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The development of the merchant marine propulsion plant before and after the oil crisis in 1973

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THE DEVELOPMENT OF THE MERCHANT MARINE PROPULSION PLANT BEFORE AND AFTER THE OIL CRISIS IN 1973

By

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DECLARATION

I certify that all the material in this dissertation that is not my own work has been identified, and that no material is included for which a degree has previously been conferred on me.

The contents of this dissertation reflect my own personal views, and are not necessarily endorsed by the University.

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ABSTRACT

Title of Dissertation: **The Development of The Merchant Marine Propulsion Plant Before and After The Oil Crisis In 1973**

Degree: **MSc**

The dissertation is a study on the events leading to the oil crisis in 1973 and its consequences. This momentous event in history is examined, taking its repercussion on global energy need, especially the need of the shipping industry into closer scrutiny. Its tremendous impact on the development of the merchant marine propulsion plant during the aftermath of the crisis is specifically analysed.

Prior to the oil crisis, maritime transportation were mainly centred on the fast and high power marine propulsion plant. Steam turbine, diesel engine and gas turbine were each competing for the supremacy of this lucrative shipping market. A brief history on the development of these propulsion plants and the outcome from the competition is evaluated.

The post oil crisis years has seen the drastic decline of high power ships. The emphasis on shipping was shifted to one on fuel economy instead. This was due to the soaring prices of bunker fuel. The innovative evolution of diesel engine propulsion plant especially, is closely examined in view of the dramatic achievements in design and performance standards accomplished within this era.

The environmental impact from the exhaust gas emission of this highly efficient modern low speed diesel engine is examined. The methods to curb this emission is also investigated. The author's views are drawn in the concluding chapter.

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LIST OF ABBREVIATIONS

BP	British Petroleum
BMEP	Brake Mean Effective Pressure
BHP	Brake Horse Power
BS	British Standards
CO_2	Carbon Dioxide
CO	Carbon Monoxide
DWT	Deadweight Tonne
EGR	Exhaust Gas Recirculation
GE	General Electric
HC	Hydrocarbon
H_2O	Water
ISO	International Organisation For Standardisation
IFO	Intermediate Fuel Oil
IHI	Ishikawajima Harima Heavy Industry
kW	Kilowatt
MDO	Marine Diesel Oil
MCR	Maximum Continuous Rating
MEP	Mean Effective Pressure
MTBF	Mean Time Between Failure
MTTR	Mean Time To Repair
MTBO	Mean Time Between Overhaul
N_2	Nitrogen
NO_x	Nitrogen oxide
OPEC	Organisation Of Petroleum Exporting Countries
O_2	Oxygen
PTO	Power Take Off
PTI	Power Take in

PPM	Part Per Million
Pmax	Maximum Pressure
Pcomp	Compression Pressure
SHP	Shaft Horse Power
SO_x	Sulphur Oxide
SIPWA-TP	Sulzer Integrated Piston Ring Wear Detecting Arrangement-Trend Processing
SCR	Selective Catalytic Reduction System
TCS	Turbo Compound System
ULCC	Ultra Large Crude Carrier
UMS	Unmanned Machinery Space
VLCC	Very Large Crude Carrier
VIT	Variable Injection Timing
VEC	Variable Exhaust Closing

CHAPTER 1

Introduction

Ever since the first ship was put to sea, it was propelled by certain devices. There were manual, wind-assisted or machine-driven types of propulsion plant. Simply, there exist a variety of different kinds of ship with varying types of propulsion plant installed. Merchant marine propulsion plant would simply mean machine-driven propulsion plant for merchant ships. These ships are generally plying the ocean of the world carrying their load of cargo to a defined destination. Hence, it is only sensible that they are propelled by a machine such as an engine of steam, diesel or even gas turbine. The demand of world trade has indeed contributed to many innovative designs in the development of the merchant marine propulsion plant such as the high power steam turbines and the diesel engines.

The aftermath of the oil crisis in 1973 has seen a tremendous upsurge in the spot prices of crude oil. In a period of less than a decade, overall oil prices were increased by a total of about 1,700%. The immense impact of the organisation of petroleum exporting countries (OPEC) in determining the oil production capacity for the world market, has since created an economic revolution world-wide. Energy conservation was seen as the main drive of most nations then.

The shipping industry was believed to be the worst hit by this energy crisis. Many of the world large oil tankers fleet were put out of service, replaced and new orders cancelled due to the upsurge of bunker fuel prices. This tragic event thus prompted the engine builders to carry out an intense search for a fuel efficient marine propulsion plant. This search was realised with the introduction of the long stroke and superlong stroke marine 2-stroke low speed diesel engine in the early 1980s. This modern marine diesel engine has indeed undergone many improvements and modifications compared to its predecessor. Its unique features are high stroke to bore ratio, high cylinder combustion pressure, high output power per cylinder, low specific fuel oil consumption, low engine revolution and the ability to digest poor quality residual fuel. The first oil crisis in 1973 was seen as the main catalyst in the further development of modern marine propulsion plant especially the low speed diesel engine.

Prior to the oil crisis in 1973, the development of marine propulsion plant was geared towards high output power per cylinder. This was the era of the high power steam turbine propulsion plant. The post oil crisis saw the emphasis shifted to fuel economy as the main theme. This lasted till the early years of 1980s. Reliability and durability of the engine were prominence in the late 1980s and early 1990s. In this later part of 1990s, the issue of environment appears to be the main catchword in the shipping world. A 'green' or 'environmental friendly' diesel engine is the way with the modern marine propulsion plant in addition to the other important features.

The dissertation seek to investigate, analyse and evaluate the numerous milestones in the development of the various marine propulsion plants over the past three decades. In the shipping industry, it is widely accepted that the repercussion from the first oil crisis in 1973 has indeed played a particularly significant role in the development of marine propulsion plants. Hence, the title of the dissertation: **The development of the merchant marine propulsion plant before and after the oil crisis in 1973.**

This dissertation is prepared mainly in a descriptive manner. The information was obtained from text books, conference and seminar papers, periodicals, engine builders catalogues, publications, lectures handouts, various reports and correspondence with engine builders. These references are shown in the bibliography with the authors and the year issued.

The dissertation is a study of those marine propulsion plants that are adversely affected by the oil crisis in 1973. It is the author's view that other propulsion plants such as the diesel electric and dual-fuel propulsion plants bear no relation to the oil crisis in 1973. Thus, these propulsion plants are not discussed here.

The dissertation is broken down into six chapters with their outlines as follows;

Chapter 1

This chapter captures the aim of the dissertation with certain background leading to the defined topic of the dissertation. The methods used in the research and a brief overview of the entire paper are also deliberated.

Chapter 2

The chapter commences with a background information leading to the oil crisis in 1973. The major oil exporting countries and their oil reserves are shown in a chart with particular emphasis on the oil cartel of OPEC. The events leading to the control of oil production by OPEC and the subsequent oil crisis in 1973 are also discussed. The later part of the chapter touches on the impact of the oil crisis on international oil prices, world economy, quality of fuel oil and the shipping industry.

Chapter 3

This chapter includes a review on the history of the various marine propulsion plants, their design concept, performance characteristics and their operation and maintenance features. It provides an insight into the development and competition between the various propulsion plants of steam turbine, diesel engine and gas turbine in their search for supremacy as the sole prime mover for the merchant marine propulsion plant. The thermal efficiency, reliability and durability of the plants are elaborated to render a comparison on the performance characteristics of the various propulsion plants. The operation and maintenance features of the plant - an economy factor that was not seriously considered prior to the oil crisis years, are also discussed in the concluding section.

Chapter 4

This chapter deals with the various impacts which resulted from the aftermath of the oil crisis in 1973. It commences with the reasons leading to the downfall of the high power steam turbine especially popular among the large tankers and container ships. The ongoing development of low speed diesel engines is particularly given a wide coverage in view of the many modifications and improvements accomplished in the engine structure, transmission components and various other fittings. A section on the propulsion plant energy recovery system is also discussed. The great leap in thermal efficiency, reliability and durability of the engine is forwarded together with its heat balance flow diagram. The chapter concludes with a view on the operation and maintenance features of a modern engine room with emphasis on the main propulsion plant. The concept of reliability, durability, simplicity and economy in the operation and maintenance are the emphasis of a modern marine propulsion plant in this post oil crisis era.

Chapter 5

This chapter deals with the impact of modern low speed diesel engine on the shipping industry and the environment. It points out the new concept of a 'layout field' for engine selection. This is one of the unique features made available for potential shipowners in their selection of an optimum marine propulsion plant. In this modern era, engine builders are also more inclined to tune to the need of shipowners and thus, the concept of a standard ship design is always in jeopardy. The chapter concludes with a section on the problems of environmental pollution created by exhaust gas emission from modern low speed diesel engine. A discussion on the various methods to curb this pollution is also forwarded.

Chapter 6

This forms the conclusion of the dissertation. It provides a summary of the entire dissertation content and the views of the author in relevant areas.

CHAPTER 2

The Oil Crisis: Its Impact on Fuel Quality, Global Shipping Market and the Engine Builders

2.1 Background to the Oil Crisis

2.1.1 World Oil Exporting and Importing Countries

Before World War II, the single most prominent oil exporting country was the United States Of America (USA), followed closely by the Former Soviet Union (U.S.S.R), and Mexico. This trend was reversed after the war. Instead, these countries have very limited reserve oil for export. The Middle East, Venezuela, north and west African countries have grown in prominence in terms of their oil reserve and dominance of the world's energy requirements.

Till today, the Middle East countries especially, are most impressive in their crude oil reserve. These countries comprise fifteen States and form a unique geographical concentration on a map (Odell, 1975, pp 65). They produce oil in large quantities not only for their own consumption but also for export both in crude and refined categories. The oil production is estimated to exceed four times or more than their own domestic consumption (Odell, 1975, pp 65). According to 'Exxon Background Series of August 1976', the Middle East region oil reserve stood at 396 billion barrels out of a world-wide total of 659 billion barrels with a production capacity of nearly 50% of the world total. In another survey by 'BP statistical review of world energy, June 1995', the region oil reserve has risen to 660.3 billion barrels out of a total world oil reserve of 1009.3 billion barrels at end of 1994 - a share of about 60% of the world total oil reserve as shown in Figure 2.1.

This clearly indicates the important role played by these Middle Eastern countries in determining the quantum of world energy supply. Any oil press headline could always be linked to some political problem in the region or a conflict of interest between the oil companies and the host governments.

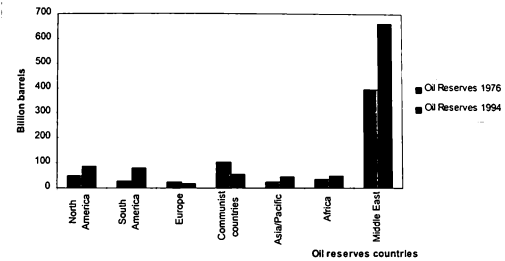


Figure 2.1 World distribution of oil reserves in Billion Barrels

"Source: Exxon background series, August, 1976, pp 3 and BP statistical review of world energy, December, 1994, pp 2".

The period after World War II has seen many countries embarking on a rapid stage of development in rebuilding their shattered economies. Oil has replaced coal as the major source of energy for development. Western Europe, Japan, Latin America, South East Asia and Africa were the major oil importing countries then. The rush for development, centred around industrialisation, has greatly increased the rate of energy consumption as shown in Figure 2.2. This is a typical trend in countries such as Malaysia and Brazil. In both these countries, the annual rate of rise of energy consumption for the past two decades was in the range of 12% (Odell, 1975, pp 145).

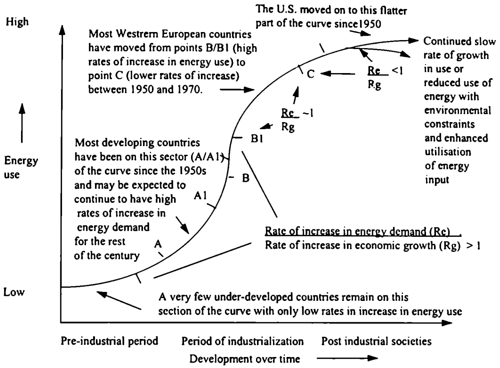


Figure 2.2 The relationship between energy use and economic development over time

The diagram demonstrates how rates of increase in energy use remain higher than the growth rates in the industrialised world and as developing countries depend mainly on oil for their energy this means that their importance as participants in the International Oil industry will steadily increase.

"Source: Odell, 1975, pp 143".

A 1994 figure of world crude imports and exports countries is shown in Table 2.1. It indicates that countries with the world's highest crude oil import are mainly dominated by the industrialised countries of the USA, Western Europe and Japan. They shared a total of 71.7% of the world total crude import. The developing countries of Asia, especially China, South Korea and South East Asian countries also showed a remarkably high energy consumption of 15.6% of world total crude import. This is due to their quest for industrial development. The crude oil import in developing countries of North and West Africa and The Middle East are merely a total of 1%. This is due to their high oil reserves and slow national growth rate. However, they made up a total of 69% of the world crude oil export - the world's highest crude oil exporting countries.

Table 2.1 Crude Imports and exports 1994

Country	Crude imports (%)	Crude exports (%)
USA	25.4	0.4
Canada	2.2	3.7
Mexico	-	4.8
South & Central America	4.4	6.1
Middle East	0.4	52.2
North Africa	0.5	7.3
West Africa	0.1	9.5
East & Southern Africa	1.4	-
South Asia	2.6	0.1
Other Asia	12.1	3.9
Japan	16.9	-
Australasia	1.4	0.5
Western Europe	29.4	3.9
Eastern Europe	1.9	5.2
China	0.9	1.4
Unidentified*	0.4	1.0
Total World	100.0	100.0

* Includes changes in the quantity of oil in transit, movement not otherwise shown, unidentified military use etc.

Notes: Bunkers are not included as exports.

Intra-area movements (for example, between countries in Western Europe) are excluded.

"Source: BP statistical review of world energy, June 1995, pp 17".

2.1.2 The Oil Crisis of 1973

After World War II, the source of energy for development switched from coal to petroleum energy consumption. This was due to the destruction of many coal mines in Europe. It was not economical to revive the coal industry then. Moreover, petroleum was the cheapest fuel available.

Many oil companies decided to embark their investment in the oil rich regions of the Middle East. The atmosphere was right for them since the regions needed revenues for development. These oil companies had a free hand in the operation of the petroleum industry. There were a few meetings with the host governments regarding the day to day running of the industry. Petroleum posted or listed prices changed depending on the market forces of demand and supply.

In most cases the host government only received the revenues based on the oil posted prices decided by the oil companies. Whenever the governments of these oil producing countries wanted to change the price of oil in order to increase their revenues, they would have to negotiate with the oil companies. The oil companies have a great role in manipulating the international oil market prices that could affect the oil producing countries' revenues.

In the late 1950s, the oil companies decided to reduce the posted price of oil in view of an imminent oil surplus and also to stimulate oil demand. This decision adversely affected the revenues of the host governments.

Thus, in 1960, the oil producing countries decided to form an organisation of petroleum exporting countries (OPEC). This organisation is comprised of Saudi Arabia, Kuwait, Iran, Iraq, United Arab Emirates, Libya, Nigeria, Venezuela, Indonesia, Algeria, Neutral Zone (Saudi Arabia & Kuwait), and Qatar. The aim of the organisation is to protect their own interest in the oil revenues. This clearly prevented the oil companies from manipulating the posted prices of oil at their own free will.

OPEC grew in strength as it represented the voice of the oil producing countries. It succeeded in preventing further reductions in oil posted prices. Consequently, the host government was ensured of a steady revenue from the barrels of oil produced. It also negotiated with the oil companies on the method of calculating profit and obtained a regular increase in revenues of oil per barrel.

However, OPEC still could not regulate and control the output of oil in member countries up till 1970. This unhealthy phenomenon was seen as a major stumbling block that could jeopardise the stability of the oil posted prices then. The main reason was due to the many 'independent' companies in addition to the 'seven majors/sisters' oil companies (Esso/Exxon, Mobil, Chevron, Gulf Oil, Texaco, Royal Dutch/Shell and BP), battling for the international oil market share. The competition was so keen and acute that a system of control of the oil production and market demand at an agreed price level was not viable then.

In December 1970, the oil producing countries finally managed to establish a system of consensus for collective action. This eventually led to the reversal of the downward trend of oil prices. The posted prices of oil went upward and thus the revenues per barrel of oil producing countries increased substantially.

In late 1973, the oil producing countries begun to exercise control over the quantity of oil output. This production control adversely affected the world energy consumption rate, thus, leading to the first oil crisis that resulted in tremendous changes to the world's international oil market system.

2.1.3. Oil Prices and Their Effects on World Economy

According to Marquez, (1984, pp 1), the economy of the world has experienced enormous changes in the last decades. The prices of oil between 1973 and 1974 - the first oil crisis, rose by 400% while in the years between 1979 and 1981 - the second oil crisis, it reached another height of 161% increment. In a period of less than a decade, the oil prices experienced an overall rise greater than 1,700%.

The economy of developed countries experienced a drop in the national growth rate by an average of 5% from 1960 to 1973 and a further reduction to less than 2% for the years 1974 to 1980. The growth rate even dropped to negative around the early years of the 1980s (Marquez, 1984, pp 1).

Developing countries of Non-OPEC members also encountered the same fate in their growth rate. Additionally, they faced an uphill task on the settlement of their increased oil bills and other foreign debts.

Inflation has turned into a world-wide phenomena that seems to stay forever. This incident of changes in the world economy has opened the eyes of many world leaders. They have finally begun to show great interest in the effects and determination of changes with oil prices. The world spot crude prices between 1972 to 1994 are shown in Figure 2.3.

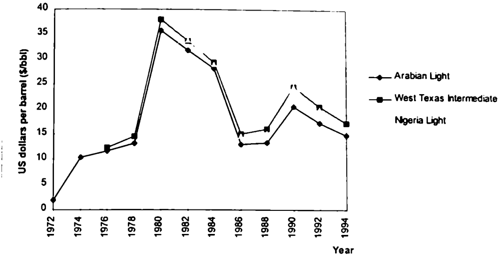


Figure 2.3 Spot crude prices (US dollars per barrel)

"Source : BP statistical review of world energy, June 1995, pp 12".

The major events which changed Middle East crude oil selling prices between 1971 to 1991 were as follows:

- February 1971. Income tax rate raise to 55% and posted price raise 30 cents under Teheran Agreement, signalling new era in which government take role in selling oil prices.
- January 1973. Governments acquired a 25% interest in the producing properties.
- October 1973. OPEC increased the posted price of oil by 70%.
- January 1974. OPEC more than doubled the new posted price. Government acquired a 60% interest in the producing properties.
- October-November, 1974. Tax rate raised to 85%. Royalty raised to 20%.
- October 1, 1975. OPEC increased price by 10%.
- January 1977. OPEC goes to two-tier pricing (Saudi Arabia and United Arab Emirates at $12.09 per barrel ; others, $12.70 per barrel average)
- July 1977. OPEC government selling prices reunified at $12.70 per barrel.

- December 1978. OPEC states intention to raise price by 14.5% in 1979 in quarterly adjustments.
- March 1979. OPEC telescopes total 1979 increase of 14.5% into first quarter, raising government selling price to $14.55 per barrel.
- December 1979. Arabian Light raised to $24 per barrel, retroactive to Nov.1, after breakdown of OPEC price unity and series of individual members' increases.
- May 1980. Saudi Arabia sets price for Arabian Light at $28 per barrel, retroactive to April.
- August-November 1980. Arabian Light raised to $30 per barrel in August and to $32 per barrel in November.
- October 1981. OPEC agreed to raise crude prices with marker crude set at $34.
- March 1982. OPEC agreed for first time to establish production ceiling, fixing total at 17.5 million barrels per day.
- March 1983. OPEC agreed on first reduction in prices, fixing marker crude at $29 per barrel and reducing production ceiling for 1983 from 18.5 million barrel per day (b/d), set in December, to 17.5 million b/d.
- September 1985. OPEC presided over the destruction of official prices and the promise of an unparalleled decrease in revenues.
- January, 1991. 'Desert storm' military operation to liberate Kuwait, and with the concomitant selling of emergency oil stocks in the USA and Europe oil prices fell to pre - crisis level

"Source: Exxon Background Series, December 1984, pp 28. Skeet, I., 1988, pp 208. BP statistical review of world energy, June 1992, pp 1".

2.2 The quality of Fuel Oil and Its Effect on Marine Propulsion Plant Performance

2.2.1 The Origin of Fuel Oil and Its Treatment

The basic form of fuel oil is in the form called crude oil. It is located beneath the earth's surface. The oil is formed by the decomposition of marine organisms. Many remains of these tiny organisms are found at sea. A certain amount come from the land that were carried to the sea by river and also those marine plants at the bottom of the ocean. These remains of the organisms mix and mesh with the sand and silt that settle to the bottom of the sea. They form rich deposits of organic materials which later become source rocks for the production of crude oil.

This process takes millions of years with the development of more abundant life each day (Bram & Dickey, 1971, pp 308). The sediments become thicker and heavier as they sink down into the seabed. As more and more deposits pile up, the one below will experience increased pressure of several thousand times with a temperature rise of several hundred degrees. The sand and mud harden forming shale and sandstone; carbonates in them precipitate and limestone is formed from the hardened skeletal shells. Eventually, crude oil and natural gas is transformed from these remains of the dead organisms.

When crude oil is extracted from the oil field, it is treated. The treatment involves heat and addition of chemical. This process separates the natural gas, removing both water and solids. The fluid is then piped or stored and later transported to the refinery. A distillation unit forms the basic refining tool for treated crude oil. In this unit the oil is heated to various temperatures in order to vaporise the fractions of hydrocarbons into naphtha, gasoline and kerosene. The residue is then further treated and passed into a steam distillation plant. Lubricant and distillate fuel are obtained from the upper regions of the plant while waxes and asphalt can be found at the lower regions.

The post oil crisis of 1973 saw some drastic changes to the oil refinery operation. Residual fuel of poorer quality was introduced because of an attempt to increase yield from the premium product of the crude. A secondary refinery process of 'thermal cracking' and 'catalytic cracking' assumed a wider role. The latter process involved lower temperature and a catalyst. One of the problems associated with these processes is the high percentage of asphaltenes formed that could jeopardise cylinder combustion due to its slow burning quality. The other problem was incompatibility when mixed with other fuels in the tank. Sludge formation and blending problems would arise and this could result in serious consequences to the operation of the marine propulsion plant.

2.2.2. The Composition and Standards of Marine Fuel Oil

Prior to the oil crisis of 1973, marine fuel oil could be categorised into 'Bunker C', residual or heavy fuel oil. The higher grade of marine diesel oil (MDO) was used for ships with medium speed diesel engines. The design of diesel engines was based on a known limit of fuel viscosity. The fuel viscosity was closely related to the combustion characteristics of the engines. 'Heavy fuel' was specified for marine use and its common maximum viscosity's limit was 1,500 seconds Redwood 1 (180 Centistokes) or less. Generally, crude oil ,irrespective of its origin, had minimum effects on the combustion characteristics of its refined products.

The fuel densities were generally below about 985 kg/m^3 @ 15 °C (Newbery, 1996, pp 241). The fuel was also compatible to other fuel bunkered and the treatment on board was simple with densities of fuel mostly below 1.0 kg/m^3. Higher densities fuel of near to 1.0 kg/m^3 or slightly higher were restricted for steam powered plants. The quality of fuel did not seem to be a particular critical issue then.

According to John Lamb, (1965, pp 711), the composition of various fuels were as shown in Table 2.2.

Table 2.2. Average composition of various fuels

Composition	Heavy Fuel (%)	Diesel Fuel (%)
Carbon	86.0	87.0
Hydrogen	10.5	11.0
Oxygen	1.5	1.0
Impurities	2.0	1.0

"Source: John Lamb, 1965, pp 711".

The composition clearly indicates the low percentage of impurities. The impurities could be classified as water, sulphur, carbon residues etc. The impurities or contaminants did not appear to be of significant importance to the user. There was no necessity for a standard with regards to the composition of fuel. The fuel was considered good by today's standards (Newbery, 1996, pp 241).

The post oil crisis saw the collapse of the standard marine fuel oil. Viscosity, density, and carbon residue readings went upwards. The presence of high percentage of sulphur and abrasive contaminants such as catalytic fines, began to pose a significant threat to engine operations. The buyer needed to take note of relevant and accurate fuel data before bunkering; this to ensure the continuous, smooth and efficient operation of the diesel engine, hence reduced operating costs.

Marine diesel engines were rising in demand then, and hence there were public awareness to curb the deteriorating trend of marine fuel quality. There was pressure for an international minimum acceptable standard for marine fuel oil. It was realised that the standard could only be effective with the participation from the oil industry, their customers, the engine builders, bunker suppliers and other related maritime sectors. The subsequent outcome was the formation of a subcommittee and working group led by the British Standards Marine Fuels Standard. Many developments on fuel oil standards took place thereafter.

A list of marine fuel standards by British Standard (BS) and the International Organisation for Standardisation (ISO) is shown in Table 2.3 (a) - (d).

Table 2.3 Marine fuel standards - grades

(a) Marine distillate fuels

Description:	BS MA 100: 1982	ISO 8217: 1996
Emergency purposes (Gas oil)	M1	DMX
No residuum (Gas oil)		DMA
Traces of residuum (Marine diesel fuel)	M2	DMB
Some residuum (Blended marine diesel fuel)	M3	DMC

"Source: Newbery, 1996, pp 242".

(b) Marine residual fuels: parameters influencing grade structure

							BS MA100 1982	ISO 8217 1996
Viscosity cST @ 100°C Max.	Viscosity cST @ 50°C (1)	Density kg/cm³ @15°C Max.	Pour point °C Max.	Carbon residue % (m/m) Max.	Ash % (m/m) Max.	Vanadium mg/kg Max.		
10	(40)	991.0	24	12	0.10	250	M4	
10	(40)	975.0	0/6	10	0.10	150		RMA10
10	(40)	981.0	24	10	0.10	150		RMB10
10	(40)	981.0	24	10	0.10	300		RMC10
15	(80)	991.0	30	14	0.10	350	M5	
15	(80)	985.0	30	14	0.10	350		RMD15
25	(180)	991.0	30	20	0.15	500	M6	RMF25
25	(180)	991.0	30	15	0.10	200		RME25
35	(380)	991.0	30	22	0.20	600	M7	RMH35
35	(380)	991.0	30	18	0.15	300		RMG35
45	(500)	991.0	30	22	0.20	600	M8	RMH45
55	(700)	991.0	30	22	0.20	600	M9	RMH55
35	(380)	1010.0	30	22	0.20	600		RMK35
45	(500)	1010.0	30	22	0.20	600		RMK45
55	(700)	1010.0	30	22	0.20	600		RMK55
35	(380)	-	30	-	0.20	600	M10	
45	(500)	-	30	-	0.20	600	M11	RML45
55	(700)	-	30	-	0.20	600	M12	

Note: (1) Approximate equivalent to viscosity at 100°C.

"Source: Newbery, 1996, pp 242"

(c) Marine distillate fuels - specifications

ISO 8216:1987 and ISO 8217:1996	DMX	DMA	DMB	DMC
Density @15°C, kg/m³ max	-	890.0	900.0	920.0
Viscosity at 40°C, cST min	1.40	1.50	-	-
max	5.50	6.00	11.0	14.0
Flash point, °C min	43	60	60	60
Pour point (upper), °C Winter quality.		-6	0	0
Summer quality. max	-	0	6	6
Cloud point, °C max	-16	-	-	-
Carbon residue, % (m/m) on 10% residue max	0.20	0.20	-	-
Carbon residue % (m/m) max	-	-	0.25	2.5
Sediment by extraction, % (m/m) max	-	-	0.07	-
Water, % (V/V) max	-	-	0.30	0.30
Cetane number min	45	40	35	-
Visual inspection	-	clear and bright	-	-
Sulphur, % (m/m) max	1.0	1.5	2.0	2.0
Vanadium, mg/kg max	-	-	-	100
Ash, % (m/m) max	0.01	0.01	0.01	-
ISO 8217:1996 only				
Total existence sediment % (m/m) max	-	-	-	0.10
Aluminium plus silicon, mg/kg max	-	-	-	25

"Source: Newbery, 1996, pp 243".

(d) Marine residual fuels - specifications

		ISO Grades	
	RMA 10 RMB10 RMC10	RMD15	All others
ISO 8217:1987 and ISO 8217: 1996			
Flash point, °C min	60	60	60
Water, % (V/V) max	0.5	0.8	1.0
Sulphur, (m/m) max	3.5	4.0	5.0
ISO 8217:1996 only			
Total sediment potential, % (m/m) max	0.10	0.10	0.10
Aluminium plus silicon mg/kg max	80	80	80

"Source: Newbery, 1996, pp 244".

Table 2.3 gives an idea of the many additional parameters for marine fuel grades, standards and specifications. It clearly indicates the vast difference in fuel quality before and after the oil crisis. It is the author's view that the shipping industry should ensure that this fuel quality and standards are upheld in order to ensure the smooth and efficient operation of marine diesel engines. The petroleum industry would not invest in this venture simply because of the small market of marine fuel as mentioned by Ewart, (1982, pp 1) :

> 'Marine fuel represent a near $40 billion market for the petroleum industry. Nevertheless, while it might appear substantial, this figure represents only a mere 5% of the total oil product market. It is therefore unlikely that the petroleum industry is going to invest in specialist production facilities for the marine industry which will guarantee the quality of the oil fuel sold as ship's bunkers.'

2.2.3 The Effects of Fuel Quality on Marine Propulsion Plant Performance

Fuel quality is closely related to the composition of the fuel with reference to the chemical and physical contaminants present. The chemical contaminants are present as free elements or in combination with other chemical compounds in the fuel.

Sulphur forms the highest chemical contaminants of fuel oil. It is present as free elements or in combination with carbon, hydrogen, nitrogen and salts. It could also be combined with other chemical compounds. In a cylinder, the product of combustion of these contaminants will be the ash compound of various categories and characteristics. Sulphur could be a possible exception. However, the presence of sulphur could lead to low temperature corrosion due to the formation of sulphuric and sulphurous acid that will corrode cylinder liners, exhaust systems and the exhaust gas boilers. At high temperature, sulphur also contributes to the phenomena of 'vanadium attack' especially common on exhaust valves (ryuuka corrosion by sodium sulphate).

In most cases marine fuel oil is contaminated with salt water. This often contributes to microbiological problems which lead to the degradation of fuel oil, corrosion of injection equipment, fouling and chokage of filter and acid attack on other components of the fuel system. The high level of sodium present is mainly responsible for the increased rate of fouling and corrosion of the engine. Hence more maintenance and repair cost for the shipowners.

A high level of conradson carbon and asphaltene will affect the combustion quality of the fuel. The contaminants will increase engine wear and foul the turbocharger. This results in reduced air supply to the cylinder which will eventually lead to a drop in engine performance due to poor cylinder combustion.

Marine fuel contaminated with sodium, vanadium and ash will form deposits in the cylinder. These deposits influence high temperature corrosion in the combustion chamber.

Aluminium oxide and silicon oxide are catalyst fines of an abrasive nature. These contaminants in fuel are attributed to many incidents of high abrasive wear in the cylinder or piston ring of marine diesel engines.

Generally, poor quality fuel will always have a marked effect on the conditions of diesel engine components, and their performance during operation. It greatly influences the wear, lifespan of engine components and reduces the mean time between overhaul of running units substantially.

2.3 The Oil Crisis: Its Impact on Global Shipping Market and the Engine Builders

2.3.1. The Impact on Global Shipping Market

The post oil crisis of 1973 also saw some high growth rates in maritime transportation. Lloyd's Register's 1973 annual report claimed that there were 12.8 million gross tons of plan approval for shipping. This figure was the largest tonnage approval in its history. It surpassed the previous order of 1971 by an estimated 2.5 million gross tons. Though this expansion of the world merchant fleet is expected to slow down in the near future, the long term growth in the numbers, sizes and types of ships is expected to continue either as replacement tonnage or new orders.

However, the energy crisis and economic recession in the industrialised countries had created less demand in shipping services. This situation eventually led to a tonnage surplus, especially prominent with the tanker market.

It is the author's view that this phenomenon clearly pointed towards the shipowners' continuous search for overall economic operation of their fleet. Afterall, at the end of the day it is the overall operational profit of the fleet that mattered most. In order to achieve this goal, the search for reduced manning and cheaper bunker were the main options in view.

The post oil crisis had seen the soaring prices of both marine bunker oil and diesel oil at the international bunker oil market. However, these oil prices varied around the many bunker markets of the world. The bunker markets mainly cover regions around the Arabian Gulf, North Europe, South East Asia, the U.S. Gulf and the Mediterranean. Their bunker prices are shown in Figure 2.4 and Figure 2.5.

The figures clearly shows the variation of bunker prices in the different regions of the world. The prices of MDO is higher compared to intermediate fuel oil (IFO). The shipowners search for cheaper bunker will need to direct their vessels to the nearest port that offered such services. In the South East Asia region, the port of Singapore is especially known for its cheap bunker and many ships enroute to Europe and East Asia called for bunker at the port. The price of bunker in South East Asia regions was among the lowest during the post oil crisis of 1973.

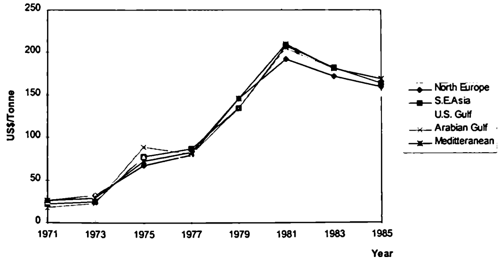


Figure 2.4 Bunker prices (intermediate fuel oil): annual averages, 1971 - 1985.
"Source: Cockett, Neil. M, 1987, pp 21".

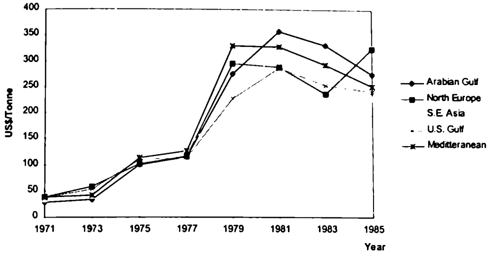


Figure 2.5 Bunker prices (marine diesel oil): annual averages, 1971-1985
"Source: Cockett, Neil. M, 1987, pp 21".

2.3.2 The Impact on the Engine Builders

The high prices of bunker fuel also resulted in the shipowners' search for economy in the operation of the marine propulsion plants. The role of the engine builders in resolving this issue was under tremendous pressure then. During the oil crisis years, the evolution of marine propulsion plant was especially speeded up. The author received a letter together with some diesel engines' catalogues dated 02.20.97, from MAN B&W stating clearly the position of the company during the oil crisis of 1973. Accordingly, the development programme of its marine low speed diesel engine was started in the late 1960s.

During and after the oil crisis years, various research programmes were carried out on the effects of light, intermediate, and heavy fuel on main and auxiliary engines. The vast different of fuel prices between marine diesel oil and heavy fuel oil especially demanded such drastic actions. The main emphasis of all engine builders were focused mainly on the design concept of a more efficient engine that could burn higher viscosity fuel in addition to reduced specific fuel oil consumption. The company succeeded in its development programme with the eventual introduction of the MAN-B&W MC engines in 1983.

The high prices of bunker fuel also contributed to numerous keen competition between the selection of a steam, diesel engine or other types of propulsion plant as the main prime mover for merchant ships. The economy of fuel consumption was seen as the main yardstick in the selection of a propulsion plant by many potential shipowners then.

Hence, many engine builders were driven by this demand of the market forces to come up with a suitable model of engine that would satisfy their need. A number of these engine builders were simply made history because of the keen and acute competition in addition to the high cost of development and testing of these propulsion plants. The oil crisis in 1973 has indeed created an economic revolution especially centred around the world's energy supplies. Nowhere has its impact been felt so severely than in the shipping world.

CHAPTER 3

The Marine Propulsion Plants before the Oil Crisis

3.1 The Various Types of Marine Propulsion Plants

3.1.1 Background to the Various Types of Marine Propulsion Plants

Marine Steam Turbine

In 1906, the *Mauretania* was powered by a total of 73,000 shp steam turbines designed by Sir Charles Parsons. These turbines were more powerful than the largest land based power generation turbine at that time. The impressive feat of Sir Charles set the pace for over 60 years where steam turbines were commonly installed for the large passenger ships, fast ferries, cargo liners and oil tankers.

The smaller vessels were generally powered by steam reciprocating engines. They were in the range of 3,000 shp. It was found that the efficiency of these engines was higher than steam turbines when operating in the lower power range. The higher efficiency was especially significant when an exhaust turbine is incorporated into the system which would expand the steam to a lower and better vacuum than the engine itself.

In the 1920s, the design of the marine impulse turbine collapsed due to rotor disc failure. Thus, Sir Charles's design of Parsons' marine reaction turbine dominated the market. Many ships around the world were fitted with these turbines then. However, the efficiency of these turbines was low mainly due to the leakages of steam across the moving blades and the casing. Additionally, the shrink fit construction of its rotor drum limited the temperature of supply steam to the turbine due to the phenomenon of vibration. Nevertheless, these turbines were known for their durability and reliability.

After the Depression of the 1930s, the development of marine steam turbines was speeded up. This was due to the initiation of an extensive naval shipbuilding programme in the USA. The development of the steam turbine with superior efficiency was pioneered by the power station turbine builder, General Electric (GE). It produced a reputable range of top class impulse turbines for the navy. This led to an era of marine steam turbine design that operated on a steam condition of 28 bar at 400°C. A groundwork standard laid down by GE that triggered the development of other steam turbine designs by De Laval and Westinghouse (Nicholas, 1990, pp 85).

GE was the pioneer in modern marine steam turbine design. It dominated the market share in the 1940s and after World War II, it was the world's largest supplier of marine steam turbines. The turbine shaft power ranged from 18,000 - 32,000 shp, (MST 13 model in 1961), 45,000 shp maximum, (MST 14 model in 1965) and 45,000 - 120,000 shp, (MST19 model in 1970). A steam condition of 59 bar at 510°C for non-reheat and 100 bar at 510°C for reheat version were the highest attainable in the MST 19 model (Nicholas, 1990, pp 87). The other major marine steam turbine builders in this modern era were Stal Laval (joint companies of De Laval and ASEA Stal) and the Japanese companies of Mitsubishi, Ishikawajima Harima Heavy Industry (IHI), and Kawasaki.

The Gas Turbine

The gas turbine as a merchant marine propulsion plant was first installed on board a Shell ship *Auris* and Marad's *John Sergeant* in the 1950s. The turbine used gas/air regenerator based on derivatives from land based gas turbines. The specific fuel oil consumption for the two vessels was 469g/kW/h and 308g/kW/h respectively (Harrold, 1989, pp 4).

In 1970, four Euroliner container ships owned by Pratt & Whitney were installed with gas turbines without gas/air regenerators. A reduced rate of fuel consumption at 295g/kW/h (Harrold, 1989, pp 5) was achieved. This was due to an improved method of fuel treatment carried out on board. The vessels were able to sail at a speed of 30 knots with an output of 44,742 kW.

Subsequently, two industrial based gas turbines were installed on board the ro-ro ships of *Iron Monarch* and *Iron Duke* owned by Broken Hill Proprietary Company in 1973. The gas turbines were made by GE. A large gas/air regenerator was installed between the compressor and the power turbines. The 17,000 shp output of the turbine drove a controllable pitch propeller through double reduction gears.

The major builders of marine gas turbine were Pratt & Whitney in the 1950s till 1970. GE came onto the scene in 1973. However, there were very few ships fitted with gas turbines for merchant marine propulsion until today. The main reasons can be attributed to the high operating cost of the plant due to material and fuel oil problems. Unless a suitable material is discovered, and clean fuel with acceptable cost is available, the role played by gas turbines as a marine propulsion plant will be insignificant (Harrold, 1989, pp 7).

Marine Diesel Engine

Rudolf Diesel was the most prominent figure in the development of large marine engines in the 19th century. He attributed much of his success to theoretical thermodynamic reasoning and detailed practical experimentation. In 1903, the first diesel powered 25 bhp engine was installed on board the French canal barge 'Petit Pierre'. The 210 mm bore and 300 mm stroke, horizontal opposed piston diesel engine was built by Dyckhoff (Brown, 1985, pp 8). This was followed by other installations of diesel engines on board inland river tankers in Europe.

The manoeuvring and reversing mechanism of the early marine diesel engines were carried out by electric transmission. In 1905, Sulzer introduced the first 2-stroke direct drive reversible engine. This was carried out mainly by the use of separate cams for the variation of fuel injection timing in both the ahead and astern running condition. Nevertheless, after the *Selandia* which was the first ocean going vessel powered by two large B&W 2-stroke diesel engines (Lebech, 1959, pp 50 & B&W Engineering, 1988, pp 1) and the *Monte Penedo*, powered by a Sulzer 2-stroke diesel engine in 1912, the development of marine diesel engines had begun to show improvement in areas of fuel consumption and thermal efficiency. By the year 1939, two out of three ships ordered were powered by diesel engines. However, diesel engines were still unable to burn the residual fuel under ship's boiler. The decisive criteria then was based on the prices of MDO to residual fuel and the efficiency of the diesel to steam system (Nicholas, 1990, pp 84).

Medium Speed Diesel Engine

The development era of marine medium speed diesel engines (engines with an operating speed range between 300 - 1000 rpm), actually began during World War II. The German navy needed a lighter engine for its light warships. This engine should have a power/weight ratio of less than one third of its pocket-sized battleship. The design of this engine was put forward by Gustav Pielstick of MAN company. He succeeded in reducing the weight of the engine by increasing the speed up to 350 rpm. Additionally, a supercharger, 'V' form of engine configuration and a multiple arrangement system was included in the design. The specific weight of this engine was about 16 kg/kW (Sair, 1981, pp 8).

Subsequently in 1963, medium speed engines operating in the range of 450 - 600 rpm were providing an impact in the merchant marine propulsion plant. It had, of course, always dominated the smaller vessels of coastal ships, ferries and tugs where headroom was a critical factor.

However, it is the author's opinion that these engines had great difficulty in burning heavy fuel oil due to their high rotational speed and the lack of technical know-how. Moreover, before the oil crisis, there were no market demand to change its normal fuel of MDO to Heavy Oil in view of the low cost of MDO; Arabian Gulf, 1970, MDO: US$ 24.59/tonne (Cockett, 1987, pp 21) compared to today. The medium speed engine also consumed higher amounts of lubricating oil and there was more maintenance work to be carried out (Harrold, 1989, pp 9) in view of the greater number of cylinder units.

Low Speed Diesel Engine

The first successful engine of Rudolf Diesel in 1897 was a slow speed, single cylinder, 4-stroke, compression ignition engine (Sair, 1981, pp 6). Ten years after the birth of this engine, there were a number of merchant vessels fitted with low speed diesel engines. The engines' power was in the range of 224 kW/cyl. They posed a threat to the steam engines from 1920 especially due to their higher thermal efficiency. After World War II, the higher power range of low speed diesel engines was developed.

The Sulzer marine low speed diesel engine introduced the RD series of engines in 1957. It had a rating of 1,193 kW/cyl. The production of this engine continued for a period of 12 years (Harrold, 1989, pp 10), before the RND series came into the limelight in 1968. The production of this unique and modern engine lasted until 1976 - 3 years after the oil crisis. It had many modifications to its design. The maximum cylinder bore size was 1,050 mm. It was the first Sulzer engine which discarded the use of rotary exhaust valves in favour of exhaust ports and longer piston skirt.

The MAN-B&W marine low speed diesel engine started its production of the VT2BF engine in 1963. It had a bore of 620, 740 and 840 mm, and a stroke to bore ratio of 2.1 - 2.3. The operating bmep was 8.4 bar. This design was upgraded in 1968 to the KEF series. This engine had the same bore and stroke length but a higher bmep of 9.4 bar. The KGF range of engine followed suit in 1973 - the year of the oil crisis. There were many modifications to the design of this engine.

3.1.2 The Market Share of the Various Marine Propulsion Plants

The market share of the various marine propulsion plants would be determined by the selection of the propulsion machinery. This selection would depend on basic environmental and functional requirements in addition to those of technology and economic nature.

Prior to the oil crisis of 1973, the market share of the propulsion plant could be clearly divided between the line of power requirement and fuel economy. The power requirement of an engine would determine the selection of the type of propulsion plant for the ship. The economy in fuel consumption was also a determining factor, especially for the shipowner.

The power availability of a propulsion plant could be defined in the range of 15 MW, 20 MW and 30 MW. Generally, the medium speed engine was preferred in the power requirement of below 15 MW, the low speed engine would be in the range of 15 MW to 20 MW while the steam turbine would be in a power range of 20 MW to 30 MW or higher (Marine propulsion systems: Proceedings of a symposium, 1974, pp 53). Moreover, at a higher power range, the cost of reciprocating components would be much higher than their rotating counterparts. Thus, the diesel engine lost out in terms of economic benefits.

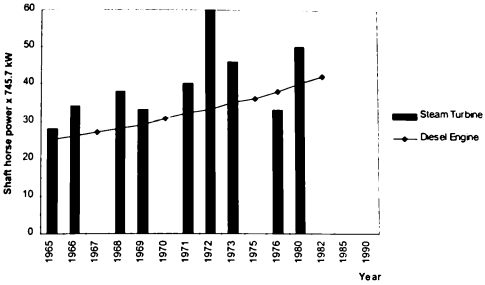


Figure 3.1 Power requirement versus diesel availability

"Source: Harrold, 1989, pp 3".

Figure 3.1 indicates the power availability of the diesel engine versus steam turbine propulsion plant installed on board large tankers in the mid 1960s and early 1970s. It clearly shows the choice and suitability of the steam turbine for the higher power requirements of the VLCCs and ULCCs.

The specific fuel oil consumption of the marine diesel engine was about 201 g/kW/h for both the medium speed and low speed range. This figure was much less than the steam turbine and gas turbine plant where consumption stood at 315 g/kW/h and 295 g/kW/h respectively. However, one important advantage with the steam turbine plant was that fuel economy improved as power increased (Figure 3.2). This was due to the constant blade clearances as the size of the engine increased. Hence, the popularity of the choice of steam turbine to power the ULCCs in the 1970s as shown in Figure 3.3. On the other hand, the fuel economy of the diesel engine remained reasonably constant over a wide range of service power. This characteristic has resulted in the popularity of diesel engines among the shipowners in reducing operating costs due to the high fuel prices.

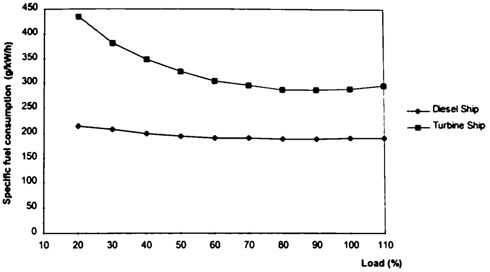


Figure 3.2 Comparison of fuel consumption performance characteristics between turbine ship and diesel ship

"Source: Seikan Ishigai , Shinsuke Akagi, 1980, pp 8".

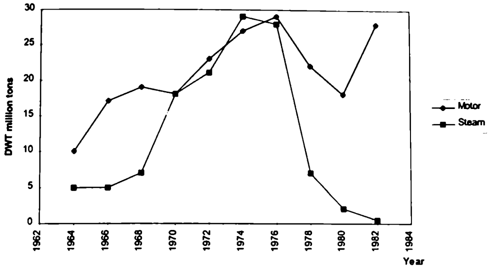


Figure 3.3 The steam ship boom in the 1970s

"Source: Nicholas, 1990, pp 86".

The market share of the various propulsion plants prior to the oil crisis of 1973 depended very much on the power requirement of the engine. This could be analysed through the demand of the various types of propulsion plant between the 1960s and 1970s. The diesel engine was in the forefront in the early years of the 1960s because of the lower power requirements of merchant marine propulsion plants. The demand for steam engines reached its height only with the introduction of high powered ships of VLCC and ULCC such as *Tokyo Maru* in 1965 et al, during the latter part of 1960s and early 1970s, as shown in Figure 3.4.

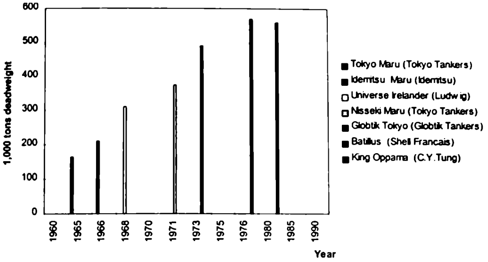


Figure 3.4 Tanker growth 1965 - 1980

" Source: Harrold, 1989, pp 2".

The price of heavy fuel oil was not a serious hindrance in the selection of a propulsion plant as the cost of residual fuel was reasonably cheap then. However, the price of distillate fuel, such as MDO, was always higher. This high fuel price of distillate fuel was a negative aspect in the shipowners' selection of a marine propulsion plant.

Nevertheless, prior to the oil crisis, steam turbines and diesel engines made up the largest market share of the merchant marine propulsion plant. This was mainly due to their varied power range availability and reasonably low operating costs.

3.2 The Design Concept of the Various Propulsion Plants

3.2.1 Specification of the Various Propulsion Plants

The space and weight occupied by a main propulsion plant on board would vary for different types of ship. In order to make a valid comparison of the different types of propulsion plants, it would be necessary to evaluate and include the relevant auxiliaries. There should be availability of clear working space for overhauling of machinery parts during routine maintenance or repair. An adequate access for control and operation of machinery should be provided.

A typical example of machinery weight for VLCCs are as follows:

- Medium speed diesel engine : 78 kgf/kW.
- Low speed diesel engine : 103 kgf/kW.
- Steam turbine : 68 kgf/kW.

"Source: Marine propulsion systems: Proceedings of a Symposium, 1974, pp 6".

The steam turbine has the lowest weight per kiloWatt as compared to the diesel engine. The length of its engine room is also shortest due to its compactness. However, medium speed diesel engine obviously has the lowest head room requirements though it may have the greatest specific length. On the other hand, low speed diesel engine has the highest weight per kiloWatt among the other main propulsion plants. The length of its engine room would also be greater than the steam turbine.

Marine Steam Turbine

The specification and development data of steam turbine design prior to the oil crisis is shown in Table 3.1. The data was obtained from General Electric Company Alsthom EI, (GEC/AEI) merchant marine designs. Over the years, the steam consumption for non-reheat turbine was reduced to 30% (Nicholas, 1990, pp 91). This was possible due to the higher steam condition in addition to better design of blade and nozzle concept. The use of low alloy steel for the casing and forging of the monobloc rotor had further enhanced this design.

The improved properties of materials used had allowed the stress on the bore to increase by two times between 1945 and 1970. It was also able to further increase the speed of the blade in the high pressure (HP) and low pressure (LP) turbines by 60% and 67% respectively (Nicholas, 1990, pp 91). The work done on each stage of the turbine had also improved. In the HP, the work done per stage had gone up three times while in the LP, it increased by 50% (Nicholas, 1990, pp 91). There was also a corresponding reduction in the number of stages. This greatly enhanced the compactness of the entire turbine system as shown in Figure 3.5.

Table 3.1: GEC/AEI design development 1945-1970

Design references	Inlet steam condition	Inlet steam condition	Power	Number of stages	Maximum blade tip speed (m/s)	Maximum blade tip speed (m/s)	Maximum blade tip speed (m/s)
-	Bar	°C	SHP	-	HP	IP	LP
National shipbuilding programme (1945)	29.63	393	7,500	11	137	-	209
Blue Funnel Line (1955)	42.37	510	8,000	30	183	158	219
Ben Line Container Ships (1970)	62.01	510	44,000	15	218	-	394

"Source: Nicholas, 1990, pp 91".

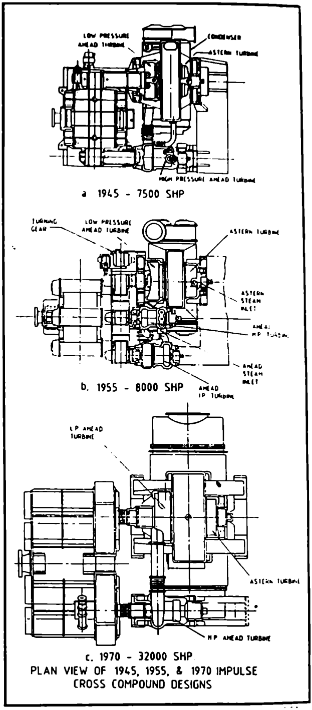


Figure 3.5 AEI/GEC cross compound design 1945-1970

"Source: Nicholas, 1990, pp 90".

4-Stroke Medium Speed Diesel Engine

In 1967, the major engine builders of this model were Sulzer, Pielstick, MAN and Werkspoor. A comparison of the various makes of the engines in terms of engine specifications and particulars are as follows:

- all engine bores were in the 400 mm class.
- the highest stroke to bore ratio was 1.35.
- the highest piston speed was 7.62 m/s.
- the highest bmep was 16.9 bar.
- maximum cylinder pressure was 113.8 bar.
- the specific weight however, varied between 9.9 - 16.45 kgf/kW.
- the lowest specific fuel oil consumption was 205 g/kW/h.

"Source: Harrold, 1989, pp 9".

The medium speed diesel engine generally has an average speed of 450 rpm. This high rotational speed would encounter minimum problem in burning distillate fuel. In 1972, the distillate fuel price of MDO was US$ 35.25 / tonne maximum, (Cockett, 1987, pp 21). It was reasonably low by today's standards. The engine specific fuel oil consumption of 205 - 217 g/kW/h was not an alarming issue to the shipowners. Thus, the specification and particulars of the above range of medium speed engines were generally acceptable to many shipowners.

2-Stroke Low Speed Diesel Engine

Prior to the oil crisis of 1973, the major engines in contention were Doxford, Gotaverken, Stork, Fiat, MAN, Sulzer, and B&W diesel engines. They were the seven leading engine designers in Europe then. However, remains today are the Sulzer and now MAN-B&W diesel engines.

A comparison of the engine specifications and parameters of these two major marine diesel engine builders could be deduced as follows:

- the maximum cylinder bore size was 1050 mm of RND engine.
- the highest stroke to bore ratio of 2.0 favoured the KGF engine.
- the maximum piston speed reached 6.6 m/s in the KGF engine.
- the highest bmep of 11.4 bar was attained by the KGF engine and
- the specific fuel oil consumption was in the range of 204 - 208 g/kW/h.

"Source: Harrold, 1989, pp 12-13".

Thus, it could be concluded that prior to the oil crisis, the specification of 2-stroke low speed diesel engine was that of large bore, short stroke, low mean effective pressure, high rpm and high specific fuel oil consumption engine. This conclusion was drawn based on their comparison to the present day low speed engine.

3.2.2 The Design Features of the Various Propulsion Plants

Marine Steam Turbine.

In the 1920s, the design of the marine steam turbine was mainly monopolised by the Parson's reaction turbine. A reaction turbine has very small running clearances between its rotor blades and casing. This is necessary in order to reduce leakages of steam across the small clearances. It is estimated that 50% of the pressure drop on each stage of the turbine takes place across the rotor or moving blades (Nicholas, 1990, pp 84). The rotor was of the large drum type constructed by shrink-fitting.

However, this type of turbine has very low efficiency due to the phenomenon of vibration when operating with steam condition of above 343°C (Nicholas, 1990, pp 84). Moreover, it is also difficult to maintain the small clearances between the rotor blade and casing.

After World War II, the design of steam turbine was switched to a combination of impulse and reaction principle. This cross compound turbine was designed with an impulse design on the HP side. This HP impulse turbine was subsequently connected to a separate HP astern turbine of the similar design. The LP turbine was a reaction design but the shrink-fit construction of its rotor drum was changed to a monobloc forging. The operating pressure and temperature range was 27.56 bar at 435°C, a limit higher than the reaction turbine. Hence, a higher efficiency was obtained from this cross compound design.

In order to further compete with diesel engine in the early 1960s, there were many developments to the design of steam turbine. A higher steam condition was obtained with improved design of reheat boiler. Research into metallurgy has also enabled improved materials to be used for the turbine system. The possibility of reduction on weight was accomplished with the use of epicyclic gears for the first reduction gearing. This had further increased the compactness of the turbine system. Furthermore, the efficiency at LP turbine increased due to the availability of a higher dryness factor of supply steam. The introduction of reheat versions with higher power capability had also resulted in reduced specific fuel oil consumption. Automation in the operation of the steam turbine system, especially on the reheat version, had also further enhanced the overall efficiency of the steam turbine propulsion plant.

4-Stroke Medium Speed Diesel Engine

The medium speed diesel engine was basically designed to burn distillate fuel. It could be operated at low revolution by the use of reduction gears. This was especially important for higher propeller efficiency. The opportunity for energy savings could also be realised by incorporating an electrical power take off (PTO) along its shaft. It had such features as ease of installation, reduced cost per horse power, and multi-engine flexibility for improved reliability of the propulsion plant.

The medium speed engine is a multi-cylinder engine with in-line or 'v' form configuration. It was designed for the low power range of a marine propulsion plant. Prior to the oil crisis it was making an impact on merchant marine propulsion plants due to its low power to weight ratio. It had replaced certain steam as well as the low speed diesel propulsion plant during this period.

2-Stroke Diesel Engine

The design feature of this engine was mainly geared towards the supercharged version replacing those unsupercharged types. The main objective was to increase the output power per cylinder with more efficient combustion from the pressurised air and fuel in the cylinder. The emphasis on simple engine design, good accessibility, better and refined components with high reliability were also the theme of engine builders then.

The introduction of improved welding on fabricated engine structure had replaced the previous cast construction, with box shaped design of welded bedplate and frame. However, due to many incidents of crack on welded seams of bedplate main bearing saddle, this portion of cross member still revert to cast or forged steel version.

The cylinder cover also undergone numerous design changes due to the increased number of units and larger engine bore sizes. These features together with higher brake mean effective pressure (bmep), called for greater attention on the selection of suitable materials in order to withstand both mechanical and thermal stresses during operation. Hence, the advocate of cast steel material in contrast to its cast irons counterpart, has contributed to increased mechanical strength of cylinder cover outer part. The reduction of wall thickness on inner cast iron insert instead contributed to lower thermal stresses. Cylinder cover with cast steel outer and cast iron insert were the feature of Sulzer engine design then.

Prior to the oil crisis, the increased output power of individual engine cylinder by supercharging has contributed to higher mechanical stresses at liner top region. This has warranted the modification of liner top flange to withstand the increased mechanical and thermal loading. In Sulzer engine, a thicker flange was used to withstand the mechanical stresses while bore-cooling concept was incorporated to minimise thermal loading in the region.

The crankshaft of the earlier design was made by forging it in one piece. The later engine design of larger bore with increased number of units demanded the use of semi-built up type of crankshaft instead. In this configuration, shrink-fitting was carried out between the webs and journal pins.

The piston crown of early Sulzer engine was thin-walled with supporting ribs internally. This design was inadequate for increased bore size and higher firing pressure of the later engines. Hence, a wider piston rod flange with piston skirt were advocated to withstand the higher gas load on the outer region of the forged steel piston crown. The cooling medium was changed from oil cooled to water cooled with stuffing box separating the combustion space from the crankcase. This permitted the employment of ribbed cast steel piston crown with thinner wall. Hence, a reduction in thermal stresses was achieved. The number of piston rings were reduced from eight to six, and the grooves top and bottom surfaces were chromium plated to reduce wear from the corrosive residue of fuel.

Prior to the oil crisis of 1973, 2-stroke diesel engine has indeed undergone numerous changes in its design and operation features compared to its predecessor. This was mainly due to the introduction of engine turbocharging system and the development of alkaline cylinder oil in the 1950s. The turbocharging system that utilised engine exhaust gas to drive the exhaust turbine has contributed to a greater mass of pressurised air for better combustion, increased cylinder output power and higher engine efficiency. On the other hand, the introduction of alkaline cylinder oil has enabled the neutralisation of relatively cheap and high sulphur content residual fuel to be carried out. This has greatly reduced the wear rate of cylinder liner from the corrosive sulphuric acid (Kilchenmann, 1973, pp 70).

3.3 Performance Characteristics of the Propulsion Plants

3.3.1 Efficiency of the Propulsion plants

The improvement of efficiency in maritime transportation was mainly due to the development of the low speed engine pioneered by the *Selandia* in 1912. Later, many ships were equipped with diesel engines as the main propulsion plant. The steam turbine dominated the higher power range from the 1950s to the early years of the 1970s in view of its better fuel economy at optimum propeller revolution. However, the introduction of turbocharging in the 1950s has further improved the efficiency of the 2-stroke low speed marine diesel engine. These engines then begun to show dominance in the higher power range occupied by the steam turbine.

Marine Steam Turbine.

The history of the steam turbine can be traced back to over 100 years ago. It was thus unlikely that this kind of old technology would have undergone much dramatic improvements. However, with the influence of better material in addition to the application of computers in design process, the steam turbine in the early 1970s was able to perform with higher operating steam pressure and temperature.

The ratio of blade speed to steam speed would determine the efficiency criterion of any turbine. A reaction turbine would achieve the maximum efficiency when the ratio of blade speed to steam speed reached 94% (Institute of marine engineers, 1965, pp 93). In the early 1970s, the efficiency of the steam turbine (propulsion power) was in the range of 31% as shown in Figure 3.6.

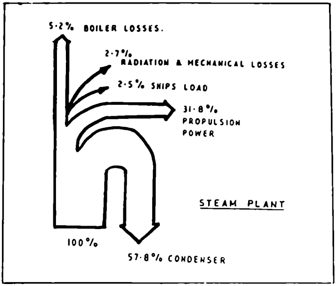


Figure 3.6 Energy flow diagram in steam plant

"Source: Marine propulsion system: Proceedings of a symposium, 1974, pp 48".

Marine Diesel Engine

The birth of the compression ignition engine principle was started by Carnot in 1824 when he conceived the idea of auto-ignition where a charge was compressed. This was followed by the invention of the first internal combustion engine burning gaseous fuel that attained a very low efficiency of 4% (Sair, 1981, pp 6). This engine was built by Etienne Lenoire in 1860. Nikolaus Otto, in 1876, using the principles of Rochas, invented a spark ignition engine where for the first time, the charge was compressed before ignition. The engine reached an efficiency of 12% (Sair, 1981, pp 6). Rudolf Diesel's invention of the diesel engine in 1897 was a combination of the idea from Carnot and Otto. The engine was tested to an efficiency of about 20% using oil as fuel (Sair, 1981, pp 6).

The advent of supercharging in the 1950s especially contributed to the achievement of higher thermal efficiency for the marine diesel engine as shown in Figure 3.7. The figure shows a gradual rise of engine thermal efficiency to the 1950s. Thereafter, quite a significant rise in the engine thermal efficiency was experienced with the introduction of engine supercharging and the turbocharging system. Prior to the oil crisis of 1973, the thermal efficiency of the marine diesel engine was about 37% (Brown, 1985, pp 6).

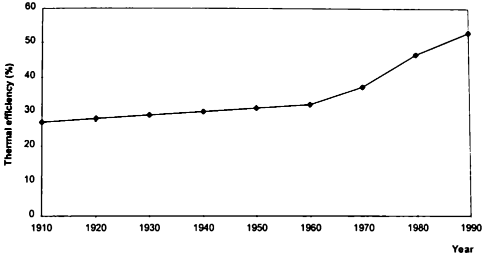


Figure 3.7 Thermal efficiency of large 2-stroke diesel engine

"Source: Brown, 1985, pp 6".

The heat efficiency and losses is an indication of the expenditure of diesel engine heat in fuel of 100%. This heat balance is shown in Table 3.2. A useful work or thermal efficiency of 37% was obtained from the diesel engine. Exhaust gases loss accounted for the highest percentage of 35%. This heat loss at the stack could be further reduced by a waste heat recovery system using an exhaust gas boiler or by other means; an area of great importance in the search to improve the heat efficiency of the marine diesel engine.

Table 3.2 Heat balance data (BHP basis)

	Per cent
Heat converted into useful work (thermal efficiency)	37
Heat carried off in exhaust gases	35
Heat carried off in circulating water, such as cylinder jacket, piston cooling	17
Heat used by turbocharger	7
Friction and other losses	4
Total	100

"Source: Bowden, Reprinted 1984, pp A41".

3.3.2 Reliability and Durability of the Propulsion Plants

Reliability and Durability

The commonly accepted definition of reliability is the probability that a machinery or equipment will perform adequately for a stipulated time and under defined operating conditions (Institute of Marine Engineers: Proceedings of a conference, 1971, pp 6). Every shipowner's main concern is the reliability of the main propulsion plant as well as its supporting auxiliaries.

The durability of a diesel engine on the other hand has a different meaning altogether. It is the feature of its well being due to the design and manufacturing characteristics. The distinction as suggested by Vulovic (The Motor ship, May, 1992, pp 32), is simply:

> 'Durability is a function of design and manufacturing whereas reliability is a matter of understanding what the durability is.'

A breakdown of any propulsion plant or its auxiliaries would mean an 'out of service' of the ship. This could result in costly delays and monetary loss to the shipowner. Hence, the main goal of a shipowner is to ensure the maximum availability, the probability that the system will be in an operating state at any time, of machinery and minimum 'out of service' time throughout the useful life of the ship.

The machinery availability of VLCC's for propulsion as well as cargo pumping operation should be 345 days and more per year. Generally, the availability of steam and motor tankers between 1962 to 1972 was in the range of 94 - 96.5 %, (Marine propulsion system: Proceedings of a symposium, 1974, pp 79). The expected 'out of service time' due to breakdown of other conventional cargo ships, passengers ships, containers and bulk carriers should be less than 24 hours per ship per annum. (Marine propulsion systems - Proceedings of a symposium, 1974, pp 6).

The total percentage of vessel stoppages due to steam and diesel plant failure were about 61% and 69% respectively (Marine propulsion systems - Proceedings of a symposium, 1974, pp 14-15). This would suggest that a diesel plant would experience more stoppages or 'out of service' time due to engine failures than a steam plant. It simply points towards the higher reliability of steam plants over diesel plants.

Steam Turbine versus Diesel Engine

During the trade boom of late 1960s and early 1970s, the increased sea trade between Japan, Europe and the United States triggered the rise of many high powered steam turbine driven VLCC's and large container ships. The threat to the Suez Canal, escalation of Middle East oil supplies and the revolutionisation of general cargo ships by containers also contributed to this phenomenon. Diesel engines were not able to penetrate the market then in view of the lack of power for the higher range. Moreover, a rotating machine like the steam turbine, had already proven its long standing popularity for reliability, and durability in design and operation. The technology of over a century old has resulted in many proven tests to its credit.

It is always an accepted fact that a rotating machine such as the steam turbine would experience less vibration under dynamic loading when compared to a reciprocating component. A reciprocating component such as the large marine diesel engine would experience large inertia forces, with increasing mass, that could contribute to the familiar phenomenon of vibration from reciprocating, and rotating out of balance forces during operation. Hence, the possibility of breakdown would be higher if careful analysis of the forces acting were not carried out. The added stability from vibration phenomenon in the steam turbine in the higher power range especially, would mean less wear down of the running parts due to rubbing or friction. Hence, less breakdown of the plant and added reliability.

The durability of the marine diesel engine plant depends very much on the wearing down of running parts during operation. Hence, the emphasis on careful monitoring and maintenance of the plant is critical. The corrosive nature of the sea environment and the quality of fuel should be checked in order to further reduce wear down of parts from corrosion and hence contribute to a reduced durability of the plant. It is widely accepted that the quality of marine fuel before the oil crisis was reasonably good by today's standards. Hence, the corrosive nature of the fuel due to sulphur especially, should not have posed a serious threat to the durability of the diesel engine then. The various research on and development of the use of improved material also enhanced its durability.

Prior to the oil crisis of 1973, the performance of the marine propulsion plant was mainly geared towards the efficient operation of the higher power range engines of VLCC's and ULCC's. The ability of its boiler to burn poor quality fuel and provide high power requirements, had won the steam turbine propulsion plant its most glamorous era of superiority over the diesel engine. It spanned for over a short period of only four years, between 1968 - 1972 where the order book for steam turbines had actually surpassed that of diesel engines for ships over 20,000 dwt. The development of the diesel engine with turbocharging system was not ready to penetrate the market then. The added advantages of reliability and durability of the steam turbine plant had also contributed to its great demand during this period.

3.4 Operation and Maintenance Features of the Propulsion Plants

3.4.1 Operation Features of the Propulsion Plants

The availability of suitable personnel to operate a plant would affect to a certain extent, the selection of the type of machinery and the degree of automation to be equipped in the entire engine design. In the late 1950s, the improved economy of the world has resulted in the increased demand for the new building of ships. This demand was greatly affected by the shortage of qualified seafarers and the high cost of manning. Hence, the idea of an 'Engine Control Room' was realised.

The advent of the 'Engine Control Room' has seen many changes to the design concept of instrumentation and monitoring equipment in the engine room. Much of this local equipment was transferred to the Control Room. This idea has greatly contributed to an improve working environment in the engine room.

The era of automated ships or unmanned machinery space (UMS), came onto the scene in the 1960s. The reasons were mainly due to the vast improvement in control system technology based on the utilisation of micro-processors, and the concept of a simplified engine room arrangement. Accordingly, the number of automated steam turbine plants was more than that of the diesel engine propulsion plant. However, many of these UMS ships did not operate with UMS at sea. The status obtained was mainly to acquire the safety standards of the installation.

3.4.2 Maintenance Concept of the Propulsion Plants

Since the first marine machinery was fitted on board, there was already a 'regular maintenance' philosophy in practise. The reason was due to the requirement for routine survey by classification societies' surveyors . The maintenance schedule of the plant was planned by the chief engineer. Generally, it was carried out using the various guidelines stipulated in the engine builder's instruction manuals. The task of carrying out the maintenance work was then being delegated to the respective engineers on board.

It was also of utmost importance to keep a record of all the maintenance work, machinery component wear, parts renewed, running hours and other details relevant to an efficient maintenance culture. A maintenance log book was generally used for this purpose. However, all these records would need to be noted down manually by the individual responsible engineers concern.

CHAPTER 4

The Marine Propulsion Plants after the Oil Crisis

4.1 Changes to the Selection of Propulsion Plant from Steam to Diesel

4.1.1 Reasons for the Change

The rate of decline of the steam turbine after the oil crisis was much more abrupt than its rate of growth prior to that momentous years of the late 1960s and early 1970s. The high crude oil prices had resulted in many developments of oil fields nearer to the sources of consumption, especially in the North Sea region. There were also greater consumer awareness in the economy of oil consumption. These factors mainly contributed to some adverse drops in the large tanker market. The trend towards higher power and faster ships was dramatically changed to one of fuel economy.

The first oil crisis of 1973 saw the quadrupling of the oil prices while in the 1979 crisis it doubled. The proportion of fuel cost had since changed from about 8% in 1970 to approximately 70% of the overall ship's operating cost by 1980 (Nicholas, 1990, pp 87). Diesel engines had already attained a fuel rate as low as 163 g/kW/h (Nicholas, 1990, pp 86), and with the complex heat recovery system, it could be operated at nearly half the total fuel rate of steam ships. The advantage of the diesel engine had simply outclassed the steam system burning similar fuel. Thus, for fuel economy, many steam ships went 'slow steaming', new orders were cancelled or taken out of service and replaced.

4.1.2 Closing State of Development of the Marine Steam Turbine Propulsion Plant

The competition between steam turbine and diesel engine had seen the former venturing into higher steam conditions. This meant the application of reheat cycles, enhanced gearing arrangement allowing greater reduction ratio, and thus lower propeller revolution leading to higher propulsion efficiency.

The adoption of the reheat cycle had always been a standard design of land based power generating plants. Its application for marine use had met with problems during manoeuvring and astern movement where the reheater could be starved of steam and was likely to burn out. The high steam condition also contributed to the increased inertia of the steam. This phenomenon created problems with steam control to turbine during sudden loss of load. These dilemmas were, however, resolved with the fitting of overspeed trip devices after the reheater and the adoption of reheater gas by-pass on the main boiler.

The reduction of about 30% in the machinery weight (Harrold, 1989, pp 4) was achieved by the use of single - plane gearing such as the planetary gears in the initial reduction. This led to an improvement in the compactness of the engine and resulted in less kinetic exhaust losses to the condenser due to its axial flow pattern. Additionally, drives for the feed pump and generator were found in the gear box.

Nevertheless, with all these efforts and development, in 1980, there still remained a gap of about 20% in thermal efficiency (Harrold, 1989, pp 4) between the steam turbine and diesel engine. In today's standard, this gap of thermal efficiency could be even greater and doubtful to be closed. It is the author's view that the main influencing factor here is the high ignition pressure and temperature of modern low speed diesel engine that catapulted its thermal efficiency to a greater height.

The remarkable era of marine steam turbine propulsion spans a memorable and short period of almost fifteen years. However, it did rise to a couple of brilliant innovative designs, and manufacture technology. These efforts were still unable to curb the progress made by diesel engine in penetrating the higher power ranges and consuming low grade fuel. Marine steam turbine propulsion eventually came to an abrupt end after only a brief appearance of superiority and dominance.

4.2 Improvement on the Design Concept of the Diesel Engine Propulsion Plant

4.2.1 Specification of the Propulsion Plant

4-Stroke Medium Speed Diesel Engine - The Tenacious Competitor

Over the years, there were many claims by engine builders that progress was made in areas of higher output power/cylinder, improved performance with heavy fuel, greater reliability and increased mean time between overhaul. These claims were met by the 2-stroke engine builders with great enthusiasm.

The introduction of the low speed, long stroke engine that operated at lower propeller revolution with higher propeller efficiency, deprived the exclusive claim of the medium speed engine in this arena. The advantage of lower power to weight ratio and the saving of space also appeared to be trivial. The ability of the low speed engine to operate at constant speed for PTO has indeed provided a rival to the medium speed engine. Its proven reliability also has countered the attraction of medium speed multi-engine flexibility. Thus, all that is left of the medium speed engine would be the advantage involving high electrical load application especially common in passenger ships. In merchant marine propulsion plant, it would be very unlikely for the medium speed engine to replace the 2-stroke low speed engine in the years to come .

2-Stroke Low Speed Diesel Engine - Era of Dominance

The post oil crisis of 1973 had a great impact on the development of the low speed diesel engine's design concept . The pre 1973 preference for larger and faster ships was drastically changed to one of fuel economy. This evolution in the design concept of low speed diesel engine was made available to the shipping market towards the latter half of the 1970s.

The popularity of the large bore low speed engine was returned to its prominence due to its ability to burn and digest poor quality fuel from the aftermath of the oil crisis. The introduction of the longer stroke engine was also a trend in engine design. The stroke/bore (S/B) ratio of the Sulzer engine was increased from its RND, RND-M version of 1.52 - 1.67 to the long stroke and 'superlongstroke' engine of the RL and RTA series with a stroke/bore ratio of 2.1 and 3.75 respectively. However, the trend towards lightness and compactness in engine design has favoured a shorter S/B ratio of 2.6 in the latest RTA engine (Table 4.1).

On the other hand, MAN-B&W engine increased its stroke bore ratio from the KGF series of 2.0 to the long stroke L-GB version of 2.5 - 2.7. The long and superlong stroke of L-MC/MCE and SMC series has a stroke/bore ratio of 3.2 - 4.0 respectively while the short stroke K-MC series has a S/B ratio of about 3.0. The latest K-MC-C model or the compact version, has a S/B ratio of only 2.4 (Table 4.2).

This advocate of greater stroke/bore ratio incorporating uniflow scavenging system, has indeed contributed to a higher fuel economy of the engine as shown in Figure 4.1, and satisfied greater power/speed requirement. Moreover, the loop scavenging was found to be at a thermodynamically disadvantage at the higher stroke bore ratio of more than 2.9 (Brown, 1985, pp 41) due to low scavenging efficiency. The high combustion space also permits freedom of compression ratio limit, better combustion from good fuel spray pattern without impinging on liner wall and space for exhaust valve opening at top dead centre with choice on valve timing.

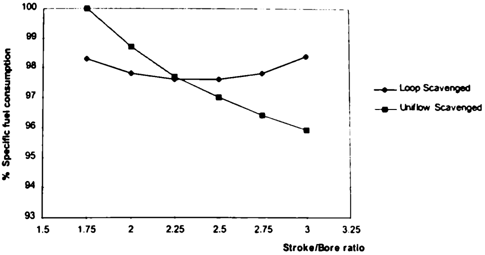


Figure 4.1 Sulzer - Stroke/Bore ratio and uniflow - loop scavenge selection
"Source: Listewnik, 1996".

The increase in thermal efficiency over the years was mainly attributed to the development of higher cylinder combustion pressure. Prior to the oil crisis, the Sulzer RND version of engine attained a Pmax of 84 bar. The post oil crisis saw the achievement of Pmax of 94 bar in the RND-M series. The subsequent version of RL and RTA engines reached a staggering height in Pmax of 118 and 130 bar respectively as shown in Figure 4.2. This indicates an increase in Pmax of about 55% since the RND version of engine.

In MAN-B&W engine, its KGF series in the early 1970s, attained a Pmax of 88 bar while the latest engine of the MC version has a Pmax of 140 bar, an increase in Pmax of about 59% within a period of about two decades.

The strive for higher cylinder combustion pressure also meant greater unit mean effective pressure and hence greater increased cylinder output power of the modern engine. Prior to the oil crisis, the average mean effective pressure (mep), and horse power/cylinder (kW/cyl), between the two engine builders was about 11 bar and 2,274 kW/cyl respectively. It has now attained an average reading of 18 bar in mep, and 5,518 kW/cyl in its output power per cylinder. An increase of mean effective pressure of more than half but a big leap of output power per cylinder of about one and a half times. This high output power would indeed require substantial modification to the engine design.

The post oil crisis has indeed brought about tremendous improvement in the specific fuel oil consumption of marine diesel engine. The search for reduced fuel consumption has also contributed greatly to the reduction of engine revolution design concept in addition to better propeller efficiency.

The average minimum speed of modern engine now is about 74 - 75 revolution per minute (rpm). A reduction of about half the speed of the engines in the early 1970s. However, there are also existing engines operating at a speed as low as 56 - 58 rpm. A great reduction of speed indeed compared to the era of the pre oil crisis. This reduction in speed also provided the engine with more time to digest the fuel. The specific fuel oil consumption has also proved to be a great saving of about 24% from an average of 205 g/kW/h to the present figure of 156 g/kW/h.

Table 4.1 Engine specifications (Sulzer)

	RD 1963	RND 1968	RND-M 1976	RLB 1979	RTA 1982	RTA 1996
Max. Bore (mm)	900	1050	900	900	840	960
Stroke/ Bore ratio	1.72	1.52	1.67	2.11	3.45	2.60
Max. piston speed (m/s)	6.3	6.48	6.48	6.4	7.45	* 8.33
Min. (rev/min)	120	122	122	90	56	90
Mep (bar)	8.49	10.4	12.1	14.0	16.3	18.2
Max. press. (bar)	75	82	92	116	123	142
kW/cylinder	1715	2237	2535	2938	3542	5490
Specific consumption (g/kW/h)	208	208	193	178	156	161

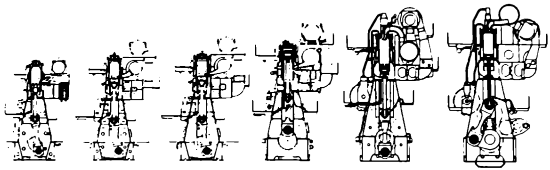
* : Mean piston speed.

"Source: Harrold, 1989, pp 12. MER, Directory of marine diesel engines, 1996 & Sulzer RTA96C catalogue, Oct. 1995".

Table 4.2 Engine specifications (MAN-B&W)

	12KGF90 1975	K98MC-C 1996
Bore (mm)	900	980
Stroke (mm)	1800	2400
Stroke/ Bore ratio	2.0	2.4
Minimum speed (rev/min)	110	94
kw/cylinder	2312	5710
Piston speed (m/s)	6.6	8.3
Bmep (bar)	11.4	18.2
Specific consumption (g/kW/h)	204	174
Weight (ton)	1135	2075
Length (m)	22.52	24.56

"Source: Harrold, 1989, pp 13 and MER, Directory of marine diesel engines, 1996".



Type		RD 90	RND 90	RND90 M		RLB 90		RTA 84		RTA 84 M	
Year		1961	1969	1976	1979	1980		1984		1985	
mep	bar	8.65	10.64	12.29	12.29	14.31	14.79	16.59	12.63	16.58	12.67
p_{max}	bar	76	84	92	94	118	118	125	125	130	130
b_e	g/kWh	208	208	203	193	182	184	171	163	165	158
p	kW/Cyl.	1892	2133	2465	2465	2940	2800	3310	1820	3460	1900
n	rev/min	119	122	122	122	102	94	90	65	78	56
Rating point		MCR	MCR	MCR1	MCR1	MCR1	MCR2	R1	R4	R1	R4

Figure 4.2 The Succession of Sulzer low speed diesel engine designs from the RD series of 1961 to the RTA 2-series engines of 1985 onwards.

"Source: Moor & Brooks, 1989, pp 3".

The tremendous improvement in the engine performance was mainly due to the development and availability of powerful computerised models or the use of sophisticated finite element methods in design. The lower specific fuel oil consumption, higher output power per cylinder and better propeller efficiency are definitely some of the great achievements in the development of modern marine propulsion plants. It is the author's view that this feat has indeed fulfilled the wishes of the shipowners in their search for an economy, greater power, and an efficient marine propulsion plant.

4.2.2 Design Features of the Modern Low Speed Diesel Engine Propulsion Plant

The post oil crisis has seen the development of marine diesel engine design concept gearing towards improved fuel economy, higher engine efficiency and greater output power per cylinder. This era lasted till the early years of the 1980s.

During the later part of the 1980s and 1990s, though engine efficiency is still an important criteria, the emphasis was shifted to focus on engine reliability, durability, extended mean time between overhaul and the environment. The lower and stable fuel prices have also contributed to this new design philosophy.

The major and latest types of marine low speed diesel engines available at the market today are MAN-B&W, MC series and Sulzer RTA series. These modern 2-stroke diesel engines are categorised into two models of large bore and small bore engine. The small bore versions are mainly designed as an alternative to the geared medium speed 4-stroke engine propulsion plants. They are mainly installed on board inland, coastal and deep-sea trades vessels from 2,000dwt to 15,000dwt (The Motor Ship, May, 1994, pp 13). The large bore versions are engine with 500mm to 980mm bore size for MAN-B&W and 580mm to 960mm bore size for Sulzer engine. These engines are mainly fitted on board ocean-going vessels.

Engine Structure

The bedplate of the two types of engines is made of fabricated steel for the large bore model while the small bore, lower power range versions are cast in one piece of cast iron material. The bedplate of the large bore model is made simpler by a single wall box - shaped construction. The high longitudinal and cross girder are fabricated and they provide excellent accessibility for high quality welding. The bearing saddle is made of cast steel and it is welded to the fabricated cross girder. Two stay bolts or tie rod tubes are retained on each side of the bearing saddle. The bedplate could be in one unit or a division of unit. The thrust block is an integral part of the bedplate and this design contributed to compactness and reduction in engine length. The aft most cross girder is specially stiffened to transmit the variable thrust load from the thrust collar to the engine seating.

In MAN-B&W, small size, lower power range version of the S-MC series engine, the chain drive mechanism is incorporated in the thrust block compartment at the aft of a 4-9 cylinder engine. In Sulzer engine, a one-piece gear double columns design is used instead. The bedplate seating is machined without taper if epoxy resin chocks are used while a taper of 1:100 is provided for cast iron chocks fitting. The seating between bedplate and frame box is placed high from the shaft centreline. This enables better compressive load distribution from the stay bolts or tie rods.

The holding down bolts are long and elastic. A single row of bolts secured the bedplate seating to the engine seating. They are tightened by hydraulic tightening tools. The simplified single wall design of the bedplate provides free accessibility for inspection and maintenance work on the bolts. Hence, higher reliability is ensured with the bedplate and the holding down bolts arrangement.

The frame box or monobloc column is basically made up of a number of columns. The frame box is a single welded unit for the large bore model and cast iron unit or monobloc column in the small model. The cast iron crosshead guide could be an integral part of the column or a section bolted between the column two fabricated side frame girders. A column or A-frame is fitted with double crosshead guides. The frame box is tightened to the bottom bedplate and the cylinder frame on top by evenly distributed elastic bolts or screws.

The cylinder frame or cylinder jacket of both engine builders are made of cast iron in unit of two to three cylinders. The one piece or monobloc version are available for the small bore model.

In Sulzer engine, the cylinder jacket of cast iron is redesigned to accommodate the new uniflow scavenging system to provide passage for scavenge ports only. The cylinder lubrication arrangement has also been shifted to the liner top. It no longer passes through the wet cylinder jacket.

The stay bolts or tie rods are generally made of one piece in the small bore model. The material used is low alloyed steel of high tensile strength. They are fitted on each side, port and starboard, of the engine structure, holding the cylinder frame,

frame box and bedplate under compression. The bolts are tightened by hydraulic tools. Each bolt is further secured at the top of the frame box to counter the phenomenon of transverse oscillations.

The modern engine structure has been designed to aim at simplicity and reliability as shown in Figure 4.3. This has contributed to a more robust, rigid, and low stress engine structure altogether.

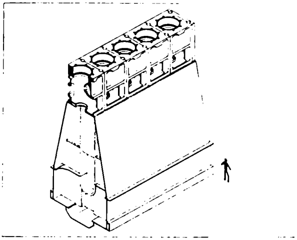


Figure 4.3 Main structure of the RTA84C: the bedplate, columns and cylinder blocks

"Source: Sulzer RTA84C engine catalogue, November 1993, pp 7"

Cylinder Cover

The cylinder cover of both types of engine is made up of one piece. The material is forged steel and bore-cooling principle is applied here. The cover is made deeper with the joint between the cover and liner lowered (Figure 4.4). This design reduces the thermal and mechanical stresses experienced by the cast iron liner. It also enables more freedom of fuel spray in the combustion space and avoided fuel impingement on cover and liner surfaces. It has a central bore for exhaust valve and other bores for fitting of fuel valves, starting air valve, indicator valve and safety valve. The cover is secured to the cylinder frame by hydraulically tightened studs and nuts.

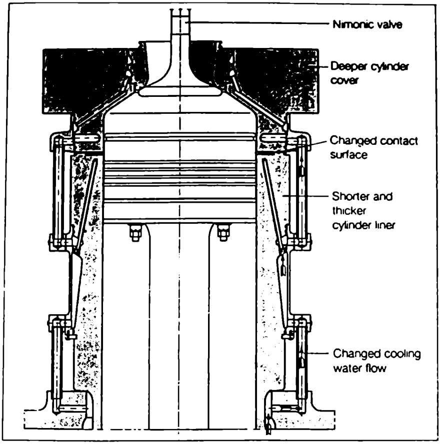


Figure 4.4 The new combustion Chamber design

"Source: Gopi, 1996, pp 87".

Cylinder Liner

The liner material is generally made up of alloyed nodular cast iron. The materials used possess properties of high mechanical reliability, good corrosion resistance, and wear resistance (Sulzer engine liner is of cast iron with hard-phase content included for reduced wear rate). The liner has scavenge ports at its lower portions and drilled holes at its upper regions for two or multi-level cylinder lubrication. The liner is bore-cooled. In order to meet the different cooling intensity of the various engine rating, a special water guide or insulating tubes keeps the cooling water away from the cylinder jacket. Load dependent cylinder liner cooling system is found in the large bore version. In Sulzer engine, the water flow to the liner could be regulated by control valve during full and part load. This design is applied to avoid the dew point of corrosive exhaust gases such as sulphur dioxide, leading to low temperature corrosion, and for the purpose of waste heat recovery. The liner's collar is also stiffened to withstand high loading of gas pressure.

In MAN-B&W engine, a dual cast liner produced by centrifugal casting is used. This is to ensure a good sliding performance on the inner surface and a high material strength on the outer. It features a ferrite-pearlite structure for the outer cast steel layer and an inner cast iron layer of pearlite structure with graphite and hard phase proper. In order to increase its tensile strength, material such as Tarkalloy C is used. An uncooled cylinder frame is applied to the lower portion of the cylinder liner to avoid cold corrosion phenomenon.

Engine Transmission Components

Piston and Piston Rings

The large and small bore pistons are made of two parts; piston crown and skirt. The piston crown of MAN-B&W engine is of chrome molybdenum steel and forged steel in small bore version while the skirt is of cast iron. In Sulzer design, the accommodation of two parts short skirt have been used. A higher piston topland as shown in Figure 4.5, has also been advocated to counter the high mean effective pressure of unit. The crown is tightened to the upper end of the piston rod flange to ensure distortion - free transmission of cylinder combustion pressure. The skirt is bolted to the piston crown underside or mounted to the flange with screws. An improve modified version of the locking arrangement of the nuts and screws has been put into practice. In the large bore model, bronze band has been incorporated to the skirt for running in purposes. The short piston is designed with mainly four ring grooves version. The grooves are chrome plated on both upper and lower surfaces.

The top piston ring of Sulzer and top two rings of MAN-B&W, are of higher height in order to improve ring wear performance under increased pressure. Plasma or chromium surface coating is applied to the top ring which has a preprofiled running face. In MAN-B&W engine, the materials of the rings are alloyed grey cast iron.

The piston crown is bore-cooled. The cooling medium is oil for MAN-B&W engine while Sulzer practises a mixture of oil or water cooled piston depending on the engine type. The jet (downward stroke) and shaker (upward stroke) method of cooling the piston crown are widely used in Sulzer engine. In the oil cooled piston, a system of nozzles fitted on the piston rod provide cooling oil supply through annular passage spraying into the cooling bores on piston crown underside. The oil cleans and cools the piston.

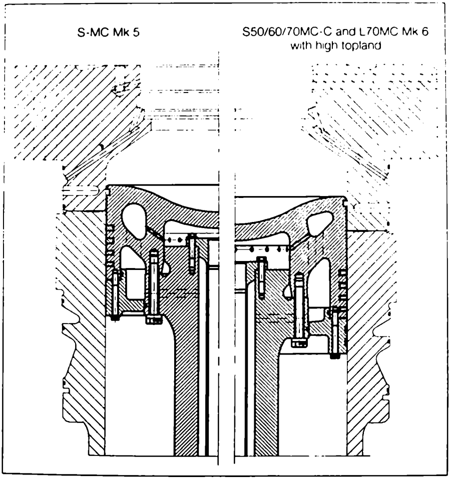


Figure 4.5 Piston pack assembly MAN-B&W MC vs MC-C

"Source: MAN-B&W catalogue: The MC Engines - News and Views, 1996, pp 4".

Connecting Rod

A short connecting rod is used in both engine types in order to reduce the height of the engine. The rod is equipped with an upper palm for the crosshead bearing while the lower portion accommodate the crankpin bearing. The guide shoes are made free or floating in order to minimise unit piston alignment common in crosshead engines.

Crankshaft

In both engine types, the crankshaft is of semi-built up type while the small bore version is made in one piece by forging. In the semi-built up type of crankshaft, shrink-fitting was strengthened to accommodate the higher torque values. The shaft diameter has been increased while its webs thickness reduced. This is to compensate for the shorter length of the engine. In order to reduce the weight of the larger diameter crankpin and journal, bore through pin and journal or welded crankshaft are used in Sulzer and MAN-B&W respectively. Improved vibration damper, tuning wheel and balancing devices such as counterweight has been fitted to the crankshaft to reduce vibration phenomenon and bearing load in the shafting system. A flange at the foremost portion of the crankshaft could be used for this purpose but if the need arise, PTO fitting could be attached thereto.

In MAN-B&W engine, the crankshaft is generally equipped with a thrust collar accommodating camshaft chain drive sprocket rim on its outer circumference. This design reduces the overall engine length. An axial vibration damper is fitted on the free end.

Bearings

The crankpin bearings of Sulzer engine are of the similar design as the main bearings. They are of thick shell, white metal lined bearing for upper and lower half. MAN-B&W engine opted for thin walled, steel shell, white metal lined bearing for both the upper and lower half.

The crosshead bearing is a single lower half full width, thin shell bearing of aluminium tin lined in Sulzer and white metal lined in MAN-B&W engine. The crosshead guide shoes or slippers are white metal lined bearings.

The crosshead bearing is lubricated by hydrostatic lubrication principle where increased oil pressure is used. The ability of the oil to lift the pin hydraustatically through pockets in the bearing for lubrication has indeed enhanced the durability of the bearing.

Camshaft and Its Driving Mechanism

In Sulzer design, the camshaft of full engine length, and increased diameter has been shifted higher in many of the long stroke engine models compared to its traditional mid height position. The main reasons are to overcome the greater pressure losses of fuel pump and exhaust valve, increased cost, and the effect on engine performance. The present design has enabled the attainment of higher injection pressure and increased combustion efficiency. The camshaft of two cylinder segments is provided with fixed cam for the exhaust valve and rotatable cams for the fuel pumps.

The camshaft is driven by a gear wheel from the crankshaft. An additional gear wheel was needed due to the higher camshaft position. The additional cost of this component is compensated by a thinner camshaft arrangement.

In MAN-B&W engine, the camshaft is built in sections and positioned higher to reduce hydraulic pipe length and timing errors of exhaust and fuel valve. Each section is in corresponding to one or two cylinders. They are assembled with flange couplings and is carried in interchangeable bearing shells mounted in the roller guide's housing. The small bore model may have a one piece camshaft depending on the number of cylinders. The cams and coupling can be adjusted, and for maintenance, hydraulic oil could be used to separate them. The cam profiles for fuel and exhaust valve have been modified. An improved and reinforced shrink-fit of cams with better bolts connection has been incorporated into the design.

The camshaft is driven by chain from the crankshaft. One intermediate wheel is used as a chain tightener by the normal chain drive. Guide rails lined with rubber is used to support the chains. Hydraulic chain tightening and damping has been introduced into the design. The auxiliary drive for the cylinder lubricator, starting air distributor and others, are provided from the camshaft.

Various other Fittings

Exhaust Valve

The valve spindle of both engine types are made of heat and corrosion resistant materials - Nimonic 80A alloy with surfaced hardened seat, while its housing is cast iron. The opening of the valve is carried out hydraulically while a special device controls or dampens the closing landing speed. In MAN-B&W engine, the dampened closing is provided by an oil cushion at the spindle top while Sulzer practises controlled valve landing speed by hydraulic 'pushrod' as shown in Figure 4.6. This is to improve valve seat life by reducing the knocking effect. The valve closing is done with an air pressure/spring. Hence, the elimination of a valve spring in the design. The valve rotates (Sulzer - vane impeller or rotation mechanism; MAN-B&W - inclined vanes), during operation and thus avoids local overheating and high temperature corrosion from vanadium. This rotation is possible due to the exhaust gas pressure acting on the vanes of the spindle. The exhaust valve seat is efficiently bore-cooled by water. The spindle guide bush is provided with air sealing and added oil mist. In MAN-B&W design, the exhaust valve housing wall thickness is increased in order to raise its temperature against cold corrosion. The exhaust valve is assembled in a valve cage and this facilitates easy replacement with a reconditioned one during maintenance.

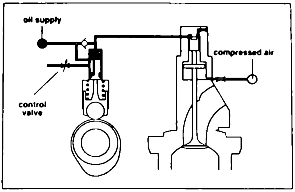


Figure 4.6 Exhaust valve actuating arrangement with the VEC (variable exhaust valve closing) device

"Source: Sulzer RTA84T catalogue, September 1993, pp 16"

Fuel Injection System

In Sulzer design, two to four fuel valves could be used depending on the size of the cylinder. Generally, three fuel valves fitting are popular due to the even distribution of cylinder temperature during combustion. The traditional shrink-fit nozzle has been replaced by a two-piece nozzle design in certain model, (RTA84C). This enables the renewal of the nozzle tip without changing the whole unit. The material of the nozzle tip has been changed to stellite to reduce corrosion on the tip and the atomiser hole edges have been further rounded up internally to avoid cracks at the holes. A modified arrangement of the atomiser holes has also been incorporated.

In both engines, the fuel valve is opened by the high pressure fuel from the fuel pump and closed by the spring. The fuel valve is uncooled but the fuel system is kept warm by fuel circulation and a vent is provided in MAN-B&W to avoid overpressure due to sticking valve spindle. This design also avoided the use of cooling water pipings around the cylinder cover. Both engine builders have also invented a new type of 'variable opening pressure' fuel valve. This fuel valve opening pressure varies with respect to the engine load.

The fuel pump of the Sulzer engine is located at a higher level due to the long stroke of the engine. It is equipped with a pump housing, valve type fuel injection pump and exhaust valve actuators for two cylinder units. The small bore model employs helix-controlled pump standard. The fuel pump is connected to the variable injection timing (VIT) device.

In the MAN-B&W engine, each cylinder is equipped with one fuel pump. The traditional helix-controlled fuel pump is still in use with some modifications. The new 'umbrella type' fuel pump has been introduced in order to provide a sealing arrangement for the pump against contamination of the camshaft lubricating oil system. This allows the employment of the uni-lube oil system for the engine. The fuel pump is equipped with a puncture valve which prevents fuel injection during stopping. It is also linked to the VIT device for optimum operation. The fuel oil high pressure pipe is provided with protective hoses or doubled walled pipe with insulation.

Cylinder Liner Multi-Level Lubrication

In the Sulzer engine the cylinder liner lubrication is equipped with a multi-level accumulator system (Figure 4.7). This system allows the supply of a very small amount of cylinder oil at every stroke of the engine. Hence, an optimum layer of oil film can always be ensured on the liner wall. A simple dual-line distributing system is applied here without the need of a drive shaft. Since the dosage of the oil is already remote controlled, the feed rate can be adjusted according to various parameters.

In the MAN-B&W engine, the cylinder lubrication is carried out with multi-stage cylinder lubricators. These have a built-in capability for adjustment of oil quantity. Each lubricating point is indicated with a sight glass and they are of the sight feed lubricator type. The cylinder feed rate will be proportional to the engine revolution once adjusted. A speed and mean effective pressure dependent lubricator is used for the controllable pitch propeller plant.

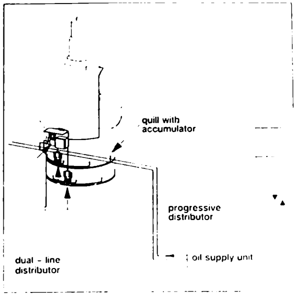


Figure 4.7 Cylinder lubrication arrangements with the Sulzer multi-level accumulator system fed from a dual-line distributor

"Source: Sulzer RTA84T catalogue, September 1993, pp 16"

Engine Reversing Mechanism

The modern engines of MAN-B&W and Sulzer utilises a simple and reliable engine reversing mechanism. In both designs, the exhaust valve cam is not displaced during reversing.

In the Sulzer engine, the fuel pump cams are combined with a small hydraulic servomotor. These fuel pump cams are rotated by the hydraulic servo motor during reversing.

In the MAN-B&W engine, the reversing mechanism incorporates an angular displaceable cam roller in each drive of the fuel pump. The starting air distributor and fuel injection timing is changed by moving the link connecting roller guide and cam roller of each cylinder pneumatically from ahead to astern and vice versa. The system is self locking in the ahead and astern positions. The engine can still be manoeuvred even if one cylinder fails to reverse. The said cylinder fuel pump index is set to zero position then.

Supercharging and Turbocharging

Turbocharging has been considered as the single most important contributor to the technical development of marine diesel engines. It has eliminated the constraint imposed on naturally aspirated engines and added increased power with saving in engine size and weight. Most of all, it has contributed to the superiority of diesel engine over steam turbine as a compact and economical prime mover for marine propulsion plant.

The first turbocharging patent was introduced by Buchi in 1905 but it was only in the 1950s that the idea was applied to large marine 2-stroke diesel engine (Brown, 1985, pp 28). The concept of turbocharging is basically to increase the mass of pressurised charge air into the cylinder. This is to ensure an adequate scavenge air supply for combustion after the scavenging process.

The enhancement of turbocharger design was ultimately aided by gas turbine development in the 1940s (Brown, 1985, pp 30). It has since replaced the use of engine mounted or independently driven reciprocating air pumps and roots type blowers on marine diesel engines. This advocate of turbocharging in pulse and constant pressure system, has indeed pioneered the numerous improvements and modifications in 2-stroke engine design and performance. Cylinders output were boosted to 170% compared to those unsupercharged engines (Brown, 1985, pp 33).

The highly efficient modern exhaust gas turbocharger of today has greatly improved the specific fuel oil consumption and thermal efficiency of marine low speed diesel engines. When incorporated with an energy recovery power turbine in a turbo-compound system, a power turbine output of up to 5% of engine power can be obtained (The Motor Ship, March 1992, pp 40). Thus, an increased in engine fuel economy and an enhanced overall engine thermal efficiency of about 53% (Brown, 1985, pp 35).

Engine Balancers

The reciprocating and rotary motion of the engine, added with a short connecting rod, contributes to high vertical force couple/moment of the second order and vertical and horizontal force couple/moment of the first order. This is especially serious in large size engines with four, five and six cylinders.

In the Sulzer engine, integrated Lanchester type of balancer gears are used. They are fitted to both ends of the engine. In the second order balancer gear arrangement, a free-end balancer driven by camshaft and an after-end balancer driven by the crankshaft are incorporated as shown in Figure 4.8.

In the MAN-B&W engine, moment compensators driven from the crankshaft are fitted. In countering or reducing moments of the second order, two chain wheels with counterweights are fitted along the camshaft chain drive. A similar compensator is employed at the front end. In the case of countering moments of the first order, a large chain wheel with counterweights is used as a chain tightener for the camshaft drive.

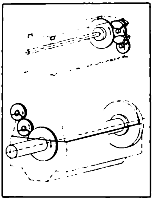


Figure 4.8 Sulzer engine second-order balancer gear arrangement
"Source: Sulzer RTA84T catalogue, 1993, pp 19"

Variable Injection Timing (VIT)

The VIT system basically functions to maintain the maximum cylinder pressure constant by advancing the fuel injection timing. This is active when the engine is operating between 65% to 80% (The Motor Ship, July 1991, pp 55) of maximum continuous rating (MCR). The system is linked to the fuel quality setting (FQS) device.

Variable Exhaust Closing Device (VEC)

The function of this device is to maintain a constant cylinder compression pressure. This is carried out by advanced closing or delayed opening of valve, at part load operation. This has the advantage of ensuring a constant Pmax/Pcomp ratio at a defined load range. The device operates between 80% to 100% of MCR (The Motor Ship, July 1991, pp 55).

The VIT, VEC and FQS are regulated electronically from the control room. The parameter that determines the control of VIT and VEC is the scavenge air pressure. Their combination have contributed to a reduced fuel oil consumption in the usual service load range (Figure 4.9), and the attainment of a higher exhaust gas temperature to be utilised for waste heat recovery system. The optimisation of the operation cycle at the various load ranges has also enabled the engine to tune well to the respective sea condition.

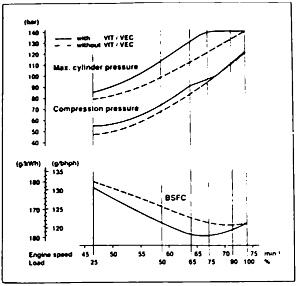


Figure 4.9 Fuel savings in the part-load range with the VIT-VEC functions. At R1 rating with exhaust, power turbine, 27,160kW, 74 rev/min

"Source: Sulzer RTA84T catalogue, September 1993, pp 21".

The development of the modern diesel engines' design concept was mainly attributed to the great effort and co-operation rendered by engine designers, builders and shipowners as described by Wolf (Listewnik, 1996, pp 6):

> 'The further development of the diesel engine is based
> on the three 'pillars' : Theory, experimental analysis,
> and service experience.'

4.2.3 Energy Recovery System

The tremendous rise of fuel prices after the oil crisis has contributed considerably to numerous innovative designs of energy recovery system from the main propulsion plant. However, the optimum operation of these systems would need the fulfilment of stipulated criteria in relation to engine speed or power.

Power Take Off (PTO)

This is a system where a generator is driven by the power taken from the main engine - main engine driven generator. This system has the advantage of saving in fuel where an auxiliary generator could be shut down at sea. However, a standard design would provide full electrical output power from 70% to 100% of engine speed at specified MCR.

In the Sulzer engine, the camshaft gear drive was specially design to accommodate an integrated flange mounted gear box for PTO. PTO driven by crankshaft at the free-end of the engine could also be accommodated. This engine driven generator could operate through a flexible coupling at a speed of 1200 rpm with a compact 'Con-speed' gear box. This design is applicable for the fixed pitch propeller with varying speed. The installation of planetary gear with variable gear ratio in the 'con-speed' gear box, will ensure an output of constant generator speed. The controllable pitch propeller with constant shaft speed will be suitable for conventional constant speed generator PTO configuration.

In MAN-B&W engine, the PTO system are provided with various options in design. The most popular version is the PTO/Renk constant frequency (RCF) type. This side-mounted generator is driven by a crankshaft mounted gear at the forward end. The generator produces a constant frequency based on mechanical hydraulic speed control which varies the gearing ratio. The various components of the system are flexible coupling, step-up gear, epicyclic gear, variable-ratio gear with built in clutch, hydraulic pump and motor, and a standard generator. This version of PTO/RCF is particularly suitable for fixed pitch propeller. However, for controllable pitch propeller application, a compact unit of PTO/GCR (Gear constant ratio) with step-up gear could be adopted.

Power Take In/Turbo Compound System/ Efficiency Booster (PTI/TCS)

The modern high efficiency turbocharger which operates with less exhaust gas, could be utilised to drive the engine and thus reduce the specific fuel oil consumption. This is carried out by the surplus exhaust gas that is being by-passed to an auxiliary gas turbine which generates mechanical energy. This energy is conveyed to the crankshaft through a gear train as shown in Figure 4.10.

The system generally works best when the engine load is approximately 50% (Lustgarten and Moore, 1987, pp 65) or more of its maximum in order to ensure sufficient scavenge air supply for efficient engine combustion during low engine load. A by-pass flap on the auxiliary exhaust gas turbine pipeline automatically closes when the engine is operating below about 50% of maximum load or an equivalent of 25% of maximum power level of TCS. The auxiliary exhaust gas turbine is installed at the forward end of the crankshaft while the gear train is connected to the crankshaft through a flexible coupling.

This system was found to have resulted in a saving of 2 - 5 g/kW/h of fuel during low load conditions in MAN-B&W engines and 5.5 g/kW/h in Sulzer during full load operation. A saving in the electrical load of auxiliary blower for scavenging air during low load, is also quite significant.

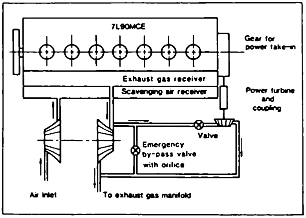


Figure 4.10 Flow diagram for the TCS system of Power Take In on 7L90MCE
"Source: Klintorp and Jacobsen, 1987, pp 4".

Waste Heat Recovery System from Exhaust Gas

The waste heat from the engine exhaust gas is used for the production of steam at the exhaust gas economizer. Traditionally, this steam is used mainly for heating. This practice has changed due to the higher heat energy of the exhaust gas produced by modern diesel engines. The steam produced could be used for the turbogenerator, especially applicable for VLCC. This waste heat recovery is effective during sea passage and it incurs no additional fuel cost. It is also reliable with low maintenance cost. In VLCC, it is estimated to provide an electrical load of about 3.5% to 6% of the main engine output power (Lustgarten, Porchet and Brown, 1992, pp 14). However, the various factors that will influence the production of electric power are the superheated steam temperature, condenser pressure and the turbine efficiency.

Waste Heat Energy Recovery System from Scavenge Air

The relatively high temperature of scavenge air provides a good source for heat energy recovery. However, the requirement for any recoverable heat energy needs to ensure that the air cooler is divided into two or more sections. This is to enable the utilisation of the high temperature section for heat energy recovery while the low temperature portion is used to cool down the scavenge air temperature to an admissible level as shown in Figure 4.11.

In low speed diesel engine, a separate recovery-circulation system instead of a combination type is preferred due to the comparatively low jacket cooling water temperature of the engine. The temperature level could be higher with a separate circulation system. The heat recovered from the scavenge air system will depend on the hot water outlet temperature. Generally, it is in the range of 10-15 per cent of engine power (Engja, 1989, pp 15). The hot water of about 90°C can be used for domestic water, fuel or bunker heating.

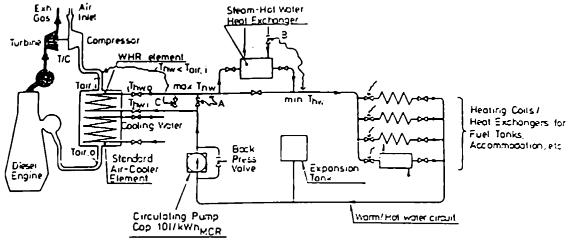


Figure 4.11 Scavenge air heat recovery - warm/hot water flow system

"Source: Engja, 1989, pp 15".

Waste Heat Energy Recovery from Jacket Cooling Water

The heat energy from jacket cooling water is usually in the range of 6-8 per cent of the heat energy supplied to the engine. It amounted to 12-15 per cent of the engine power (Engja, 1989, pp 16). In order to avoid contamination of the jacket cooling water, a separate waste heat circulation system is commonly used. The cooling water temperature of 60°C to 90°C is widely used as heat source for fresh water generator, accommodation and fuel tanks.

The post oil crisis has seen many developments in the areas of energy recovery system. An effective energy recovery system must incorporate a careful plan of the type of equipment needed and its location; most of all, an analysis of the energy balance of the installation and the interval of its operation (Bjorkqvist, August 1985, pp 65).

It is widely accepted that the energy recovery system has indeed contributed to the reduction of ship operating costs. However, with the stable fuel prices today, the trend towards energy recovery concept could be challenged. Additionally, the author believes that the modern engine has indeed reached its peak of output power and fuel efficiency. Thus, the limitation of its surplus energy is inevitable.

Any further effort in the development of energy recovery system could prove futile unless improved methods in curbing heat losses from the engine are carefully scrutinised. The use of ceramic to reduce engine heat loses to coolant could be one of the solution (Ostergaard, August 1985, pp 29).

4.3 Performance Characteristics of the Propulsion Plant

4.3.1 Efficiency of the Modern Low Speed Marine Diesel Engine

The improvement of modern marine low speed diesel engine design concept has indeed contributed to its increased efficiency. Since the oil crisis, we observed a reduction in specific fuel oil consumption of approximately 24% and an increased in propulsive efficiency of about 12% as in Figure 4.12 due to higher stroke bore ratio and low propeller revolution. Hence, a total rise in efficiency of 36% was achieved.

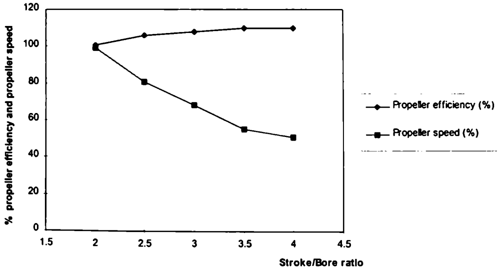


Figure 4.12 Relationship between stroke/bore ratio and corresponding reduction in engine speed from 1970 to 1985

"Source: Ostergaard, 1985, pp 28"

The thermal efficiency of modern diesel engine however, is in the range of 50% to 55% (Lustgarten, Porchet and Brown, 1992, pp 3). This range of efficiency will depend on the design of the engine with reference to engine rating point, type of turbocharger in use and the installation of turbo compound system or efficiency booster power turbine.

Figure 4.13 illustrates the heat balance diagram of a Sulzer low speed diesel engine, operating at a rating of 75% MCR at full speed. The engine is equipped with an efficiency booster system, PTO generator and waste heat recovery system. The engine attained a thermal efficiency of 51.6 % and the total useful energy available was 71.7 %.

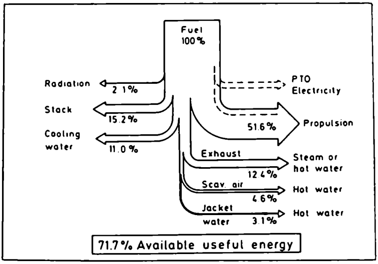


Figure 4.13 Heat balance diagram of a Sulzer RTA84M

"Source: Lustgarten and Moore, March 1987, pp 68 ".

When the first diesel engine was tested in 1897, Mr. Rudolf Diesel forecasted an engine overall efficiency of about 58% (Shipping World & Shipbuilder, March 1997, pp 33). Thermodynamically, the modern diesel engine has indeed reached its peak performance and limit of its efficiency. It is the author's view that the attainment of such high efficiency compared to other thermal engines will put diesel engines in the forefront of all marine propulsion plants irrespective of the output power. Diesel engine will continue to dominate the market share of merchant ship propulsion plant in the foreseeable future.

4.3.2 Reliability and Durability of the Marine Diesel Engine

Reliability and safety came on the scene of the shipping industry mainly due to the many accidents and problems encountered by ships at sea. However, reliability of diesel engines are influenced by many factors. Mr. Lustgarten (The Motor Ship, May 1992, pp 32), stated that:

> 'Reliability can be a mechanical consideration, but it can also be influenced by the type of operation involved, by maintenance systems, by the crew and by the owner's philosophy.'

Reliability and durability will certainly affect the performance of any propulsion plant eventually. The factors that influenced engine reliability and durability are shown in Figure 4.14. The trend towards an 'intelligent engine' concept today with minimum manning and 'high tech' equipment incorporating electronic control on board, will definitely demands an engine with high reliability and durability in its entire set-up.

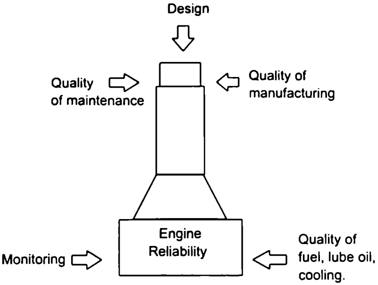


Figure 4.14 Factors influencing reliability and durability

"Source: Lustgarten, 1988, pp 21".

The features of high reliability and durability can be seen in the strong and rigid engine structure where the cylinder, frame and bedplate are firmly kept under compression by the tie rods or stay bolts. The compressive stresses are strictly limited to these vital components.

They have also extended the mean time between overhauls of many of the major engine components to about four years. The exhaust valves design of conventional and nimonic material with steel bottom piece and surface hardened seat, has achieved a mean time between overhaul of about 25,000 running hours without overhaul. The cylinder liner wear rate has been reduced to 0.05 mm/1,000 hours. This was mainly attributed to multi-level cylinder lubrication, improved materials and optimisation of liner surface temperature. A reasonable unit overhaul could be extended to 6,000 - 8,000 hours due to improved wear resistant of plasma coated, pre-profiled thicker top ring and low wear rate of piston with higher topland and an improved method of separation and drainage of moisture in scavenge air. The highly efficient design concept of bore-cooling has also contributed immensely to reduced mechanical and thermal stresses in spite of increased output power.

The improvement in material technology, engine lubricant properties and fuel treatment equipment were the few main criteria that contributed to the reliability and durability of modern marine diesel engines. An improved engine monitoring system with sophisticated electronic equipment has also played a vital role in this context.

The reliability and durability factors have indeed contributed to an era of simple engine maintenance concept, and longer overhaul intervals of main components. It is an enormous achievement of modern engine design and manufacturing philosophy compared to the pre oil crisis years.

Nevertheless, it is the author's opinion that the reliability and durability of modern marine diesel engine remain an issue of basic engine design. A sound basic design will always ensure the attainment of reliability and durability of the propulsion plant.

4.4 Operation and Maintenance Features of the Propulsion Plant

4.4.1 Operation Features of the Propulsion Plant

The post oil crisis has seen a greater use of automation and sophisticated shipboard computers in the operation of the propulsion plant. The plant's operations are made more reliable, simple and easy to operate by shipboard staff.

The performance of the engine could be monitored by the use of a computer such as that of the Sulzer or MAN-B&W design of computer aided performance analysis device (CAPA). This has the advantage of relieving the chief engineer of his manual calculation of the engine performance parameters and also provide a uniform report in all circumstances. When presenting the performance results with the expert diagnosis system, it enables the engineers to detect any deviation in the operation parameters. The Sulzer design of integrated piston ring wear detecting arrangement with trend processing, (SIPWA-TP), provides information on average ring wear, and other details of ring condition. This invaluable information could assist the engineers in the planning of maintenance intervals and optimisation of cylinder oil consumption of the propulsion plant.

The trend towards reduced manning in modern ships has resulted in the greater emphasis on UMS. However, the operation of UMS ships has always been looked upon with scepticism. The reason could be due to the lack of reliability of the automation system installed in the engine room. According to D.Paro (IMAS 89, 1989, pp 21):

> 'The future engine room is, of course, unmanned. Today, operational staff and the shipowner's technical departments disagree about just how unmanned the engine room should be. The major reason for these differences is probably that the system and devices in the engine room do not work as reliably as expected'.

However, in this modern era of the 1990s, the suspicion of the concept of UMS has gradually been overcome. The reason could be attributed to the improvement in the reliability of machinery components and the availability of a sophisticated condition monitoring system for the main propulsion plant and other engine room machinery. This has especially been proven with the practice of reduced manning in majority of the modern ships today.

The manning level of the crew today would lie between eighteen to twelve persons depending on the type, trade and other particulars of the ship. The great leap in technology development in terms of machinery automation demands higher qualifications from the crew members. The introduction of modern integrated bridge control and operation of the propulsion plant, together with the employment of dual purpose crews are strategies geared towards reduction of shipboard manning costs. Simply, the 'high tech' ships of today are manned by a small group of highly qualified and experienced seafarers.

4.4.2 Maintenance Concept of the Propulsion Plant

The modern propulsion plant maintenance concept has shifted enormously to the use of computer-based packages. This concept provides support for the maintenance, spare parts planning and administration of engine room machinery. The features include spare parts inventory control, purchasing, statistical reporting, work orders and maintenance history recording.

The greater demand in the use of on board computers has contributed to a reduction in the chief engineer's workload. The philosophy of 'paperless administration' especially on the shipboard maintenance concept, could indeed be realised with the modern advance data communication systems. The application of modern sophisticated conditions and performance monitoring system on board would also result in the shift of shipboard maintenance concept to shore based maintenance. However, this concept will be influenced largely, by the type and trade patterns of the ships.

It is the author's view that, the drawbacks of the modern operation and maintenance concept lie mainly in the computer system itself. Many of the systems are not user friendly and are inflexible to the owner's requirements. The advancement of software development also renders most of the system obsolete by the time they are fitted on board. Nevertheless, the system does contribute to certain improvement as stated by Kyrtatos (1996, pp 143):

> 'However, it is undoubtedly true that such computer-based machinery maintenance support systems, assist the improvement of availability, by decreasing mean time to repair (MTTR) and the mean

preventive maintenance time and increasing mean time between failure (MTBF)'.

CHAPTER 5

Impact of the Present Marine Propulsion Plant on Global Shipping and the Environment

5.1 Selection of the Propulsion Plant

5.1.1 Shipowner's Option

The prime concern of a shipowner in the area of ships' management is to ensure that the fleet has a long life record and lowest life long cost. This characteristic simply demands not only a good quality ship but also, up-to-date technical and operational capabilities. Simply, the features should not render the propulsion plant obsolete within a short period of the ship's lifetime. In today's shipping market, the owners are given the freedom in the selection of engine ratings based on the contract maximum continuous rating, power and speed. The wide range of engine speed has enabled the precise matching to the optimum propeller speed and thus contributes to the higher propeller efficiency. The availability of the wider scope of engine power also provides the preferred choice between lowest specific fuel oil consumption and lowest initial cost. These information could be easily obtained from the 'layout field' shown in Figure 5.1, provided by the engine builders. According to Lustgarten and Moore (1987, pp 63):

> 'Taking the example of a 40,000 dwt bulk carrier to sail at 14 knots with a maximum possible diameter of 6.75m, a total of 15 different RTA engines are available. Of these, the solutions with lowest daily fuel consumption (6RTA62), lowest capital costs

(4RTA62), lowest combined fuel and capital cost (7RTA52) or as a basis, the most popular engine model (6RTA58) can be selected for closer study, depending upon the specific priorities of the potential customer'.

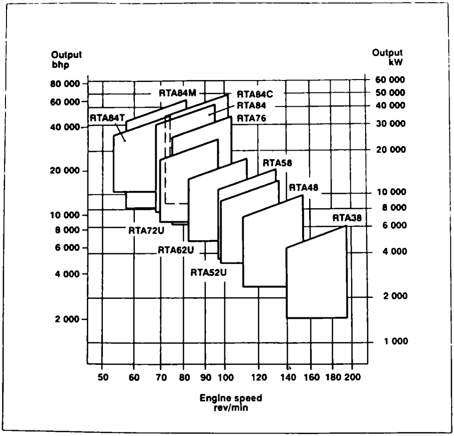


Figure 5.1 Power and speed range of Sulzer RTA-series
"Source: New Sulzer Diesel catalogue, November 1993, pp 4".

5.1.2 Engine Designers and Builders' Philosophy

The post oil crisis of 1973 has seen many changes to the development of the marine low speed diesel engine's design philosophy. The modern engine is quite different from its predecessor in the late 1970s (Lustgarten, 1988, pp 3). The factors that contributed to this rapid and intensive development is illustrated in Figure 5.2.

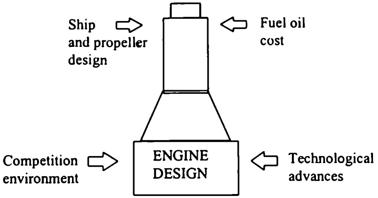


Figure 5.2 Factors influencing the engine design

"Source: Lustgarten, 1988, pp 13".

The modern engine designers and builders are driven by the above factors to seek a more efficient, reliable and appropriate modern manufacturing technology against a ships built in the 1970s. A ship of today can be propelled at the same sailing speed as its predecessor in the 1970s but with much lower specific fuel oil consumption. The basic overall economy of the plant remains the main emphasis. The overall plant economy covers various aspects of fuel economy and investment costs, reliability of operation at sea accompanied with durability and minimum maintenance.

In a modern engine designer's philosophy, the customers' needs are always taken into consideration. Various steps have been taken to accommodate the customers' requirements and thus the concept of a standard ship design and building is always in jeopardy.

In the search for economy and cost effectiveness, engine designers have also piloted various concepts of re-rated engines for higher output power. These concepts of modification of the combustion chamber, scavenging air and turbocharging system have contributed to a lower engine production cost. Hence, a larger and more expensive engine for higher power is frequently not an acceptable solution today.

5.2 Environmental Impact of the Propulsion Plant

5.2.1 Exhaust Gas Emission

Marine low speed diesel engines have been criticised for their noxious exhaust gas emission world-wide. The contents of this emission are mainly composed of the by-products from combustion. The composition of exhaust gas at the stack are made up of nitrogen, oxygen, carbon dioxide, water vapour, sulphur oxide (SO_x), nitrogen oxide (NO_x), partially-reacted and non-combustible hydrocarbon and particulates. Their detail constituents are illustrated in Figure 5.3.

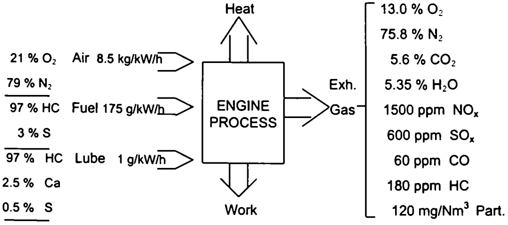


Figure 5.3 Typical emissions from a MAN-B&W MC low speed engine
"Source: MER, February 1997, pp 14".

Among the various compositions of exhaust gases emission, NO_x, SO_x and carbon dioxide are considered harmful to mankind, vegetation and the environment. The ability of the modern low speed engine to burn low grade fuel at a high cycle temperature has further contributed to this problem of emission. This has resulted in the introduction of various innovative designs of the main propulsion plant component parts and additional devices to curb the 'high' emission of nitrogen oxide especially. The emission of carbon dioxide could be controlled with high thermal efficiency of the engine. Bunker, with reduced sulphur concentration, and the use of water washing in the scrubber, would contribute to a reduction in SO_x emission. However, the drawbacks are the availability of space and high cost of the scrubber. Moreover, the price of bunker with low sulphur content would be much higher.

The control of NO_x emission could be carried out by primary and secondary methods. The primary method involves emission control taking place inside the engine or during the combustion process, while the secondary method is carried out by passing the exhaust gas through a gas treatment 'plant'.

The various options available for primary emission control are :

- retarding the injection timing (Reduced Pmax).
- the use of water emulsified fuel.
- injection of water into the intake air.
- increasing air boost pressure.
- higher compression ratio.
- decreasing air intake temperature.
- the use of special fuel valve nozzle (Slide valve).
- recirculation of exhaust gas (EGR).

The objective of the primary control method is to ensure the reduction of NO_x through cylinder combustion control. The generation of NO_x is influenced by temperature and oxygen concentration during the combustion process. A higher reading of NO_x is obtained with increased cylinder temperature and longer resident time of the exhaust gas at high temperature. Hence, the modern low speed engine generates more NO_x compared to other heat engines (medium speed engine, steam and gas turbines) of a similar output. The obvious reason is due to its higher combustion temperature since NO_x generation is mainly a function of the maximum cylinder combustion temperature. A comparison of NO_x emission against the medium speed engine is shown in Table 5.1.

Table 5.1: Emission factors (g/kW/h) for marine engines under steady-state conditions

	Low speed engines	Medium speed engine
Nitrogen oxide (NO_x)	18.7	13.8
Carbon monoxide (CO)	2.1	1.8
Hydrocarbon (HC)	0.5	0.6
Sulphur dioxide (SO_2)	21.0 X S	21.0 X S

Note: (S: Sulphur content of fuel/ % by weight)

"Source: MER, February 1997, pp 14".

The disadvantages of practising or installing certain options of the primary control method on the main propulsion plant are an increased in specific fuel oil consumption and high carbon emission. There could also be a reduction in engine reliability and durability. However, a combination of the various options in the primary control method is shown in Figure 5.4. This would generally be sufficient to meet the requirements stipulated by IMO speed-dependent NO_x emission limit shown as follows:

1) 17.0 g/kW/h ..... when n is less than 130 rpm

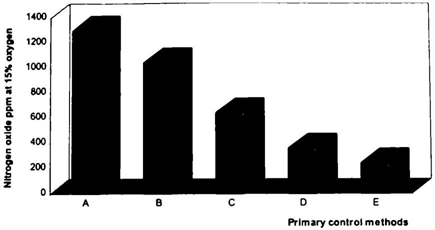
2) $45n^{(-0.2)}$ g/kW/h ..... when n is 130 rpm or more, but less than 2,000 rpm

3) 9.8 g/kW/h ..... when n is 2,000 rpm or more

Note: n is rated engine speed (crankshaft revolution per minute)

The test procedure and measurement methods to be in accordance with the NO_x Technical Code.

"Source: The Motor Ship, February 1997, pp 39".



Note: **A**: Reference. **B**: Slide valve. **C**: Slide valve + 50% water + Reduced Pmax.

D: Slide valve + 50% water + 20% EGR.

E: Slide valve + 50% water + 20% EGR + Reduced Pmax.

Figure 5.4: Effect of combining various NOx reduction measures on a MAN B&W 5S70MC engine

"Source: MER, February 1997, pp 16".

The secondary emission control method uses an external gas processing 'plant' commonly termed the selective catalytic reduction (SCR) system (Figure 5.5). In this system, ammonia is added to the hot exhaust gas at the upstream of the catalytic converter. The subsequent reaction of the NOx gas with ammonia, in the presence of a catalyst in the converter, will result in the formation of nitrogen and water vapour. Both of these gases exist naturally in the atmosphere and thus they are harmless to human health. Additionally, the SCR reactor also removes certain amounts of soot and hydrocarbon through the process of oxidation.

The most effective method to curb the problem with shipboard exhaust emission would be the adoption of a combination of various primary and secondary emission control methods. A combination of these control methods were found to have further reduced the amount of nitrogen oxide emission to a much lower limit. This is especially critical when a stricter emission control is imposed by the local or regional authority (MER, February,1997, pp 19). The economic selection of a suitable emission control method will eventually lie with the shipowner. The decision is indeed vital in view of the proposed forthcoming IMO regulation (Annex VI of Marpol) and also the States' legislations on marine exhaust gas emission.

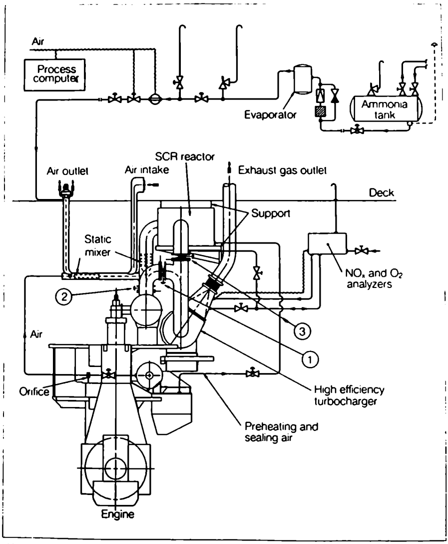


Figure 5.5 Schematic layout of SCR system for a low speed engine

"Source: MAN-B&W catalogue, December 1996, pp 10".

CHAPTER 6

Conclusions

The development of merchant marine propulsion plant has a history of more than a few centuries old. The main prime movers for the merchant ships are steam turbines and diesel engines. Gas turbines came into the scene in the 1950s. However, they are generally popular among the navy ships due to their compactness and higher power capability.

At the beginning of this century, we saw the dominant role of steam turbine as the main prime mover for the merchant ships. Rudolf Diesel's invention of the diesel engine in 1897 was not able to compete against the supremacy of steam turbine then, though it has already achieved an engine efficiency of more than twice that of the steam engine and with better fuel consumption. However, in 1912, the *Selandia* created history for her unique role as the first ocean going ship that was successfully powered by two large marine diesel engines. It was as a result of this great feat that diesel engines began to pose a serious threat to the supremacy of steam turbine as the leading marine propulsion plant.

In the late 1950s, the advent of supercharging and turbocharging added another milestone to the development of diesel engines. The switch to turbocharging has further increased the engine output power per cylinder. This also marked the era of diesel engines entry into the higher power range of marine propulsion plants. During the trade boom of the 1960s, many of the world VLCC, ULCC and large container ships were dominated by the high power steam turbine. Diesel engine still does not have the power availability for this range of vessels then.

The steam turbine dominance on the high power ships was brought to a close with the dawning of the oil crisis in 1973. The high cost of marine fuel simply made it too expensive to operate these ships due to the high specific fuel oil consumption of steam turbines as mentioned in Chapter 3. This technology of over a century old could not have seen much further improvement in areas of fuel economy and efficiency. The many technological advances achieved has since benefited the land based steam turbines instead.

Prior to the oil crisis in 1973, the development of marine propulsion plant was aim at higher output power per cylinder. An increase in engine power (EP) will require the same increase in either the mean effective pressure (P), stroke length (L), cylinder bore area (A), or engine speed (N). Their relationship can be simply expressed as; EP = PLAN. The emphasis on higher cylinder output power was continued after the oil crisis years though engine's fuel economy was view as the main agenda then. The high bunker fuel's prices from the aftermath of the oil crisis have particularly contributed to this development trend. After a lapse of about ten years, the introduction of long stroke and superlong stroke low speed diesel engines with high stroke to bore ratio was seen as the ultimate solution to counter the high bunker fuel's prices. This modern diesel engine has managed to attain a reduction in the specific fuel oil consumption by about 24%, an increase in the engine mean effective pressure and output power per cylinder of over 60% and 100% respectively compared to its predecessor. It is also able to digest the poor quality residual fuel from the aftermath of the oil crisis. The lower engine revolution has further enhanced the propeller efficiency and provide more time for an efficient cylinder combustion to take place.

However, this long stroke and superlong stroke diesel engine demand a higher engine headroom due to its high stroke/bore ratio of about 3.75 - 4.0. This is the main drawback for ships where headroom is a critical factor. The engine structure is robust, rigid and low stress with emphasis on simplicity and reliability. Engine balancers are installed to counter the phenomenon of vibration as explained in

Chapter 4. The improvement in material technology for piston crown, cylinder liner and cylinder cover with accompanying efficient bore-cooling concept and multi-level cylinder liner lubrication has also managed to ensure the reliability and durability of the combustion chamber component parts. The engines are all fitted with a central exhaust valve, hydraulically and pneumatically operated. The concept of uniflow scavenging is strictly adhered to in order to ensure its thermodynamic advantage at high stroke/bore ratio. The modern low speed diesel engine has indeed undergone many tremendous improvements and modifications compared to its predecessor. The numerous feats that it has achieved in areas of fuel economy, output power per cylinder, propeller efficiency, reliability and durability have since contributed to its dominant role as the major prime mover for merchant ships.

These immense achievements could be attributed to the availability of advance technological know-how especially in areas of computer modelling or finite element methods in determining the stresses and strength of the engine component parts. The endeavour of sophisticated engine test-bed trial, improvement in material technology and high quality manufacturing techniques were also some of the major contributing factors leading to the evolution of the modern low speed diesel engines.

The oil crisis in 1973 was seen as the main driving force behind this entire evolution. It is the author's view that the conservative nature of the shipping industry has somehow contributed to this development trend. The industry is not pro-active to the changing trend of technological development but only react after the strike of an incident or disaster such as the energy crisis.

The thermal efficiency (η) of heat engines which convert heat energy into mechanical work, can be expressed as; $\eta = 1 - (T_{low} / T_{high})$, which is closely related to the cylinder operating temperature (T). The modern low speed diesel engine has a thermal efficiency of about 51%. This is mainly attributed to its high operating temperature. Any further increases in the operating temperature will require an improved material technology in order to withstand the greater thermal stresses imposed on the engine component parts. Unless this dilemma is resolved, the modern low speed diesel engines would not be able to attain a higher range of thermal efficiency in the foreseeable future even though it has an advantage over the steam turbine in the limitation of temperature. However, a higher overall engine efficiency is obtained by its accompanying energy recovery system from the exhaust gas, cylinder cooling water, scavenge air and the turbo-compound system as explained in Chapter 4. The modern low speed diesel engine is indeed regarded as the most efficient thermal engine for marine propulsion plant.

Lately, the development of the merchant marine propulsion plant has also undergone some dramatic and interesting changes in the latest version of the Sulzer RTA96C and MAN-B&W K98MC-C high power diesel engines of 5490 - 5710 kW/cyl as shown in Table 4.1 and 4.2 of Chapter 4. The shift towards lower stroke/ bore ratio of 2.4 - 2.6 with a higher engine minimum speed of 90 - 96 rpm is advocated in these engines. It is the author's view that the changing trend is mainly geared towards compactness by the reduction in engine specific weight. Hence, a reduction in the engine headroom requirements with greater cargo space led to the possibility of increased revenues. The higher engine speed advocated is closely related to the lower stroke/bore ratio employed as indicated in the engine power (EP) expression earlier. Nevertheless, the stable fuel prices of today is also another factor which could has prompted to this development trend of the latest marine low speed diesel engine.

The oil crisis in 1973 has also subsequently triggered another alarming issue of the environment after the development of the low speed diesel engine. The engine is found to has a 'high' concentration of NO_x and SO_x emission from the exhaust. It is the author's view that the environment is indeed a critical issue to address. Hence, no effort should be spare to curb the exhaust emission from the engine even though it would jeopardise the performance characteristics of the highly efficient marine low speed diesel engine.

Bibliography

Aeberli, Kaspar (1994). 'Towards The Intelligent Engine'. *16th Annual International Marine Propulsion Conference* (10-11 March 1994: Mount Royal Hotel London). Sutton, Surrey: Reed Business Publishing Ltd & First Europe Communications.

'Air Pollution: Exhaust Gas Monitoring' (1997). *Marine Engineers Review (MER), February 1997*, page 12.

B&W Engineering Publication, (1988). *B&W Constant Pressure Turbocharging.* Copenhagen: B&W Engineering.

Brown, David T (1985). *A History of The Sulzer Low Speed Marine Diesel Engine.* Winterthur, Switzerland: Sulzer Brothers Ltd.

British Petroleum Company (June 1992-June 1995) . *Statistical Review of World Energy.* London: British Petroleum Company.

Bram, Leon. L & Dickey, Norma. H (1971). 'Petroleum'. *Funk & Wagnalls New Encyclopaedia*, Volume 20. United States of America: Funk & Wagnalls Corporation.

Bowden, J. K (Reprinted 1984). *Southern's Marine Diesel Oil Engines.* Tenth Edition. Great Britain: James Munro & Company Ltd.

Bjorkqvist, T (1985). 'Energy Recovery Lower Cost'. *The Motor Ship, August 1985.* pages 64-67.

Bina, Cyrus (1985). *The Economic of The Oil Crisis: Theories of Oil Crisis, Oil Rent, and Internationalization of Capital In The Oil.* New York: St. Martin's Press

'Choice Widens for Small Ship Propulsion', (1994). *The Motor Ship, May 1994*, pages 13-17.

Cockett, Neil. M (1987). 'The Bunker Market In Perspective'. *The Motor Ship Ninth International Marine Propulsion Conference*, (March 12-13 1987: Royal Lancaster Hotel London). Sutton, Surrey England: Reed Business Publishing.

'Directory Of Marine Diesel Engine' (1996). *Marine Engineers Review*, pages 1-36.

'Diesel Technology', (March, 1997). *Shipping World & Shipbuilder*, pp 32 -36.

Donald, K. M. B (1977). *Marine Steam Turbines*. Marine Engineering Design and Installation Series. London: Marine Media Management Ltd.

Ewart, W. D (1982). *Bunker: A Guide for The Ship Operator*. Ship Management Series. London: Fairplay Publication.

Engja, Hallvard (1989). 'Waste Heat Energy Recovery'. *Western European Graduate Education Marine Technology*, Vol. 8, pages 1-17.

'Engine Developments: Four-Legged Testbed Solves Sulzer's Problems', (1991). *The Motor Ship, July 1991*, pages 53-55.

'Engine Developments: Designers anticipate engine emission controls' (1992). *The Motor Ship, August 1992*, pages 26-31.

'Engine Emissions: The Green Diesel' (1997). *Marine Engineers Review, February 1997*, pages 14-19.

Exxon Background Series (1976, 1980, 1984). *Middle East Oil*. New York: The Corporation.

Gopi Mohan, Saha (1996). *The Technical and Economical Aspects of Marine Engine Selection.* Msc dissertation. Malmö Sweden: the World Maritime University.

Harrold, A. F (1989). 'Development of Merchant Ship Propulsion Machinery Over The Past 25 Years'. *Transaction Institute Of Marine Engineers,* Vol. 101, pages 1-16

'IMO Target Marine Exhaust Emissions', (February, 1997). *The Motor Ship,* pp 39.

Institute of Marine Engineers, (1971). *Proceedings Of a Conference On The Theme Factors In The Selection Of Marine Machinery and Plant With Particular Reference To Reliability, Maintenance, and Cost.* London: Institute of Marine Engineers.

Institute Of Marine Engineers, (1965). *The Running and Maintenance of Marine Machinery.* Fifth Edition. London: The Institute of Marine Engineers.

Ishigai Seikan, Akagi Shinsuke (1980). 'Marine Fuel Conservation'. *Bulletin of The Marine Engineering Society of Japan* , Volume 8 No.4, pages 311-318.

Ishii, I (1996). 'A Half Century With Marine Diesel Engine - Preprint'. *The Institute Of Marine Engineers.* Pages 1-15.

Knack, Christen (1990). *Diesel Motor Ship's Engine and Machinery.* London: Marine Management (Holdings) Ltd. for the Institute Of Marine Engineers.

Kilchenmann, W.A. (1973). '75 Years Continuous Development of Diesel Engines'. *Sulzer Technical Review* - The 42nd Andrew Laing Lecture 1973, Vol. 4/1974, pages 61-76.

Kyrtatos, Nicholas. P (1996). 'Improved Availability of Ships Through Advanced Communication-Based Technical Support From Shore'. *The Motor Ship 18th Annual International Marine Propulsion Conference* (21-22 March 1996: Radisson SAS Hotel Hamburg). London: First Conference.

Klintorp, Henry & Jacobsen, Birger (1987). ' Engine Room Energy Optimisation'. *The Motor Ship Ninth International Marine Propulsion Conference* (March 12-13 1987: Royal Lancaster Hotel- London). Sutton, Surrey England: Reed Business Publishing.

Lamb, John (1965). *The Running and Maintenance of The Marine Diesel Engine.* Sixth Edition-Rewritten. London: Charles Griffin & Company Ltd.

Lebech, Mogens (1959). *B&W Museum.* Copenhagen: O.G. Olsen & Co. A/S.

Listewnik, Jerzy (1996). 'Development Aspects of Sulzer Marine Diesel Engines'. *Ship Machinery and Automation.* Hand-outs. World Maritime University, Malmö, Sweden.

Lustgarten, G. A, Porchet, Federic & Brown, David. (1992). 'Optimum Propulsion Solution for VLCCs'. *14th International Marine Propulsion Conference* (12-13 March, 1992: London, UK). Sutton, Surrey: Reed Business Publishing Group.

Lustgarten, G. A & Moore, M (1987). 'Low Speed Diesel Engines for Overall Propulsion Economy'. *The Motor Ship Ninth International Marine Propulsion Conference* (March 12-13 1987: Royal Lancaster Hotel- London). Sutton, Surrey England: Reed Business Publishing.

Lustgarten, G. A (1988). 'The Sulzer 2-Stroke Low Speed Engine'. *The Motor Ship 10th International Marine Propulsion Conference* (March 10-11 1988: Royal Lancaster Hotel-London). The Motor Ship.

Marquez, Jaime. R (1984). *Oil Price Effects and OPEC's Pricing Policy: An Optimal Control Approach.* Lexington Massachusettes, Toronto: Lexington Books D.C. Heath and Company.

Marine Propulsion Systems Proceedings of a Symposium (23 and 24 May 1974: Liverpool Polytechnic, England). Edited by Wakefield, S. B. London: Marine Media Management Ltd.

MAN B&W Publication, (1996*). Emission Control Two-Stroke Low Speed Diesel Engines*. Copenhagen: MAN B&W Diesel A/S.

MAN B&W Publication, (1996*). The MC Engines - News and Views.* Copenhagen: MAN B&W Diesel A/S.

MAN B&W Publication, (1994). *Two-Stroke MC Engines*. Copenhagen: MAN B&W Diesel A/S.

Moor, M. & Brooks, C.R. (1989). 'Economy & Reliability- Who Pays'. *The Motor Ship 11th International Marine Propulsion Conference* (March 9-10 1989: Hotel Russell, London). The Motor Ship.

New Sulzer Diesel Publication, (1995). *RTA Two-Stroke Marine Diesel Engines.* Winterthur, Switzerland: New Sulzer Diesel Ltd.

New Sulzer Diesel Publication, (1993). *Sulzer RTA84C Engines Upgraded for Modern Containerships.* Winterthur, Switzerland: New Sulzer Diesel Ltd.

New Sulzer Diesel Publication, (1993). *Sulzer RTA84T The Modern VLCC Engine.* Winterthur, Switzerland: New Sulzer Diesel Ltd.

Newbery, P.J (1996). 'Ongoing Development of The International Marine Fuel Standard'. *Transaction Institute of Marine Engineers,* Vol.108, Part 4, pages 241-257.

Nicholas, D. G (1990). 'Review of The Marine Steam Turbine Over The Last 40 Years'. *Transaction Institute of Marine Engineers*, Vol. 102, Part 1, pages 83-104.

Nicholas, D. G (1995). 'A Brief History of The Marine Steam Turbine'. *Transaction Institute of Marine Engineers*, Vol. 108, Part 1, pages 67-88.

Odell, Peter. R (1975). *Oil and World Power: Background To The Oil Crisis.* Fourth Edition. Harmondsworth; Baltimore: Penguin Books

Ostergaard, A (1985). 'Limits To Efficiency'. *The Motor Ship, August 1985,* pages 28-29.

Paro, D (1989). 'A New Era in Ship Propulsion'. *The Institute of Marine Engineers International conference IMAS 89* (24-26 May 1989: Athens). London: Institute of Marine Engineers by Ma.

Petersen, Erik. A (1997). Letter to The Author. Marketing Department, MAN-B&W, Copenhagen, Denmark.

'Reliability Remains Owners' Priority' (1992). *The Motor Ship, May 1992*, pages 30-32.

Sair, K. F (1981). 'Development of Marine Diesel'. *Marine Engineers Review, December, 1981*, pages 6-8.

Skeet, Ian (1988). *OPEC. Twenty Five Years of Prices and Politics.* Edited by Eden, Richard. Cambridge; New York: Cambridge University Press.

The Motor Ship (1979). 'Marine Propulsion and Future Fuel'. *Proceeding of International Conference* (March 1-2 1979: London). London: IPC Business Press for The Motor Ship.

'Turbochargers: Increased Pressure Ratios, Reduced Capital Cost', (1992). *The Motor Ship, March 1992*, pages 36-42.