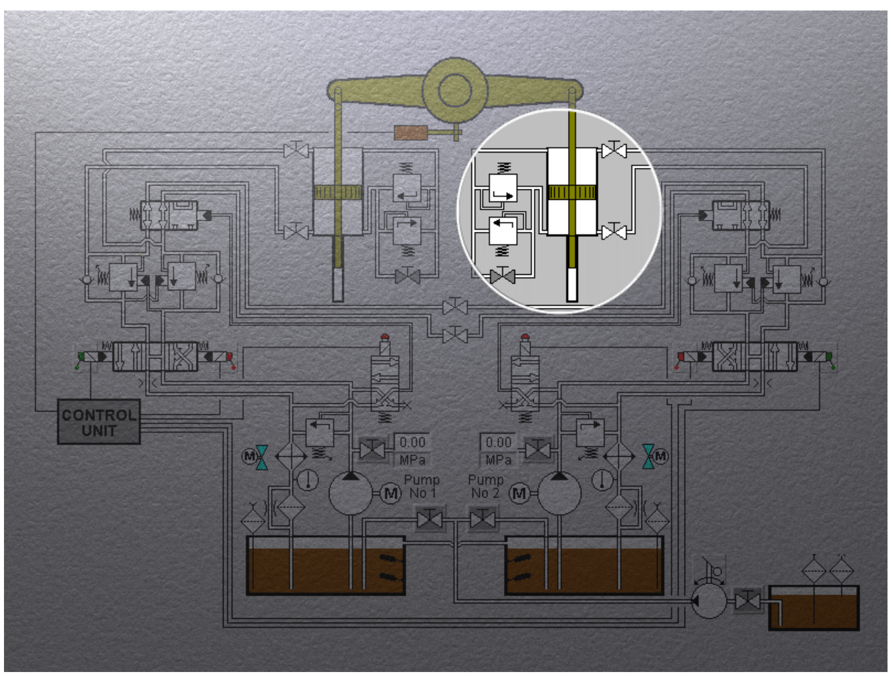


圖 6.33 兩柱四缸之油壓系統

2. 一號油壓泵起動運轉中，供油至油壓缸，使舵右轉 20°，如圖 6.34 所示

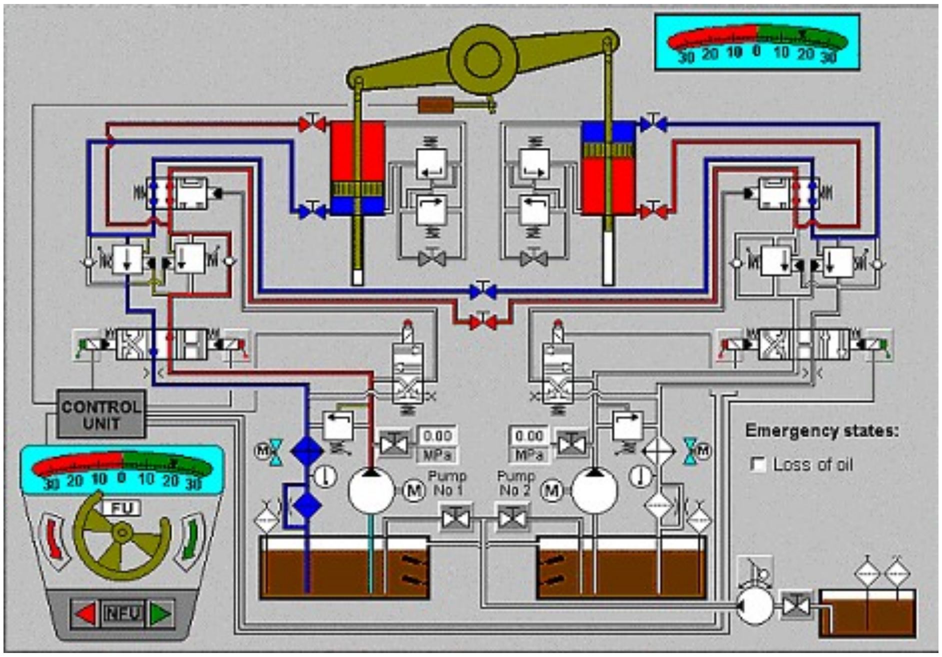


圖 6.34 舵右轉 20°

3. 進出港或於河道中，二台油壓泵同時起動運轉，供油至油壓缸，使舵右轉，圖 6.35 所示

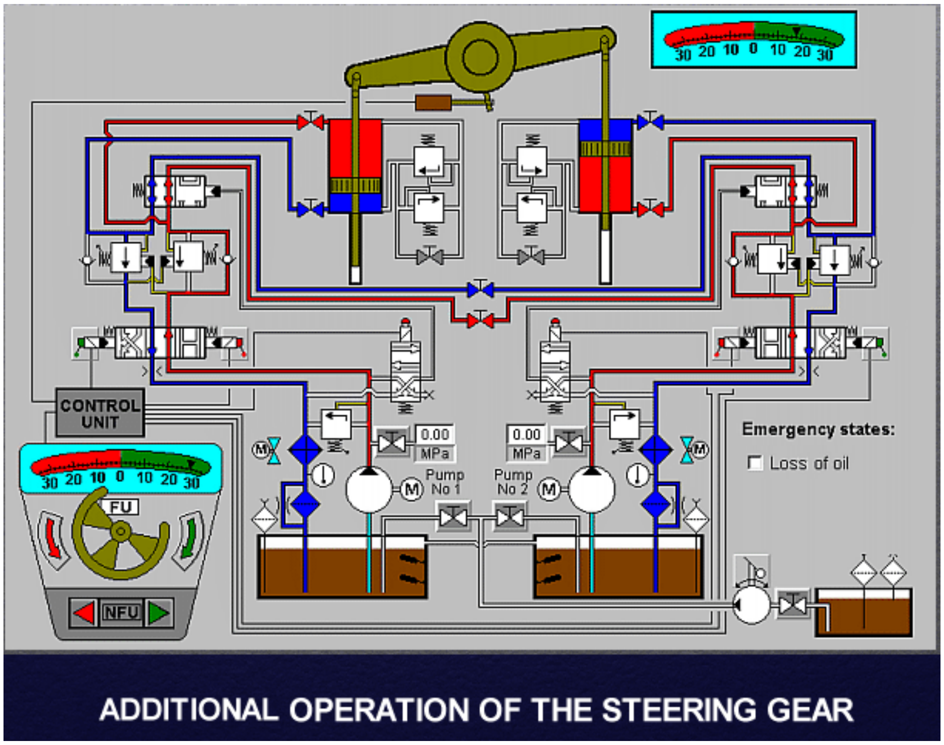


圖 6.35 二台油壓泵作動

4.此乃緊急狀況，一號油壓泵運轉中，但僅供油至左側油壓缸，右側油壓缸互相旁通，此時操舵能力約50%，如圖6.36所示。

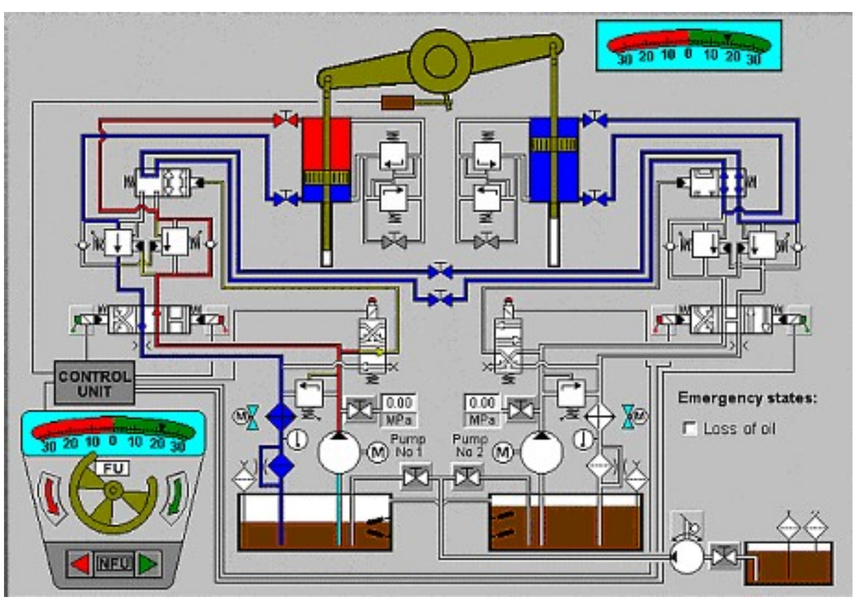


圖 6.36 供油至左側油壓缸，右側油壓缸互相旁通

2.14 船首推進器(Bow Thruster, Side Thruster)

2.14.1 船首推進器之原理

船首推進器，位於船頭吃水線下方，使用電動機來轉動螺旋槳，目地是為了進出港時，增加船舶轉向的靈活度。

1.下圖 6.37，為船首推進器之構造[2]

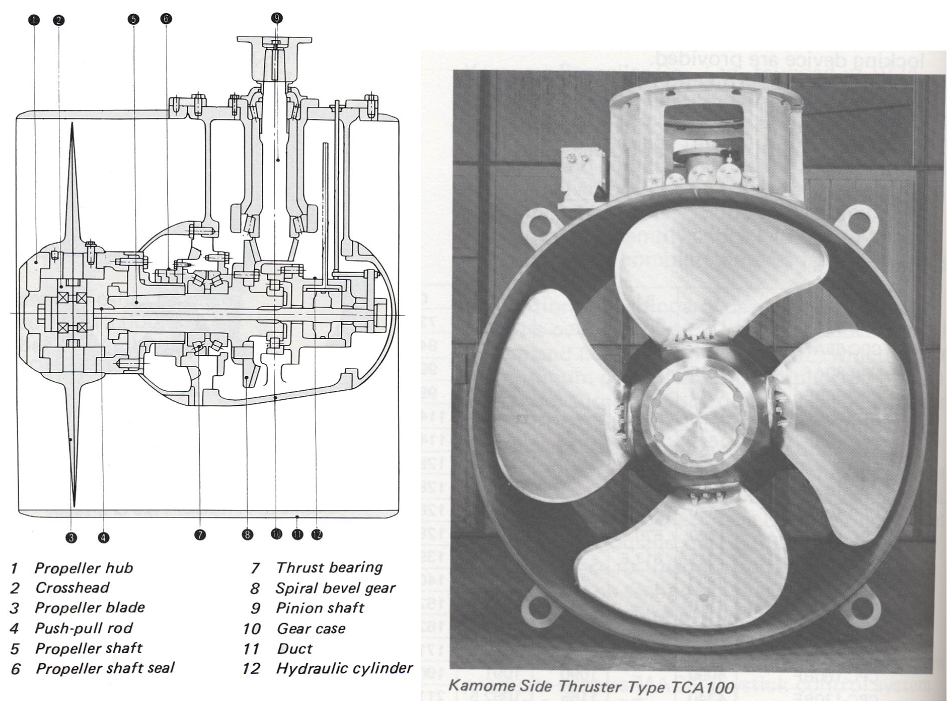


圖 6.37 船首推進器

2. 圖 6.38 乃船側(Side Thruster)推進器，適用於船舶吃水較淺時

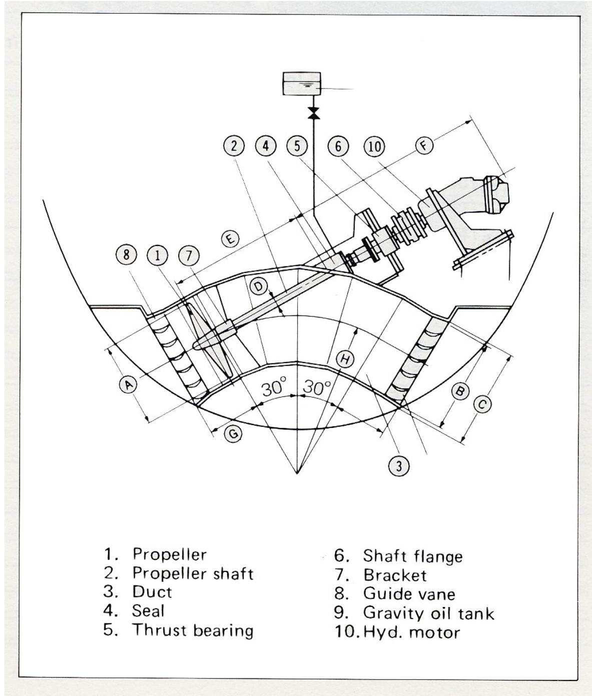


圖 6.38 船側推進器

2.15 振動平衡器(Vibration balancer)

2.15.1 振動平衡器之原理

為減輕船舶的振動，可於船尾加裝振動平衡器，振動平衡器所產生之振幅和船舶振動振幅相反，故可將船舶振動減至最低。

1. 圖 6.39 振動平衡器[2]

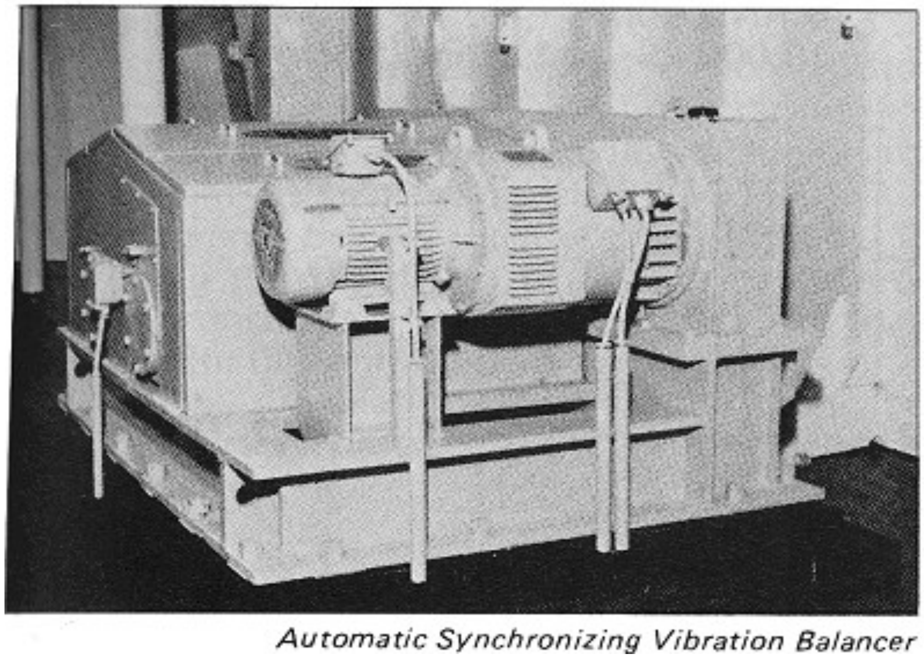


圖 6.39 振動平衡器

2. 圖 6.40，振動平衡器所產生之振幅之情形

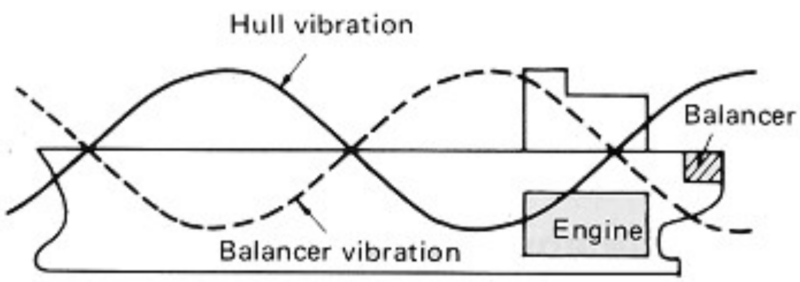


圖 6.40 振動平衡器所產生之振幅

3. 圖 6.41 顯示，15500DWT 貨櫃加裝振動平衡器後，加速度減少之情形

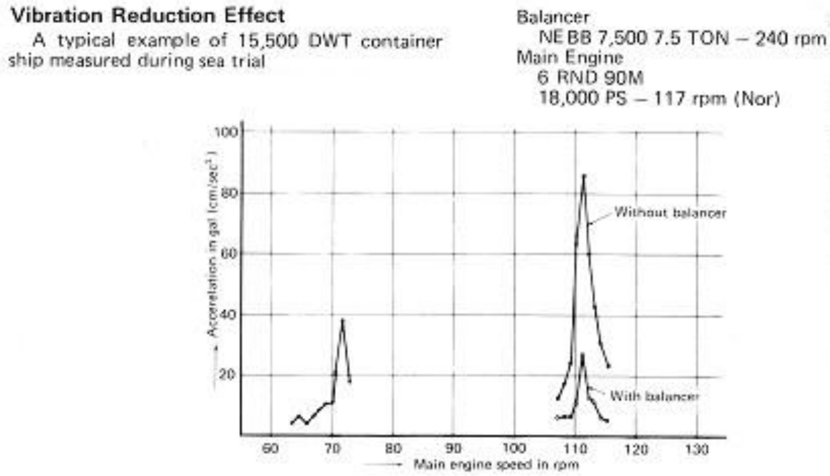


圖 6.41 加裝振動平衡器前後，加速度之比較

Content

- 2.10.3 淨油機的燃油系統
- 2.10.4 淨油機的排渣構造
- 2.10.5 淨油機新型排渣機構
- 2.10.6 離心式潔油機(Decanter centrifugal)

2.11 油水分離器

- 2.11.1 油水的基本原理
- 2.11.2 起動油水分離器前
- 2.11.4 油水分離器排出污油
- 2.11.5 油水分離器結束排出污油

2.12 污物處理器(Sewage treatment)

- 2.12.1 污物處理器的基本原理
- 2.12.2 污物處理器的系統

2.13 舵機(Steer Gear)

- 2.13.1 舵機基本原理
- 2.13.2 舵機油壓系統

2.14 船首推進器(Bow Thruster, Side Thruster)

- 2.14.1 船首推進器之原理

2.15 振動平衡器(Vibration balancer)

- 2.15.1 振動平衡器之原理

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