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Technical and economic aspects about marine engine selection

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WORLD MARITIME UNIVERSITY

Malmo, Sweden

TECHNICAL AND ECONOMICAL ASPECTS ABOUT MARINE ENGINE SELECTION

By

LIU FU-SHENG

THE PEOPLE'S REPUBLIC OF CHINA

A dissertation submitted to the World Maritime University in partial fulfillment of the requirement for the award of the:

Degree of Master of Science

in

Marine Education and Training in Engineering

Year of Graduation

1991
DECLARATION

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

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

Supervised and assessed by:

M. Kimura
Professor
World Maritime University

Co-assessed by:

J. Listewnik
Visiting Professor
World Maritime University
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Most special heartful thanks to my wife Lang Dan-Ping and my daughter Liu Yue for their endless love and support.
The selection of a marine engine depends on the target market and the expectations for ship design trends.

Marine propulsion plants have, in recent years, been the subject of rapid development. Not only has thermal efficiency been increased to well over 50%, but reliability has likewise been improved. The operational behaviors and environmental factors have also been enhanced. Therefore, the selection process is mobile and complicated.

It can be said that the two-stroke slow speed engine and four-stroke medium speed engine have all come to perfection on the technological side. Their development is along the lines of the improved reliability, simplification and consideration of the environment. The overall evaluation between slow and medium speed engines can get the results that the slow-speed engine is still the optimum in most of the comparable propulsion range.

This dissertation is based on the description above regarding the development of the marine diesel engine, considering in some respects of the technical and economical aspects, in order to give the shipowner general guidance in marine engine selection, based on the MAN B&W MC Programme. The project can also be used for marine engineers and technicians on board ships, in shipyards and other relevant units.
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CHAPTER 1

INTRODUCTION

1.1 Necessity

1.1.1 Ship-building market

The future volume of shipbuilding and the types of vessels to be built are dependent first and foremost on world economic development and the resultant transport requirements.

As various studies show, there is little cause to anticipate a significant increase in freight volume in the next few years despite the sharp increase in the world's population. If we also bear in mind the existing over-capacities in tonnage, there are no grounds for expecting a significant increase in new-building activity in the near future.

However, a certain upswing can be expected in the 90's when the large numbers of vessels built in the 70's will be gradually replaced.

The statistics of the installed power in ships above 2000 DWT show that diesel engines are now the exclusive propulsion machinery for merchant ships, with only a few exceptions. Assuming that no radical invention completely revolutionizes the propulsion sector, it can be assumed with considerable certainty that the diesel engine will continue to remain the first choice as a propulsion engine for merchant ships for the foreseeable future.

Therefore, it can be certainly said that the prospect for ship-building, with the diesel engines as the main propulsion plants, is magnificent and bright.
1.1.2 Main propulsion plants

With the development in high technology and science what are the new standards and requirements for the next generation of ships and their main propulsion plants? Studies in recent years have shown that the next generation of ships will probably be designed and built for quite drastically reduced manning. Anything in excess of "the key and the throttle" will be considered unnecessary for operating the main propulsion system.

This generation of ships demands main propulsion engines with the following characteristics:
- Reliability
- Simplicity
- Cost-effectiveness
- Comfort
- Environment

Catching up with the modern requirements of marine engines for the new generation of ships has been the target of all engine-makers. A lot of statistics have also shown that most of the engine-makers have even been making a series of technical improvements to meet the rigorous market requirements. It also creates, in engine-making, a competitive situation between slow speed and medium speed engines.

At present the low-speed two-stroke engine is dominant, accounting for approximately 75% of the installed output; with respect to the number of ships built, the slow-speed two-stroke engine boasts a figure of approximately 70%. In small sized ships, the medium speed engines have taken over the largest proportion in main propulsion plants.

Between the "reserved" categories is a territory of small-to-medium sized cargo ships, with propulsion demands from 1,500 kw to 8,000 kw (2,000 bhp to 11,000 bhp), which can be met by two stroke or four stroke engines.
This is the hot point that both slow-speed and medium-speed engine-builders have been exploring to optimize design and improvement in both low-speed two-stroke and medium-speed four-stroke marine engines, for particular applications. The overlap is becoming wider and wider.

Otherwise, different ships and their requirements from the modern marine engine design produce a complicated engine programme. It is difficult for a shipowner to select a marine engine, which can be satisfactory both on economical and technical grounds, based on the engine programme.

Therefore, the upswing in the ship-building market, along with increased competition, will undoubtedly require the ship-owners to have their own opinions and methods to evaluate and select the optimum next generation of main propulsion plants for their ships.

1.2 Importance

Traditionally, the choice of either direct or indirect drive of a ship is governed more by the operating profile than by the characteristics of a particular make or type of main propulsion diesel engine. There are, of course, many ship owners who may prefer to remain with a particular make or type of engine for various reasons such as reliable past operating experience, crew familiarity, spares control, good service back-up, and so on.

However, new designs of engines are being rapidly revealed these days because of extended competition between builders. The modern standards of marine engine building require the shipowner and marine engineers to evaluate the new types of engines, much more depending on the opinion of the overall economy.

Sometimes, the owner who has not built a new vessel for possible arrangements, with which he is unfamiliar. Much
decision making about the choice of engine is governed by cost considerations regarding the type of fuel burned, maintenance costs, manning levels, and the initial purchase price of the engine.

There is, however, more of a tendency to consider total life cycle costs rather than simply the purchase price of the main engine. The total purchase and operating costs over 15 years, for example, can be greatly influenced by the initial choice of main engine.

In any event, the selection and installation of a propulsion plant for a given ship is connected with the different aspects of technology, economy, and operating behavior. It is by no means an easy one. Therefore, it is reasonable to establish a set of systematic methods concerning modern marine engine selection.

1.3 Directions

The purpose of this paper is to present a technical and economical evaluation, a comparison between low and medium speed engines, and an engine selection program based on the MAN B&W engine MC-programme.

The objectives of the project are:
- to discuss new requirements and the potential development market of future ship’s machinery;
- to analyze technical features and the most modern concepts in the fields of slow and medium speed diesel engines;
- to compare basic factors in the engine selection range and their application;
- to provide a systematic method to select the main engine for a given ship;

It must be clear that the main thrust of this project is different from traditional opinions in marine engine sele-
ction, which are used to choose specific ships such as ferries, ro-ro ships and passenger ships.

The key difference in the selection will not only be in slow and medium speed engine types, but also inside the two-stroke slow speed engines themselves, which is in the territory of MAN B&W Engine Programme, used on bulk carriers, container ships, or other competitive types of ships.

The reason is that the economy is always in the forefront of consideration in the selection. Sometimes, one can choose the best engine type, but you cannot get the optimum engine for a specific vessel. The selection inside the same engine types often gives a more important difference, compared with the different engine types, in technology and economy.

The project will aim to consider the following items:
- Engine type: MAN B&W
- Power range: MC-Programme
- Ship size: Small and middle
- Used period: 15 years

1.4 Chapter layout

Chapter 1 is about the general introduction of the dissertation. It includes the previous situation in marine propulsion markets, the necessity of engine selection and the arrangement of the dissertation: why is it, what is happening, and how it can be done.

The engine is used on board a ship, therefore it is necessary to look at the essential requirements of ships and their propulsion. What are the relationships between a ship and its machinery, the machinery arrangement, the technical and economical features and the market share. Chapter 2 will review the field.

The machinery concepts for the new types of marine engines will be dealt with in Chapter 3. The key factor is
the overall economy, which includes the following factors:

- **reliability.**
  - this is as realized the first factor for most shipowners;
- **simplicity.**
  - this is an important addition to reliability, the cost and others;
- **economy.**
  - this consists of investment costs, operating costs (manning, maintenance and others);
- **operational flexibility.**
  - this is important, not only regarding adaptability because of the various environments that ships and engines will face, but also regarding technical criteria, such as economical ratings...what all marine engines have to meet.
- **operational behaviors.**
  - this is a modern criterion in engine evaluation, which deals with the comfort that is important to reduced manning on board ships; gas emission that will also be more and more restricted in the future in the field due to the regulations of IMO and other related organizations.

Recalling the history and development of both low and medium speed, it can be said that both engines have completed the common technical revolution, aimed at their different design features for specific ships. Therefore, in Chapter 4, I will highlight the developments engine-building technology of the MC series in the 90's.

Economic comparison has been a topic in marine engine selection. In Chapter 5, the traditional comparison of initial costs, operating costs and other relative costs will be discussed. The S26MC series is compared with the competitive medium speed marine engines. The objective is that the slow speed engines are also the optimum choice, even in
comparably small sized ships, like their application in medium and large size types of ships.

The selection of marine engine is connected with many factors. With the improvement and development in engine building, the comparison between the different engine types are not now the only consideration from economical and technical aspects. In most situations, the comparison between the same types of engines is more important and practical, because many of the shipowners like selecting the marine engines that only easily depend on man-power, experience, and short term targets. Therefore, in Chapter 6, the selection programme will be introduced.

Chapter 7 will deal with some aspects that should be considered in engine selection, such as the engine room programme, gas emissions, vibration and noise. Especially, in some requirements, of certain ports, countries and international organizations, so the comparison between them is necessary.

Chapter 8 will conclude the technical and economical aspects that have been listed in previous chapters.

1.5 The definitions

Reliability:
-An expression of the probability that an engine will continue operating for a given period.

Durability:
-An expression of the length of time for the period between overhauls or life of either individual components or whole engines.

Behavior:
-The typical way in which an engine functions according
to the principles of diesel engines. Aimed at engine operation, here it specifically indicates gas emission, noise and vibration.

Simplicity:
- An expression that the plant is uncomplicated and can be operated and maintained easily, with total functions that are absolutely necessary for general diesel engines.
In any discussion of a ship's machinery, it should be reasonable to begin by taking an overall look at the ship. The various features of marine machinery all relate to its operation in a safe, reliable, efficient and economic manner. The main propulsion machinery installed will influence the machinery layout. This will determine the operational and maintenance requirements of the ship, the relative knowledge required and the significance of engine selection.

Basically, ship propulsion means that a certain power needs to be transmitted from a machinery, via a propelling device, to the water. With diesel machinery this can be done in several ways, e.g. one engine coupled to a propeller, one engine coupled via a gear to a propeller, or a multi-engine plant coupled via gears to one or more propellers. In this section, I just discuss the first two items.

2.1 Ship and machinery

2.1.1 Ships

A ship is a large, complex vehicle which must be self-sustaining in its environment for long periods of time with a high degree of reliability.

In general, the ocean-going vessels, depending on their functions, can be classified in to the following types:

- Tankers
- Container Ships
- Bulk Carriers
- Specific Vessels
The different types of ships have different requirements in size, cargo space, design speed, etc. In general, it should meet the requirements in the hull, its construction, form, habitability and ability to endure its environment.

A ship might reasonably be divided into three distinct areas: the cargo-carrying holds of tanks, the accommodation and the machinery space. Depending upon the type, each ship will assume varying proportions and functions. In general, the machinery space size will be decided by the particular machinery installed and the auxiliary equipment necessary.

Therefore, machinery space is an important factor in a ship's machinery selection, especially in the main propulsion plant compared with other machinery.

2.1.2 Machinery

Two principal types of machinery installation are to be found in merchant ships today. The direct-coupled slow-speed diesel engines and medium-speed diesels with a gearbox are the two layouts. Their individual merits change with technological improvements, advances and economic factors such as the changes in oil prices, manning and technology. However, here it is intended only to describe the layouts from an engineering point of view to know the geometric relationship between the ship and its machinery.

1) Slow-speed diesel

In general, the machinery installation is of a compact and complicated nature. The main items in the installation are the main engine and boiler.

The more usual plan and drawings of a typical slow-speed diesel installation are shown in Figure 2.1. A six-cylinder direct-drive diesel engine is shown in this machinery arran-
The only auxiliaries visible are a diesel generator on the upper flat and an air compressor below. Other auxiliaries within the machinery space would include additional generators, an oily-water separator, an evaporator, numerous pumps and heat exchangers. An auxiliary boiler and an exhaust gas heat exchanger will be located in the uptake region leading to the funnel. Various workshops and stores and the machinery control room will also be found on the upper flats.

Fig 2-1. The arrangement of a six-cylinder direct-drive diesel engine room

2). Geared medium-speed diesel

Four medium-speed (500 rev/min) diesels are used in the machinery layout of the rail ferry shown in Figure 2.2. The
gear units provide a twin-screw drive at 170 rev/min to controllable-pitch propellers.

Fig. 2-2. The machinery layout of a geared medium-speed diesel engine

The gear units also power take-offs for shaft-driven generators which provide all power requirements while at
sea. The other machinery can also be seen from this figure. Obviously, the machinery space is more compact than the slow-speed diesel considering the size of the main propulsion plant.

2.1.3 The relationship

The goal of shipping is cargo transportation. From the point of view of the machinery, the requirements for engines are efficiency, function, security, compactness and systemization.

From the architectural side, the cargo holds should take a larger proportion. The machinery space has to be restricted by the cargo space or suitable for the requirements of a specific ship.

2.2 Evaluation about the engine arrangements

The direct drive of a larger ship's propeller by a slow-speed two-stroke engine and a small ship's four-stroke, medium-speed engine still remain the most popular methods in marine propulsion.

2.2.1 Direct drive

At one time a slight loss of propulsive efficiency was accepted for the sake of simplicity, but the introduction of 'long stroke' and very recently 'super long stroke' crosshead engines has done much to reduce propulsive losses. For a large ship, a direct coupled speed of, say 110 rev/min is not necessarily the most suitable, as it is proven that a larger propeller turning at speeds even as low as 60 rev/min is more efficient than one of smaller diameter absorbing the same horsepower at around 110 rev/min.
The long and super long stroke engines now on the market develop their rated outputs at speeds ranging from as low as 65 rev/min up to around 180 rev/min for the smallest bore (around 260 mm) models. It is now possible to install a direct drive diesel engine which will achieve very nearly the optimum propulsive efficiency.

Direct coupled engines develop high outputs per cylinder, particularly large bore models, and it is easy to obtain the power required from an engine of a small number of cylinders. An owner will prefer an engine with as few cylinders as possible, provided that problems of vibration, balance, etc. do not ensure, because this directly affects the maintenance workload, spare parts carried, the overall size of the engine and hence the machinery space.

In most vessels height is less of a problem than length, so a larger bore engine with fewer cylinders will inevitably result in a shorter machinery space and more space for cargo.

It is also proven in practice that larger bore engines have a better specific fuel consumption than smaller engines and there seems to be a better tolerance to burning heavy fuels of poor quality.

A direct coupled main propulsion engine cannot operate unaided as it requires service pumps for cooling, lubrication, and fuel and lubricating oil handling and treatment. These items of auxiliary machinery need a power source, which is usually provided by generators driven by four-stroke, medium-speed diesel engines which normally burn fuel of a lighter quality than that used by the main engine.

Manufacturers of small auxiliary engines have stepped up their efforts to produce machines capable of burning not only the same heavy fuel as main engines, but marine diesel fuel of blended fuel (heavy fuel and distillate mixed in various proportions, usually 70:30) either supplied as an
intermediate fuel or blended on board.

The cost of auxiliary power generation can weigh heavily on the choice of main machinery, so developments recently have tended to maximize the use of waste heat recovery for auxiliary power generation, the use of alternators directly driven from the main engine through speed increasing gears and the drive of certain auxiliary machinery items from the main engine.

The designers of slow-speed direct coupled power plants are moving towards the 'one fuel ship', whereby low quality heavy fuel is used aboard ship for all purposes using the methods previously mentioned. Examples of such vessels are now in service.

2.2.2 Indirect drive

The most satisfactory and economical form of indirect drive of a ship's propeller is one or more four-stroke, medium-speed diesel engines coupled through clutches and couplings to a reduction gearbox to drive either a fixed or controllable pitch propeller.

The latter obviates the need for a direct reversing main engine while the gear allows the best propeller speed to be obtained with certainty. There is, nevertheless, a loss of efficiency in the transmission, but this in most cases would be cancelled out by the improvement in propulsive efficiency when making a comparison of direct and indirect drive engines of the same horsepower. The additional cost of the transmission can also be compensated for the lower cost of marine main engines as two strokes, being large and heavier, inevitably cost more.

In general terms, the following advantages are claimed for geared medium-speed engines:
for vessels that have more than one main engine reliability if improved as the vessel can still operate with one engine if a breakdown at sea occurs.

when vessels are running light, partially loaded or slow steaming, one engine can operate at its normal rating and others be shutdown rather than a direct coupled engine which may be called upon to operate for long periods at reduced output, at which it is inefficient. This has indeed been the case in the last few years with many large oil tankers.

maintenance is easier because the engine components are of a more manageable size, and generally major components such as cylinder covers, pistons, liners, etc. are much cheaper than corresponding two-stroke engine components.

engines can be overhauled at sea by steaming on other engine(s) rather than is the case for the direct coupled engine which can only be overhauled in port. Ships today spend little time in port, so the facility of repair can be an important factor.

by modifying the number of engines per ship and cylinder numbers per engine to suit individual horsepower requirements, the propelling machinery for a fleet of ships can be standardized on a single cylinder bore size with subsequent savings in spares costs, availability, etc.

With certain vessels with high auxiliary loads, such as ferries, passenger ships, offshore service craft and so on, so called 'uniform machinery' can be employed with engines of the same bore and stroke used for both main and auxiliary duties.
-the weight of the machinery and space required, particularly headroom, is much reduced—an important feature for vehicle deck vessels such as car ferries.

The above are true advantages for the geared medium-speed engine and are much exploited by manufacturers in promoting their sales.

However, there are many shipowners who are reluctant to burn heavy fuels (say 3500 seconds Redwood I viscosity) in many types of four-stroke engines, while none of the direct coupled two-stroke engines appears to have great problems.

Lubricating oil consumption is another factor to be considered, as the specific oil consumption of, for example, a trunk piston engine is inevitably higher.

2.3 The market requirement and share

2.3.1 Market requirement

To summarize, the choice of main machinery for a given ship will depend on many factors, each of which needs to be carefully examined by a shipowner or engine builder.

The ship’s overall size, tonnage and range to perform its required duties are first established and the choice of direct drive, geared drive or electric drive will depend on these requirements, but much more on degree of reliability, initial cost of the machinery, cost of maintenance over the life cycle, quality and cost of fuel that can be burnt in both main and auxiliary engines, the cost of lubricating oil and, to some degree, passengers or crew comfort through excessive noise and/or vibrations.

However, a diesel propulsion plant is favorite for virtually all merchant ship new-buildings today. In practice, the final decision is commonly made without detailed con-
sideration. The choice between the direct and indirect options being influenced by 'fashion'-their current market strengths in the specific ship sector-and often a focus on one parameter: for example, the lowest optimum specific fuel consumption at economy rating. "Feeling" rather than "facts" still holds way for some operators.

2.3.2 Market share

The direct-coupled two-stroke low speed engine has established itself as the prime candidate for medium-to-large oceangoing cargo ships like bulk carriers, tankers and containerships of 1500 TEU-plus capacity.

The geared four-stroke medium speed engine is entrenched in the smaller ship sectors-coastal cargo vessels, gas and chemical tankers, and containerships of under 700 TEU capacity, as well as tugs, supply vessels, fishing craft and ferries. Cruise ship operators can opt for either a two-stroke or four-stroke solution.

The statistics for installed power in ships above 2000 DWT show that diesel engines are now the exclusive propulsion machinery for merchant ships, with only very few exceptions. In bhp installed, one can establish a steady position for the low-speed diesel engine with about 70-75% market share, whereas the remainder is taken by its medium-speed counterpart.

2.3.3 Analysis

This remarkably stable situation keeps up with some predictions which have often projected a majority share for the low-speed engine. Analyzing this situation in an objective way, it can be listed as following:
a). In two-stroke slow speed engines:
- Fewer moving parts in the engine for a given power;
- Lower specific fuel consumption;
- No gear lossed;
- Simplicity in construction;
- More favourable inertia-forces on moving parts;
- Always a one directional pressure on the pistons giving a less sensitive connecting-rod to the crankshaft;
- Long lifetime meaning longer economical life and higher second-hand value of ships with slow speed engines;
- Low lubricating oil consumption.

b). In four-stroke medium speed engines:
- First cost;
- Engine weight and overall dimensions;
- Waste heat recovery utilization;
- Low cylinder liner wear rate.
CHAPTER 3

MACHINERY CONCEPTS AND SELECTION GUIDELINE

The machinery design has to obey certain principles to meet the requirement in functions. It is pretty clear that Overall Economy is the key point in machinery design, operation, maintenance and comparisons.

Overall economy means:
- Reliability at sea, with durability and low maintenance
- High fuel economy
- Low investment costs
- Low manning level

Therefore, it will be the principle consideration for future propulsion plants.

3.1 System engineering concepts

For the project engineer, whether working for a ship-owner, a shipyard or an engine builder, a good engine programme is one that allows the engine to be selected without having to compromise on something significant, still fulfilling the basic requirement of simplicity, and offering a number of engine alternatives for easy installation and overall energy conservation. A good engine programme in this context also means that answers to unavoidable questions are easily accessible and easily applicable.

The numerous possibilities that exist for combining systems and engines, together with the inherent flexibility of the individual units in an engine programme, provide the means for finding ideal solutions to virtually any conceivable specification requirement.

It is clear from statistical results that a two-stroke
direct-coupled solution to a propulsion task is the first choice and that such a solution is, in fact, also the final choice except in such specialized cases where all two-stroke possibilities have been exhausted.

These days engine builders are drawn into project work at a much earlier stage in a ship project than was previously the case. A first estimate on a suggested propulsion system, or even an entire propulsion package, will often have to be given on the basis of very preliminary physical data of a ship.

The preliminary data for a certain ship and an outline of the engine selection process can be seen in Fig. 3-1. Fig. 3-2 shows a calculated speed/power curve with corresponding propeller speed. When this data is entered in the engine programme, a number of engine alternatives will be result. Estimations of ship's load profile, electricity and heating requirements will offer a further study to select the most economical engine/propulsion plant for the job. Such system engineering analysis comprising economic analysis of first cost and operating cost of a number of alternative engines and electricity producing equipment has been introduced.

This was also the case for vibration aspects, which is another item analyzed in the course of a project. Also an analysis of auxiliary equipment for the engine is carried out.

The auxiliary power retirements of the engines have been significantly reduced compared to previous types.

In the course of analysis and preparatory systems engineering for the entire engine room, it is better to look especially into an evaluation of engines with and without turbo-compound system, with and without main engine driven generators.
Fig. 3-1. Engine selection process on the basis of ship particulars
3800 TDW Container ship

Ship speed (KTS)

Power (BHP)

Incl. 15% Sea margin

Excl. Sea margin

Ship speed (KTS)

Propeller speed (r/min)

Incl. 15% Sea margin

Excl. Sea margin

Fig. 3-2. Speed and power curves calculated from ship particulars
Such analysis work provides an unbiased evaluation of possibilities based on an owner's input. At the time of now, many engines have, as a result of such analysis, been equipped with PLO/PTI systems.

It is also noteworthy that such systems as the ICS (Integrated Charge System), which allows an auxiliary engine to run under idling conditions using heavy fuel, has now been in operation for several years.

The system allows a practically 100% utilization of a turbogenerators, without having of a power turbine on the auxiliary engine thereby maintaining the full simplicity of the main engine, and still benefiting from the economies of turbo compound systems.

System engineering will be further developed and provides an increasingly important additional feature of the engine programme.

3.2 Reliability aspects

Freight rates, which without doubt can be expected to remain low in the years ahead, demand more than ever a high degree of aggregate efficiency of the vessel, and thus a high degree of efficiency of the propulsion plant. Therefore a high reliability is an important plus as machinery design guidelines.

3.2.1 Reliability and durability

In a ship's engine, we are surely looking for more durability. Reliability is, by itself, meaningless. For example, a racing engine can be fully reliable, being expected to run without problem for only two hours in order to win the race. Both reliability and durability, however, can only be quoted, with any confidence, if the engine
concerned has a proven record of service experience.

There are two definitions about the reliability: a reliable engine is reliable when well maintained; and a reliable engine is reliable even when not well maintained. Most of the components in a modern diesel engine are expected to work for long periods without any maintenance of attention at all. The main areas where mistakes in operation or maintenance can affect the engine are:

- lubrication of main and big end bearings;
- in the case of piston rod engines, lubrication of crosshead bearings, crosshead guides and maintenance of piston rod gland;
- lubrication of piston, piston rings and cylinder liner;
- and fuel cleaning and preparation.

The main and big bearings: for improving the reliability of bearings against mistakes in maintenance, the direction is quite clear. The film thickness must be increased.

The piston and liner: the improved lubricator and pressure lubricated piston was introduced.

The application of modern manufacturing methods to improve the reliability and competitiveness of the engines is always a necessary and permanent goal. The technological developments such as new casting, welding and machining methods, NC or CNC-controlled machines, surface treatment have improved a lot in reliability.

Typical examples are the revision of castings and machining technology for the newest types of engine liners, and the research into coating and surface hardening technology.

3.2.2 Relative factors

The reliability and durability of a propulsion plant is influenced by several factors; the most important being:
Fig. 3-4. Factors influencing the engine reliability

Consequently, the related standards and requirements about the reliability are:

- Extending the time between overhauls (TBO) for the main components (piston, exhaust valve) to exceed 8000 - 12,000 hours and thereby allow two years' maintenance-free operation.

- Keeping down wear rates, for example the liner and ring wear to below 0.1 mm/1000 hours and 0.4 mm/1000 hours respectively with average wear rated as being even lower, and at the lowest possible lubricating oil consumption.

- Reducing the amount of intermediate maintenance for fuel nozzles and other components needing regular maintenance.

- Easing further the maintenance work by adequate and...
simple-to-handle tool systems.

-Developing software and hardware certain for an efficient condition monitoring system ("early warning devices")

3.3 simplification

As mentioned, a lot of improvement have been done by marine engine builders. However, there are still not any dramatic changes in the near future that can be predicted. Needless to say, the reliability and maintainability of these systems must be improved since they represent the major workload of the operators.

Therefore, it can be seen that the pressure on manpower and economic competitions will force developments towards better reliability and maintainability of conventional systems.

In practically all areas where damage can occur, the low speed engine concept is inherently far less vulnerable than any other propulsion system. The low speed diesel has, in spite of its technologically advanced stage and its engineering complexity, never lost its inherent simplicity.

To the users this simplicity means that practically no damage is fatal. The internally accumulated energy that needs to be absorbed in a particular part of the engine in case of a mechanical mishap is moderate due to the low speed, and the risk of consequential damage is therefore limited. In a design principle of the low speed engine, a damaged engine can practically always operate, for which reason low speed diesel installations are designed as single-engine installations with no "take-home" power. Therefore a simple low speed engine is always been chosen, where great reliability must be ensured under any circumstances.

The simplification is just the best guaranty in relia-
bility. The obvious example seems that there is a need for a reduction of the number of sensors and alarm points used today.

Experience shows that even if the ships have been built for years, there is still a great challenge for the future. Development and use of good quality control and test systems is vital, as too many problems seem to occur because of poor quality control.

The monitoring component in a reliable engine does not need a lot of monitoring function. Also in system design, the trend today is towards simpler and more reliable layouts.

This is partly a reflection of the competitiveness in the marine business. There is no reason, any more, to think of what is technically possible, rather the necessity to figure out what is really necessary.

In fact, the monitoring functions according to the classification societies are more than enough to guarantee safe operation of good machinery. In some cases there can even be room for reduction of the monitoring functions stipulated today by the classification societies.

3.4 Economical aspects

3.4.1 Capital cost.

1) Cost per unit power

The most important aspects with respect to engine manufacturing today is the competitive commercial environment calling for still further reductions in manufacturing costs per unit power.

It is the index for every marine engine factory to compete with other manufacturers.
2) Relative design

In the engines of MAN B&W MC-Pregramme, the alternative standard designs for certain components are in order to facilitate production. This, however, does not apply to wear parts for which exchangeability must be ensured.

An exemplified result of the above is the bedplate where an advantage in reducing the width compared to previous designs was obvious. For this reason the holding-down bolts were moved from an outside mounting to small compartments inside the bedplate.

With regard to the frame box design, this is available with integral as well as with bolted-on crosshead guides.

In a similar way, each large and small component is continuously surveyed before fitted so as to make it possible to adjust to present and new facilities, for production cost control.

By utilizing production experience as well as service feedback from previous engine designs in the design process, the number of production hours has been reduced by approximately 20% compared to previous engines (MAN B&W).

The number of components used for certain engine sections has been appreciably reduced, thereby achieving a reduction of production cost as well as assembling cost.

Efforts are being made to design the engine in units to an even greater degree, with a view to reducing the number of components and joint surfaces. This means that components are welded, cast, and built in sizes which are suited to actual production and transport capacities.

The adaptation of the design to suit modern production facilities and the integrated computer systems, linking the entire process together, and new technologies within manufacturing, will call for further reduce in capital costs.
3) Installation cost

Except the engine manufacturing cost is decisive in today’s hard, competitive situation, the engine builder also has to take into consideration the engine related installation costs, such as auxiliary power requirements, the cost of the fuel and lubricating oil treatment plant, and cooling equipment. This means the cost of the whole propulsion plant. There are no signs that these main market requirements will change in the near future.

3.4.2 Operational costs

This cost generally includes the four items: fuel; lub oil; spare; and manning.

1) Fuel consumption rates of engines

Spurred on by high fuel prices in the past, engine development has been expanding with considerable efforts to find ways of reducing the fuel consumption rates of engines. Up to now, the thermal efficiency of current low-speed engines has reached values considered close to the theoretical limits set by their basic parameters (Pmax, BMEP, stroke/bore ratio, turbocharger efficiency, and so on).

Even though fuel prices are comparatively low at present, and they can without doubt be expected to remain so in the immediate future, fuel costs still take a share of 40% to 65% of the ship’s running costs. So it can be said that high economic efficiency means first and foremost low fuel consumption rates.

Therefore, SFOC is still the key point for engine builders to pursue.

Fuel costs are connected with following factors (shown as
- power/speed selection (propeller efficiency);
- engine optimisation over the load range;
- recoverable waste heat;
- auxiliary power need.

**Fig. 3-5**: Factors influencing overall fuel costs

2) Lube oil consumption

A further factor influencing the overall economic efficiency of a propulsion plant is the lube oil consumption rate.

Using modern engine designs and advantage lubricator, it has been possible to significantly reduce lube oil consumption rates.

However, in medium speed engines, very low lube oil consumption rates can also have a corresponding contaminating effect on the system oil, which in some cases can make replacement of the oil charge necessary after 12,000 operating hours, especially as the tank need not be topped up as
cylinder lubrication is effected via a separated fresh oil system.

3) Spare parts

The service period and the price of spare parts are two of the most important factors.

4) Manning

Shipowner are meeting stiffer competition as time goes by. This results in an increasing desire to cut costs to improve a competitive position. One answer to this problem is crew reduction.

The trend towards further reductions in manning levels will continue. This also means that the reliability of engine must be further improved, and the simplification of propulsion plants.

3.4.3 Compound system

In addition to fuel consumption rate lowering measures on the diesel engine itself, the propulsion system as a whole is increasingly being included in economic efficiency considerations.

As the thermal efficiency of the engine as such today exceeds 50%, and by appropriate use of "energy optimizers" it is possible to achieve approx. 60% in mechanical/electrical output. On top of this, further possibilities exist in waste heat recovery.

Extensive utilization of the various sources of engine heat, such as exhaust gases, charge air, etc., as well as the use of turbo compound systems in their numerous variants, offer here a whole series of opportunities for
optimum measure. By re-engineering, it was possible to achieve an overall system efficiency of approx. 75% by consistently utilizing various waste heat sources.

By using Turbo Compound Systems, the fuel consumption rate can be considerably reduced, lowered by up to 7 g/KWh.

It will without doubt be necessary to ensure that the use of Turbo Compound Systems does not make the propulsion system appreciably more complicated.

3.5 Operating behavior

3.5.1 Comfort

In most areas of life the trend has been towards increased comfort, and so it has been on board ships. Vibration and noise are the issues that have been paid attention.

The introduction of superlongstroke crosshead engines with a few cylinders, high combustion pressures and huge rotating and reciprocating masses made vibration a problem for ship designers.

The following items should be considered:
a). Natural frequencies versus excitation frequencies
b). The ignition frequency

In many of today’s passenger vessels direct resiliently mounted engines are being installed in order to keep disturbing engine vibrations away from the body of the ship, thus making life more comfortable for the passengers and crew as regards the noise and vibration levels in the vessel.

3.5.2 Heavy fuel capability

One can always hope for improvements in fuel quality, but engine-builders must be well prepared for further deterior-
ration in bunker fuel quality. The selection will cope with fuels that are more difficult to burn.

The rotating piston, the use of bore cooling for all combustion space components and the particular exhaust valve design combine to give the selected engine a unique capability for sufficient time between overhauls when burning really poor quality fuel oils.

3.5.3 Partial load operation

In future it will no longer be possible to view neat-load operations solely with regard to its economic efficiency, but it will, to an increasing degree, be necessary to direct efforts towards the optimum of the plant, as well as meeting the demands for reductions in pollutant emissions. In this respect a considerable improvement can be expected as far as the injection system is concerned.

1) Spill timing

In this respect, optimizing the spill time dependent upon fuel type and the operating point in the operating curves will be of significance. MAN B&W began concerning with the optimum of spill timing depending on fuel type and operating point in the performance curves many years ago. With the aid of this controllable mechanism it will in future be possible to set the optimum spill timing automatically, which will have a positive effect on the economic efficiency of the engine and on engine emissions.

2) Turbocharging

There is no doubt that sequential turbocharging will also be an interesting possibility for improving partial load
behavior for certain application.

On the other hand efforts will also be directed towards rendering the wear parts less sensitive to contamination.

3.6 Environmental concern

The environment problem will come into focus more, and requirements for exhaust emissions will increase, especially for traffic on the coast and in port. This may lead to increased requirements for fuel oil properties, or for engines to be developed with better combustion with catalysts of flue gas cleaning.

It can be expect that in the design of fuel systems in the future, the health aspect will, to a large extent, have an influence on the marine engine selection.

In particular this will include:

a) Optima of the engine specifications so as to improve ignition and combustion.

b) Selection of fuel type.

c) Reduction of lub oil consumption and/or lub oil TBN decreases the particles originating from lub oil ash and lub oil hydrocarbons.

d) Exhaust gas cleaning techniques
   - Nitrogen oxides
   - Sulphur oxides
CHAPTER 4

THE DEVELOPMENT IN MARINE PROPULSION IN THE 90's

The market for two-stroke low speed diesels is a moving target. Therefore, the development of such engines must focus on product refinement and adaptation, both with a view to comply with altered operating conditions and production facilities and to meet requirements raised by changing trade patterns and by changing legislation.

Fig 4-1 shows the development of the B&W (since 1960 MAN-B&W) two-stroke engine types since 1959 up to now.

The first major step in the development of the present engine programme was the mid-February 1985 introduction of the K-MC and S-MC engine series together with the introduction of the Turbo-Compound System (TCS) on the large bore MC-engines (from 50MC and upwards).

The S-MC engine series was with its 17 bar mep, 130 bar firing pressure, and 7.6 m/s mean piston speed, at the forefront of two stroke engines.

In 1986 since the adoption of and increased mean piston speed for the L-MC types, the major event was the announcement of a new design concept, the mini-bore S26MC.

4.1 MC Engine Programme

4.1.1 Programme

MC engines were introduced in 1963, with a total of more than 1700 units, of which around 1200 are in still service. Fig.4-2 shows the current MC programme, comprising engine with all relevant combinations of speed and power for ship propulsion, with unit size up to 67,080 BHP.
Fig 4-1: Developments in firing pressure $P_{\text{max}}$, mean effective pressure $P_e$, Scavenge air pressure $P_s$, Turbocharger efficiency $\eta_{TC}$ and specific fuel oil consumption.
The programme has been constantly updated to meet, at short notice, requirements as they have emerged. Originally the programme comprised the L35MC, and the L50, L60, L70, L80 and L90 MC, all of which had been tested and entered service a couple of years after introduction.

In the MC engine programme, the K-MC engine and S-MC engine series with shorter and longer strokes, and correspondingly higher and lower speeds were introduced as a major step.

The S-MC engines were successfully targeting the tanker market and today comprise the S50, S60, S70 and S80 MC. The small-bore S26MC aims at the small cargo vessel sector.

The K-MC types were developed to specifically cater for the growing container ship market.

4.1.2 Thermal Efficiency and the Total Energy Concept

The traditional design tools used in the MC-engines in order to reduce SFOC, are higher stroke-to-bore ratios, a higher ratio between firing pressure and mean effective pressure, and higher turbo-changer efficiency.

As seen in Fig 4-3, the present MC-engines used the effect of the first two parameters to the extent that changes in the current range will be marginal.

The effect on SFOC of the actual spread of stroke-to-bore ratios is only about one g/BHP/hr different between the largest "short-stroke" engine, the K-90MC-2 and the largest long-stroke , the S-80MC. Any major changes to the stroke-to-bore ratios, except that will be dictated by different applications, are not foreseen, and the potential for further reduction of SFOC is limited.
With regard to the $P_{\text{max}}/\text{mep}$ ratio, all engines are on the same level except the smallest S-26MC, where a higher $P_{\text{max}}$ at the same mep had made a really attractively low SFOC possible for such a small engine.
For the time being, no changes are foreseen by the company within the remaining programme, as this would require major design change.

The third parameter used is the turbo-charger efficiency. Basically the MC-engines are specific with a turbo-charger efficiency of about 64%, with efficiency saveable in the range of 70%, this extra margin could be utilized in the engine, but would result in only a slight reduction of SFOC, as can be seen in Fig 4-3. Moreover, the resulting exhaust gas temperature would be very low, causing difficulties for the exhaust gas boiler and, most likely, non-optimal plant economy if an additional oil burner becomes necessary.

As can be seen from the figure, the only potential left for further reductions of the SFOC by traditional methods is a significant increase in the combustion pressure for derated engines. For fully rated engines, practically the same reduction of SFOC can be obtained in most cases from the present engine programme by choosing a derated version of the same engine type, which means an engine with two or three more cylinders in order to give the same MCR power.

In both cases (increased $P_{\text{max}}$/ or increased cylinder number/derated), the engine will be significantly more expensive, and such development will only be worthwhile if fuel prices are high.

This illustrates that the relevant question is not"How low an SFOC is it possible to achieve?" but "which engine development is worthwhile?". In general, the future development of the low-speed two-stroke marine diesel engine will, to a much lesser extent, concentrate on a reduction of SFOC of the engine than witnessed during the last decade.

The pace of development as well as the final level to be reached will be controlled by economic factors and not by Carnot and other technical factors.
Fig 4-3: Effect on SFOC of stroke-to-bore ratio, ratio of maximum firing pressure to mean effective pressure, and turbocharger efficiency.
4.2 The development

4.2.1 Short-term development targets

Although the major design changes are not foreseen in the near future, interest has recently concentrated on fully rated engines.

The moderate fuel oil prices and improved freight rates together with the concurrent tendency towards higher ship speeds, have placed increasing emphasis on reliable operation of ships and, consequently, on the main engines reliability. To complete the picture, the trend towards reduced manning is gaining further momentum.

As a consequence, the most important short-term development target for the MC-engines is to enhance their reliability and, wherever possible, to reduce production costs without endangering reliability and engine performance.

4.2.2 New Developments in Reliability

1) Exhaust valve

Fig 4-4 shows the MC-exhaust valve with a considerable improvement compared with previous designs, in that seat-burning has not been a problem and the average interval between overhauls has been increased, for the large engines to 6,000 hours and for the smaller engines to 4,000 hours.

a) Valves

The underside and seat area of the spindle consisted of Nimonic 80A, while the rest is normal SNCrW-material. The compound spindle is produced by means of the Hot Isostatic Pressure (HIP) method, where powder of the two mentioned materials is sintered together in a single process at a
A protecting layer of Inconel 625 is welded on to the spindle’s underside, as shown in Fig 4-5, for engines with 60 cm cylinder diameter and upwards. For smaller engine types, the present valve design remains the standard. The Inconel 625/SNCrW spindle is somewhat cheaper than the compound Nimonic 80A/SNCrW spindle and can be produced by most exhaust valve manufacturers. The HIP-spindle, however, has the advantage that the sintered material is superior to forged material in structure and mechanical properties, and the production procedure without seat armoring being welded-on eliminates residual stresses in the high-loaded seat area.
b) Valve Bottom-Piece

The exhaust valve bottom-piece in its present bore-cooled version is a reliable, but also rather expensive component. Encouraged by its very low rate of burnt seat failures, a simpler design, the semi-cooled, cast-iron bottom piece, has been developed, as shown in Fig 4-6, together with the present bore-cooled bottom piece.

The introduction of the semi-cooled bottom piece on 90MC-type engines shows, as standard, that it gives a reduction in production costs, for this component, of around 35%. The seat can, if necessary, be ground on board.

c) Valve Housing

The exhaust valve housing has, in a number of cases, presented problems with cold corrosion in the upper part of the duct. The final solution for new engines is the 'hot duct' exhaust valve housing. In this design, there is cooling water only around the spindle guide, which is the only part that actually requires cooling. The duct itself is not cooled.

2) Cylinder Liners

It is not easy to cast material of the required quality for the very large liner dimensions. In order to facilitate casting, and to reduce the loss in case scrapping becomes necessary. A two-part liner, which is bolted together before the final machining of the running surface, has been designed. Fig.4-7 shows the upper part with cast-in cooling pipes, which is a much smaller component than a complete liner, and therefore easier to cast.
Fig 4-5: Exhaust valve with Inconel 625 welded on to the spindle underside

Bore-cooled bottom piece  Semi-cooled bottom piece

Fig 4-6: Comparison between the present bore-cooled 90MC bottom piece and the semi-cooled cast-iron bottom piece

Longer-term development has been concentrated on eliminating the casting problems with the pipes by utilizing bore-cooling instead. Two types of liner have been designed: a cast-iron type and a compound liner type.
Fig 4-7: Upper part of the two-part cylinder liner with cast-in cooling pipes for an S80-MC

The general design of the compound liner type has a two-part liner, and has to be so because the upper, bore-cooled part is made from steel and iron part as not doing surface. The lower, cast iron part is not doing subjected to high mechanical or thermal loads and can, consequently, be made from ordinary Tarkalloy. The four component liners, put in to service, have logged up to 4300 service hours, an excellent result.
K90MC-C Bore-cooled Piston

Fig 4-8 Bore-cooled piston for the K90MC-c with welded-on high temperature protection layer of Inconel 625

3) Pistons

The MC-pistons have not caused any problems worth mentioning, but some development has, nevertheless, been carried out for the purpose of increasing design margins. The outcome, shown in Fig 4-8 is the bore-cooled piston introduced as standard on the K90MC-C.

The main reasons for the introduction are the ability of the bore-cooled design to sustain as significantly as po-
ssible, and a reduction in production costs compared with the welded version of the present MC-piston.

The piston has a welded-on layer on the outer part of the top, covering the areas where the highest temperatures are usually found. The reason for applying Inconel 625 is to increase the margin against hot corrosion on K-type engines, and engines fitted with a TCS.

4). Fuel System

The fuel system, shown in Fig 4-9 has functioned well, and has given good, reliable service results.
The puncture valve has contributed to improved starting and stopping performances. Therefore, it is now the standard on all engines.

In a number of cases, a satisfactory VIT (Variable Injection Timing) function was hindered by the formation of leak-oil sludge on the top of the fuel pump barrel, which prevented the upward movement. This was counteracted by improving the drainage conditions.

5) Chain Drive

The hydraulic, self-adjusting chain tightener, see Fig-10, has been introduced on the K90MC-C type engine.

The new chain tightening device, which has now been in service for more than 7,000 hours, with very satisfactory results, ensures sufficient chain tension and reduction and reduces the necessary maintenance work. The new arrangement can also be retrofitted on plants in service, if the need arise.

![Fig 4-10 Hydraulic chain tightener](image-url)
4.3 Turbocharger and Other Systems

4.3.1 Turbocharger

On the whole, all turbochargers have functioned mechanically satisfactorily, and the main cause of concern today is thermo-dynamical rather than mechanical.

The point of primary importance is the negative influence of in-service decreases in turbocharger efficiency, which, on modern engines, has become the dominating factor for the level of thermal loading.

In this respect, cover ring (shroud ring) erosion on the turbine side has given problems on several plants, in that enlarged clearances and blade geometry alterations drastically reduced the turbocharger efficiency, and thereby greatly increased the thermal load on the engine.

Dry cleaning is now the recommended cleaning method to prevent the erosion, caused by water washing and/or particles in the exhaust gas.

4.3.2 Power take-off System

The system was introduced in 1984, as shown in Fig 4-11, which is the RCF (Renk Constant Frequency) Power Take-Off gear arrangement for the main engine driven generator. It has been running very well from the service feed-back.

4.3.3 Turbo Compound System

The Turbo compound system (TCS) was introduced in 1985, with the design shown in Fig 4-12, comprising a power turbine and a transmission gear. Since some teething troubles with a new system have been reported, appropriate countermeasures have been introduced in the general system,
and the situation has been improved.

Fig 4-11 Renk BW111/RCF gear for main engine driven generator

Fig 4.12: BW/RTCS - Turbo-compound system
4.4 Future development

It is quite obvious that use of computer technology is increasing also on board the merchant fleet, e.g. for performance evaluation, engine control, etc. A step in this direction is the Computer Aided Performance Analysis software introduced some years ago.

The ultimate aim is the "intelligent engine", i.e. an engine which not only constantly monitors its own condition but which, on the basis of computerized performance analyses, automatically adjusts its parameters to provide the optimum engine operation at all times, regardless of the sea, weather or draught conditions.

MAN B&W has developed a system which facilitates the measurement of the actual combustion process. In principle, this can be carried out by fitting strain gauges to any combustion chamber component for the studs assembling these component. A decision has been made to take the advantage of the fact that the tension of the cylinder cover studs reflects the average cylinder pressure, and not a local pressure influenced by pressure wave motion in the combustion chamber. The influence on stud tension from the pressure variations in neighboring cylinders is only modest (and can be corrected for).

The complete measuring system is shown in schematic form in Fig 4-13. Two cylinder cover studs on each cylinder are fitted with strain-gauges placed in grooves, covered by a protecting layer. These two studs are placed close to the centerline of the engine in order to reduce the influence from guide forces, and the signals from the two studs are combined in the computer in order to reduce noise effects. The signal to commence data sampling is given by two optical transducers - the one scanning TDC on cylinder 1, the other the 'Zebra' lines of a plastic ribbon attached to the engine.
crankshaft. This combination allows the reading of 600 pulses per revolution, providing data sampling at intervals of approx. 0.6 degree CA, or less, for each cylinder.

![Diagram of cylinder pressure measuring system](image)

**Fig 4-14** Cylinder pressure measuring system based on instrument cylinder cover studs

Data sampling is carried out by means of a multi-controller and an IBM compatible PC, which may be part of the general computing equipment of the ship. Subsequently, mean indicated pressure and firing pressure are calculated for each cylinder, and the engine speed is determined. The results of these calculations may be used as the direct input to a condition monitoring system, and, if desirable, a number of other parameters such as compression pressure, maximum rate of pressure rise, ignition time, etc., may also be determined.

The relative behavior of the cylinders can be determined precisely, and the effects from neighboring cylinders and from different sensitivity of the strain gauge/cover stud
Another important brick in the development towards the "intelligent engine" is cylinder condition control and monitoring.

Each cylinder is fitted with its own camshaft-driven lubricator, which consists of an oil container and a pump block with a row of small built-in pumps, each supplying oil to a lube oil point in the cylinder wall. This system gives a manually adjusted lube oil amount proportional to the engine r/min.

It has been the standard practice to increase the dosage of cylinder oil during start and load change (manoeuvre) of the diesel engine, as experience has shown that risk of wear on the cylinder liners first and foremost occurs in these situations.

The system has also been developed, which automatically registers any load change requiring extra lubrication for a period of approximately 30 min. after the load change.

The nominal lube oil amount can, at part load, be reduced in proportion to the effective mean pressure (m.e.p.), and an arrangement has been developed to make this procedure automatic.

As a natural consequence of the general market requirements, the above systems can be said to be a top priority development area for the future.

The MC-engine, developed in the days when the battle for promoting the reliability and reducing SFOC rated, have evolved into a very reliable and very fuel-efficient engine with a low spares consumption.

In view of the increasing costs of obtaining even lower SFOC for the engine itself, and the current modest level of fuel oil prices, future development will continually, besides concentrating on means to optimize total economy, also focus
on achieving even better reliability, by means of, for instance, introduction of components with a larger margin to cope with deterioration, and systems to help early identification of maintenance needs.

Fig. 4-15 shows the expenses related to major damage of main engines.

<table>
<thead>
<tr>
<th>TYPE OF ENGINE</th>
<th>PERCENTAGE OF TOTAL COSTS</th>
<th>NUMBER OF ENGINES (%)</th>
<th>AVERAGE COST PER DAMAGE (SEK)</th>
</tr>
</thead>
<tbody>
<tr>
<td>2-stroke</td>
<td>21.8</td>
<td>68.9</td>
<td>350,000.-</td>
</tr>
<tr>
<td>4-stroke</td>
<td>78.2</td>
<td>31.1</td>
<td>750,000.-</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>1987</th>
<th></th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>2-stroke</td>
<td>19.8</td>
<td>68.7</td>
<td>350,000.-</td>
</tr>
<tr>
<td>4-stroke</td>
<td>80.2</td>
<td>31.3</td>
<td>1,000,000.-</td>
</tr>
</tbody>
</table>

Fig. 4-15 Expenses related to major damage of main engines (statistics from all Swedish insurance companies)

It is very striking that only 31% of all engines - i.e. medium-speed engines - are responsible for around 80% of all expenses referred to major damage.

The reason is partly that the frequency of damage is higher for medium-speed engines, and partly that such damage is more expensive on medium-speed engines, despite the fact that the low-speed engines in the statistics have higher power and are therefore larger. It should be noted that the profit lost due to off-hire is not included in the statistics.

The statistics are self-explanatory, but a safe conclusion is that the well-known relationship between price and quality exists in the marine diesel engine
business as well. Consequently, the MC-programme is a reliable and also an economical choice.

Further items such as operation behavior and environment questions will be discussed in subsequent chapters of this dissertation.
CHAPTER 5

ECONOMICAL COMPARISON

The comparison presented here concentrates on the economical side and considers the MAN B&W 2-stroke S26MC engine and some competitive 4-stroke engines used frequently in small or specialized ships.

The concept behind the S26MC engine is to allow operators of medium sized and smaller ship types—oceaning, coastal and inland—the opportunity of exploiting the commercial and technical benefits of the low speed crosshead engine.

5.1 Scope of Comparison

Among the parameters involved in the operation of a ship, a reasonable limitation for the study must be evaluated and decided first. If the charter prices, port charges, etc. are assumed that are independent to the main engine choice, it is then sufficient to concentrate on operating costs and first cost of the propulsion system.

A typical breakdown of operating costs is shown in Fig 5-1, as follows:

<table>
<thead>
<tr>
<th>Crew expenses</th>
<th>35%</th>
</tr>
</thead>
<tbody>
<tr>
<td>Port charges</td>
<td>24%</td>
</tr>
<tr>
<td>Fuel &amp; Lub. oil expenses</td>
<td>21%</td>
</tr>
<tr>
<td>Maintenance (ship + engine room)</td>
<td>17%</td>
</tr>
<tr>
<td>Other</td>
<td>3%</td>
</tr>
</tbody>
</table>

The main engine choice primarily affects the fuel.
lubricating oil and maintenance expenses, i.e. 38% of the operating costs.

It is more difficult to estimate the influence of the main engine on crew expenses. At the present time the degree of automation is low on smaller ships. A great part of the crew expenses cover non-engine related purposes, and the main engine has little influence on crew expenses. However, as the manhours used for overhaul are, to some degree, quantifiable, they are included in this chapter.

The following elements are taken into consideration in this comparison:

Capital costs
- propulsion system related capital investments.

Operating costs
- Fuel oil
- Lubricating oil
- Spare Parts
- Maintenance related with the engine

5.2 Capital costs

Prices are difficult to assess, being dependent on time, place and suppliers. The calculation in operating expenses must also be evaluated in relation to differences in capital investments.

It is estimated that the S26MC is approx. 10% more expensive than 4-stroke alternatives (including gear and clutches) with the same propeller equipment. As the S26MC has a fixed propeller, this difference will actually be reduced.

Fig 5-2 shows a price estimate for a 6S26MC (1986):
Fig 5-2: 6S26MC with Shaft Generator and Fixed Pitch Propeller

The four-stroke alternative propulsion systems are assumed to be priced at 617,800 USs.

5.3 Operation costs

5.3.1 Basic Assumptions

If an economical comparison is to be reliable, the different propulsion alternatives must be compared on equal terms.

Therefore, this comparison will be made on the basis of a standard ship - a general cargo carrier of 4000 tdw - and assume that the basic ship data are the same irrespective of the propulsion system chosen.

The data of the ship are as follows:
<table>
<thead>
<tr>
<th>Length</th>
<th>= 90.0 m</th>
</tr>
</thead>
<tbody>
<tr>
<td>Breadth</td>
<td>= 14.9 m</td>
</tr>
<tr>
<td>Draught</td>
<td>= 5.7 m</td>
</tr>
<tr>
<td>Displacement</td>
<td>= 4586 m</td>
</tr>
<tr>
<td>Block coefficient</td>
<td>= 0.60</td>
</tr>
<tr>
<td>Ship speed</td>
<td>= 13.5 knots</td>
</tr>
<tr>
<td>Effective power *</td>
<td>= 1400 BHP</td>
</tr>
</tbody>
</table>

* The effective power is the power needed to move the ship through the water at knots. It is very important that this figure is kept constant throughout the calculation.

To illustrate the situation in a very simple way, the calculations in this study have been restricted to the normal service point (normal ship speed, trim and weather conditions).

1) Propulsion systems

The following propulsion systems are included in this investigation:

6S26MC - MAN B&W Diesel 
6R32 - Wärtsilä 
6M453C - MaK

The two 4-stroke propulsion systems comprise engine, reduction gear, and controllable pitch propeller, whereas the S26MC has a direct coupled propeller.

All alternatives are assumed to have a continuous delivery of 150 kw electrical power from a shaft generator. When taking generator efficiency into account, this means approx. 220BHP.
2) Propeller type

The MCR speed range of the S26MC is 185-250 r/min. In this case the direct coupled S26MC is assumed to operate at 230 r/min. This corresponds to an effective power of 1400 Bhp at 230 r/min, and the propeller can be optima accordingly.

4-stroke alternatives can be operated at the largest propeller diameter allowed by the hull, which is assumed to be:

Max. propeller diameter = 0.67 * ship draught = 3.83 m.

Propeller calculations are made in accordance with the Wageninge B-series propeller characteristics.

As the S26MC is reversible and does not have built-on pumps, there is nothing to prevent the use of a fixed pitch propeller. Therefore, the S26MC is assumed to have a fixed pitch propeller, whereas the 4-stroke engines are assumed to have controllable pitch propellers.
When a shaft generator is installed, it is common practice to install either a constant frequency converter, or to use a controllable pitch propeller so as to be able to run at constant speeds. As a viable alternative, a concept with a variable frequency system is recommended, which allows the propeller/power curve to be a regular cubic curve for highest propulsion efficiency. In the regular speed range of the ship, the electrical frequency will consequently vary from 50 to 60 Hz, which is acceptable. This shaft generator concept gives a simple and inexpensive solution.

Practical experience with this concept has been accumulated in recent years. Some ships in service with "floating frequency" have been followed and have shown satisfactory performance. Only the navigation equipment is bound to use a separate (and inexpensive) frequency converter.

5.3.2 Necessary Propulsion Power

As an S26MC is operated at a propeller speed that differs from that of 4-stroke alternatives, it is very important to get an appropriate estimate on the variation in necessary propulsion power with propeller speed.

However, as a own disagreements between existing empirical theories exist, the effect of two theories can be compared:

1. Harvard
2. Holtrop & Mennen

In principle, the necessary power delivered to the propeller from the shaft can be calculated from the formula:

\[ P_p = \frac{P_e}{\eta_o \cdot \eta_h \cdot \eta_r} \]
Where

\[ \begin{align*}
\text{Pp} & = \text{propulsion power} \\
\text{Pe} & = \text{effective power (here 1400 BHP)} \\
\eta_0 & = \text{open water efficiency of propeller} \\
\eta_h & = \text{hull efficiency} \\
\eta_{rr} & = \text{relative rotative efficiency}
\end{align*} \]

\( \eta_0 \) is calculated from Wageningen B-series.
\( \eta_{rr} \) is assumed to be constant in the study (\( \eta_{rr} = 1.00 \))

\[ \eta_h = \frac{1-t}{1-w} \]

Where "t" is the thrust deduction fraction, i.e. the fraction of the propeller thrust which is not used for propulsion purposes due to propeller/hull interactions.

"W2 is the wake fraction, which is the relative difference between the velocity of the ship (V) and the velocity with which the water flows to the propeller (Va).

The controversy between the two above-mentioned methods lies in the estimation of \( w \) and \( t \). Therefore, in the determination of hull efficiency (\( \eta_h \)).

Harvard's method, which uses the diagram shown in Fig.5-4, predicts a considerable variation in hull efficiency with propeller diameter, whereas Holtrop & Mennen's method, which is based on empirical formulas, gives only a slight variation.

The results of both methods are shown in Fig.5-5.

It can be seen that Harvard's method predicts a difference of only 75 BHP = 4% when the propeller diameter is varied from 3.83 m to 3.10 m, whereas Holtrop & Mennen estimates a difference of 195 BHP = 9%

Few model tests have been carried out which can enlighten this disagreement.
Fig. 5-4: Wake fraction according to Harvaid

\[ W = W_1 + W_2 + W_3 \]

<table>
<thead>
<tr>
<th>Method</th>
<th>Harvaid</th>
<th>Holtrop &amp; Mennen</th>
</tr>
</thead>
<tbody>
<tr>
<td>Engine</td>
<td>6S26MC 4T</td>
<td>6S26MC 4T</td>
</tr>
<tr>
<td>Propeller diameter (m)</td>
<td>3.10</td>
<td>3.10</td>
</tr>
<tr>
<td>Propeller speed (r/min)</td>
<td>230</td>
<td>230</td>
</tr>
<tr>
<td>Wake fraction w</td>
<td>0.290</td>
<td>0.232</td>
</tr>
<tr>
<td>Thrust deduction fraction t</td>
<td>0.195</td>
<td>0.178</td>
</tr>
<tr>
<td>Propeller-efficiency ((\eta_o))</td>
<td>0.567</td>
<td>0.591</td>
</tr>
<tr>
<td>Hull efficiency ((\eta_h))</td>
<td>1.134</td>
<td>1.070</td>
</tr>
<tr>
<td>(\eta_o \eta_h)</td>
<td>0.643</td>
<td>0.632</td>
</tr>
<tr>
<td>Propulsion power (P_p) BHP</td>
<td>2177</td>
<td>2215</td>
</tr>
</tbody>
</table>

Fig. 5-5: Calculation of propulsion power
Recent investigations indicate, however, that Harvaldās method is the most reliable. Lips states that half the change in open-water-efficiency from variation in propeller diameter is cancelled by a counteracting change in hull-efficiency. In the above example a change of approx. 10% occurs in open-water-efficiency. Harvard predicts a change in total efficiency of 4%, i.e. approx. half the change in open-water-efficiency. It remains to be said that the difference shown is not directly transferable to large vessels with high block coefficients.

5.3.3 Calculating Total Brake Power

The propulsion power found above must be corrected to find the necessary brake power to be delivered by the engines.

\[ P_0 = P_p + P_g + P_{cpp} + P_s \]

Where:
- \( P_0 \) = brake power
- \( P_g \) = reduction gear losses
- \( P_{cpp} \) = controllable pitch propeller losses
- \( P_s \) = shaft losses

1). Gear loss

Gear manufacturers claim that gear losses only amount to 1-1.5%, whereas shipyards often use 3-4% to be on the safe side. ship-builders use the figure is often used in economical comparisons 2% (50 BHP).

2). CPP loss
Some CPP manufacturers claim that no losses occur from using CPP. This is unlikely because of the relatively larger hubs of CP-propellers. Some yards even consider a figure of 3-4% to be realistic. 2% (50%) is used in this comparison.

3). Shaft loss

Shaft losses are equal for all alternatives, and are estimated to be 2% (50%).

Based on the above, the following brake powers can be calculated:

It should be noted that according to Harvard, a 6S26MC will need less brake power than the 4-stroke alternatives, despite the higher propeller speed. No gear losses and no CPP losses occur in the FPP direct coupled solution. This more than compensates for the higher propeller speed. According to Holtrop & Mennen, the 6S26MC needs 95 BHP = 4% more power than the 4-stroke alternatives. The comparison is shown on Fig.5-6.

<table>
<thead>
<tr>
<th>Method</th>
<th>Harvald</th>
<th>Holtrop &amp; Mennen</th>
</tr>
</thead>
<tbody>
<tr>
<td>Engine BHP</td>
<td>6S26MC</td>
<td>4T</td>
</tr>
<tr>
<td>Pprop BHP</td>
<td>2177</td>
<td>2102</td>
</tr>
<tr>
<td>Gear loss BHP</td>
<td>-</td>
<td>50</td>
</tr>
<tr>
<td>CPP loss BHP</td>
<td>-</td>
<td>50</td>
</tr>
<tr>
<td>Shaft loss BHP</td>
<td>50</td>
<td>50</td>
</tr>
<tr>
<td>Ppto 150 kW BHP</td>
<td>220</td>
<td>220</td>
</tr>
<tr>
<td>Po, Brake Power</td>
<td>2447</td>
<td>2472</td>
</tr>
</tbody>
</table>

Fig 5-6: Calculation of Brake Power
5.3.4 Fuel Oil Consumption

Having calculated the necessary brake power, the fuel oil consumption of the different engines can be calculated (Fig. 5-7), based on a fuel price of 120 s/t and 6000 hours service/year.

<table>
<thead>
<tr>
<th>Harvold</th>
<th>6R32 CPP</th>
<th>8M453C CPP</th>
<th>6S26MC FPP</th>
</tr>
</thead>
<tbody>
<tr>
<td>Engine power $P_o$ (BHP)</td>
<td>2472</td>
<td>2472</td>
<td>2447</td>
</tr>
<tr>
<td>SFOC **) (g/BHPh)</td>
<td>139.8</td>
<td>132.2</td>
<td>128.4</td>
</tr>
<tr>
<td>HFO ***) t/Year</td>
<td>2203</td>
<td>2082</td>
<td>2003</td>
</tr>
<tr>
<td>HFO ($/Year)</td>
<td>264360</td>
<td>249840</td>
<td>240360</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Holtrop &amp; Mennen</th>
<th>6R32 CPP</th>
<th>8M453C CPP</th>
<th>6S26MC FPP</th>
</tr>
</thead>
<tbody>
<tr>
<td>Engine power $P_o$ (BHP)</td>
<td>2390</td>
<td>2390</td>
<td>2485</td>
</tr>
<tr>
<td>SFOC **) (g/BHPh)</td>
<td>140.0</td>
<td>132.6</td>
<td>128.3</td>
</tr>
<tr>
<td>HFO ***) t/Year</td>
<td>2132</td>
<td>2019</td>
<td>2032</td>
</tr>
<tr>
<td>HFO ($/Year)</td>
<td>255840</td>
<td>242280</td>
<td>243840</td>
</tr>
</tbody>
</table>

**) Lower calorific value for MDO is 42700 kJ/kg

***) Lower calorific value for HFO is 40200 kJ/kg

Fig 5-7: Calculation of Fuel Oil Consumption

It can be seen that due to lower SFOC, the 6S26MC will, in spite of a higher propeller speed, use less HFO than the 4-stroke engines (except for the 8M453c calculated by Holtrop & Mennen).

Hence, a lower propulsion efficiency at higher propeller speed for the 6S26MC is outweighed and more than eliminated by gear losses and controllable pitch propeller losses for
4-stroke engines, combined with a lower specific fuel consumption of the 6526MC.

5.3.5 Other Operating Costs

1) Lub oil consumption

In this comparison, the generally accepted lub. oil consumption of 1.0 g/BHP/h for all the 4-stroke engines is used. The lub. oil consumption for 4-stroke engines is assumed to be constant at part load, although a rising tendency at part load is well known. The lub. oil consumption of all MC-engines, including the 526MC, includes cylinder oil, which is tightly controlled by timed lubricators at 0.6-0.9 g/BHP/h, and system oil, which is about 0.15 g/BHP/h. This means that the actual lub. oil consumption of the 2-stroke engine is less than the claimed figures for the 4-stroke engines. As cylinder oil is a little more expensive than medium speed engine oil, the lubricating oil consumption will be considered as being equal.

2) Spare Parts Consumption

The calculation of spare parts consumption is based on lists from engine manufacturers stating times of service life and prices of spare parts. The spare parts consumption will, normally, vary much during the service period. In this comparison, the spare parts consumption as an average over approx. 60,000 hours is calculated. Only spare parts which are expected to be used under normal conditions are taken into consideration, which means that the gear wheel drive, camshaft, cylinder cover, etc. have been omitted.

Data on prices, life-time and overhaul intervals are shown in Fig 5-8.
<table>
<thead>
<tr>
<th>Parts</th>
<th>6R32</th>
<th></th>
<th></th>
<th>8M453C</th>
<th></th>
<th></th>
<th>6S26MC</th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Price</td>
<td>Life</td>
<td>Over-</td>
<td>Price</td>
<td>Life</td>
<td>Over-</td>
<td>Price</td>
<td>Life</td>
<td>Over-</td>
</tr>
<tr>
<td></td>
<td>(US$)</td>
<td>time</td>
<td>haul-</td>
<td>(US$)</td>
<td>time</td>
<td>haul-</td>
<td>(US$)</td>
<td>time</td>
<td>haul-</td>
</tr>
<tr>
<td></td>
<td>(Hours)</td>
<td>(Hours)</td>
<td>ing</td>
<td>(Hours)</td>
<td>(Hours)</td>
<td>ing</td>
<td>(Hours)</td>
<td>(Hours)</td>
<td>ing</td>
</tr>
<tr>
<td>Main piston</td>
<td>5560</td>
<td>50000</td>
<td>8000</td>
<td>7395</td>
<td>60000</td>
<td>20000</td>
<td>2627</td>
<td>64000</td>
<td>12000</td>
</tr>
<tr>
<td>Piston rings/scraper rings</td>
<td>234</td>
<td>16000</td>
<td>8000</td>
<td>393</td>
<td>10000</td>
<td>20000</td>
<td>61</td>
<td>12000</td>
<td>12000</td>
</tr>
<tr>
<td>Sealing rings</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>466</td>
<td>-</td>
<td>12000</td>
</tr>
<tr>
<td>Lamella rings</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>71</td>
<td>-</td>
<td>12000</td>
</tr>
<tr>
<td>Cylinder head</td>
<td>-</td>
<td>-</td>
<td>8000</td>
<td>-</td>
<td>-</td>
<td>20000*</td>
<td>-</td>
<td>-</td>
<td>12000</td>
</tr>
<tr>
<td>Cylinder liner</td>
<td>3050</td>
<td>48000</td>
<td>16000</td>
<td>1761</td>
<td>40000</td>
<td>10000</td>
<td>2294</td>
<td>50000</td>
<td>12000</td>
</tr>
<tr>
<td>Inlet valve</td>
<td>344</td>
<td>24000</td>
<td>8000</td>
<td>184</td>
<td>20000</td>
<td>10000</td>
<td>-</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>Exhaust valve</td>
<td>692</td>
<td>20000</td>
<td>8000</td>
<td>384</td>
<td>15000</td>
<td>5000</td>
<td>1666</td>
<td>40000</td>
<td>4000</td>
</tr>
<tr>
<td>Fuel pump element</td>
<td>1003</td>
<td>24000</td>
<td>8000</td>
<td>900*</td>
<td>24000*</td>
<td>8000*</td>
<td>1094</td>
<td>32000</td>
<td>16000</td>
</tr>
<tr>
<td>Fuel valve nozzle</td>
<td>230</td>
<td>4000</td>
<td>6000</td>
<td>179</td>
<td>3000</td>
<td>6000*</td>
<td>113</td>
<td>16000</td>
<td>8000</td>
</tr>
<tr>
<td>Main bearing</td>
<td>300</td>
<td>32000</td>
<td>16000</td>
<td>513</td>
<td>20000</td>
<td>10000*</td>
<td>613</td>
<td>96000</td>
<td>8000</td>
</tr>
<tr>
<td>Crankpin bearing</td>
<td>252</td>
<td>24000</td>
<td>8000</td>
<td>513</td>
<td>20000</td>
<td>10000*</td>
<td>585</td>
<td>96000</td>
<td>8000</td>
</tr>
<tr>
<td>Bearings for turbocharger</td>
<td>3666</td>
<td>15000*</td>
<td>24000*</td>
<td>3600*</td>
<td>15000*</td>
<td>24000*</td>
<td>2026</td>
<td>24000</td>
<td>24000</td>
</tr>
<tr>
<td>Crosshead bearing</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>1200</td>
<td>64000</td>
<td>8000</td>
</tr>
</tbody>
</table>

* Estimated

Fig 5-8 Spare part data from Engine Manufacturers

Based on these figures above, the yearly expenses in US$ have been calculated and shown in Fig 5-9.

It can be seen that the spare parts expenses for the S26MC are only half of those of the 4-stroke alternatives.
Calculation of the time consumption for maintenance is based on the overhaul figures provided by engine manufacturers, shown as Fig 5-10.

<table>
<thead>
<tr>
<th></th>
<th>6R32 Spare parts/ year</th>
<th>Expens./ year (US$)</th>
<th>6H453C Spare parts/ year</th>
<th>Expens./ year (US$)</th>
<th>6S26MC Spare parts/ year</th>
<th>Expens./ year (US$)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Main piston</td>
<td>6</td>
<td>0.72</td>
<td>4003</td>
<td>8</td>
<td>0.80</td>
<td>5916</td>
</tr>
<tr>
<td>Piston rings/ scraper rings</td>
<td>6</td>
<td>2.25</td>
<td>527</td>
<td>8</td>
<td>4.80</td>
<td>1896</td>
</tr>
<tr>
<td>Sealing rings</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>Lamella rings</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>Cylinder head</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>Cylinder liner</td>
<td>6</td>
<td>0.75</td>
<td>2288</td>
<td>8</td>
<td>1.20</td>
<td>2113</td>
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<tr>
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<td>12</td>
<td>3.00</td>
<td>258</td>
<td>16</td>
<td>4.80</td>
<td>883</td>
</tr>
<tr>
<td>Exhaust valve</td>
<td>12</td>
<td>3.60</td>
<td>2491</td>
<td>16</td>
<td>6.40</td>
<td>2458</td>
</tr>
<tr>
<td>Fuel pump element</td>
<td>6</td>
<td>1.50</td>
<td>1505</td>
<td>8</td>
<td>2.00</td>
<td>1800</td>
</tr>
<tr>
<td>Fuel valve nozzle</td>
<td>6</td>
<td>9.00</td>
<td>2070</td>
<td>8</td>
<td>16.00</td>
<td>2864</td>
</tr>
<tr>
<td>Main bearing</td>
<td>7</td>
<td>1.31</td>
<td>393</td>
<td>9</td>
<td>2.40</td>
<td>1231</td>
</tr>
<tr>
<td>Crankpin bearing</td>
<td>6</td>
<td>1.50</td>
<td>378</td>
<td>8</td>
<td>2.40</td>
<td>1231</td>
</tr>
<tr>
<td>Turbocharger bearings</td>
<td>1</td>
<td>0.40</td>
<td>1466</td>
<td>1</td>
<td>0.53</td>
<td>1920</td>
</tr>
<tr>
<td>Crosshead bearing</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>Total</td>
<td></td>
<td>15379</td>
<td></td>
<td>22312</td>
<td></td>
<td>7764</td>
</tr>
</tbody>
</table>

Fig 5-9: Calculation of yearly Spare Parts Expenses
For the reasons of comparison, a number describing the time consumption necessary is introduced, for the overhauling of a given component. As a base, the time consumption for piston overhaul on an S26MC to index 1.00 is set.

<table>
<thead>
<tr>
<th>Component</th>
<th>6832</th>
<th>8MA53C</th>
<th>6526MC</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Time index</td>
<td>Time index</td>
<td>Time index</td>
</tr>
<tr>
<td></td>
<td>Overhaul time*</td>
<td>Overhaul time*</td>
<td>Overhaul time*</td>
</tr>
<tr>
<td></td>
<td>Overhaul/year</td>
<td>Overhaul/year</td>
<td>Overhaul/year</td>
</tr>
<tr>
<td>Main piston</td>
<td>6 4.50 1</td>
<td>8 2.40 1</td>
<td>6 3.00 1</td>
</tr>
<tr>
<td>Piston rings</td>
<td>6 4.50 1/10</td>
<td>8 2.40 1/10</td>
<td>6 3.00 1/10</td>
</tr>
<tr>
<td>Sealing rings</td>
<td>- - -</td>
<td>- - -</td>
<td>- - -</td>
</tr>
<tr>
<td>Lamella rings</td>
<td>- - -</td>
<td>- - -</td>
<td>- - -</td>
</tr>
<tr>
<td>Cylinder head</td>
<td>6 4.50 1</td>
<td>8 2.40 1</td>
<td>6 3.00 1</td>
</tr>
<tr>
<td>Cylinder liner</td>
<td>6 2.25 1/3</td>
<td>8 4.80 1/3</td>
<td>6 3.00 1/3</td>
</tr>
<tr>
<td>Inlet valve</td>
<td>12 9.00 1/2</td>
<td>16 8.60 1/2</td>
<td>- - -</td>
</tr>
<tr>
<td>Exhaust valve</td>
<td>12 9.00 1/2</td>
<td>16 9.60 1/2</td>
<td>6 9.00 1/2</td>
</tr>
<tr>
<td>Fuel pump element</td>
<td>6 4.50 1/4</td>
<td>8 6.00 1/4</td>
<td>6 2.25 1/4</td>
</tr>
<tr>
<td>Fuel valve nozzle</td>
<td>6 6.00 1/10</td>
<td>8 8.00 1/10</td>
<td>12 9.00 1/10</td>
</tr>
<tr>
<td>Main bearing</td>
<td>7 2.60 1/3</td>
<td>9 3.40 1/3</td>
<td>7 5.25 1/3</td>
</tr>
<tr>
<td>Crankpin bearing</td>
<td>6 4.50 1/3</td>
<td>8 4.80 1/3</td>
<td>6 4.50 1/3</td>
</tr>
<tr>
<td>Turbochr. bearings</td>
<td>1 0.25 1</td>
<td>1 0.25 1</td>
<td>1 0.25 1</td>
</tr>
<tr>
<td>Crosshead bearing</td>
<td>- - -</td>
<td>- - -</td>
<td>- - -</td>
</tr>
<tr>
<td>Total</td>
<td>23.55</td>
<td>21.02</td>
<td>19.46</td>
</tr>
</tbody>
</table>

Fig 5-10 Calculation of Overhauling Time
Furthermore, it is assumed that a given component needs the same time for overhaul on the S26MC as on the chosen 4-stroke engines. Thereafter, the time consumption for individual components is estimated as being relative to index 1.00 for one piston. The results obtained and shown in Fig 5-10 are a total index, which is a measure of relative time consumption.

4) Total maintenance expenses

Estimating a yearly time consumption of 450 hours for S26MC, the yearly time consumption is calculated for the 4-stroke alternatives according to the indexes.

The cost per hour can differ considerably from country to country. 25 US$/h is used in this comparison.

<table>
<thead>
<tr>
<th>Engine</th>
<th>Time index</th>
<th>Hours/year</th>
<th>Personnel expenses US$/year</th>
<th>Spare parts expenses US$/year</th>
<th>Total maintenance US$/year</th>
</tr>
</thead>
<tbody>
<tr>
<td>6S26MC</td>
<td>19.46</td>
<td>450</td>
<td>11250</td>
<td>7764</td>
<td>19014</td>
</tr>
<tr>
<td>6R32</td>
<td>23.55</td>
<td>545</td>
<td>13625</td>
<td>15379</td>
<td>29004</td>
</tr>
<tr>
<td>8M453C</td>
<td>21.02</td>
<td>486</td>
<td>12150</td>
<td>22312</td>
<td>34462</td>
</tr>
</tbody>
</table>

Fig 5-11 Calculation of Maintenance Costs

It is normally accepted that spare parts and maintenance costs on 4-stroke engines are approx. twice as high as on 2-stroke engines. Some sources even report a ratio of 3:1.

From the above calculations, it is seen that the S26MC has significantly lower spare parts and maintenance costs than the 4-stroke engines, even when using the engine manufacturers' own figures!

It can be concluded that the S26MC will help the owner to reduce these costs.
5.4 Economical Comparison

It is often seen that conclusions in economical comparisons are based on very few parameters. In this comparison, all the cost parameters discussed in this paper will be taken into account.

In order to evaluate the extreme situations, the calculation will be built on the basis of two scenarios:

5.4.1 Scenario I:

1) Harvard's method for estimation of brake power
2) The expenses for spare parts and maintenance for 4-stroke engines are twice the expenses for S26MC.

This scenario represents the maximum expected savings for S26MC relative to the chosen 4-stroke engines.

<table>
<thead>
<tr>
<th></th>
<th>6R32</th>
<th>8M453C</th>
<th>S26MC</th>
</tr>
</thead>
<tbody>
<tr>
<td>HFO *)</td>
<td>264360</td>
<td>249840</td>
<td>240360</td>
</tr>
<tr>
<td>Lub. oil **)</td>
<td>16315</td>
<td>16315</td>
<td>16150</td>
</tr>
<tr>
<td>Spare parts</td>
<td>15528</td>
<td>15528</td>
<td>7764</td>
</tr>
<tr>
<td>Maintenance</td>
<td>22500</td>
<td>22500</td>
<td>11250</td>
</tr>
<tr>
<td>Total</td>
<td>318703</td>
<td>304183</td>
<td>275524</td>
</tr>
<tr>
<td>Difference</td>
<td>43179</td>
<td>28659</td>
<td>0</td>
</tr>
</tbody>
</table>

Fig 5-12 Scenario I (Expected Operating Costs)

5.4.2 Scenario II:

1) Holtrop & Mennen's method for estimation of brake power
2) Expenses for spare parts and maintenance from Fig 5-9.

This scenario represents the minimum expected savings on an S26MC Relative to 4-stroke engines.
<table>
<thead>
<tr>
<th></th>
<th>6R32</th>
<th>6M453C</th>
<th>S26MC</th>
</tr>
</thead>
<tbody>
<tr>
<td>HFO *)</td>
<td>255640</td>
<td>242280</td>
<td>243840</td>
</tr>
<tr>
<td>Lub. oil **)</td>
<td>15774</td>
<td>15774</td>
<td>16401</td>
</tr>
<tr>
<td>Spare Parts</td>
<td>15379</td>
<td>22312</td>
<td>7764</td>
</tr>
<tr>
<td>Maintenance</td>
<td>13625</td>
<td>12150</td>
<td>11250</td>
</tr>
<tr>
<td>Total</td>
<td>300618</td>
<td>292516</td>
<td>279255</td>
</tr>
<tr>
<td>Difference</td>
<td>21363</td>
<td>13261</td>
<td>0</td>
</tr>
</tbody>
</table>

*) HFO-price = 120 USs/t  
**) Lub. oil price = 1100USs/t

Fig 5-13 Scenario II (Expected Operating Costs)  
Net Present Value (NPV) Curves

To illustrate the financial consequence of the above extra costs on the total economy, the net present method (NPV). The results are illustrated in Fig 5-14 & 15.
TOTAL ECONOMY

SAVINGS FOR 6S26MC COMPARED TO 6R32

NET PRESENT VALUE
1000 USD

DISCOUNT RATE PERCENT: 8.0
RATE OF INFLATION PERCENT: 4.0
FUEL OIL PRICE (ME) USD/T: 120.0

Fig.5-14
TOTAL ECONOMY

SAVINGS FOR 6S26MC COMPARED TO BM453C

NET PRESENT VALUE
1000 USD

-150 -100 -50 0 50 100 150 200 250 300 350 400 450 500 550 600 650 700 750

YEARS AFTER INVESTMENT

MAXIMUM SAVINGS
AVERAGE SAVINGS
MINIMUM SAVINGS

DISCOUNT RATE PERCENT: 8.0
RATE OF INFLATION PERCENT: 4.0
FUEL OIL PRICE (ME) USD/T: 120.0

Fig.5-15:
ENGINE SELECTION PROGRAM

As mentioned before, today the MC-programme composes power range from 1,100 BHP to 70,440 BHP and speed from 59 r/min to 250 r/min.

Fig 4-2 shows, by means of superimposed diagrams for all engine types, the entire layout area for the MC-programme in a power/speed diagram. As it can be seen, there is a considerable overlap of power/speed combinations so that for nearly all applications, there is a wide selection of different engines to choose from, all of which meet the individual ship's requirements.

6.1 The Engine Selection Process

The engine selection programme is based on the MC-engine series, in order to select the optimum marine engine. Therefore, it is necessary to get the detailed information in the particular engine. The information contains all the relevant data for MC-engine types. It facilitates comparison and selection of the main engine at the initial stage.

The main points in an engine selection process can be seen in Fig.3-1. Although the process concentrates on a simple method, it is practice aimed at estimating the ship main particulars, the power/speed combination, and then to choose the main engine that fulfills the ship's requirements.

6.2 SHIP'S POWER REQUIREMENT

6.2.1 Power requirement

At the initial stage of the process, the shipowner or
shipyard usually stipulate the ship type, ship size and the design speed of the ship. From this very limited data, the estimation of power/speed requirement can only be an approximation. The curves in this chapter show such an estimate based on the installed power of a large number of ships in service.

Fig. 6-1 reflect the main particulars of the ships in question. These figures are only valid for a service speed of 14 knots for general cargo, tankers and bulk carriers. For other service speeds than those assumed, it can be said that the length will increase with increased service speed, whereas the block coefficient will decrease.

Fig. 6-2: Main particulars of bulk carriers, tankers and general cargo ships

The following parameters are estimated in the figures:

Lpp = Length between perpendiculars in meters
B = Breadth in meters
T = Draught in meters
$s = \text{Block coefficient}$

Fig 6-2 shows the power of typical bulk carriers, tankers and general cargo ships.

These curves reflect the power installed in existing ships as a statistical average.

6.2.2 Propeller Speed

Once the power requirement has been found the optimum propeller speed can be found, by using the optimum propeller diameter. As fig 6-3 shows (which is a statistically based average propeller diameter installed in existing ship):
Fig. 6-3 Draught and proposed maximum propeller diameter

By means of this propeller diameter the optimum propeller speed can be found, by using the figure 6-4.

6.2.3 Example
The example illustrates how the above curves are used.

Given:

30,000 dwt tanker,
service speed 14.0 knots

a). Estimate: Main particulars (Fig. 6-1)

Lpp = 170 m
B = 26 m
D = 11 m
s = 0.775 m

b). Estimate: Installed power (Fig 6-3)

P = 8,000 BHP

c). Estimate: Propeller diameter (Fig. 6-4)

D = 6.0 m

---

**Fig. 6-4:** Engine power, propeller diameter and propeller speed for 4-bladed propellers
6.3 Engine selection

Engine selection is the next step in the procedure, just after the optimum power/speed combination of the ship has been found.

6.3.1 Engine type

The normal procedure is to start by finding the relevant engine types and sizes in a certain engine speed range, such as the range between 100 and 150 r/min.

1) a-Curve

Fig. 6-5: shows the selection. Curves I; II; III are called a-curves, also ‘equal ship speed curve’, which expresses the relationship between increasing the power requirement and propeller speed.

The correlation can be described as follows:

\[
P = \frac{N}{N_{\text{ref}}} = \frac{P_{\text{ref}}}{a}, \quad N > N_{\text{ref}}
\]

\( N_{\text{ref}} \): reference propeller speed
\( N \): selected propeller speed
\( P_{\text{ref}} \): power at reference propeller speed (\( N_{\text{ref}} \))
\( P \): necessary power at propeller speed \( N \)

There are different values for different ship's applications. For general cargo, bulk carriers and tankers, the following data applies:

\( a = 0.15 \) for a ship up to 10,000 dwt
\( a = 0.20 \) for a ship from 10,000 to 30,000 dwt
\[ a = 0.25 \text{ for a ship of more than 30,000 dwt} \]

Fig. 6-5  Engine choice for a 30,000 dwt tanker with 8,000 BHP at 100 R/min
The a-curve can be used to compare different engine alternatives at varying propeller speeds. In Fig 6. 9 three different propeller curves are shown:

I. Propeller curve corresponding to reference propeller layout
II. Propeller curve corresponding to minimum propeller speed with 5S50 MC
III. Propeller curve corresponding to minimum propeller speed with 5L50 MC.

6.3.2 Engine type

The way of finding the relevant engine types can be seen from Fig.6-6, where the relevant five cylinder engines have been found. Usually, no more than two or the cylinder numbers are relevant for a project, which is why the preferred number of cylinders has been used as the basis for the curve. Using the power requirement per cylinder at a given cylinder number, the relevant engine can be found by drawing the relevant a-curve through the reference power/speed combination.

Example 1:
The following example illustrates how to choose the relevant engines.

Power/speed requirement:
8,000 BHP at 100 r/min
Choice of six cylinder engines:
Power/cal = 1,333 BHP/CAL
In Fig.6-7, the relevant engines would be 6L60, 6L50MC, 6S50MC.

Choice of five cylinder engines:
Power/cal = 1,600 BHP/cal
Relevant engines: 5S60MC, 5L60MC, 5S50MC, 5L50MC.

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Fig. 6-6: Engine selection based on engine power or cylinder power requirement
Choice of four cylinder engines:
Power/cal = 2,000 BHP/cal
Relevant engines: 4L60MC, 4S60MC, L70MC.

6.3.3 Operation costs

Once the relevant engine types have been found, a more detailed engine comparison can be made, which means the operation costs, viz:
- Fuel oil
- Lubricating oil
- Spare parts
- Overhaul work

a). Fuel oil

For comparison of the fuel consumption it is necessary to know the exact power requirement for the different engine alternatives depending on the actual propeller speed and efficiency. This can be found from the a-curve as explained above.

EXAMPLE 2:
Pref = 8,000 BHP, Nref = 100 r/min, a = 0.20
Engine type chosen: 5L50MC
Necessary propeller speed: N = 140 r/min
Necessary engine power: P = 8,557 BHP

b). Lub.Oil

The lubricating oil consumption and the cylinder oil consumption can be found by using Fig. 6-7 and 6-8.

The price level is about 1,000 USD/ton for system lubricating oil and about 1,300 USD/ton for cylinder lubricating oil (price basis 1991).
Fig. 6-7 Consumption in tons/6,000 hours of system lubricating oil

Cyl. oil consumption

\[
\begin{array}{|c|c|c|}
\hline
\text{tons/6000 hours} & \text{50} & \text{100} \\
\hline
\text{5,000} & \text{30} & \text{60} \\
\text{10,000} & \text{60} & \text{120} \\
\text{50,000} & \text{120} & \text{240} \\
\text{100,000} & \text{240} & \text{480} \\
\hline
\end{array}
\]

Fig. 6-8 Consumption in tons/6,000 hours of cylinder lubricating oil
c). Spare part

The estimated average spare part expenses over a long period of operation time (6,000 hours) can be found by using Fig.6-9 (price basis 1991).

![Graph showing spare part expenses per cylinder/6,000 hour, average price for 60,000 hours.]

**Fig.6-9** Spare part expenses per cylinder/6,000 hour, average price for 60,000 hours.

d). Maintenance

The estimated hours for maintenance and the corresponding expenses with a labour cost of 322 DKK/hour can be
calculated by means of Fig. 6-10.

Fig. 6-10  Labour expenses for overhaul work

6.4 Specific Fuel Oil Consumption

To calculate the exact operating cost it is, of course, necessary to calculate the total fuel consumption for the four alternative shown above.

The calculation of the SFOC for each engine type can be carried out by means of Fig. 6-11 to 6-16.

The SFOC of the engine in the whole load area depends on
where the optimising point is chosen, and whether a Turbo Compound System (TCS) is to be used.

The SFOC curves in Fig. 6-11 to 6-16 are valid for engines with and without TCS and refer to ISO ambient conditions:
ISO 3046/1-1986:
1.000 mbar ambient air pressure
25°C ambient air temperature
25°C cooling water temperature.
and are based on a fuel oil with a lower calorific value of 10,200 kcal/kg (42,707 kJ/kg).

6.4.1 Optimising point

The optimising point 0 is drawn into diagram 1 in the above-mentioned Fig. 6-11 to 6-16. A straight line along the constant mep curves (parallel to L1-L3) is drawn through the optimising point 0. The line intersection with the heavy lines (without TCS) and the dashed lines (with TCS) indicates the reduction in SFOC at 100%, 80% and 50% of the optimising power, related to the SFOC at nominal MCR(L1) rating which are stated in the figures.

The procedure of such a service optimising of an L60MC engine optimised at 01 or at 02 = 85% power of 01 are illustrated in Fig 6-17. This shows that the lowest SFOC is obtained at about 80% of the rating in the optimising point.
Diagram 1

<table>
<thead>
<tr>
<th>Engine type</th>
<th>Data at nominal MCR:</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>BHP</td>
</tr>
<tr>
<td></td>
<td>r/min</td>
</tr>
<tr>
<td></td>
<td>g/BHPPh</td>
</tr>
</tbody>
</table>

Diagram 2

Fig. 6-11 SFOC for engine with fixed propeller Valid for 90MC-80MC-70MC-60MC-50MC and 42MC
Engine type:

Data at nominal MCR:
- 100% Power: BHP
- 100% Speed: r/min
- Nominal SFOC: g/BHP

Data at specified MCR
- 100% Power: BHP
- 100% Speed: r/min
- SFOC: g/BHP

Please note that the output of the TCS, if any, shall be added to the engine's output, if the power turbine is not coupled to the engine's shaft.

It is assumed that the TCS power is fed back to the engine shaft.

**Fig. 6-12**  SFOC for engine with fixed pitch propeller

Valid for K90MC-C and K80MC-C
Fig. 6-13 SFOC for engine with fixed propeller valid for L35MC and S26MC

<table>
<thead>
<tr>
<th>Engine type:</th>
</tr>
</thead>
<tbody>
<tr>
<td>Data at nominal MCR:</td>
</tr>
<tr>
<td>100% Power:</td>
</tr>
<tr>
<td>100% Speed:</td>
</tr>
<tr>
<td>Nominal SFOC:</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Data at specified MCR</th>
</tr>
</thead>
<tbody>
<tr>
<td>100% Power:</td>
</tr>
<tr>
<td>100% Speed:</td>
</tr>
<tr>
<td>SFOC:</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>SFOC in g/BHPh at nominal MCR (L1)</th>
</tr>
</thead>
<tbody>
<tr>
<td>L35MC</td>
</tr>
<tr>
<td>S26MC</td>
</tr>
</tbody>
</table>
Diagram 1

Diagram 2

Please note that the output of the TCS, if any, shall be added to the engine's output, if the power turbine is not coupled to the engine's shaft.

It is assumed that the TCS power is fed back to the engine shaft.

**Fig. 6-14** SFOC for engine with constant speed Valid for
90MC-80MC-70MC-60MC-50MC and 42MC
Diagram 1

<table>
<thead>
<tr>
<th>Power (%)</th>
<th>100% Power</th>
<th>100% Speed</th>
</tr>
</thead>
<tbody>
<tr>
<td>50%</td>
<td>BHP</td>
<td>r/min</td>
</tr>
<tr>
<td>60%</td>
<td></td>
<td></td>
</tr>
<tr>
<td>70%</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

Diagram 2

<table>
<thead>
<tr>
<th>Power (%)</th>
<th>0%</th>
<th>10%</th>
<th>20%</th>
<th>30%</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

Please note that the output of the TCS if any shall be added to the engine's output, if the power turbine is not coupled to the engine's shaft.

It is assumed that the TCS power is fed back to the engine shaft.

Fig. 6-15 SFOC for engine with constant speed Valid for K90MC-C and K80MC-C
Fig. 6-16 SFOC for engine with constant speed. Valid for L35MC and S26MC.
Diagram 1

**Engine type:** 6L60MC

<table>
<thead>
<tr>
<th>Data at nominal MCR:</th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>100% Power:</td>
<td>15,600 BHP</td>
</tr>
<tr>
<td>100% Speed:</td>
<td>123 r/min</td>
</tr>
<tr>
<td>Nominal SFOC:</td>
<td>128 g/BHP</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Data at specified MCR</th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>100% Power:</td>
<td>12,800 BHP</td>
</tr>
<tr>
<td>100% Speed:</td>
<td>110.7 r/min</td>
</tr>
<tr>
<td>SFOC:</td>
<td>126 g/BHP</td>
</tr>
</tbody>
</table>

Please note that the output of the TCS, if any, shall be added to the engine's output, if the power turbine is not coupled to the engine's shaft.

It is assumed that the TCS power is fed back to the engine shaft.

**Fig. 6-17** Example of SFOC for 6L60MC with fixed pitch propeller
Fig. 6-18 Example of SFOC at part load for a L60MC as a function of the optimising point.
It should be noted that the 100%, 80% and 50% graphs refer to the optimising point 0, and only gives the specific fuel oil consumption for loads equal to or lower than the optimising point.

For calculation of the SFOC between the optimising point and the specified MCR (M) and outside the optimising curve, it should be referred to the description below.

The optimising point must be between 85% and 100% of the specified MCR (M).

By optimising the engine at a lower power than the specified MCR (M), a saving of 1-3 g/BHPh can be obtained in the range between 50% and 85% of M, which is the normal operating range for most ships. It is, however, important to note that the SFOC is increased above the optimising point 02 - this penalty of SFOC in point M will be about 4 g/BHPh relative to the SFOC in 0.2

The virtues of such a service optimising made at 100%, 90%, 80% or 70% of nominal MCR (L1) appears from Fig.6-19. This shows that the fuel penalty for obtaining savings in SFOC by optimising at lower loads is a limitation to the maximum available power.

6.4.2 Fuel consumption at Arbitrary Load

Once the engine has been optimised in point 0, Fig.6-19, the specific fuel oil consumption in and arbitrary point S1, S2 of S3 can be estimated based on the SFOC in point '1' ans '2'.

These can be calculated by using the graphs in Fig.6-12 to 6.14 for the propeller curve I and Fig.6.15 to 6.17 for the constant speed curve II, obtaining the SFOC in point 1 and 2, respectively.

Then the SFOC for point S1 can be calculated as an interpolation between the SFOC in points '1' and '2', and
for points S2 and S3 as an extrapolation.

The above-mentioned method provides only an approximate figure. A more precise indication of the expected SFOC at any load can be calculated by using our computer program. This is a service which is available to our customers on request.

![Fig.6-19 SFOC at an arbitrary load](image)

**6.4.3 SFOC with Turbo Compound System (TCS)**

The engine can be delivered with TCS, leading to an
overall reduction of the fuel oil consumption on board the ship. For the MAN B&W standardized system, the power produced by the power turbine is led back to the crankshaft through a reduction gear thereby giving a decrease in the specific fuel oil consumption of the main engine.

The SFOC with and without TCS is shown in Fig. 6-11, 6-12, and 6-14, 6-15 for the respective engine types. The engine types L35MC and S26MC can not be delivered with TCS.

Below 50% load the TCS is cut off, leading to increased scaveng air pressure and combustion pressure and decreased SFOC.

6.5 Engine Room Packages

The engine room packages have evolved from MAN B&W's long experience as designers of diesel engines and controllable pitch propellers for the lower power range of MC-engines.

The extent of the package could include all or some of the following units:

. Main engine
. Controllable pitch propeller
. Remote control system
. Power take off
. Auxiliary units
. Other auxiliary equipment

( Item 1 is described in previous chapter).

6.5.1 Power take off

Electricity production is the second largest fuel consumption on board. The electricity is produced by using one or more of the following types of machinery, either running along or in parallel:

. Auxiliary diesel generating sets
Main engine drive generators
Steam driven turbogenerators
Emergency diesel generating sets

The machinery installed should be selected based on an economic evaluation of first cost, operating costs, and the demand of man-hours for maintenance.

1). PTO

With a main engine driven generator coupled to a Power Take Off (PTO), the electricity can be produced based on the main engine's low SFOC and use of heavy fuel. Several standard PTO systems are available (Fig.6-20).

For the smaller engine types, especially the L35 and the S26 the step-up gear and generator have to be located on a separate seating i.e. the BW II or BW IV system is to be used.

2). Gear Constant Ratio, PTO/GCR

The Power Take Off type BWII/GCR, Fig.6-21 alternative 11, is mainly used together with engine types L42MC, L35MC of S26MC, but it can also be applied on the larger engines.

The system has been developed together with the British manufacturer Newbrook, for the purpose of generating electrical power on board ships equipped with a controllable pitch propeller, running at constant speed.

The PTO unit is mounted on the tank top at the fore end of the engine, and the system has been developed as a short, compact design in order to minimize the installation space requirement.
<table>
<thead>
<tr>
<th>Alternative generator positioning</th>
<th>Design</th>
<th>Seating</th>
<th>Total efficiency</th>
<th>Applicable on MC-types</th>
</tr>
</thead>
<tbody>
<tr>
<td>1a</td>
<td>BW I/RCF</td>
<td>At engine (Vertical generator)</td>
<td>88-92</td>
<td>90-50</td>
</tr>
<tr>
<td>2a</td>
<td>BW II/RCF</td>
<td>On tanktop</td>
<td>88-92</td>
<td>90-26</td>
</tr>
<tr>
<td>3a</td>
<td>BW III/RCF</td>
<td>At engine</td>
<td>88-92</td>
<td>90-42</td>
</tr>
<tr>
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BW III/RCF (3b) and BW II/GCR (11) are our standard solutions, all others are available on request.

**Fig.6-20: Types of PTO**
The PLO generator is activated at sea taking over the electrical power production on board when the main engine speed has been stabilized at a level corresponding to the generator frequency required on board.

The BW II/GCR cannot, as standard, be mechanically disconnected from the main engine, but a hydraulically activated clutch, including hydraulic pump, control valve and control panel can be fitted as an option.

6.5.2 CP-propeller

1). programme

The current production programme of controllable pitch equipment includes the VB/VSA and VBS types, designed for the power range from 250 kw (≈ BHP) to more than 10,000 kw (13,600 BHP), see Fig 6-22, and for CP propellers of up to 6m diameter see Fig 6-23.
Fig. 6-22: CP-propeller, without ice class notion, for installation together with engine types L42, L35 and S26.

2). Optimizing the complete propulsion plant

In the case of two-stroke direct drive engines having a flexible layout diagram, the diameter chosen should be as large as the hull can accommodate, allowing the propeller revolutions to be selected so as to provide optimum efficiency. The optimum propeller revolutions corresponding to the diameter chosen can be found from Fig. 6-23 for a given reference condition (ship speed 12 knots and wake fraction 0.25).
Fig. 6.24 shows the CP propellers which can be applied on the L42MC, L35MC and S26MC type engines if they are laid out in points L1 or L3.

6.5.3 Remote control

Alphatronic IIA is an electronic control system with two modes of operation, viz. combined mode or separate mode.

1). Combined mode

Single lever control, which means that one handle controls both engine speed and propeller pitch, through a
load control system. This results in optimised operation, thus minimizing the fuel consumption, and protecting the engine against overload.

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<th>Engine type</th>
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<th>Propeller speed r/min</th>
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Fig.6-24 CP-propellers for L42, L35 and S26, open propeller without ice class
Fig. 6-25 Alphatronic IIA remote control system
2). Separate mode

In this mode of operation, the engine speed and the propeller pitch are controlled individually by two levers. The system is shown schematically in Fig.6-25. Electric signals from the handles are transmitted to electric actuators on the main engine and servo unit for the propeller. The actuators adjust the pitch and engine speed proportionally to the lever setting. The load control receives feedback from the speed sensor and fuel pump rack indicator. If these values increase or decrease, the load control automatically sets the engine load in accordance with the optimised load curve.

6.5.4 Auxiliary Units

In recent years the trend has been towards modulization of many shipboard installations, as it has been proved that the preparation of individual auxiliary machinery installations often has been particularly time-consuming for the shipyard. Therefore, on the basis of many years of experience in combining selected optimum components for auxiliary machinery, MAN B&W Diesel A/S have designed units for:

- Fuel oil treatment
- Camshaft lubrication, and
- Stuffing box drain oil filtration
- Others

6.5.4.1 Fuel oil supply unit

The fuel oil supply unit consists of:

- Two fuel oil supply pumps
- Two fuel oil circulating pumps
. Two steam preheaters, plate type (electrical heating for the S26MC only)
. One automatic full flow filter
. Alarm sensors and control box
. One viscosity controller

6.5.4.2 Piston rod unit

The piston rod unit consists of:
. One drain tank
. One circulating tank with steam heating coil
. One circulating pump
. One CJC fine filter
. Pertaining alarm sensors
(Except the S26MC for which it is not required)

6.5.4.3 Camshaft lubricating oil unit

The camshaft lubricating oil unit consists of:
. One magnetic filter
. One drain tank
. Two circulating pumps
. One oil cooler
. One duplex full flow filter
. One CJC fine filter with pump in by-pass
. Alarm sensors and control box
(except the S26MC for which it is not required).

6.5.4.4 Others

The engine room package can be extended with, e.g. the below-mentioned equipment, or parts of it, adapted to the yard's requirements for the individual project:
. Pumps, coolers and filters for lub. and fuel oil
Centrifuges for HFO and lub. oil
Pumps and coolers for cooling water
Starting air receiver and compressors
Exhaust gas silencer
MAN B&W double jib crane—for use in ships with restricted engine room height
CHAPTER 7

OPERATING BEHAVIORS

7.1 Vibration Aspects

The vibration characteristics of the two-stroke low speed diesel engines can for practical purposes be spilt up into four categories, and if the adequate countermeasures are considered from the early project stage, the influence of the excitation sources can be minimized or fully compensated.

In general, the marine diesel engine may influence the hull with the following:

EXTERNAL MOMENTS

These can be classified as unbalanced 1st and 2nd order external moments, which need to be considered only for certain cylinder numbers.

- Guide force moments
- Axial vibrations in the shaft system
- Torsional vibrations in the shaft system

7.1.1 External moments

The inertia forces originating from the unbalanced rotating and reciprocating masses of the engine create unbalanced external moments although the external forces are zero.

Of these moments, only the 1st order (producing one cycle per revolution) and the 2nd order (two cycles per revolution) need to be considered, and then only for engines with a low number of cylinders. The inertia forces on
engines with more than six cylinders tend, more of less, to neutralize themselves.

The natural frequency of the hull depends on the hull's rigidity and distribution of masses, whereas the vibration level at resonance depends mainly on the magnitude of the external moment and the engine's position in relation to the vibration nodes of the ship.

1) 1st order moment

1st order moments act in both the vertical and the horizontal direction. In two-stroke engine with standard balancing these are of the same magnitudes.

For engines with five cylinders of more, the 1st order moment is rarely of any significance to the ship. It can, however, be of a disturbing magnitude with four-cylinder engines.

As standard, four-cylinder engines from the 90 to 50 bores are fitted with adjustable counterweights, as illustrated in Fig 7.01.

The fitting of such counterweights can reduce the vertical moment to and insignificant value (although simultaneously increasing the horizontal moment), so this resonance is easily dealt with.

For the L42, L35 and S26 types these counterweights can be ordered as an option.

In rare cases, where the 1st order moment will cause resonance with both the vertical and the horizontal hull vibration mode in the normal speed range of the engine, a 1st order compensator can be introduced as an option in the chain tightener wheel, reducing the 1st order moment to a harmless value.

The standard engine is not prepared for the fitting of such a compensator.
2) 2nd order moment

The 2nd order moment acts only in the vertical direction. Precaution need only be considered for four, five and six cylinder engines.

Resonance with the 2nd order moment may occur for multi-node hull vibrations of more than three nodes.

Even the 2nd order moment compensators are provided, which are not included in the basic extent of delivery, experience has shown, that vessels of a size propelled by the S26MC, L35MC and L42MC engines are less sensitive to hull vibrations for which reason engine-mounted 2nd order compensators are not applied on these smaller types.

A decision regarding the vibrational aspects and the possible use of compensators must be taken at the contract stage. If no experience is available from sister ships, which would be the best basis for deciding whether compensators are necessary or not, it is advisable to make calculations to determine which of the evolutions mentioned should be employed.
3). Power related unbalance (PRU)

To evaluate if there is a risk that 1st and 2nd order external moments will excite disturbing hull vibrations, the concept Power Related Unbalance can be used as a guidance:

\[
PRU = \frac{\text{External Moment}}{\text{Engine Power}} \quad \text{Nm/kW}
\]

With the PRU-value, stating the external moment relative to the engine power, it is possible to give an estimate of the risk of hull vibrations for a specific engine, Fig.7-02.

According to 1st and 2nd order
External moments in layout point L, 

Fig.7-02: Power related unbalance, PRU Nm/kW
External forces and moments in layout point L₁

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External forces in kN for all types

|                | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 |

External moments in kNm

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<td>237</td>
<td>0</td>
<td></td>
<td>0</td>
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</tr>
</tbody>
</table>

Fig. 7-03: Table

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The table, Fig. 7.03, shows the external moment \( M_1 \) at
the speed \( N_1 \) and MCR rating in point L1 of the layout
diagram. For other speeds \( N_a \), the corresponding external
moments \( M_a \) are calculated by means of the formula:
\[
M_a = M_1 \times \frac{N_a^2}{N_1}
\]
(Tolerance on the calculated values is 2.5%).

7.1.2 Guide force moments

The so-called guide force moments are caused by the
transverse reaction forces acting on the crossheads due to
the connecting rod/crankshaft mechanism. These moments may
excite engine vibrations, moving the engine top toward ships
and causing a rocking (excited by H-moment) or twisting
(excited by X-moment) movement of the engine as illustrated
in Figs 7.04 and 7.05, respectively. The guide force moments
are harmless except when resonance vibrations occur in the
engine/double bottom system.

Fig.7-04 H-type and X-type guide force moment
The top bracing is recommended as standard, comprising stiff connections (links) with friction plates which allow adjustment to the loading conditions of the ship. As an option, a hydraulic top bracing the above-mentioned natural frequency will increase to a level where resonance will occur above the normal engine speed.

7.1.3 Axial vibrations

When the crank throw is loaded by the gas pressure through the connecting rod mechanism, the arms of the crank throw deflect in the axial direction of the crankshaft, exciting axial vibrations. Through the thrust bearing, the system is connected to the ship's hull. Generally, only zero-node axial vibrations are of interest, and only the effects of the vibrations regarding additional bending stresses in the crankshaft and possible vibrations of the ship's structure due to the reaction force in the thrust bearing are to be considered.

An axial damper is fitted as standard to all MC-engines, eliminating the effects of the axial vibrations.

7.1.4 Torsional vibrations

In general, only torsional vibrations with one node need to be considered. The main critical order causing the largest extra stresses in the shaft line is normally the vibration with order equal to the number of cylinders, i.e. five cylinders per revolution on a five cylinder engine.

Experience has shown that 8 and 12 cylinder engines for fixed pitch propeller plants and practically all plants coupled to CP-propellers may require a torsional vibration damper.

Four, five and six-cylinder engines require special
attention. On account of the heavy excitation, the natural frequency of the system with one-node vibration should, preferably, be situated away from the normal operating speed range, to avoid its effect. This can be achieved by changing the masses and/or the stiffness of the system so as to give a much higher, or much lower, natural frequency, called undercritical or overcritical running, respectively.

An undercritical system is normally characterized by:

. Relatively short shafting system
. Probably no tuning wheel
. Turning wheel with relatively low inertia
. Large diameters of shafting, enabling the use of shafting material with a moderate ultimate tensile strength, but requiring careful shaft alignment, (due to relatively high bending stiffness)
. Without barred speed range

The torque (propeller torsional amplitude) induces a significant varying propeller thrust which, under adverse conditions, might excite annoying longitudinal vibrations on engine/double bottom and/or deckhouse. The yard should be aware of this and ensure that the complete aft body structure of the ship, including the double bottom in the engine room, is designed to be able to cope with the described phenomena.

Undercritical layout is normally applied for four-cylinder engines.

An overcritical system is characterized by:

. Tuning wheel may be necessary on crankshaft fore end
. Turning wheel with relatively high inertia
. Shafts with relatively small diameters, requiring shafting material with a relatively high ultimate tensile strength
. Barred speed range of about ±10% around critical engine speed.
Overcritical layout is normally applied for engines with more than four cylinders.

7.2 ENVIRONMENTAL ASPECTS

The sensitivity of the human ear is closely related to frequency. The sensitivity is low at low frequencies, for which reason it is often necessary to take measurements at different frequency ranges.

Normally, these measurements are made in the so-called octave band frequencies which are named according to their average frequencies: 31.5, 63, 125, 250 etc. up to 16,000 Hz.

7.2.1 Exhaust gas noise

The constant pressure turbocharged two-stroke MC-engines are equipped with a large gas-receiver located between the cylinder's gas outlets and the turbocharger(s).

Due to its ideal location, i.e. close to the noise source, the gas-receiver also functions as a kind of exhaust gas silencer, particularly dampening the low frequency gas pulsations which are inherent to the exhaust gas from the cylinders.

Fig.7-6 shows an example of a 6L80MC engine, where the calculated octave band analysis of the exhaust gas noise valid for an exhaust gas system without boiler and without and external silencer has been drawn in.

The noise level is based on an actual distance of 15 meters from the top of the funnel to the bridge wing. The curve sheet shows that the noise level for curve 1 in the octave band frequencies between 125 and 1,000 Hz is decisive for the total noise level of noise rating NR 81, and the A-weighted sound level corresponds to 85 dB(A).
7.2.2 Airborne noise

Engine room noise is primarily generated by emissions from the individual engine components and their surfaces, which cause the air to pulsate.

The engine's average noise levels measured e.g. in accordance with 'CIMAC's Recommendations for Measurements of the Overall Noise for Reciprocating Engines', or other similar standards, are used to express the engine's typical
Airborne sound pressure level.

In other words, the calculated average sound level corresponds to the average value of sound intensity measured at different points around the engine. Measuring points are located at two or three height levels around the engine depending on the engine size and at a distance of approximately 1 meter from the engine's surface.

In general, but naturally depending on the type of engine, the average noise level for a nominal rated engine will be around 100 dB(A), whereas the maximum level measured around the engine will be approximately 105 dB(A).

Curve 2 in Fig. 7.07 shows the corresponding average noise level calculations for a 6S26MC engine. Due to the reflections of sound which occur in the engine room being a so-called 'near field', the noise levels measured in the vessel may be 1-3 dB higher than the calculated figures.

7.2.3 Structure-born noise excitation

Vibrational energy in the engine is propagated, via the engine structure, to the engine bedplate flanges i.e. the engine's 'feet'. From here, the energy is transferred to the ship's tank top, from where it propagates outward to the ship's structure which starts to vibrate and, thus, to emit noise.

An example of sources which can generate vibrational energy are the pulses caused by the engine's combustion process and the reciprocating movement of the pistons.

Similar to the sound pressure level, the level of velocity is best expressed in dB.

There are, as yet, no internationally determined reference values but, at rangey enough, 10 m/sec is often used.

Curve 3 in Fig. 7.07 show examples of the structure-borné
noise excitation levels from a 6S26MC engine, given as a vertical vibration velocity level on the engine feet.

Incidentally, the velocity level of a two-stroke engine is, on average, approximately 15-20 dB lower than of a medium speed four-Stroke engine which, therefore, may sometimes have special vibration isolators (resilient mountings) built-in between the engine feet and the tank top of the ship. The achieved structure-borne sound insulation is of some 15-20 dB, which means that the final result corresponds to the level of solid mounted two-stroke MC-engines.

Generally, the noise emitted by the two-stroke MC-engines' exhaust gas, and the structure-borne noise excited from the engine, are at such a low level that it is possible to keep within the noise requirements valid for the bridge wing and accommodation.

7.3 EXHAUST GAS EMISSION CONTROL

In various parts of the world the interest in air quality is growing, the emission from the marine and stationary diesel engines is being quantified, and rules are being prepared.

Key items are the emission of soot particles, SOx and Nix (oxides of sulphur and nitrogen).

The low speed diesel engine generally has a very clean combustion, meeting the soot and particle emission limits but, as a consequence of its high thermal efficiency, the emission of Nix is comparatively high.

SOx control will normally be effectuated by limiting the sulphur content of the fuel.

MAN B&W can offer diesel engines for the use of low sulphur natural gas injected at high pressures, for stationary engines and for LAG carriers.
Nix control will, dependent on the possible limits, require some additional equipment.

Although water emulsification of fuel oil will reduce Nix by up to 30%, a new equipment for control the emission of Nix, by means of a technique using Selective Catalytic Reduction (SC) by ammonia has been introduced as Fig 7.08 showing.

Such equipment makes it possible to comply with virtually potential legislative Nix emission limits.

On account of the still relatively few vessels in service with the SC equipment.

Fig. 7-8 Schematic layout of SCR system for a low speed diesel
CHAPTER 8

CONCLUSION

It can be assumed with considerable certainty that the diesel engine will continue to remain the first choice as a propulsion engine for merchant ships for the foreseeable future.

In the development of low-speed two-stroke marine diesel engines during the last decade, attention has been focused on low propeller speed and low SFOC. Virtually any type of ship can today demonstrate fuel consumption figures at least 25 per cent lower than just 15 years ago.

Minor reduction of SFOC for the engine may result from ongoing work in the fields of scavenging, cycle optimization, ceramics and combustion potential. Much more significant reductions are to be expected from "System Engineering" investigations; such as Turbo Compound System (TCS), which can give substantial energy savings, but involving more complex systems.

Future development will, besides concentrating on means to optimize total economy, also focus on achieving even better reliability, such as introduction of components with a larger margin to cope with deterioration, and systems to help early identification of maintenance needs.

Marine engine selection is a system engineering. Although the criteria in simplification, first cost and propeller speed etc. must be given high priorities, the relationship and other factors also have a great influence on total
The conclusion about the comparison between two-stroke slow speed and four-stroke medium speed engines can be list as following:

- Modern low-speed compared with medium-speed diesels is, even in terms of fuel economy and for similar cylinder power output, more economical.
- For the usual deep-sea ships with sufficient space for the engine room, the low-speed diesel provides the simplest possible propulsion arrangement without gears and with a minimum of cylinders.
- The biggest advantages of the low-speed diesel is its excellent historical record of proven reliability, simplicity, durability and safety at sea, especially on heavy fuel oils. These are the main reasons why, whenever technically and economically possible, a low-speed engine is, and will be, the first choice for a ship.
- The medium-speed engine, on the other hand, has clear advantages for applications such as flexibility, multi-engine propulsion plants, or auxiliary power generation, and where low head-room, multi-engine installations are required.
- Some criterion, such as reliability, durability and safety at sea have been improved greatly in medium speed engines, because the major parts have been modified and the overhaul intervals have been lengthened.

The MC-engine has evolved into a very reliable and very fuel-efficient engine, with a low spare parts consumption. There is now little scope with further increase in stroke/bore ratios, the ratio between fire pressure and mean effective pressure, and the turbocharger efficiency.
The smallest S26MC, with a higher $P_{\text{max}}$ at the same mean effective pressure, has made a really attractively low SFOC possible for such a small engine, as attractive alternatives to four-stroke diesel engines, enabling more operators to exploit the commercial and technical benefits of the two-stroke low speed diesel engine for a wider range of ships.

Just as important as direct cost is the impeccable performance of the prime mover. For MC-Engines, this is ensured by the proven very high degree of reliability and the extensive service back-up.

Engine costs—about half of which are fuel oil costs—represent some 40% of the ship's total costs. It should be noted that the increase in pay-back time for the higher price of a two-stroke engine is a mere two to three years.

Except the consideration of the economical aspect, the selection in same types of marine engines also is important. The selection depending on the engine programme provided by engine-builder must be set on the base of justification.

Environment aspects are increasing the importance in marine engine selection. If the ship-owners consider the using periods for a given ship, it is an important parameter.

The Slow Speed two-stroke engine is the most well established of all marine prime movers and the development in terms of output period has been tremendous. This trend is likely to continue so that the two-stroke engine will retain its market lead for some time to go.
REFERENCE

1. Marine Engineers Review

2. Two-Stroke Propulsion Package for Ship in the 1,000 to 15,000 DWT Range (MAN B&W A/S).


4. Textbooks of Ship Machinering Lectures.

5. Emission Control of Two-Stroke Low Speed Diesel Engines. (MAN B&W A/S).


10. Ships and their propulsion systems.
    by: Gallin/Hiersig/Heiderick

11. Vibration control in ships, A.S VERITEC


14. Ship Design and Construction
   by: Robert Taggart

15. Modern Ship Design
    by: Thomas C. Gillmer

16. IMAS 88-The Design and Development of Passenger Ships
    May, 1988

17. The Institute of Marine Engineers
    Transactions Volume 100, 1988-1991