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# Guidelines for the shipowner in choosing main and auxiliary engines

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# GUIDELINES FOR THE SHIPOWNER IN CHOOSING MAIN AND AUXILIARY ENGINES

BY

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

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# ABSTRACT

Selection of the machinery plant is a process of great importance, since the degree of success or failure of the commercial venture depends upon it; however, freedom to choose main and auxiliary engines is not as complete as some would imagine. Economical and technical consideration have to be recognised. Failure to do so early in the deliberations will only result in frustration and could even create problems which could otherwise be avoided.

This thesis is an attempt for studying and explaining some of the recent developments of marine diesel engines with some respect to the technical and economical aspects for the shipowner in choosing main and auxiliary engines.

## ABSTRACT

Selection of the machinery plant is a process of great importance, since the degree of success or failure of the commercial venture depends upon it; however, freedom to choose main and auxiliary engines is not as complete as some would imagine. Economical and technical consideration have to be recognised. Failure to do so early in the deliberations will only result in frustration and could even create problems which could otherwise be avoided.

This thesis is an attempt for studying and explaining some of the recent developments of marine diesel engines with some respect to the technical and economical aspects for the shipowner in choosing main and auxiliary engines.

## ACKNOWLEDGEMENTS

Praise be to Almighty God the most gracious and the most merciful.

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Special heartful thanks are due to my parents in Yemen for showering me with their endless love.

# CHAPTER ONE

## INTRODUCTION

There are a variety of engines that are expected to cover the complete range of foreseeable requirements for output and speed. Shipowners have a choice between these varieties according to ship type. There is huge competition amongst engine builders in the battle to gain the largest share in the market and this evolves around the need to restrict ship operation cost to a minimum.

Prior to 1973, attention had been given to higher specific power per cylinder, high power-to-weight ratios and more power per unit cost, and since then the preoccupation has been with fuel consumption.

The two most important measurements in the overall plant efficiency are fuel consumption and thermal efficiency. New designs will go a long way in achieving this, but equally as important is the need to modify existing installations to conserve fuel.

Following the 1973-1974 oil crises, comprehensive efforts were made by all diesel engines manufacturers to lower the specific fuel consumption of their engines. For example, the trend to lower rotational speeds for higher propulsive efficiency led to the longer stroke engines in 1979. Then also was the introduction of uniflow scavenging (in the RTA series) as it was felt that loop-scavenging could not achieve very low fuel consumptions at high stroke-bore ratios.

For the diesel engines remarkable improvements have been achieved in the last ten years to reduce the specific fuel consumption. Not less than a 20 percent reduction of this consumption has been reached and a total efficiency of over 50 percent has now been accomplished. Much work has gone into utilizing waste energy for heating and electrical generation. Also by using more economical methods of machinery operation, better use of waste heat, elimination of leaking glands and joints, overall efficiency can be improved.

Finally, the shipowners have to select an optimum ship and engine by taking into consideration the following main technical and economical aspects:

1. Choice of Speed/Flexibility
2. Vibrations
3. Reliability
4. Competition (between the engine manufacturers)
5. Initial Cost
6. Running Cost:
    - crews
    - maintenance
    - lube oil consumption
    - fuel oil consumption

Towards these intentions, an attempt will be made in this study to define "guidelines for the shipowner in choosing main and auxiliary engines".

This study has been prepared using a descriptive method obtained by reference to books, literature and interviews during my various field trips.

A substantial part of this study is a reflection of my professional and past experience as a seagoing Marine Engineer coupled with my field of study of Maritime Education and Training (Marine Engineering) at the World Maritime University, Malmö, Sweden.

The second chapter gives an insight into the development of the slow-speed and medium-speed diesel engines with particular reference to thermal efficiencies.

It is quite clear that the main purpose of turbo charging is to reduce the overall cost per KW of a diesel engine installation. The cost reduction is mainly attained by boosting the output of naturally - aspirated diesel engines of given size and speed by a factor of up to three or even four. This brings not only direct reductions in the capital cost of the engine, but also in the size and weight of the whole engine installation.

Also there is no decrease in the engine reliability or life expectancy, nor in a higher grade fuel required.

In the third chapter some considerations are given to the "development of turbo chargers".

There are two ways to improve the overall diesel plant efficiency:

1. Increasing the thermal efficiency of the diesel engine.
2. Improving the plant efficiency.

In the fourth chapter some consideration are given to the "fuel saving systems" and their considerable effect on improving the plant efficiency.

The stringent economic constraints facing today's ship operations demand maximum efficiency and reliability with minimum maintenance throughout the life of the vessel. A considerable saving can be made if there is an effective maintenance system.

Generally speaking, maintenance falls into two broad categories: necessity and efficiency. An attempt will be made to explain maintenance and condition monitoring techniques in chapter five.

The last chapter is under the main important title "Economical and Technical Aspects of Main Engine Choice". Comparison between two-stroke and four-stroke will be made to introduce the technical and economical performance of the two.

Finally this thesis will end with some conclusions and recommendations.

# CHAPTER TWO

# THE DEVELOPMENT OF MARINE DIESEL ENGINES
# TWO-STROKE AND FOUR-STROKE

## 2.1 THE HISTORY OF MARINE DIESEL ENGINES

The history of marine diesel engines is very young when compared with the history of shipbuilding. The diesel engine was developed in 1900, after Diesel formulated his original engine cycle concept as set out in his 1892 paper "Theory and Construction of a Rational Heat Engine", he envisaged a prime-mover of unprecedented efficiency. For this he saw the need for high cycle tempratures, achieved by very high firing pressures, a thermo-dynamic relationship which is equally valid today.

In 1904 the first diesel engine was built for the Russian tanker "Scormar", 810 DWT. The tanker was driven by two, four-stroke diesel engines each of 365 HP at 240 RPM. The brake specific fuel consumption reached was 240 g/HP hr., with an efficiency of 27%. The first MAN Diesel engine was built for the cargo ship "Secoundu" in 1910. The engine was four-cylinder, four-stroke with b.s.f. consumption of 220 g/HP hr., with an efficiency of 30%. The diesel engine was born while the steam turbine was advancing forward with wide steps. The steam reciprocating engine was descending slowly.

The diesel engine was able to enter the marine field at that time due to its higher efficiency, lower b.s.f. consumption, and smaller space occupied due to the absence of boilers. In 1912 the world's first ocean-going diesel motorship was launched by Burmeister and Wain, after that the marine diesel engine was greatly developed to be used in the German submarines. After the Second World War the experience and confidence gained with diesel engines, caused the demand of the marine diesel engine to increase rapidly.

The steam turbine began to make use of the difference in the fuel consumption by using cheaper heavy fuel oils which could not be used in marine diesel engine. But sooner it was possible to burn heavy oils of the same grade in low and medium speed diesel engines. After the Second World War, the ship builders began to think of building vessels of higher capacities to reduce the required freight rates. The increase capacity of vessel required an increased in the propulsive power. The steam turbine was ready to accept the new demands by increasing the pressure and temperature. But the diesel engine tried hard to coincide with the new demands of power. In 1960s, the maximum output of diesel engine was about 900 HP/ cylinder. Now it is possible to develop 5000 HP/Cylinder (Sulzer RTA 84M).

But sooner a total output per engine of 60,000 HP, has been reached e.g. (Sulzer 12 RTA 84M, for APL Company USA) with specific fuel oil consumption 123g/BHPh and thermal efficiency above 50%. But still the diesel manufacturers are pressing on toward more power to cover all the ranges.

More power can be produced by using medium speed diesel engines, by using different engine arrangements containing one to four or more, e.g. (MAN-B & W 9 x 9 L 58/64 QUEEN ELIZABETH 2 - 130000 HP).

The different output power for different machinery types are shown in Figure 2.1.

## 2.2 THE INCREASE OF DIESEL ENGINES DEMAND

The diesel engines demand was increased from year to year as shown in Figure 2.2 and Figure 2.3, where it is seen that diesel engines have already covered all the merchant ship demands, except a few ships which are still driven by steam turbines (most of steam turbine ships have been re-engined by diesel engines).

In Figure 2.3 it can be seen that the output per engine in case of slow-speed engines is nearly double that of the output power of high- and medium-speed engine.

It is clear that there is a great competition between slow- and medium-speed engines, where the medium-speed engines share is increased from year to year.

The relation between engine specific weight and engine power is shown in Figure 2.4 where it can be seen that the specific weight of engine in case of medium-speed engines is less than that of slow-speed engines.

The relation between specific fuel consumption and power output is shown in Figure 2.5 where it can be seen that the specific fuel consumption in case of marine diesel engines is less than that of other engines. At the same time there is a tendency towards the reduction of engine output by reducing vessel speeds for fuel saving reasons.

It is clear now that the increased demand of marine diesel engines is due to the following advantages:

1. The higher thermal efficiency (over 50%) and the lower specific fuel consumption, which reaches about 114 g/BHPh. The diesel engines are still more economical concerning fuel consumption, but on the other hand it˘s lube oil consumption is more than steam turbine, 0.4-1.5 g/BHph for diesel engines and 0.1-0.4 for steam turbine.

   As shown in Figure 2.5 the slow-speed diesel engines have lower consumption and the change of the consumption at different loads is minor.

2. The lower specific weight of marine diesel engine plant than that of the steam plant especially at lower ranges as shown in Figure 2.4. In general, the equal weight point depends on the type of machinery used. At higher power ranges, the specific weight of the steam turbine is less than that of slow-speed diesel engines. The reduction in steam turbine plant weights rather than diesel plants in case of higher power is compensated by the higher fuel consumption (190 g/BHPh).

3. The reduced capital cost due to reduced machinery cost and crew will compensate for the increased repair and oil costs.

4. The reduced operating cost due to use of heavy fuels, the use of heavy fuels together with the reduced fuel consumption and reduced capital cost lead to a reduction in the transport cost.

## 2.3 THE MAIN DIFFERENCE BETWEEN THE SLOW-SPEED AND MEDIUM-SPEED DIESEL ENGINES

The difference can be summarized as follows:

1. Construction:
   The medium-speed engines are of trunk piston, in-line or V-type, while slow-speed are mainly of the cross-head engines in-line.

2. Dimensions:
   The fixed dimensions of an engine are important in many cases. In general, a shorter engine means a less expensive engine room and more cargo space which gives a large freight rate to the shipowner. In special cases there may be restrictions on the size of engine that can be accommodated. This explains why more compact engines

are more competitive than large engines especially when a large power range is required. In Figure 2.6 it can be seen that the specific dimensions increase by decreasing the engine speed.

3. Weight:
   The specific weight of the slow-speed diesel engine is higher than that of the medium-speed engine as in Figure 2.4.

4. Fuel Oil Used:
   Nowadays, medium-speed diesel engines can burn the same heavy fuel as that used for slow-speed marine diesel engines. Concerning the brake specific fuel consumption, slow-speed engines consume less than of medium-speed engines, but the difference is small.

5. Lube Oil Used:
   Medium-speed engines consume more lube oil than that of the slow speed engines.

6. Power Transmission:
   The major difference between slow-speed and medium-speed engines is that the slow-speed engines are connected to the propeller shaft, where the speed is very low and is suitable for the propeller. In the case of medium-speed engines where the speed is higher than that required for the propeller, a reduction gear 3 to 4 reduction ratio is provided between shaft and engine.

7. Generally slow speed engine is more reliable than medium speed engines.

## 2.4 DEVELOPMENT OF MARINE DIESEL ENGINES

The aim on the development of diesel engines was to compete with steam turbine and other propulsion machineries in all aspects. At the same time it was necessary to increase the output power to cover all the ranges required for the world merchant fleet.

The development can be summarized as follows:

1. increasing the thermal efficiency of the marine diesel engines

2. increasing the plant efficiency

### 2.4.1 Increasing the Thermal Efficiency of the Cycle

The important area on the development of marine diesel engine is the pressure-volume diagram. The engine manufacturers have devoted effort, time and money for the intensive research and experiments which lead to the successful achievement of diesel engines in the marine field. The theoretical studies have also useful information to the designer. This useful information guides the path of marine diesel engine development and also expalins many facts.

Thus it is necessary to discuss the theoretical studies in diesel engines and the influence on thermal efficiency. The P-V diagram shape is affected by different parameters as:

1. the rate of heat release
2. excess air factor
3. compression ratios
4. inlet temperature and pressure

The effect of these parameters on the:

- thermal efficiency
- indicated mean effective pressure
- maximum pressure

will be explained. The relation between the maximum pressure and the indicated mean effective pressure will also be explained.

1. Effect of Inlet Air Temprature (Figure 2.7):
   The results show that the indicated mean effective pressure increases by reducing the inlet temprature. On the other hand the maximum pressure inside the cylinder and the thermal efficiency increases by reducing the inlet temperature but not as much as the increase on the indicated mean effective pressure.

2. Effect of Inlet Pressure (Figure 2.8):
   The inlet pressure has a great effect on the maximum pressure inside the cylinder as well as on the indicated mean effective pressure. The two are increased as the inlet pressure increases.

3. Effect of Compression Ratio (Figure 2.7)
   The theoretical calculations are carried out considering different compression ratios and inlet temperatures. The results show that the maximum pressure inside the cylinder and the thermal efficiency increases by increasing the compression ratio and reducing the inlet temprature.

4. Effect of Heat Flow Rate

   Generally, the heat added at constant volume leads to increases in the thermal charmal efficiency higher than that attained at constant pressure.

5. Effect of Excess Air Factor

   i. The indicated thermal efficiency increases by reducing the excess air factor.

   ii. The maximum pressure inside the cylinder increases by reducing the excess air factor.

   iii. The mean effective pressure increases by reducing the excess air factor.

6. Mean Effective Pressure (Pe) and the Maximum Cylinder Pressure (P max) (Figure 2.10):
   This shows the relation between mean effective pressure and the maximum cylinder pressure for different excess air factors, inlet pressure and compression ratio for a given heat release rate (V=45%).

Figure 2.10 shows the relation between maximum cylinder pressure (P max) and effective pressure (Pe) which indicates the different parameters affecting the higher P max/Pe ratio.

It can be seen that higher maximum cylinder pressure is attained at higher compression ratio and inlet pressure. It is also seen that by reducing the heat release at constant volume, reasonable values may be attained. The main factors affecting the mean effective pressure are excess air and inlet pressure. The compression ratio has minor effect.

The improvement in the overall thermal efficiency of the marine diesel engine can be summarized as follows:

i. Improving the cycle by using suitable injection systems with high injection pressures.

ii. Using large stroke/bore ratios (long stroke) to improve the scavenging efficiency.

iii. Using Turbo Compound System (TCS)

The thermal efficiency can be greatly improved and the brake specific fuel consumption is reduced to about 114g/BHPh.

### 2.4.2 Increasing the Plant Efficiency

The marine diesel power plant is improved by making use of the:

1. Exhaust gases energy losses.
2. Cooling water heat losses.

This part of energy is connected in producing useful work as follows:

1. Waste Heat Recovery:
   Generally, the exhaust gas leaves the turbo charger with considerable amount of heat. To make use of some of the exhaust heat, an exhaust gas boiler is installed at the chimney.

   The exhaust boiler is used to produce hot water used in domestic needs or as steam used in fuel heating. As the diesel engine power increases, the products from the exhaust gas boiler are also increased. Superheat was used to produce steam at reasonable pressure (7 bar and more) which is to be used to drive a steam turbogenerator.

   This installation of turbine - generator unit is not used for small diesel power plants due to economical factors, such as initial cost, cost of the KW produced and steam available.

2. Fresh Water Generators:

   To make use of the heat available as the steam produced from the exhaust boiler, water generators can be used. Sea water is evaporated at very low pressure and then allowed to condense to produce fresh water. The advantage of using a fresh water generator is not only to make use of the heat available in the exhaust boiler, but also to reduce the size of the fresh water tank which increases the cargo capacity.

3. Cooling Water Heat:

   Some amount of heat is absorbed from the cooling water for use in heating purposes, or in fresh water generator sets.

   The gain in the plant efficiency by using the available exhaust and cooling water heat is shown in Figure 2.12 which shows the heat balance for an RTA 84M engine with (TCS) turbo compound system, engine-driven power take off (PTO) generator and waste heat recovery. The engine is rated for 75% (MCR) maximum continuous rating power at 100% speed. The minimum exhaust stack temperature is assumed to be 160$^{o}$C while heat is also recovered from the scavenging air and jacket water down to coolent temperature of 90$^{o}$C and 75$^{o}$C respectively.

## 2.5 MARINE DIESEL ENGINES MANUFACTURERS DEVELOPMENT TRENDS

A diesel engine is purchased to perform a definite task. Fundamentally every effort is made to keep the total outlay needed to do this as low as possible. This is one of the basic principles of economic action which always applies.

The important development trends may be derived as follows:

1. Increasing the thermal efficiency of the engine by improving the engine cycle i.e. lower specific fuel consumption to reach lower running costs for the installation.

2. Improving the power plant efficiency by making use of the exhaust and cooling water energy.

3. Increasing the supercharging pressure while keeping maximum pressure within ranges.

4. Higher power density (kw/kg) of the engine system to reduce the first costs for the installation.

5. Reduction in the engine room size, which allows more space for the cargo and increases the profit.

6. Greater flexibility of the installations to operate with minimal costs.

7. Increasing the reliability and durability designs of the installations under all operating conditions to avoid lay by time.

8. The ability to burn even the lowest quality fuels to reduce fuel price and hence running costs. Even if there are some more repairs expected, it is still economical to use cheap heavy fuels.

9. Increasing the interval period between overhauls which increases the operating times and lowers the overhaul costs.

10. The possibility to use pulverised coal as fuel for marine diesel engines. Intensive research and experiments are under consideration for burning coal instead of fuel oil.

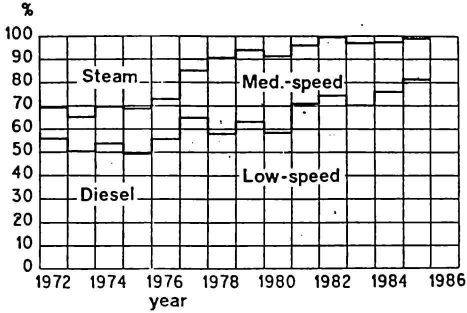
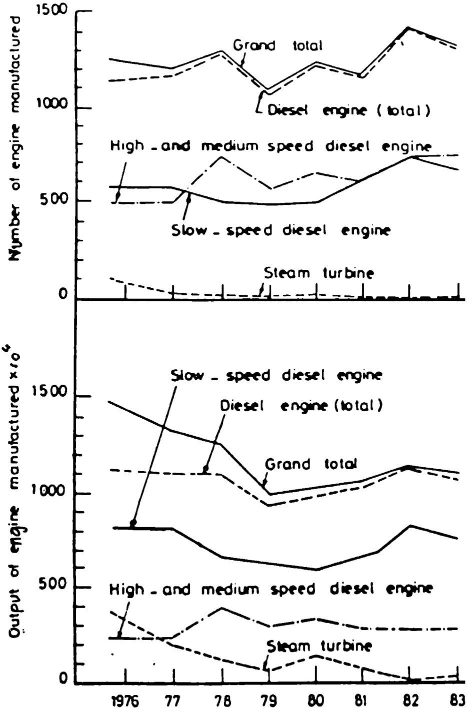
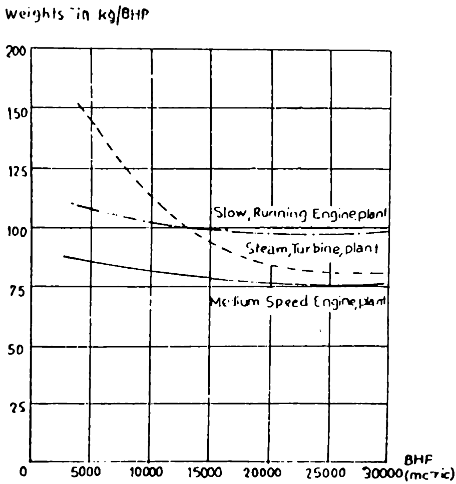

%
100
90
80
70
60
50
40
30
20
10
0
Steam
Med.-speed
Low-speed
Diesel
1972
1974
1976
1978
1980
1982
1984
1986
year

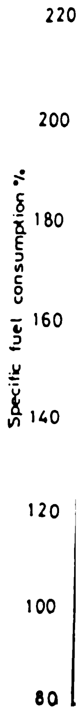
FIGURE 2.1




FIGURES 2.2 AND 2.3

FIGURE 2.4


Weights in kg/BHP
200
150
125
100
75
50
25
0
5000
10000
15000
20000
25000
30000
BHF (metric)
Slow, Running Engine plant
Steam, Turbine, plant
Medium Speed Engine plant



220
200
180
160
140
120
100
80
Specific fuel consumption %

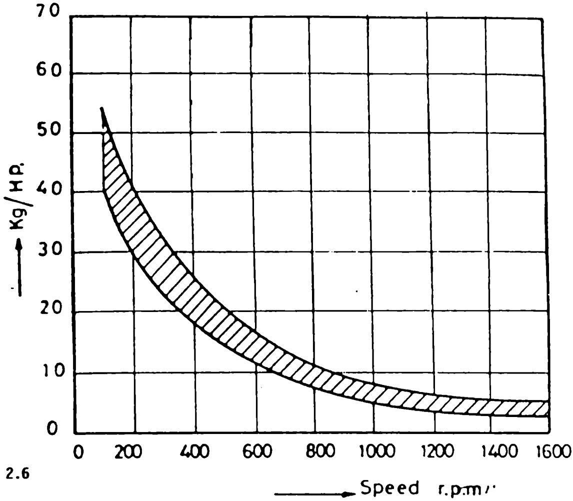



FIGURE 2.6

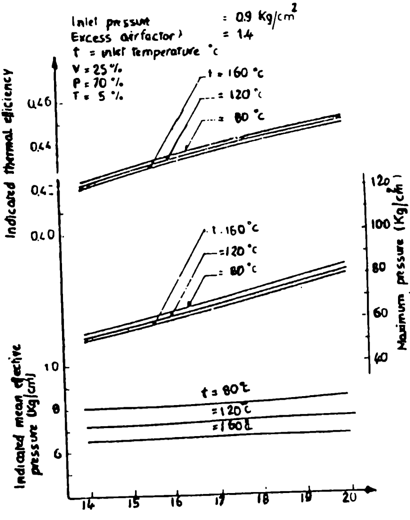



FIGURE 2.7

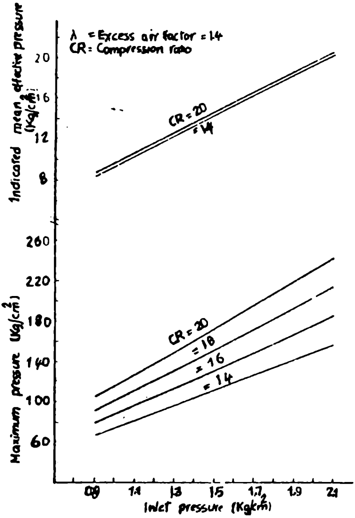

λ = Excess air factor = 1.4
CR = Compression ratio
Indicated mean effective pressure (Kg/cm²)
20
16
12
8
CR = 20
= 14
Maximum pressure (Kg/cm²)
260
220
180
140
100
60
CR = 20
= 18
= 16
= 14
0.9
1.1
1.3
1.5
1.7
1.9
2.1
Inlet pressure (Kg/cm²)

FIGURE 2.8

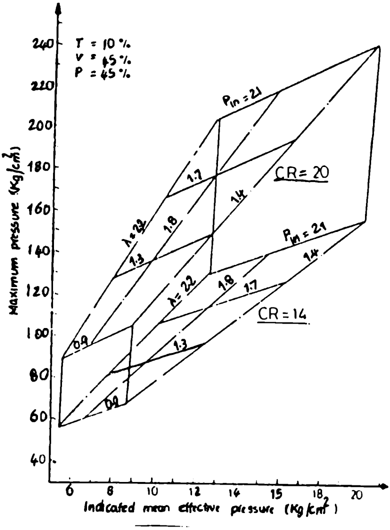



FIGURE 2.9

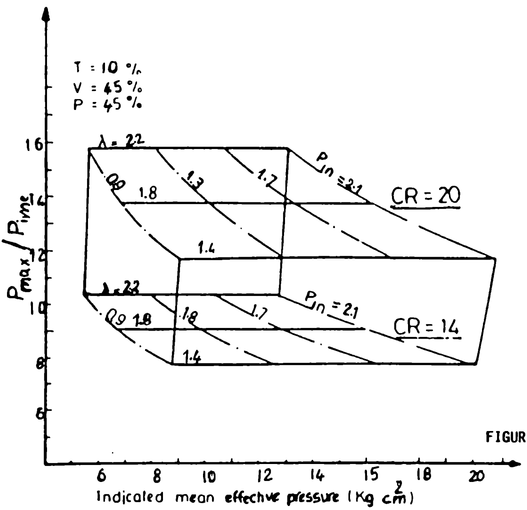
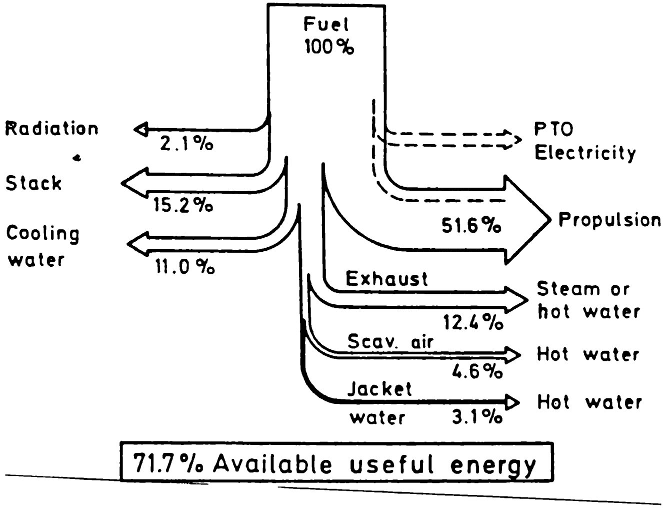



FIGURE 2.10



Fuel
100%
Radiation
2.1%
Stack
15.2%
Cooling
water
11.0%
PTO
Electricity
Propulsion
51.6%
Exhaust
Steam or
hot water
12.4%
Scav. air
Hot water
4.6%
Jacket
water
3.1%
Hot water
71.7% Available useful energy


FIGURE 2.11

# CHAPTER THREE

# THE DEVELOPMENT OF TURBO CHARGERS

## 3.1 IMPROVEMENTS IN SCAVENGING METHODS

Turbo charging has been the greatest single technical contribution to diesel engine progress and eventually did more for the high-powered marine diesel than double-action could ever have done. It removed the constraints of natural aspiration, enabling great increases in power output, and has considerably helped the diesel engine to displace the steam turbine and become the compact, economical prime-mover it is today.

All naturally-aspirated diesel engines required some supplementary means of ensuring an adequate scavenging air supply, boosting engine output by some 10-20%. With scavenging pumps becoming superfluous, the opportunity was taken in the Sulzer RD design, introduced in 1957, to combine the charge air receiver, turbo chargers and charge air coolers in one group at the side of the engine, hence improving the scavenging airflow through the ports.

A change was eventually made from cross-flow to loop flow and thereafter engine development was mainly influenced by improvements in the performance of turbo chargers, and was aimed towards ever higher unit powers and lower unit costs.

The change to constant-pressure instead of pulse-pressure turbo charging for the RND engine series in 1968 (Sulzer) allowed a much higher braek mean effective pressure with a lower specific fuel consumption as well as simplified engine design, eliminating the exhaust rotary valves incorporated in RD engine. Giving more freedom to the number of turbo chargers in 1983 was the introduction of uniflow scavenging (in RTA series) as it was felt that loop-scavenging could not achieve very low fuel consumptions at higher stroke to bore ratios. B.W. engines was the leader in using the uniflow scavenging system.

## 3.2 IMPROVEMENTS IN TURBOCHARGING METHODS

Various turbo charging methods have been developed since the work on exhaust gas turbo charging of diesel engines commenced. The turbocharging method chosen has an influence on the economy of the propulsion system. The main difference between the various exhaust gas turbo charging systems is the connection of exhaust gas flow from cylinders to the turbine.

### 3.2.1 Constant-pressure Turbo Charging

The constant pressure turbo charging system is now largely preferred for turbo charging low-speed two-stroke and medium-speed four-stroke engines, particularly in ship operation.

With constant-pressure turbo charging, any number of cylinders are connected to an exhaust manifold, as shown in (Figure 3.1). This simple and obvious system was used for the first exhaust gas turbo charging tests but, at the low-pressure ratios and efficiencies of that time, the turbo chargers then available did not permit adequate scavenging of the cylinder.

The higher pressure ratios reached during upgrading and improvement of the turbo changes have years ago permitted a return to constant-pressure turbo charging with large two-stroke engines. This return was successfully effected years ago. In the mean time manufacturers have adopted the constant-pressure turbo charging system for their new, large four-stroke engine too.

### 3.2.2 Pulse-Pressure Turbo Charging

With pulse-pressure turbo charging Figure 3.2 an attempt is made to transport as much of the energy contained in the exhaust gas as possible through short, narrow pipes from the cylinder to the turbine in the form of pressure pulses. A point of equal importance is that the pressure drops after the pressure pulse is utilized for scavenging. To prevent the pressure pulse of the following cylinder from interfering with the scaverging of the preceding cylinder, the firing interval must not be below a minimum value.

The pulse-pressure turbo charging system is the turbo charging system mostly used for four-stroke diesel engines. As already mentioned, there is a considerable difference in exhaust gas flow from the cylinders to the turbine between constant-pressure and pulse-pressure turbo charging. The turbo charging methods using pulse converters, multi-pulse and MPC (Figure 3.3) permit grouping of more cylinders to one turbine inlet than pure pulse-pressure turbo charging. They combine the characteristics of pulse-pressure and constant-pressure turbo charging and thus yield operating results which are somewhere between these two methods. Pulse converters, multi-pulse and MPC shall not be dealt with in detail here in.

## 3.3 IMPROVEMENTS IN TURBO CHARGER OVERALL EFFICIENCY

In recent years considerable progress has been made in improving the turbo charging system and raising the overall efficiency.

The criterion governing the thermodynamic efficiency of the exhaust gas turbo charger is its overall efficiency, that is the product of turbine and compressor efficiency, the mechanical loss being added to the turbine efficiency. Overall turbocharger efficiencies now attainable are plotted against the compressor pressure ratio for turbo chargers with radial-flow and axial-flow turbines. Figures 3.4 and 3.5 show the overall efficiency of NA-M.AN turbo charger.

Depending on their size, turbo chargers with radial-flow turbines now reach overall efficiencies in excess of 60%. The largest single stage turbo chargers with axial-flow turbines have reached values of about 70%. Figure 3.6 shows the performance graph of a modern compressor whose blades have backward leaning trailing edges. Distinguishing features of this compressor are a wide performance range, a compressor maximum efficiency of 85% and an adequate distance between the optimal parabala and the surge line.

The favourable distance between optimal efficiency and surge line is one of the factors that permits the engine operating point to be located in the range of high compressor efficiency without the hazard of the surge line being reached in part-load operation or under varying operating conditions. Another point of importance for economical engine operation is the curve of the turbo charger efficiency plotted against the pressure ratio the diesel engine output. Measures to be taken on the engine, e.g. keeping the firing pressure (PMX) constant over a certain load range, are positively supported by an increase in turbo charger efficiency as the load decreases, as may be noted in Figure 3.6.

### 3.3.1 Increasing Compressor Pressure Ratios

The pressure ratio of the compressor and the overall efficiency of the turbo charger will continue to increase. Higher power densities of the engines demand higher pressure ratios of the compressors. On the other hand the sharp rise in the price of oil forced the turbo charger manufacturers to increase the overall efficiency. Figure 3.7 shows three cases with different full load charging pressure. The figure shows the curve of the charging pressure which were necessary to enable the drive of a fixed pitch propeller to achieve constant air/fuel ratio throughout the whole load range, and thus to attain constant temprature.

Figure 3.8 gives an indication of what the efficiencies should be for a reasonable cylinder exhaust temperature in typical engines. It shows the ratio of compressor pressure ratio to turbine pressure ratio as a function of the former, with several levels of overall turbo charger efficiency.

Engines of the late 1970s require a compressor pressure ratio of about 3 to 1, with consequent compressor/turbine ratio in the range 1.1 to 1.2. The figure indicates that an overall efficiency in the range of 0.65 (65%) is required.

Engines of the late 1980s require a compressor ratio of about 4 to 1, with overall turbo charger efficiency of about 0.70 to 0.74. This explains why, for large installations with high engine output and definite service profiles, different turbo charging systems may be better suited than those used for small installations with low engine outputs.

### 3.3.2 Matching Turbo Chargers to Engines

With today's technology and with the emphasis laid on the reduction of fuel costs, it is very essential that owners and/or yards specify the engine layout with due regard to service power, maximum required power, engine and sea margins, and optimum propeller revolutions. Figure 3.9 presents typical compressor characteristics, these being a plot of pressure ratios against corrected mass flow rates, with rotational speed as a parameter. Contours of compressor efficiency are also included. Also shown is a typical air consumption curve for an engine at constant RPM.

The air consumption curve position is a result of careful matching of turbo charger to engine by the engine designer, this means that a compressor size is choosen to meet the air characteristics of the engine or of the group of cylinders served by the turbo charger.

From the above mentioned, there are two favourable points of importance for matching of the turbo chargers and economical engine operation:

1. The position for the engine air consumption curve referred to the Figure 3.9: it passes through the region of highest compressor efficiency. Obtaining this position in matching turbo charger to engine was doubtless the engine designer´s first objective.

2. A second aspect of favourable position is avoidance of the compressor surge limit as shown in the figure.

## 3.4 INFLUENCE OF TURBO CHARGING SYSTEM ON FUEL ECONOMY

The main purpose of turbo charging is to reduce the overall cost per KW of a diesel engine installation. This brings not only direct reductions in the capital cost of the engine, but also in the physical size and weight of the whole engine installation. There is no decrease in engine reliability or life expectancy, nor in a higher grade fuel required. Because of the efficient utilization of the exhaust gas energy, the higher mechanical efficiency and the reduction of the heat losses to the coolant improved fuel economy can be achieved with increasing post pressure.

Apart from the need for fuel injection and combustion to be carefully matched to supercharging conditions, this also demands a turbo charger and a turbo charging system with a high overall efficiency.

As already mentioned, the turbo charging technique has a considerable influence on fuel consumption. It is an accepted opinion among experts that constant-pressure turbo charging results in optimal fuel consumption rates for engines with higher specific ratings operating at full load. Consequently, the larger medium-speed engines operating at high mean effective pressures have been turbo charged to the constant-pressure method for several years. The operating results of these engines have clearly demonstrated the advantages of constant-pressure turbo charging in particular operation on board, too. Constant-pressure turbo charging supplies exhaust gas to the turbine continuously, thus enhancing its efficiency considerably. The strikingly simple design of the exhaust gas pipe system is the basis for small-scale and simple maintenance.

There are few engines with a charge air flow arrangement ensuring, on the one hand, a similarly high energy recovery and subsequently, on the other hand, a similarly uniform admission to the charge air coolers. This is important because the temperature of the compressed air downstream of the charge air cooler influences the fuel consumption considerably. A charge air temperature decrease of $10^{o}$ C leads to an average reduction in fuel consumption of 1 g/kwh.

Finally, for economical diesel engine operation the following is required of the exhaust gas turbo charger:

- high exhaust gas turbo charging efficiency
- simple maintenance
- long service life

## 3.5 OPERATIONAL RELIABILITY OF THE EXHAUST GAS TURBO CHARGER

The reliability measurement of any component is the long term satisfaction of that component without operational failure. Since exhaust gas turbo charging largely contributes to an increase of the overall efficiency of the diesel engine it has gained enormous importance since the energy crisis. The successful designing and operating of turbo charging methods and of the exhaust gas turbo chargers have reduced the specific fuel consumption rate of the diesel engines considerably. Improvement of the turbo charging system was an essential requirement for the success of other measures, such as the optimization of combustion with a view of reducing fuel consumption. This successful development in recent years has further enhanced the reputation of the diesel engine as the most economical engine.

In fact, it should be pointed out that, during the last 20 years, the charge air pressure generated by turbo-compressors has risen from 1.6 bar to about 4.0 bar and more. Furthermore most of the engines now in service on ships burn heavy fuel oil, which results in additional turbo charger stresses due to fouling and imbalance. The exhaust gas turbo charger is still a reliable element of a turbo charged diesel engine.

Table 3.1 shows the power output in the event of one constant-pressure turbo charger failing in fixed-pitch propeller operation.

In the case of failure of one of two turbo chargers, the engine will still develop 40% to 50% MCR. At about 5% MCR, the power output that can be developed in the event of all turbo chargers failing is the lowest with the two-stroke engines.

## 3.6 TURBO CHARGERS AND TURBO COMPOUND SYSTEMS

The two standardized turbo compound systems (TCS) are:

1. The Turbo Charger/Generator System, and
2. The Power Turbine System

From the energy extracted from the exhaust gas, as already mentioned, the high-efficiency turbo chargers are capable of delivering more compressed air than the engine needs. The exhaust gas energy which remains, after providing the engine with the required amount of air can be utilized by using either of the two standardized TCS with these systems. TCS can improve the overall economy in several different modes, the most interesting of which are:

### 3.6.1 Turbo Charger/Generator System (Figure 3.10)

This system uses the excess-power from the turbo charger's turbine to produce electricity in a direct coupling generator, built into the turbo charger's air inlet silencer. The electrical power produced is used to:

1. Drive an electric motor coupled to the engine shaft. The mechanical power delivered in this way can be used as an efficiency booster, thereby increasing the engine output without increasing fuel consumption.

2. Replace electricity produced by a main engine driven generator. This mode is equivalent to (1).

3. Replace electricity produced by auxiliary diesel generators, thus reducing the fuel oil consumption of auxiliary engines.

4. Supplement electricity produced by a steam-driven turbo charger (waste heat recovery system), thus, on larger plants without a main engine driven generator, eliminating the need for running auxiliary engines at sea.

   The system is only available in connection with some turbo chargers.

### 3.6.2 Power Turbine System

The power turbine unit consists of a small high-speed turbine, and a reduction gear which reduces the output shaft speed to 1800 RPM. Both MAN and ABB power turbine units can be supplied. To illustrate the general design, Figure 3.11 shows a cross section view of the ABB NTC 214 power turbine unit.

The power turbine is always used in parallel with the turbo charger turbine. Since the overall efficiency of the turbine is high, around 70%, the optimum switch-on point is between 50-60 percent of maximum engine power (MCR) as shown in Figure 3.12.

The mechanical power produced by the power turbine system is used to:

1. Supplement the engine shaft power (efficiency booster) thereby either increasing the engine output without increasing fuel consumption or replacing part of an engine driven generator reducing the fuel oil consumption at uncharged power. See Figures 3.13, 3.14 and 3.15.

2. Drive a generator coupled directly to the power turbine.

3. Combined with steam turbine (waste heat recovery system) for driving a turbo generator (Figure 3.16).

4. Combined with auxiliary engine (Turbo Diesel Generator Set) for driving generator. The diesel engine will work as a governor at low load, the exhaust gas pressure will be more than the charge air pressure to prevent the possibility of fouling and bad combustion. The "integrated charge air system" is used. (See Figures 3.17, 3.18 and 3.19)

It must be emphasized that the TCS used in combination with a main engine driven generator (PTO) or with an auxiliary diesel engine are extremely advantageous.

1. Because of the higher engine load with generator working.

2. Because of the increased scavenge air supply to the engine when the TCS is cut off, and

3. Because of the quick response of load sharing between the diesel engine and turbo-compound, the formerly working as a governor for the latter.

The pay back period of the power turbine can be recovered within about two years and maintenance only takes a few hours and can be carried out at the same time as routine turbo charger maintenance.

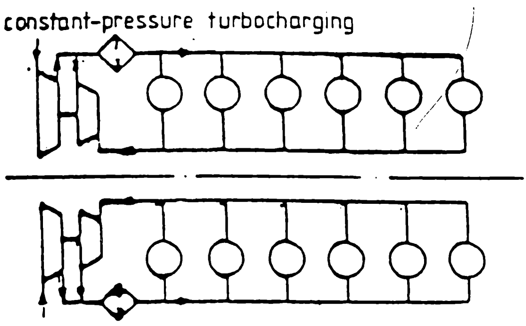



FIGURE 3.1

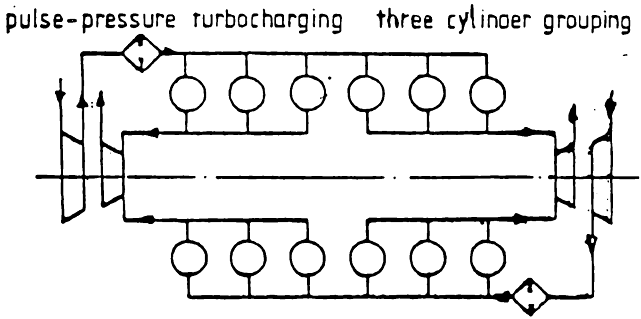



FIGURE 3.2

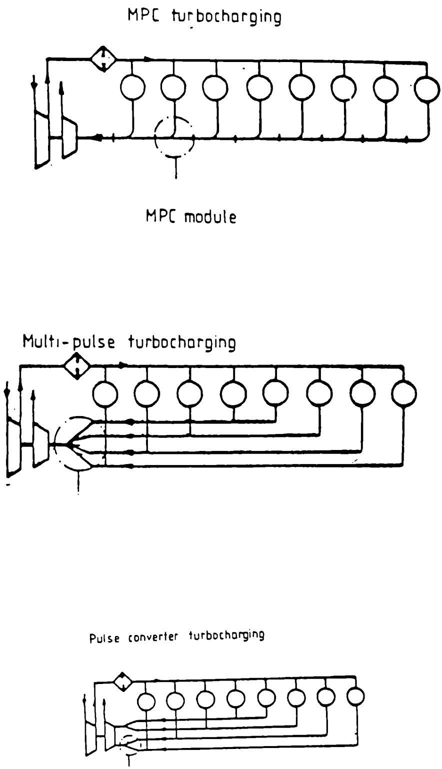

MPC turbocharging
MPC module
Multi-pulse turbocharging
Pulse converter turbocharging

FIGURE 3.3

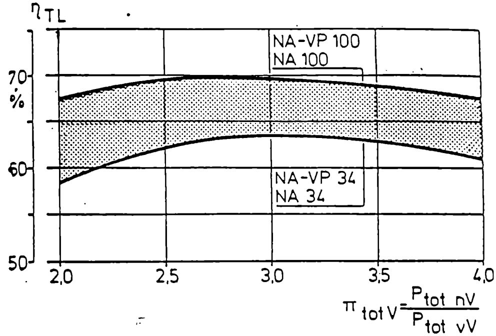



FIGURE 3.4

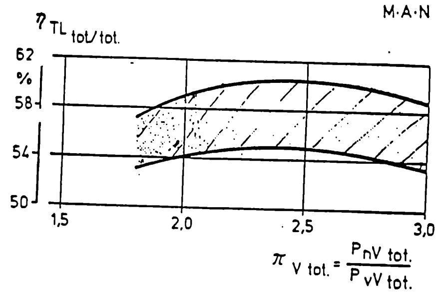
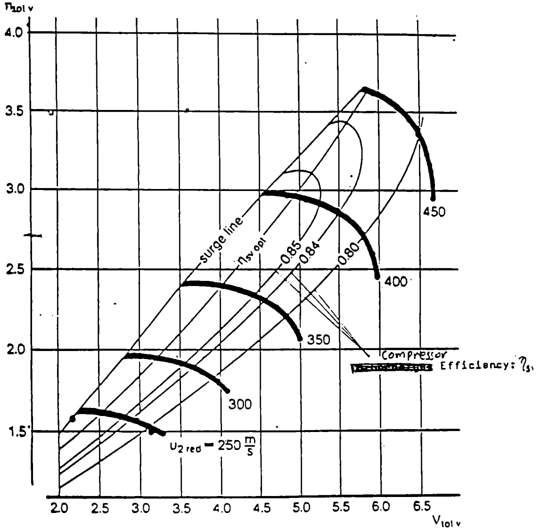



FIGURE 3.5




$U_{2red}$ : Speed of Compressor

$V_{tot\,v}$ : Air flow rate

$\pi_{tot\,v}$ : Compressor Pressure ratio

FIGURE 3.6

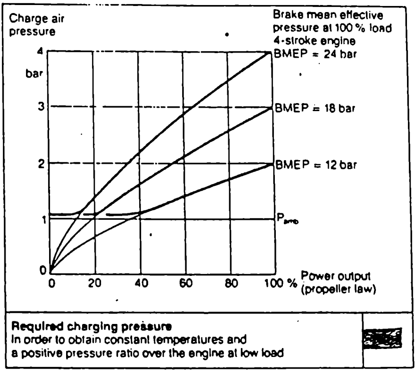



FIGURE 3.7

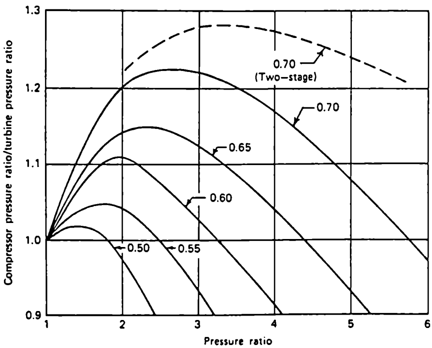



FIGURE 3.8

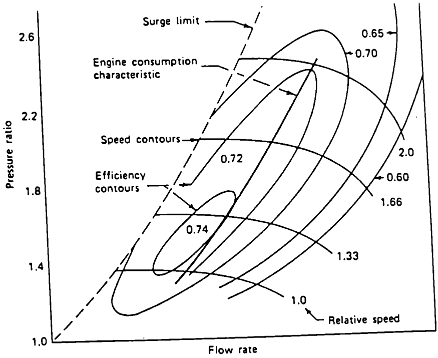



FIGURE 3.9

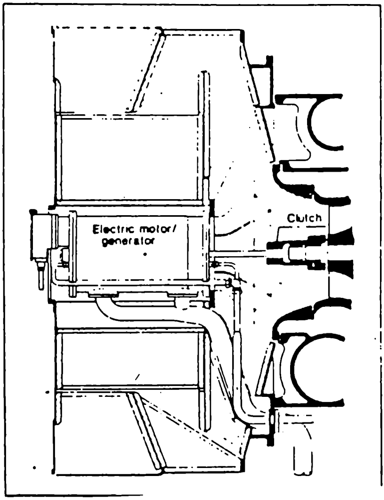



FIGURE 3.10

Power output in the event of one constant-pressure turbocharger failing in fixed-pitch propeller operation

| | Four-stroke engine | | | Two-stroke engine (with auxiliary blowers) | | |
|---|---|---|---|---|---|---|
| | m.e.p. | Propeller speed | P | m.e.p. | Propeller speed | P |
| MCR | 19.5 | 100 | 100 | 14.5 | 100 | 100 |
| Failure of one of two turbochargers, connection between air and gas sides | 11.0 | 73 | 40 | 6-9 | 73-79 | 40-50 |
| Failure of all turbochargers | 6.0 | 55 | 17 | 2 | 37 | 5 |

TABLE 3.1

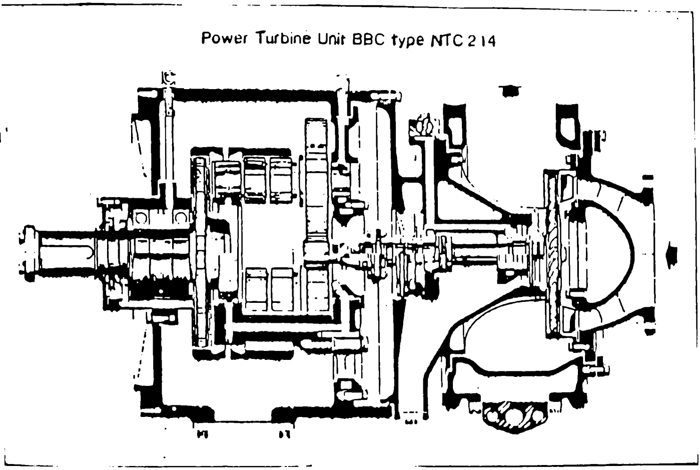
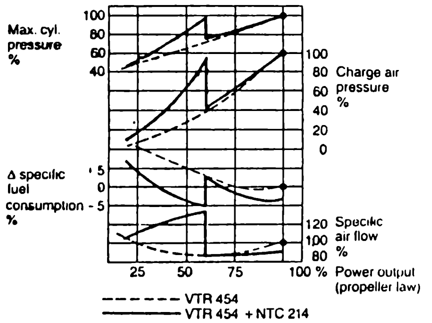



FIGURE 3.11




**Measured test results**
on a SEMT-Pielstick
4 PC 4.2

FIGURE 3.12

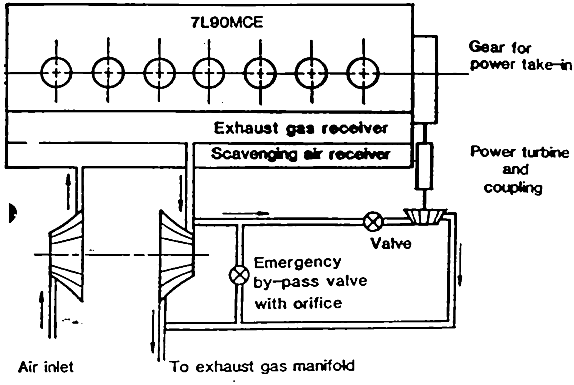
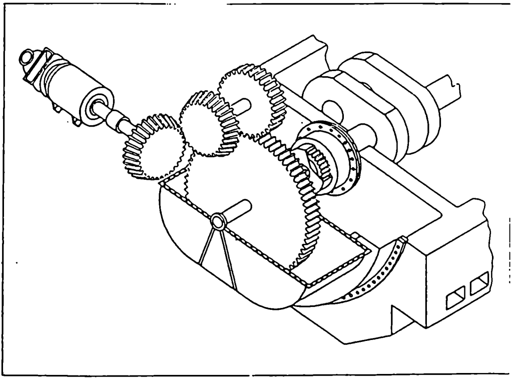


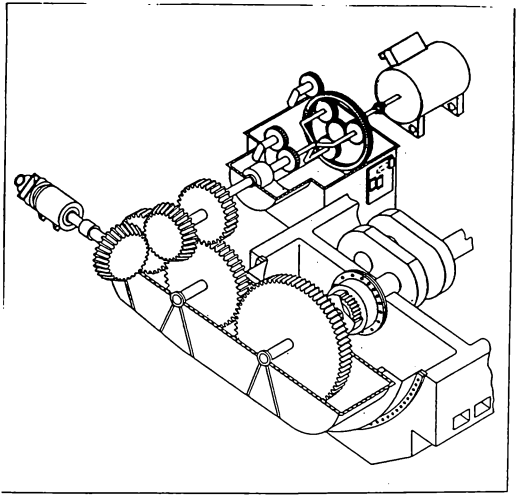

*diagram for the TCS system of "Power Take-In" L90MCE at Hyundai Engine & Machinery Co. Ltd.*

FIGURE 3.13



*Front mounted turbo compound system (TCS) drive*

FIGURE 3.14

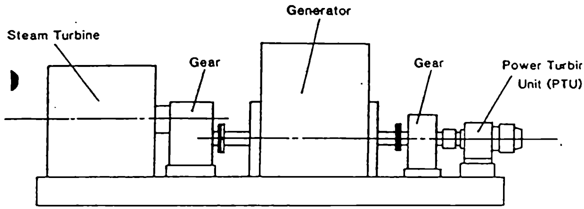


*Front mounted turbo compound system (TCS) drive combined with BW/RCF shaft generator*

FIGURE 3.15

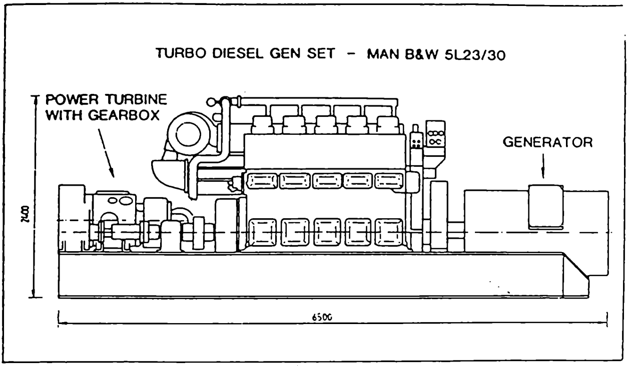
# TURBOGENERATOR–COMBINED WITH POWER TURBINE UNIT

## TURBO COMPOUND SYSTEM



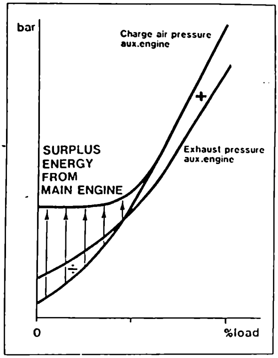

*Turbo generator with built-on main engine driven TCS*

FIGURE 3.16



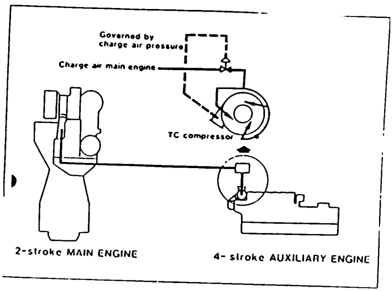

*Auxiliary engine with built-on main engine driven TCS*

FIGURE 3.17




*Integrated charge air system.*
*Differential pressure: charging air versus exhaust gas pressure*

FIGURE 3.18




*Integrated charge air system - Piping principle*

FIGURE 3.19

# CHAPTER FOUR

# FUEL SAVING SYSTEMS

## 4.1 POWER TAKE-OFF METHODS AND THEIR POSITIONING

A number of different alternatives are possible for shaft-driven generators, they can be:

1. integrated within the propeller shaft
2. arrangement power take-off (PTO)
3. integrated with a riding step-up or step-down gear

The arrangement power take-off location allows more flexibility in layout and can be combined with variable ratio speed drives and power take-in (TCS), usually a generator driven by a gear box fitted on the front of the engine. A space saving method is to place the generator adjacent to the engine e.g. integral power take-off PTO as Sluzer RTA and B & W MC engines. Figure 4.1 shows the arrangements for Sulzer RTA series.

The main advantages of this arrangement are:

- no influence on engine maintenance

- no influence on the installation of dampers at the front end

- no lengthening of engine plant, and thus shortest possible engine room

- suitable location from vibration point of view

- standardized equipment as the PTO is part of the engine builder's supply

- no separate foundation needed for the generator

The only disadvantage may be the limitation of the PTO´s power output range.

### 4.1.1 Comparison with Auxiliary Diesel Engine:

Assumptions:

- Maine Engine: 6RTA 58, rating R, 8460 kw/11520 BHP at 123 RPM
- Specific fuel consumption at 100% load = 175 g/kwh (129 g/BHPh)
- Heavy fuel price: 170 US$ per ton (1984)
- Total electric power consumption: 550 kw
- Marine diesel oil price: 270 US$ per ton (1984)
- Fixed-pitch propeller
- Specific fuel oil consumption of auxiliary engine: 210 g/kwh
- Generator and thyristor converter efficiency: 0.86
- Auxiliary diesel engine generator efficiency: 0.94
- PT-gear efficiency: 0.97 and the efficiency of the generator itself must also be considered in this case so the final efficiency shall be about 0,92.

For the PTO Generator:

The generator will absorb:

550 divided by 0.86 = 639.5 = 640 kw

The engine crankshaft supply approximately:

640 divided by 0.97 = 660 kw

The total mechanical power = 660 kw

The fuel consumption per day:

660 (kw) x 175 (g/kwh) x 24 (h)=2772 kg per day

This consumption with fuel oil have lower calorific value (LCV) of 42707 kj/kg. If we convert to fuel oil with (380 CST) and (LCV) of 40610 kj/kg,

The correct fuel consumption becomes

42707 kj/kg divided by 40610 kj/kg x 2772 kg
= 2915 kg/day

Cost of fuel consumption per day:

170 x 2915 kg divided by 1000 kg = 495 US$/day

The investment needed for: PTO-gear, coupling (with gear), clutch coupling, generator and support and complete thyristor convertor system. The price of this equipment, without erection and cabling may amount to approximately US$ 300,000.

<u>For the auxiliary diesel engine generator:</u> mechanical power needed: 550 divided by 0.94 = 590 kw
daily fuel consumption = 590 (kw) x 210 (g/kwh) x 24 (h) = 2974 kg/day

Cost of fuel per day: 2974 kg x 270$ divided by 1000 kg
= 803 US$ per day

The investment is as follows:

The price of an auxiliary diesel engine rated at approximately 700 kw with a synchronous generator and the usual engine driven pumps, without erection, cabling, may be about 180,000 US$.

<u>Comparison:</u>

The additional investment for the PTO generator amounts to 300,000 - 180,000 = 120,000 US$

On the other hand the PTO generator saved every day:

803 - 495 = 308 US$ / day.

Approximate recovery period for the additional investment:

120,000 divided by 300 = 400 days

Remarks:

The example implies that one auxiliary diesel can be saved by installing a PTO generator; this may indeed be possible. If the PTO generator is installed in addition to the auxiliary diesels the return on investment will obviously take longer.

The figures assumed may of course vary somewhat, without basically altering the fact proven by the example, namely that PTO generators represent an economic alternative to the generation of electric power onboard ship.

The example does not take into account that on a ship with a PTO generator the auxiliary diesels will accumulate less service hours per year and thus cost less in terms of service and spare parts.

The difference of fuel expenses is considerably reduced if the auxiliary diesel set burns heavy (or blended) fuel, maintenance will, however, increase.

Figure 4.2 shows the available power at PTO clutch for contract maximum continous rate (CMCR) for Sulzer RTA engine series 38, 48, 58, 68, 76 and 84. Also Figure 4.3 shows the available power at PTO clutch for part load of choosen CMCR.

## 4.2 ALTERNATIVE ARRANGEMENTS OF GENERATORS (Figure 4.4)

A generator, integral with the intermediate shaft with the poles mounted on the shaft and the stator on the ship's tank top and surrounding the intermediate shaft, is said to provide a simple robust arrangement. Where the shaft speed may be slow, the output frequency of the generator is restricted to about 15-20 Hz, thus a convertor may be required even if a controllable pitch propeller (CCP) is used. Furthermore it would not be possible to interpose flexible coupling, as shown on (Figure 4.4) Solution 3.

This arrangement can be qualified as follows:

- suitable location from vibration point of view
- normally no negative influence on engine room length
- power, in principle, unlimited
- not very suitable for very short shafts
- maintenance of the shaft system is more difficult

The arrangement of directly mounted on the fore end of the main engine, (Figure 4.4) Solution 4, where the generator is driven directly by the crankshaft which is extended and fitted with a flange for mounting the rotor. The disadvantages of this arrangement:

- unfavorable location from vibration point of view
- if dampers have to be mounted at the front end this solution becomes more complicated

## 4.3 CONSTANT FREQUENCY GENERATORS

The main engine driven generator, for ships with fixed-pitch propellers (FP), has a system to feed the main electrical power with three phase current at constant speed and frequency. To achieve a constant speed, the constant speed gear is used e.g. variable gear ratio as shown on Figure 4.5.

On the other hand the use of a thyristor bridge is widely used with shaft integrated generators.

## 4.4 IMPROVEMENTS IN PLANT EFFICIENCY

The marine diesel power plant is improved by making use of the following:

1. exhaust gas energy losses
2. cooling water heat losses
3. charging air heat

### 4.4.1 Steam Turbo Generator

Electrical energy could be gained from the exhaust gas of some engines. For slow-speed engines with exhaust gas temperatures of $300^{\circ}$ C the exhaust boiler could convert about 35% of the exhaust energy, the exhaust gas temperature cooling down to about $180^{\circ}$C in the boiler.

However, taking away the steam required for heating purposes and the losses during the process in the steam circuit of the turbogenerator, and taking into account the cycle thermo-dynamic steam efficiency (0.25), meant that the electric power actually produced amounted to only about 5% of the slow-speed diesel´s output.

The amount of heat energy available in the exhaust gas has decreased considerably over the past decade. Exhaust gas temperatures have fallen from $375^{\circ}$C to about $245^{\circ}$C for two-stroke main engines due to:

1. The improvements in engine cycle efficiency e.g.

from 40% to about 52% with efficiency boosters (TCS).

2. The improvement in supercharging efficiency.

On the other hand the heating surface of the heat recovery unit has been increased to maintain the same heat recovery. This has meant

- an expensive waste heat recovery installation,
- large installation space, and a
- large amount of equipment which needs more maintenance.

The amount of heat that can be recovered is limited by the "pinch point" temperature difference. If we assume a pinch point of $5^{o}C$, a steam saturation pressure of 8 bar and a gas inlet temperature of $245^{o}C$, the lowest gas outlet temperature is approximately $158^{o}C$ even with an economizer inlet temperature of $50^{o}C$ (feed water) as shown in Figure 4.6. By lowering the steam pressure/ medium temperature, or using a dual-pressure system, the outlet gas temperature can be decreased. But lowering the gas and heating surface temperatures will result in an increased risk of acid corrosion.

The quality of the fuel oil on the market has deteriorated giving an increased particulate content in the exhaust gas with more demands on the soot cleaning equipment, and a rise in the acid dew point due to the increased sulpher content as shown in Figure 4.7. The safest lowest gas outlet temperature is $160^{o}C$ at stack.

There are many types of exhaust waste heat recovery systems, some use a single pressure steam system and some use dual pressure steam system. It depends on requirements for steam consumption for heating services and turbogenerator electrical power production. Figures 4.8 and 4.9 show a single pressure and dual pressure system respectively.

Some of the single pressure steam systems utilize the scavenge air for heating the feed water. Figure 4.10 shows the heat recovery system layout with three stage charge air cooler for Sulzer RTA.

Recovery of the heat rejected in the exhaust gases and cooling water of the main engine charge air is extremely attractive. Although the investment costs for the heat recovery plant can be relatively high, they can be offset by fuel savings if maximum use is made of the steam output in a turbogenerator to meet all shipboard electrical requirements of the ship at sea.

If calculations for the actual plant show that the recoverable waste heat normally available for a turbogenerator does not provide an output sufficient to comply with required electrical power consumption at sea, the installation of a turbogenerator should be rejected unless special measures can be introduced, e.g. by connecting the other end of the turbogenerator to:

a. a DC motor (PTO) - See Figure 4.11.
b. a turbo compound system (TCS) - See Chapter Three.

A computer program is available to advise customers on the most economical method of electrical generation.

## 4.5 FACTORS INFLUENCING SELECTIONS OF GENERATORS

In merchant ships, the parallel method of operation is generally accepted. Normally, three generators are installed, one being sufficient to supply the ships normal sea-going load and two being operated in parallel when docking and undocking and when manoeuvring in confined waters as well as in cargo-handling in port.

The selection of the number and size of generators is influenced by a number of factors, which can be outlined as follows:-

- Economical, minimal and unbalancing of loading.
- Requirement to keep loss of capacity to a minimum.
- Requirement to install a reserve capacity.
- Flexibility of operation to cater for failure, maintenance and breakdown.
- High vulnerability of ships.
- Economical use of machinery space.
- Necessity to keep switchboard arrangement simple.
- Transfer of load via hand-operated or automatic change over switches.
- Running hours and refit intervals.
- Availability and reliability.

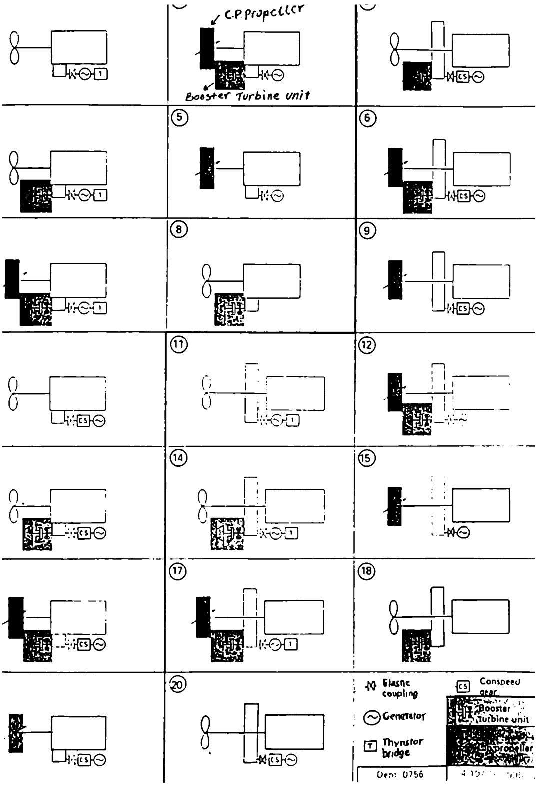
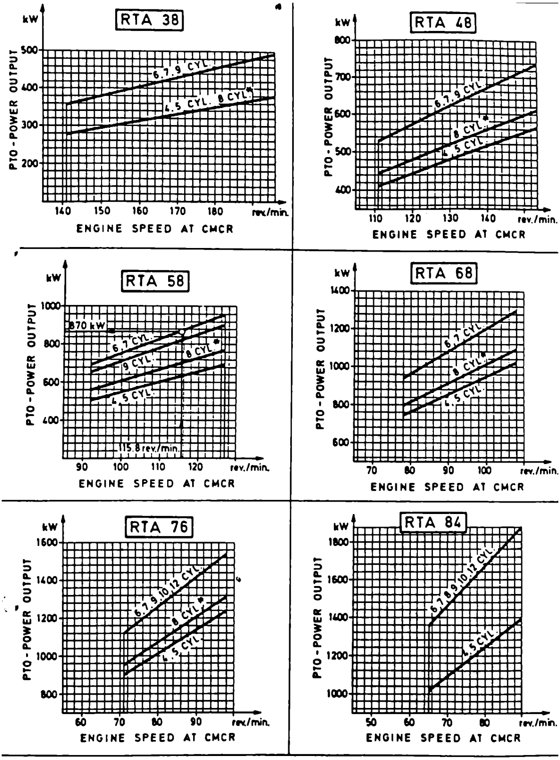

C.P. Propeller
Booster Turbine unit
5
6
8
9
11
12
14
15
17
18
20
Elastic coupling
Conspeed gear
Generator
Booster turbine unit
Thyristor bridge
Den: 0756

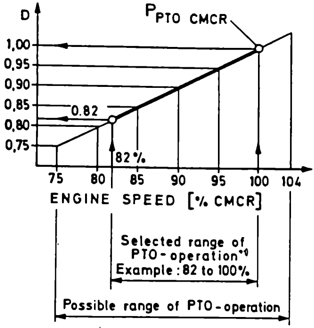
FIGURE 4.1




*Power figures are valid if a torsional damper on crankshaft-end is fitted



FIGURE 4.2




*1) Depending on the operation profile to be determined by the shipowner/yard.

Example:

Engine 6 RTA 58

- $P_{CMCR}$ = 87 % P (R1) = 8300 kW } Point Rx
- $n_{CMCR}$ = 92 % n (R1) = 115,8 rev./min. } Point Rx
- Selected range for PTO-Operation: from 100 % to 82 % of $n_{CMCR}$

To estimate:

1.) PTO-power output for 100 % CMCR: $P_{PTO\ CMCR}$
2.) PTO-power output for selected range of PTO operation: $P_{PTO}$ (82 %)
3.) PTO-power output for possible range of PTO operation: $P_{PTO}$ (75 %)

1.) PTO-power output for 100 % CMCR: $P_{PTO\ CMCR}$

Step 1: See page D 1–13 → Engine RTA58
Step 2: Number of cyl. 6 → Engine speed 115,8 rev./min.

$\boxed{P_{PTO\ CMCR} = 870\ kW}$

2.) PTO-power output for selected range of PTO operation: $P_{PTO}$ (82 %)

Step 1: See 1.) → $P_{PTO\ CMCR}$ = 870 kW
Step 2: See page D 1–14 → Engine speed = 82 % CMCR
Step 3: D = 0.82

$\boxed{P_{PTO}\ (82\ \%) = P_{PTO\ CMCR} \cdot D = 713\ kW}$

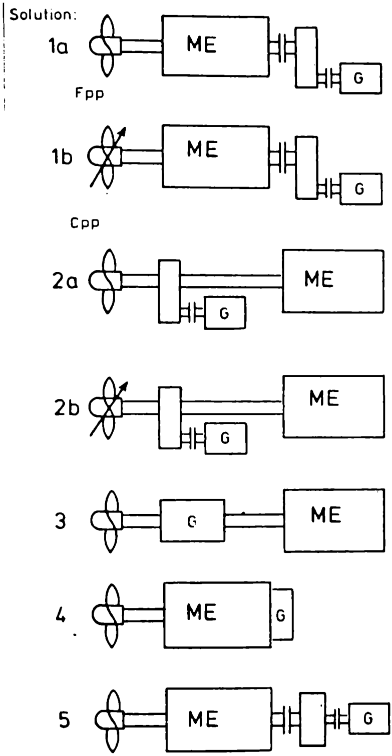
3.) PTO-power output for possible range of PTO operation: $P_{PTO}$ (75 %)

Step 1: See 1.) → $P_{PTO\ CMCR}$ = 870 kW
Step 2: See page D 1–14 → Engine speed = 75 % CMCR
Step 3: D = 0.75

$\boxed{P_{PTO}\ (75\ \%) = P_{PTO\ CMCR} \cdot D = 653\ kW}$



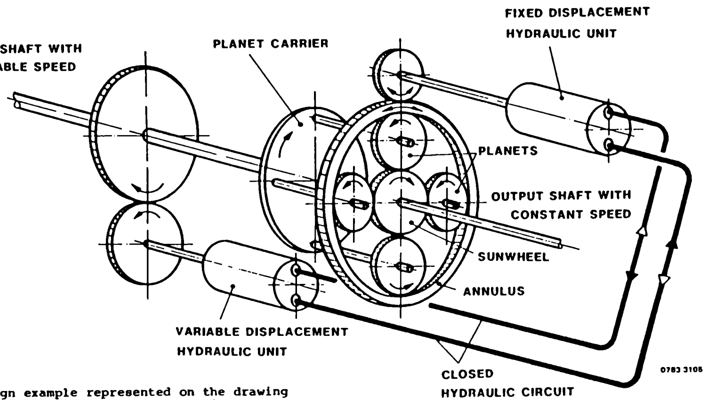
FIGURE 4.3




Alternative positionings for main engine driven

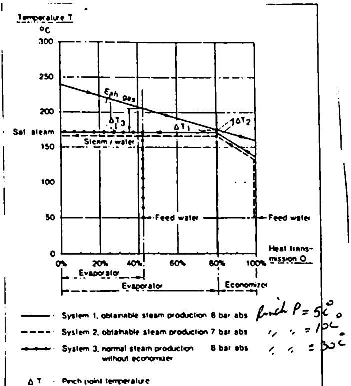
FIGURE 4.4

FIGURE 4.5



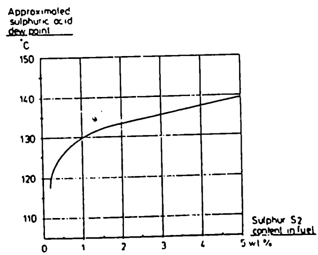

Design example represented on the drawing
The annulus of the variable-ratio planetary gear
is driven by variation of the direction of rotation

FIGURE 4.6



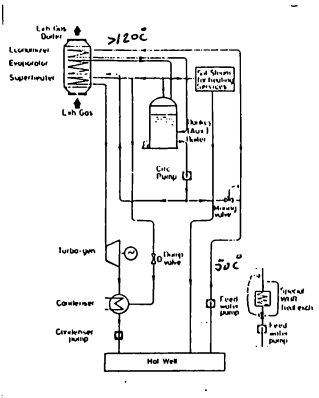

The temperature development through an exhaust gas boiler, valid for traditional single pressure steam systems, is, by way of example, shown for a nominal rated L-MCE engine, running at 80% MCR, at ISO reference conditions.

FIGURE 4.7



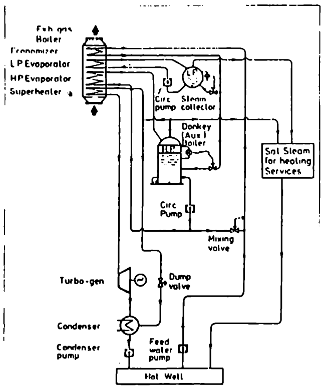

The curve shows the approximated sulphuric acid dew point of the exhaust gas as a function of the suphur content (weight %) in fuel oil combusted by an M.A.N.-B&W MC-engine.

FIGURE 4.8




Single pressure steam system with exhaust gas economizer and mixing valve

FIGURE 4.9




Dual pressure steam system with a single pressure turbo-generator.

| TECHNICAL DATA **SULZER** RTA MARINE DIESEL ENGINE | GENERATION OF AUXILIARY ENERGY<br>Heat Recovery System Layout<br>Three Stage Charge Air Cooler | B1-3<br>Issue 1987 |
|---|---|---|

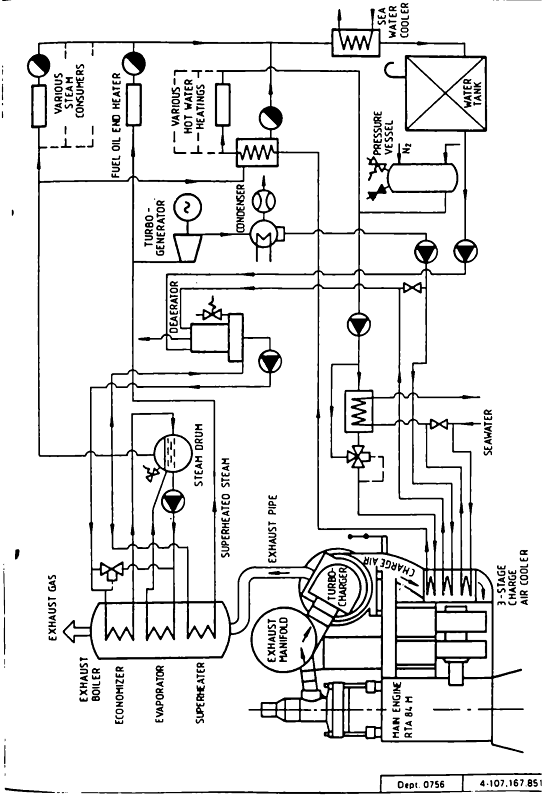
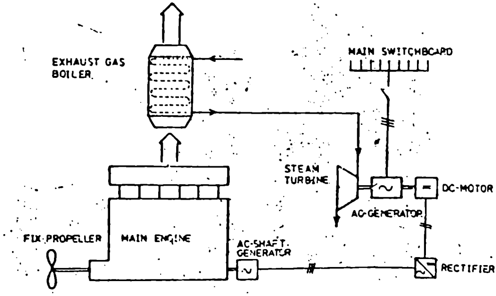



FIGURE 4.10




Turbo-generator, with engine-driven back-up generator.

FIGURE 4.11

# CHAPTER FIVE

# FUEL ECONOMY AND CONDITION MONITORING

## 5.1 THE FUEL ECONOMY FUNDAMENTALS

While fuel oil prices have declined over recent years there has been a considerable decline in the quality of fuel oil available to the shipowner and it is almost certain that the price of oil will rise again without a corresponding rise in quality, unless a dramatic change occur at the refineries and the so-called secondary refining process. Since the demand for light oil products is now so great, petroleum companies use sophisticated refining methods to extract an ever growing proportion of light fractions. As a result the quality of bunker oil, the residual product of the refining process, is steadily declining leading to high maintenance costs for diesel engines.

Irrespective of engine make, the consequences have included turbocharger surging, heavy fouling of ports and pistons and abnormal wear of fuel injection equipment, cylinder liners and piston rings. Extreme cases have led to broken rings, cracked piston skirts and cracked cylinder liners within remarkably brief running periods.

It is not the intention of this chapter to outline in great detail information that already exists, but merely to state the case for change in the description of the fuels to be bought for burning in ships. As fuel sources change and refining patterns alter to meet commercial needs the future trend in the diesel engine field is towards fuel of higher viscosity, lower cetane number, higher specific gravity and low viscosity. Hence the following problems can be significant:

- burnability in diesel engines

- stability
- compatibility
- densities and in excess of that of water
- destructive contaminants
- refinery catalytic fines

While the condition of the fuel oil as delivered to the ship is of importance to the owner, much more emphasis is placed on the condition of the fuel oil as delivered to the engine after shipboard treatment and cleaning. Obviously there have been cases of completely unacceptable bunkers, with properties outside the recommended limits for the particular engines concerned, together with fuel stability problems. But it is too simplistic to attribute all these problems merely to deteriorating bunker supplies. In too many cases, the troubles have stemmed from inadequate or incorrect fuel treatment. In reality there has been a number of factors involved, such as:

- inadequate fuel treatment equipment being installed in the ship
- wrong layout of the fuel handling and treatment system
- maloperation of the treatment plant
- lack of understanding about the vital importance of good fuel management by the ship staff
- in some cases perhaps, poor direction from the owner's shore staff.

The overall concept is simply prudent engineering practice and is already well established in many shipowning companies but in too many instances, it is overlooked and downgraded in importance. To this end the provision of an efficient fuel system is merely the first step towards the successful utilization of marine fuel oil, the second being the proper handling of these facilities.

### 5.1.1 The Mixing or Blending of Fuels

The mixing or blending of fuels in unfortunate proportion can lead to unacceptable sludge leading to gross overloading of filters and centrifuges. Although it is possible to state a policy of not mixing two different types of fuel in one bunker tank, in practice this may have to be done.

To mix these fuels to produce a homogenous bunker may be tempting but to do so may involve the risk of producing unacceptable blends. It is an operation that demands knowledge of the fuels being blended if incompatibility is to be avoided.

## 5.2 BUNKER TESTING AND MONITORING

A recommended bunker delivery note developed by the International Chamber of Shipping for ship operators and fuel suppliers is shown in Figure 5.1. Bunker fuels may not conform to the attached delivery note in numerous ways. Although the most obvious example is an incorrect extractive process at the refinery, instability during storage, microbial growth, corrosion in the storage tanks, sludge formation, oxidation, gum accumulation, degradation, and contamination can take their toll during storage and transport to shipside. It should be realized that bunkers are often blended aboard the barge during the loading operation, so that a testing programme will protect against problems with the blending system.

The essential elements of a successful fuel testing programme include correct sampling techniques, adherence to standard procedures, repeatability, a reasonable price, and timely feedback as well as proper documentation of the result.

The fuel sample must be truly representative of the entire bunker load for the testing program to be worth-while. A continuous sampling device should be attached to the bunker manifold during bunkering and a portion of this should be retained along with the analysis results, in case of controversies in the future.

Unfortunately, shipowners usually purchase an unpredictable fuel oil which conforms to only one main specification - an upper limiting viscosity. The monitoring of fuel quality other than viscosity, however, needs to be done on a historical and day-to-day basis. Monitoring quality historically is achieved by the use of shore-based independent laboratories. This monitoring ashore can be aided by regular use of fuel testing services either on worldwide or centralized basis, with full test analyses being returned and collated. Such services are offered by <u>Det Norske Veritas</u> and other private companies such as <u>Drew Ameriod Marine.</u> Daily monitoring, however, provides an immediate on-the-spot assistance to those on board ship. At present there are very few simple, compact and reliable on-board test kits available. These tests would enable sea-staff to ascertain the value of the controllable parameters of the fuel and thereby maximizing, separating and handling efficiency.

## 5.3 <u>PROPER FUEL TREATMENT TO ACHIEVE PROLONGED MAINTENANCE PERIODS</u>

Since the removal of water, sediment and sludge becomes more difficult as fuel viscosities and densities increase, security against contamination should assume greater importance. Poor siting and security of vent pipes, ullage pipes and caps, and filling connections are common causes of contamination from breaking seas or even washing down hoses. The potential rust and scale problems that can result from salt water contamination are obvious. Freshwater contamination is less likely, but can come from condensation, especially in slack tanks where the air volume and moisture it contains will be greater.

### 5.3.1 Storage Tanks

It is essential that outflow heaters should be operated by adhering strictly to the recommended temperatures. Storage tank heating should be capable of raising the fuel temperatures to $45^{\circ}C$, which should be sufficient to make it pumpable. If the fuel has been allowed to cool, however, extended heating time may be needed to dissolve all wax to avoid the blockage of filters.

### 5.3.2 Settling Tanks

As with so many aspects of ship construction, settling tanks are often a compromise with space and other limitations. This results in many tank configuration that are not conducive to good settling and draining. Suctions and drains are sometimes poorly sited. If they are too far apart, suctions may be lower than drains when vessel are listed or rolling.

Low suctions should always be high enough to clear adequately any normal sludge or water accumulations, with high suctions as an emergency safeguard.

Settling times and temperatures are more critical for higher fuel densities and viscosities and since this is a function of oil depth as well, tank depths should be as small as practicable.

The service tank temperature must be kept acceptable taking into account all the procedures for the settling tank.

### 5.3.3 Fuel Oil Centrifuging

The shipowner must equip his ships with the necessary tools to cope with the escalating problems of degraded fuel quality. Before any fuel can be passed through the fuel injector equipment of any diesel engine either large bore or medium speed, they must be cleaned by centrifuging or filteration or a combination of both. To accomplish this, heating the fuel to its acceptable temperature is necessary in order that its viscosity can be reduced to a value at which it can be centrifuged and pumped easily. The capital cost of all this equipment or its depreciation must be set against a saving in the cost of the fuel and prolonged maintenance period for the engine component.

Efficient purifier operations demands careful setting of the machine. These machines should preferably be capable of handling fuels with densities up to 1.010 $g/cm^3$ and their throughput capacity to be in accordance with manufacturers latest recommendations. It is advised that each centrifuge should have a capacity of more than five times the engine's fuel rate to allow removal of catalytic fines and other abrasive particles by using the slowest possible throughput. Normal amount of combustible sludge discharged from a total discharged purifier are around 2-3% of the total fuel throughput. For example: at rate of 60 tonnes a day, fuel price 100$/ton, this loss represents $120-$180 every day the separator plant is in operation.

## 5.4 CONDITION MONITORING

It should be remembered that machinery should not be stripped down just for the sake of inspecting them but this should be done at the right time to avoid extensive damage and cost. Thus a balance should be achieved between inspection cost, consumption of parts cost and machinery downtime cost.

The condition of ascertaining the condition of machinery in its operating mode, rather than having to dismantle its components for visual examination, which is a costly exercise in operational and labour terms, would enable engineers to conduct maintenance only when it is necessary, that is, when the condition of the machine deteriorates to a level where its performance drops off or it becomes unreliable in operation.

All machines wear naturally with time due to friction, looseness, imbalance, misalignment and so on. If the level of deterioration could be monitored at regular intervals and a trend established, the engineer could reliably predict when the machine required corrective maintenance and plan the schedule in advance. This technique is know logically as predictive maintenance through condition monitoring (or condition based maintenance).

Condition monitoring implies a systematic monitoring of characteristic parameters in order to assess the condition of a component or item of machinery and compare it with a reference or with accepted limits beyond which a failure or breakdown is more likely to occur, the objective being to minimize maintenance costs. Monitoring of machinery performance is vital to fuel economy as quite small deviations in key performance indicators (instrumentation) can lead to substantial fuel and therefore, financial waste. Hence key monitoring instruments have to be checked and re-calibrated at regular intervals and also replaced were necessary.

Various condition monitoring systems have been developed for accurate assessment of engine components condition and maintenance prediction. Combustion parameters, injection and fuel pump operation data can be used to optimize performance, while ignition delay, prevalent in burning today's low quality fuels, can be identified quickly and absolutely by the correct interpretation of an oscilloscope picture.

### 5.4.1 Objectives of a Contemporary Condition Monitoring System

- To carry out a critical examination of the combustion process and to tune the engine so that an acceptable balance is obtained.

- To carry out a critical examination of the engine's fuel system - making required adjustements to ensure correct operation.

- To provide long-term monitoring of engine performance by plotting performance data on "trend graphs".

- To record engine conditions at pre-set running-hours intervals for reference purposes.

- To establish that the engine is operating within design limits and that bearing loads are safe.

- To determine specific fuel consumption and thus monitor engine performance against a known standard.

- The overriding objective that the system should be simple - little operational paperwork involved.

### 5.4.2 The MIP Calculators

Mean Indicated Pressure (MIP) calculators:

- Norcontrol - DETS (Diesel Engine Tuning System) Norway

- Autronica - MIP - NK - 5 Norway

are operated by signals from transducers fitted to the appropriate ports of the engine as shown in 5.2.

The signals are proportional to:

- scavenge pressure
- crank angle
- cylinder pressure
- fuel oil pressure before the injector

The MIP is derived by integration over the compression and expansion strokes. The following data is taken by the MIP calculators:

- a cylinder pressure / time graph (Figure 5.3)
- a fuel pressure / time graph (Figure 5.4)
- measurement of the timing of fuel injection
- measurement of engine speed and top dead positions

#### 5.4.2.1 Combustion Efficiency

The cylinder pressure/time graph is perhaps the most useful in that it gives an accurate picture of what actually is happening inside the engine cylinder with respect to crank angle position, and it is very easy to see the effects and efficiency of the combustion process. From this graph (Figure 5.3) six very important parameters can be measured.

- MIP, Mean Indicated Pressure
- P max, Maximum Combustion Pressure
- P comp, Compression Pressure
- P exp, Pressure on the Expansion Curve, $36^{o}$ after T.D.C.
- $\propto$ P max, the angle where P max occurs, referred to T.D.C.
- Load, Cylinder Load in KW

5.4.2.2 Fuel Injection Efficiency

The fuel pressure/time graph is possibly of secondary importance though it gives a good check on the condition of engine fuel pumps. From experience, the condition of pump plungers and barrels is of prime importance in the ability of an engine to manoeuver reliably, and that performance monitoring is the only true test of their efficiency. From this graph (Figure 5.4) four very important parameters can be measured:

- FP max, maximum fuel injection pressure
- FP open, fuel oil injection pressure when the valve opens
- $\alpha$ P open, the angle where FP open occurs referred to T.D.C.
- G, fuel oil delivery in crank angle degrees.

Obviously in measuring parameters with respect to time, reference points in the crank cycle are utilized and so, highly accurate proximity sensors fitted in the engine flywheel were used to give all the top dead position marks and also a read out of engine speed.

5.4.2.3 Indicated Horse Power (IHP) - Figure 5.5

The indicated horse power can be found easily by the following formula.

Indicated horse power per cylinder =
MIP x RPM x Constant

The total indicated power is the sum of all cylinders power.

5.4.2.4 Fault Finding

This monitoring system can also be used for fault finding. Two references can be used to commence fault finding.

- comparison of the values of fuel pressure distribution from MIP calculator with normal values
- comparison of photographs of actual fuel traces with the "ideal trace" (Figure 5.6)

Tuning of the fuel system to achieve normal readings across all cylinders within the prescribed limits is associated by sketches that show "faulty" fuel traces, typical of engine defects compared with "ideal traces" (Figure 5.6).

A fault finding matrix (Figure 5.7) is also provided, to aid in identifying worn engine parts or incorrect setting of fuel system components.

If a ship´s staff decided to make adjustments at this stage without completing the whole performance check the "initial engine condition" is not known and there is no basis on which to judge any improvements.

INTERNATIONAL CHAMBER OF SHIPPING

RECOMMENDED BUNKER DELIVERY NOTE

NAME OF VESSEL · (PORT)

| | |
|---|---|
| Product name or general designation | |
| Quantity in tonnes | |
| Volumetric Quantity ($m^3$) | |
| Temperature of Product (°C) | |

| | |
|---|---|
| Density at 15°C (kg/litre) | |
| Kinematic Viscosity (cSt) at corresponding temperature (°C) | |
| Flash Point (P–M) Closed Cup (°C) | |
| Pour Point (°C) | |
| Conradson Carbon Residue (percentage by mass) | |
| Sediment (percentage by mass) | |
| Water Content (percentage by volume) | |
| Cetane Index (distillate fuels only) | |
| Sulphur (percentage by mass) | |
| Vanadium (mg/kg) | |

*Note:* Specific energy (calorific value) (MJ/kg) can be calculated from the known density and sulphur contents in the formulae quoted in BS 2869 Amendment 2 "Petroleum Fuels for Oil Engines and Burners".

Bunker Supplier: ______________________

Signature of Suppliers' Representative or Agent: ______________________

Date of Delivery: ______________________

FIGURE 5.1

2 Block diagram of MIP calculator

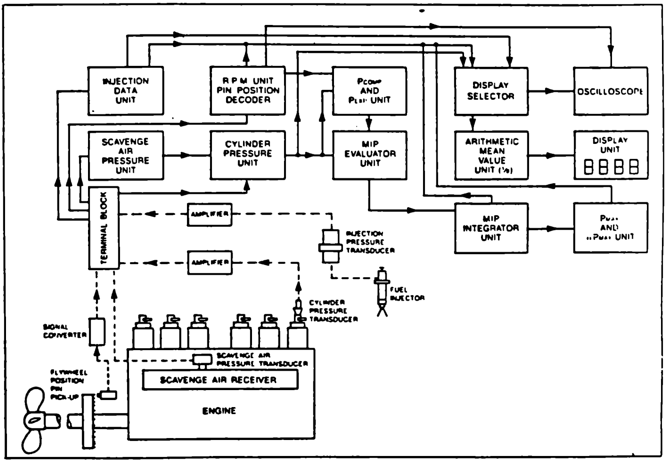
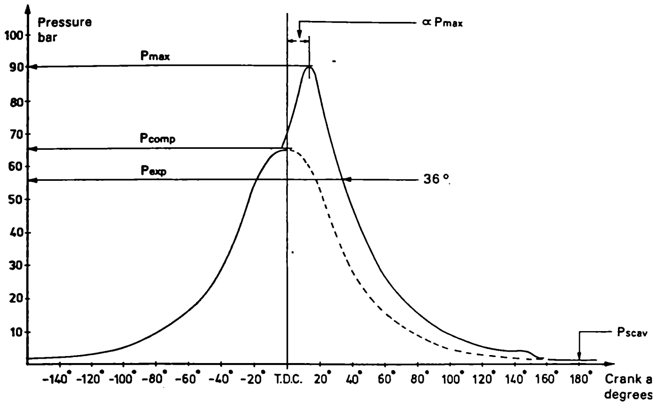



FIGURE 5.2

## Measured and calculated parameters and pressure curves

### Cylinder pressure curve



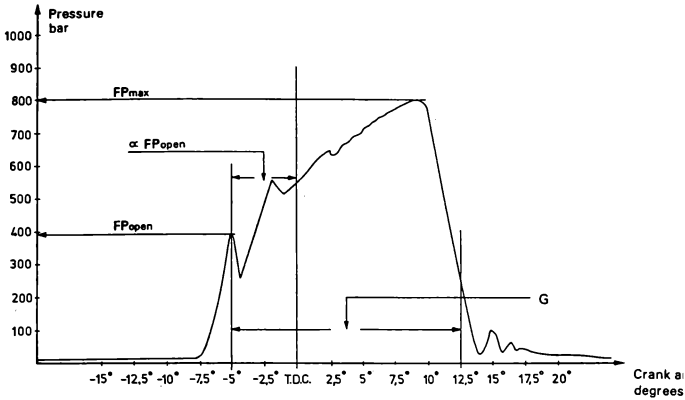

### Calculated parameters

| | |
|---|---|
| MIP | : Mean indicated pressure |
| $P_{max}$ | : Maximum combustion pressure |
| $P_{comp}$ | : Compression pressure |
| $P_{exp}$ | : Pressure on the expansion curve, 36 ° after T.D.C. |
| $\alpha$ $P_{max}$ | : The angle where $P_{max}$ occurs, referred to T.D.C. |
| LOAD | : Cylinder load in Kw |

FIGURE 5.3

## Measured and calculated parameters and pressure curves

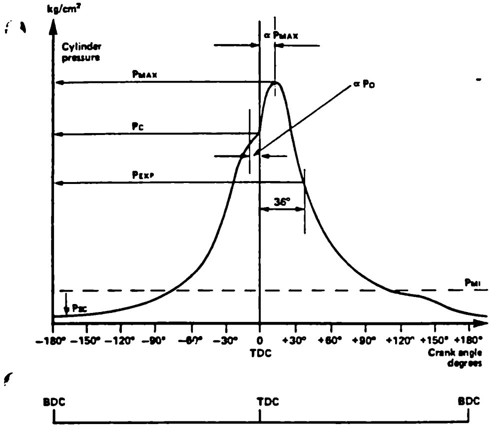
**Injection pressure curve**




**Calculated parameters**

| | | |
|---|---|---|
| $FP_{max}$ | : | Maximum fuel oil injection pressure |
| $FP_{open}$ | : | Fuel oil injection pressure when the valve opens |
| $\propto FP_{open}$ | : | The angle where $FP_{open}$ occurs, referred to T.D.C. |
| G | : | Injection period in crank angle degrees |

FIGURE 5.4




.4 Typical trace of cylinder pressu
at approximately full power

| | | |
|---|---|---|
| BDC | Bottom dead centre | |
| TDC | Top dead centre | |
| SO | Scavenge ports open | 142.5 deg approx AT |
| SC | Scavenge ports close | 142.5 deg approx BT |
| FVO | Fuel valve opens | 4 deg approx BTDC |
| EO | Exhaust opens | ... deg ATDC |
| EC | Exhaust closes | ... deg BTDC |

FIGURE 5.5

**Effects on fuel-pressure curve of: worn fuel-pump internals; leaking valve on fuel pump**

INDICATIONS

- Reduced maximum fuel pressure.
- Low rate of pressure rise.
- Reduced injection period
- Opening angle before TDC too short.
- Reflecting pressure wave damped.

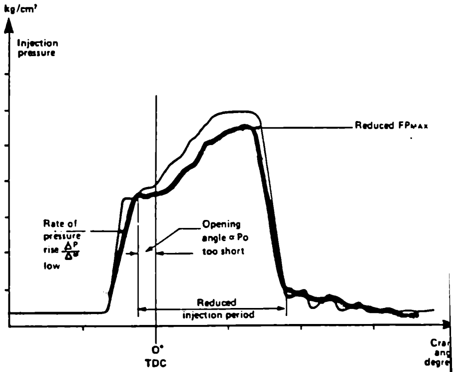



FIGURE 5.6

| FAULTS | INDICATION | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|
| | $\frac{\Delta P}{\Delta \alpha}$ LOW | FPo LOW | FPo HIGH | $P_{MAX}$ HIGH | $P_{MAX}$ LOW | $\alpha$ Po EARLY | $\alpha$ Po LATE | Injection period LONG | Injection period SHORT | Reflected pressure wave DAMPED |
| Worn fuel pump internals | ■ | | | | ■ | | ■ | | ■ | ■ |
| Leaking valve on fuel pump | ■ | | | | ■ | | ■ | | ■ | ■ |
| Fuel valve spring set too HIGH | | | ■ | | | | ■ | | ■ | |
| Fuel valve spring set too LOW or broken | | ■ | | | | ■ | | ■ | | |
| Fuel valve nozzle worn excessively or damaged | | | | | ■ | | | | | |
| Fuel pump index setting too HIGH | | | | ■ | | | | ■ | | |
| Fuel pump index setting too LOW | | | | | ■ | | | | ■ | |
| Choked fuel valve nozzle | | | | ■ | | | | | | |
| Fuel pump timing too ADVANCED (LEAD) | | | | | | ■ | | ■ | | |
| Fuel pump timing too RETARDED (LEAD) | | | | | | | ■ | | ■ | |
| Fuel viscosity too HIGH | | | | ■ | | | | | | |
| Fuel viscosity too LOW | | | | | ■ | | | | | |

Pre-performance check: fault-finding table

*Fuel- Pressure curve*

FIGURE 5.7

# CHAPTER SIX

# ECONOMICAL AND TECHNICAL ASPECTS OF MAIN AND AUXILIARY ENGINES CHOICE

## 6.1 GENERAL CONSIDERATION

Irrespective of whether the shipowner is running his business at a profit or loss, he should pay attention to reducing his expenditures.

Within the overall concept of the internal combustion engine, no new diesel engine design can be termed "revolutionary" but what designers aim for is an amalgamation of ideas and improvements which, when the new engine is introduced, represents an advance on the progress of other manufacturers. It may be imagined that so much experience has now been gained in design and operation of diesel engines that a computerized design is almost inevitable yet, as recent events have shown, there is still no assurance that operational and design experience plus advanced development and research techniques will ensure automatically a new, efficient and reliable engine. In view of the reluctance of most shipowners to accept the problems of maintaining many cylinders, the move has been towards larger cylinder sizes with increased output per cylinder, e.g. slow speed two-stroke engines.

## 6.2 THE MOST SIGNIFICANT FACTORS FOR THE CHOICE OF MAIN AND AUXILIARY ENGINES

### 1. Choice of Speed/Flexibility

The choice of speed is dependent on some parameters like freight rate and bunker prices. This is why shipowners are, more than ever, ready to work with shipyards and engine manufacturers in order to design and adapt propulsive installations of new and existing ships to better flexibility. Under such circumstances, it has become desirable for a marine engine design to cover a wide range of outputs and, in addition, to allow a free choice of propeller revolutions. On the other hand the production of the new version derated engines gives a better flexibility in choosing optimum speeds.

2. Vibration

Two traditional sources of vibration excitation have always been the propeller and the main engine. Reduced engine RPM (60-100) has put emphasis on the main engine as the main source of excitation. On the other hand engine manufacturers install effective engine balancers and vibration dampers to reduce the effect of excitation forces. Exciting forces of medium speed engines are of smaller magnitude than slow speed engines.

3. Reliability

Low specific fuel oil consumption of the last generation of engines (slow and medium speed) are very satisfactory for the shipowner but this raises an important question:

If any compromise is to be made between higher reliability and lower fuel consumption, a shipowner will never hesitate and will choose reliability due to the subjective fact that "a ship down time is more expensive than any spare parts for the engines".

The lack of reliability disturbs not only the schedule of the damaged ship but also the schedule of the entire fleet. Figure 6.1 shows that medium speed engine have higher cost claims than low speed engines.

Furthermore, we should not overlook the influence of the methodology of tradition on machinery choice. It is effective, if not decisive, to have a conventional study before final selection of machinery. However, even if considerable differences are found between the compared machinery, tradition may still have a decisive influence on the final machinery choice whatever differences in criteria such as fuel consumption, simplicity of operation and maintenance, may exist. This is why shipowners favour slow speed engines in spite of lower initial cost of medium speed engines.

4. Competition

Competition between engine manufacturers is the only efficient way to compare the quality of engines, prices, service, etc.

5. Initial Cost

The real differences in initial costs of the engines are hidden in the ship price or in the received subsidy. Nevertheless, no economical study can neglect the first and operating costs. Neither the shipowner nor the shipyard neglects the economical study.

Generally speaking, when a choice is given between medium speed main engines and slow speed main engines, the traditional thinking of the shipowner will favour slow speed engines in spite of lower initial cost of medium speed engines. Technical and economical comparison between the alternatives should be made. Emphasis on costs and difference of revenue should be kept in mind.

## 6. Running Cost

**Crews:**

National shipping companies and shipowners are today operating ships in two ways:

1. Under national flag with reduced crews so that more and more diesel maintenance is being done on shore.

2. Under economical flag with crews greater in number, but less skilled so that maintenance load on board has to be kept within their capabilities. In both cases when maintenance must be done on board, tasks have to be kept as simple as possible. This is one reason why in line engines are preferred by owners over vee-form engines.

In the past, generally speaking, the first and operating costs were the principal factors influencing the selection of machinery. However, the selection of machinery has become a more complicated matter in recent times. At the same time, it is true that the availability of experienced personnel, trained to run and maintain given machinery has been, and will continue to be an intangible yet major factor in selection of the machinery. A fleet engined by one particular series of engines makes is easier to change operating staff within the company and consequently can be of benefit from the preventive maintenance view point. Such standardisation of engine types will also increase the efficiency of personnel training.

**Maintenance:**

By experience the maintenance costs for a medium speed diesel engine are definitively higher than for the equivalent slow speed engine.

Marine propulsion plants, whatever type they are, have to fulfill the following condition:

- Number of crew and manhours needed for operation and maintenance of the plant should be a minimum.

One of the most serious problems for shipowners is the shortage of crew as well as high personnel expenses, and these factors will increase in the near future. Easy maintenance is of deep concern to shipowners in relation to these problems, because it is generally considered that overhauling of the engine components is expensive due to the greater number of cylinders installed in the case of medium speed engines. On the other hand, to the engine manufacturer it is necessary to make efforts to reduce the manhours needed for operating and maintenance of the plant. Keeping the spare parts stock to a level appropriate to expected damages should not only be the owner duty but also the engine manufacturers responsibility as a particular aspect of reliability.

**Lube Oil Consumption:**

Now-a-days the medium and slow speed engines have approximately the same specific lube oil consumption.

**Fuel Oil Consumption:**

The medium speed engines now burn the same fuel but with some particular difficulties in maintenance and lube oil consumption. Most of the medium speed engines use heavy fuel oil with lower viscosities. The spread in heavy fuel oil and marine diesel oil prices should theoretically still justify using heavy fuel oil in auxiliary engines; in fact extra costs due to excessive maintenance, overtime and lube oil consumption are a strong incentive to come back to marine diesel oil.

Now-a-days the medium and slow speed engines have approximately the same specific fuel consumption. On the other hand, slow speed engines usually use heavy fuel oil with higher viscosities.

## 6.3 TECHNICAL AND ECONOMICAL COMPARISON OF MAIN AND AUXILIARY ENGINES

In choosing engines for ship propulsion subjective fact still remain. Shipowners do not like risks and shipping is technically and economically a risky business. Comparison between alternatives two-stroke and four-stroke will be made to introduce the technical and economical performance of the two.

### 6.3.1 Investment and Operating Costs

It is assumed for all examples that:

1. Contracted Maximum Continuous Rating (CMCR) for all the engines is equal to 11000 kw.

2. Machinery prices are based on an average cost/kw and not on quoted manufacturers prices and are thus for comparison only.

3. Negative costs are regarded as equivalent to fuel savings.

4. The Continuous Service Rating (CSR) is estimated at 85% of Maximum Continuous Rating (MCR).

5. The ship is assumed to be a multi-purpose cargo ship.

6. When using prices for an economic survey all assumptions have to be made on an equal basis and for this comparison, therefore, prices are based on those for engines manufactured in Japan at the end of 1985.

7. Fuel oil and lube oil prices are based on the prices of May 1990.

8. The intention from this study is to show only the methods used in economical comparison.

| | |
|---|---|
| 9. Sailing days of ship | 250 days per year |
| 10. Interest Rate | 8.0% |
| 11. Inflation Rate | 4.0% |
| 12. Required Life Time | 10 years |

## ASSUMED MACHINERY COSTS

| Type | $/KW |
|---|---|
| Two Stroke Engine | 350 |
| Four Stroke Engine | 250 |
| Diesel Generator (MDO) | 200 |
| Diesel Generator (HFO) | 275 |
| Shaft Generator with CSG | 350 |

## ASSUMED MACHINERY INVESTMENT

| Machinery Type | Power KW | Investment Cost ($) |
|---|---|---|
| 4 RTA 76 Sulzer | 11480 | 4,018,000 |
| 6 RTA 62 Sulzer | 12180 | 4,263,000 |
| 6 S 60 MC MAN - B & W | 11220 | 3,927,000 |
| 5 L 70 MC MAN - B & W | 11800 | 4,130,000 |
| 9 PC-40 SEMT Pielstick | 11925 | 3,000,000 |
| Shaft Generator with CSG | 500 | 263,000 |
| TCS/PTI | - | 154,000 |
| PTO/PTI | - | 417,000 |
| Diesel Generator (MDO) | 500 | 100,000 |
| Diesel Generator (HFO) | 500 | 137,000 |
| Coupling or Clutch and Reduction Gear | - | 229,000 |

## FUEL AND LUBE OIL PRICES - MAY 1990

| Fuel Type | Cost $/Tonne |
|---|---|
| Heavy Fuel Oil 380 CST $50^{\circ}C$ | 112 |
| Heavy Fuel Oil 180 CST $50^{\circ}C$ | 117 |
| Marine Diesel Oil | 212 |
| Lube Oil | 1000 |

## COST OF MAINTENANCE AND SPARE PARTS

| Type | Cost ($) | Cost ($)/Year |
|---|---|---|
| Diesel Generator (MDO) | 1.33/h | 7,980 |
| Diesel Generator (HFO) | 2.49/h | 14,940 |
| Shaft Generator (PTO) | 2.0/KW | 1,000 |
| TCS/PTI 247 KW | 10.0/KW | 2,470 |

## COST OF LUBE OIL CONSUMPTION

| Type | Specific Consumption | Cost ($)/Year |
|---|---|---|
| Diesel Generator (MDO) or (HFO) | 3.0 g/kwh | 9,000 |
| Shaft Generator (PTO) | 0.5 g/kwh | 1,500 |
| TCS/PTI | 0.5 g/kwh | 741 |
| PTO/PTI-747 kw | 1.0 g/kwh | 4,482 |

MDO = Marine Diesel Oil
HFO = Heavy Fuel Oil
PTO = Power Take-Off
CSG = Constant Speed Generator
TCS = Turbo-Compound System (booster efficiency)
PTI = Power Take-In

### 6.3.2 Example I

## MAIN ENGINE CHOICE

Five Alternatives Assumed:

| Type | MCR KW | CMCR KW | MCR-RPM | CMCR-RPM | CMCR (%) | RPM (%) |
|---|---|---|---|---|---|---|
| 4RTA 76 | 11480 | 11000 | 104 | 100 | 96 | 96 |
| 6RTA 62 | 12180 | 11000 | 109 | 100 | 90 | 92 |
| 6S60 MC | 11220 | 11000 | 102 | 100 | 98 | 98 |
| 5L 70 MC | 11800 | 11000 | 100 | 100 | 93 | 100 |
| 9 PC-40 | 11925 | 11000 | 375 | 100 | 92 | 100 |

From engines layout fields we get: Fig (6.2)

| Type | CMCR KW | SFOC g/kwh | Fuel Cost ($)/Year |
|---|---|---|---|
| 4 RTA 76 | 11000 | 172.2 | 1,336,547 |
| 6 RTA 62 | 11000 | 171.4 | 1,330,338 |
| 6 S 60 MC | 11000 | 172.2 | 1,336,547 |
| 5 L 70 MC | 11000 | 170.2 | 1,321,024 |
| 9 PC-40 | 11000 | 172.7 | 1,340,428 |

For difference in lower calorofic value between MDO and HFO we multiply by 1.05 for the fuel cost calculation.
Medium speed engine used the same HFO as slow speed.
250 Sailing days a year.

Correction of the Rate of Interest (i)

| | |
|---|---|
| Rate of inflation (e) | = 4% |
| Rate of interest (i) | = 8% |
| Required life time (n) | = 10 years |
| The effective discount rate (v) | = ? |

The equation: $$(1 + v) = \frac{(1 + i)}{(1 + e)}$$

then $$(1 + v) = \frac{1.08}{1.04} = 1.038$$

$$v = 3.8\%$$

We will assume $v = 4\%$

Calculation of the Series Present Worth Factor (SPWP)

$$SPW = \frac{1 - (1 + v)^{-n}}{v}$$

$$= \frac{1 - (1 + 4.0)^{-10}}{0.04}$$

$$SPW = 8.1$$

Calculation of the Capital Recovery Factor (CRF)

CRF = The ratio between uniform savings per year and the difference investment

- Simple pay back period $= \frac{1}{CRF}$

- Pay out period (POP):
  From table 6.1, for a calculated CRF and a given rate of discount v = 4%, the POP can be found.

- Net present value (NPV):
  NPV = (SPW x savings per year) - (difference investment)

- Internal Rate of Return (IRR):
  From table 6.1, for a calculated CRF and a given required life time (n) = 10 years, the IRR can be found.

### 6.3.3 Economical Comparison - Example I

**TABLE I - EXAMPLE I**

| Alternative | A | B | C | D | E |
|---|---|---|---|---|---|
| Type | 4RTA76 | 6RTA62 | 6S60MC | 5L70MC | 9PC-40 |
| CMCR kw | 11000 | 11000 | 11000 | 11000 | 11000 |
| CMCR/MCR % | 96% | 90% | 98% | 93% | 92% |
| SFOC g/kwh | 172.2 | 171.4 | 172.2 | 170.2 | 172.7 |
| Fuel Cost $/year | 1336547 | 1330338 | 1336547 | 1321024 | 1340428 |
| Savings $ | 0 | 6209 | 0 | 15523 | -3881 |
| M/E Investment dif. | 0 | 245000 | -91000 | 112000 | -1018000 |
| Reduction gear,etc. | - | - | - | - | 229000 |
| Total Investment | 0 | 245000 | -91000 | 112000 | -789000 |
| Diffe. Investment | 0 | 245000 | -91000 | 112000 | -789000 |
| CRF | 0 | 0.025 | N/A | 0.138 | N/A |
| POP | 0 | N/A | N/A | 8.5 Yrs | N/A |
| NPV | 0 | -194707 | 91000 | -13736 | 757564 |
| IRR | 0 | N/A | N/A | 6% | N/A |

N/A = not applicable

The basic alternative "A" propulsion is a Sulzer 4RTA76 two-stroke engine developing 11480 kw at 104 RPM. Alternatives "B", "C" and "D" are both two-stroke engines. Alternative "E" is a four-stroke engine. It is assumed that all alternatives have the same propeller diameter, speed and propulsive efficiency. The points of interest here are the number of engine units and the lowest investment. Alternative "A" has the lowest number of units "4", on the other hand its investment cost is higher than alternatives "C" and "E" by 91000 US$ and 789000 US$ respectively. Alternative "D" gives savings in fuel amounting to 15523 US$ per year, but on the other hand has a negative NPV due to the higher investment cost; also the POP is about one and a half times the assumed ship life. Alternative "B" has a higher investment cost than alternative "A" and the same fuel consumption; it also has a negative NPV and differs from "A" by having two more cylinder units. Alternative "C" has a lower investment cost than "A" but with the same fuel consumption; it is different from "A" by having two more cylinders. Alternative "E" has the highest number of units and the better investment cost including the coupling or clutch and reduction gear. From an econmical point of view alternative "E" is the better choice due to the lowest investment cost, 800,000 US$ cheaper than "A".

When taking into consideration all the significant factors influencing main engine choice, we consider that alternative "A" is the better choice due to the following advantages:

1. Lower number of cylinders which means lower maintenance cost.

2. Directly coupled to the propeller shaft, which reduces the maintenance due to the absence of reduction gear and coupling or clutch.

3. Simplicity on maintenance and overhauling.

4. More reliable than alternative "E" due to the lowest RPM.

To obtain the minimum fuel consumption the engine (4RTA76) has to be run at 85% of the (MCR). The Continuous Service Rating (CSR) will be then:

$$CSR = 0.85 \times 11480 = 9758 \text{ kw}$$

By using the engine layout field, (Figure 6.2) the specific fuel consumption at (CSR) is:

SFOC without TCS = 166.9 g/kwh

SFOC with TCS = 162.7 g/kwh

Fuel Cost US$/year without TCS = 1,149,147 US$

Fuel Cost US$/year with TCS = 1,120,229 US$

$$\text{Power delivery by TCS} = \frac{3.1 \text{ g/kwh}}{166.9 \text{ g/kwh}} \times 9758 = 247 \text{ kw}$$

### 6.3.4 Economical Comparison Example II

Alternative "A" is the basic 4 RTA 76 without TCS/PTI
Alternative "B" is 4 RTA 76 with TCS/PTI

**TABLE II - EXAMPLE II**

| Alternative | A | B |
|---|---|---|
| Type | 4 RTA 76 | 4RTA 76 with TCS/PTI |
| CSR kw | 9758 | 9758 |
| CSR/MCR % | 85 | 85 |
| SFOC g/kwh | 166.9 | 162.7 |
| Fuel Cost US$/year | 1,149,147 | 1,120,229 |
| Fuel Savings US$/year | 0 | 28,918 |
| TCS/PTI Cost US$ | 0 | 154,000 |
| TCS/PTI LOC Cost US$ | 0 | - 741 |
| TCS/PTI Maintenance Cost US$ | 0 | - 2470 |
| Total Savings US$/Year | 0 | 25707 |
| Total Investment US$ | 0 | 154,000 |
| Difference Investment | 0 | 154,000 |
| CRF | 0 | 0.167 |
| POP | 0 | 7 years |
| NPV | 0 | 54,226 |
| IRR | 0 | 11% |

From the Table II, alternative "B" has a seven years payout period which in most cases is unfavourable period for the shipowners, but from economical point of view the TCS/PTI is better due to the positive value of NPV 54226 US$.

### 6.3.5 Economical Comparison Example III

Alternative "A" is the basic:
3 diesel generator sets using marine diesel oil.

Alternative "B":
1 shaft generator PTO 500 kw with two diesel generator
calculation of fuel oil cost/year US$:
Fuel cost by PTO = 58,882 US$/year
Fuel cost by diesel generator = 131,652 US$/year

**TABLE III - EXAMPLE III**

| Alternative | A | B |
|---|---|---|
| Type | 1 DG Using MDO | 1 Shaft Generator PTO |
| Power/set kw | 500 | 500 |
| SFOC g/kwh | 207 | 166.9 |
| Fuel Cost/year US$ | 131,652 | 58,882 |
| Savings in Fuel Cost US$/Year | 0 | 72,770 |
| Savings in LO Cost US$/Year | 0 | 7,500 |
| Savings in Maintenance US$/Year | 0 | 6,980 |
| Difference Investment | 0 | 163,000 |
| Total Savings US$ | 0 | 87,250 |
| CRF | 0 | 0.535 |
| POP | 0 | 2 years |
| NPV | 0 | 543,725 |
| IRR | 0 | over 30% |

From Table III, the payout period is only 2 years which in most cases is a favourable period for the shipowner, also the positive NPV about 0.6 million. Alternative "B" is economically a better choice.

### 6.3.6 Economical Comaprison - Example IV

Alternative "A" is the basic:
3 diesel generator using heavy fuel oil, 180 CST, 117 US$/tonne.

Alternative "B":
1 shaft generator PTO 500 kw with two diesel generator using HFO calculation of fuel oil cost US$/year.
Fuel cost by diesel generator = 78,133 US$/year.

**TABLE IV - EXAMPLE IV**

| Alternative | A | B |
|---|---|---|
| Type | 1 DG using HFO | 1 PTO |
| Power/set kw | 500 | 500 |
| SFOC g/kwh | 212 | 166.9 |
| Fuel cost US$/year | 78,132 | 58,882 |
| Fuel savings US$/year | 0 | 19,250 |
| Savings in LOC US$/year | 0 | 7,500 |
| Savings in maintenance US$/year | 0 | 13,940 |
| Total Savings | 0 | 40,690 |
| Difference Investment | 0 | 125,500 |
| CRF | 0 | 0.324 |
| POP | 0 | 3.5 years |
| NPV | 0 | 204,089 |
| IRR | 0 | 30% |

From Table IV, alternative "B" has a payout period of 3.5 years which is a favourable period for the shipowner. On the other hand because alternative "B" has a positive NPV about 0.20 million, it is economically better.

### 6.3.7 Economical Comparison - Example V

Alternative "A" is the basic:
3 diesel generator sets using marine diesel oil.

Alternative "B":
1 shaft generator with TCS/PTI and two diesel generator.

Calculations of fuel oil cost US$/year:
Fuel oil cost for PTO = 57,400 US$/year
Fuel oil cost for diesel generator = 131,652 US$/year
Fuel saving by TCS = 28,918 US$/year

**TABLE V - EXAMPLE V**

| Alternative | A | B |
|---|---|---|
| Type | 1 DG using MDO | 1 PTO/PTI |
| Power/set kw | 500 kw | 500 kw |
| SFOC g/kwh | 207 | 162.7 |
| Fuel cost US$/year | 131,652 | 57,400 |
| Savings in fuel cost US$/year | 0 | 74,252 |
| Savings in TCS US$/year | 0 | 28,918 |
| Savings in LOC US$/year | 0 | 4,518 |
| Savings in maintenance US$/year | 0 | 5,510 |
| Total Savings US$ | 0 | 112,198 |
| Difference Investment | 0 | 317,000 |
| CRF | 0 | 0.354 |
| POP | 0 | 3 years |
| NPV | 0 | 591,803 |
| IRR | 0 | < 30% |

From Table IV, alternative "B" PTO/PTI has a payout period of 3 years and a positive NPV about 0.6

million. From economical point of view it is a better choice.

### 6.3.8 Economical Comparison - Example VI

Alternative "A" is the basic:
3 diesel generator sets using heavy fuel oil 180 CST, 117 US$/tonne.

Alternative B:
1 shaft generator with TCS/PTI and two diesel generator.
Calculation of fuel cost US$/year:

Fuel cost for PTO = 57400 US$/year
Fuel cost for diesel generator = 78132 US$/year
Fuel saving by TCS = 28,918 US$/year

**TABLE VI - EXAMPLE VI**

| Alternative | A | B |
|---|---|---|
| Type | 1 DG using HFO | 1 PTO/PTI |
| Power/set kw | 500 | 500 |
| SFOC g/kwh | 212 | 162.7 |
| Fuel cost US$/year | 78,132 | 57,400 |
| Saving fuel cost US$/year | 0 | 20,732 |
| Saving in TCS US$/year | 0 | 28,918 |
| Savings in LOC US$/year | 0 | 4,815 |
| Savings in maintenance US$/year | 0 | 11,470 |
| Total Savings US$ | 0 | 65,935 |
| Difference Investment | 0 | 279,500 |
| CRF | 0 | 0.235 |
| POP | 0 | 4-5 years |
| NPV | 0 | 254,573 |
| IRR | 0 | 20% |

From Table VI, alternative "B" PTO/PTI has a payout period of 4.5 years and a positive NPV about 0.25 million. It is economically a better choice.

### 6.3.9 Conclusions for Chapter Six

By comparing all the alternatives:

1. Power take-off generator PTO.
2. Turbo-compound system TCS/PTI.
3. Power take-off generator with turbo-compound system PTO/PTI.

The author˜s conclusions are:

Choosing the power take-off with turbo-compound system Example V is the best choice from economical point of view due to:

1. The payout period (POP) is only 3 years which is a favourable period for the shipowner.

2. The net present value (NPV), about 0.6 million which is the best amount for the shipowner compared with the payout period.

3. The internal rate of return (IRR), greater than 30%, is higher than the assumed rate of interest (i) 8%.

**Swedish Club main engine claims, 1981-August 1986**

**(Figures in US$ millions*)**

| Year | Claims > US$0.14 million each** | | | Total, incl smaller claims |
|---|---|---|---|---|
| | Medium speed | Low speed | Total, incl steam turbines | |
| 1981 | 2 52 | 0.81 | 3.33 | 5.14 |
| 1982 | 0.86 | — | 2.26 | 3.83 |
| 1983 | 1.97 | — | 3.13 | 5.27 |
| 1984 | 1.12 | 0.22 | 2.03 | 3.54 |
| 1985 | 4.45 | 1.67 | 6.53 | 8.72 |
| 1986 | 3.74 | 0.53 | 4.28 | 6.23 |
| | 14.66 | 3.23 | 21.56 | 32.73 |

* US$1 = SKr6.942

** SKr1 million each

FIGURE 6.1

# CAPITAL RECOVERY FACTOR

DISCOUNT RATE P.C.

| YEAR | 1 | 2 | 3 | 4 | 5 | 6 | 7 | 8 | 9 | 10 |
|---|---|---|---|---|---|---|---|---|---|---|
| 1 | 1.010000 | 1.020000 | 1.030000 | 1.040000 | 1.050000 | 1.060000 | 1.070000 | 1.080000 | 1.090000 | 1.100000 |
| 2 | 0.507512 | 0.515050 | 0.522611 | 0.530196 | 0.537805 | 0.545437 | 0.553092 | 0.560769 | 0.568469 | 0.576190 |
| 3 | 0.340022 | 0.346755 | 0.353530 | 0.360349 | 0.367209 | 0.374110 | 0.381052 | 0.388034 | 0.395055 | 0.402115 |
| 4 | 0.256281 | 0.262624 | 0.269027 | 0.275490 | 0.282012 | 0.288591 | 0.295228 | 0.301921 | 0.308669 | 0.315471 |
| 5 | 0.206040 | 0.212158 | 0.218355 | 0.224627 | 0.230975 | 0.237396 | 0.243891 | 0.250456 | 0.257092 | 0.263797 |
| 6 | 0.172548 | 0.178526 | 0.184598 | 0.190762 | 0.197017 | 0.203363 | 0.209796 | 0.216315 | 0.222920 | 0.229607 |
| 7 | 0.148628 | 0.154512 | 0.160506 | 0.166610 | 0.172820 | 0.179135 | 0.185553 | 0.192072 | 0.198691 | 0.205405 |
| 8 | 0.130690 | 0.136510 | 0.142456 | 0.148528 | 0.154722 | 0.161036 | 0.167468 | 0.174015 | 0.180674 | 0.187444 |
| 9 | 0.116740 | 0.122515 | 0.128434 | 0.134493 | 0.140690 | 0.147022 | 0.153486 | 0.160080 | 0.166799 | 0.173641 |
| 10 | 0.105582 | 0.111327 | 0.117231 | 0.123291 | 0.129505 | 0.135868 | 0.142378 | 0.149029 | 0.155820 | 0.162745 |
| 11 | 0.096454 | 0.102178 | 0.108077 | 0.114149 | 0.120389 | 0.126793 | 0.133357 | 0.140076 | 0.146947 | 0.153963 |
| 12 | 0.088849 | 0.094560 | 0.100462 | 0.106552 | 0.112825 | 0.119277 | 0.125902 | 0.132695 | 0.139651 | 0.146763 |
| 13 | 0.082415 | 0.088118 | 0.094030 | 0.100144 | 0.106456 | 0.112960 | 0.119651 | 0.126522 | 0.133567 | 0.140779 |
| 14 | 0.076901 | 0.082602 | 0.088526 | 0.094669 | 0.101024 | 0.107585 | 0.114345 | 0.121297 | 0.128433 | 0.135746 |
| 15 | 0.072124 | 0.077825 | 0.083767 | 0.089941 | 0.096342 | 0.102963 | 0.109795 | 0.116830 | 0.124059 | 0.131474 |
| 16 | 0.067945 | 0.073650 | 0.079611 | 0.085820 | 0.092270 | 0.098952 | 0.105858 | 0.112977 | 0.120300 | 0.127817 |
| 17 | 0.064258 | 0.069970 | 0.075953 | 0.082199 | 0.088699 | 0.095445 | 0.102425 | 0.109629 | 0.117046 | 0.124664 |
| 18 | 0.060982 | 0.066702 | 0.072709 | 0.078993 | 0.085546 | 0.092357 | 0.099413 | 0.106702 | 0.114212 | 0.121930 |
| 19 | 0.058052 | 0.063782 | 0.069814 | 0.076139 | 0.082745 | 0.089621 | 0.096753 | 0.104128 | 0.111730 | 0.119547 |
| 20 | 0.055415 | 0.061157 | 0.067216 | 0.073582 | 0.080243 | 0.087185 | 0.094393 | 0.101852 | 0.109546 | 0.117460 |
| 21 | 0.053031 | 0.058785 | 0.064872 | 0.071280 | 0.077996 | 0.085005 | 0.092289 | 0.099832 | 0.107617 | 0.115624 |
| 22 | 0.050864 | 0.056631 | 0.062747 | 0.069199 | 0.075971 | 0.083046 | 0.090406 | 0.098032 | 0.105905 | 0.114005 |
| 23 | 0.048886 | 0.054668 | 0.060814 | 0.067309 | 0.074137 | 0.081278 | 0.088714 | 0.096422 | 0.104382 | 0.112572 |
| 24 | 0.047073 | 0.052871 | 0.059047 | 0.065587 | 0.072471 | 0.079679 | 0.087189 | 0.094978 | 0.103023 | 0.111300 |
| 25 | 0.045407 | 0.051220 | 0.057428 | 0.064012 | 0.070952 | 0.078227 | 0.085811 | 0.093679 | 0.101806 | 0.110168 |
| 30 | 0.038748 | 0.044650 | 0.051019 | 0.057830 | 0.065051 | 0.072649 | 0.080586 | 0.088827 | 0.097336 | 0.106079 |

| YEAR | 11 | 12 | 13 | 14 | 15 | 16 | 18 | 20 | 25 | 30 |
|---|---|---|---|---|---|---|---|---|---|---|
| 1 | 1.110000 | 1.120000 | 1.130000 | 1.140000 | 1.150000 | 1.160000 | 1.180000 | 1.200000 | 1.250000 | 1.300000 |
| 2 | 0.583934 | 0.591698 | 0.599484 | 0.607290 | 0.615116 | 0.622963 | 0.638716 | 0.654545 | 0.694444 | 0.734783 |
| 3 | 0.409213 | 0.416349 | 0.423522 | 0.430731 | 0.437977 | 0.445258 | 0.459924 | 0.474725 | 0.512295 | 0.550627 |
| 4 | 0.322326 | 0.329234 | 0.336194 | 0.343205 | 0.350265 | 0.357375 | 0.371739 | 0.386289 | 0.423442 | 0.461629 |
| 5 | 0.270570 | 0.277410 | 0.284315 | 0.291284 | 0.298316 | 0.305409 | 0.319778 | 0.334380 | 0.371847 | 0.410582 |
| 6 | 0.236377 | 0.243226 | 0.250153 | 0.257157 | 0.264237 | 0.271390 | 0.285910 | 0.300706 | 0.338819 | 0.378394 |
| 7 | 0.212215 | 0.219118 | 0.226111 | 0.233192 | 0.240360 | 0.247613 | 0.262362 | 0.277424 | 0.316342 | 0.356874 |
| 8 | 0.194321 | 0.201303 | 0.208387 | 0.215570 | 0.222850 | 0.230224 | 0.245244 | 0.260609 | 0.300399 | 0.341915 |
| 9 | 0.180602 | 0.187679 | 0.194869 | 0.202168 | 0.209574 | 0.217082 | 0.232395 | 0.248079 | 0.288756 | 0.331235 |
| 10 | 0.169801 | 0.176984 | 0.184290 | 0.191714 | 0.199252 | 0.206901 | 0.222515 | 0.238523 | 0.280073 | 0.323463 |
| 11 | 0.161121 | 0.168415 | 0.175841 | 0.183394 | 0.191069 | 0.198861 | 0.214776 | 0.231104 | 0.273493 | 0.317729 |
| 12 | 0.154027 | 0.161437 | 0.168986 | 0.176669 | 0.184481 | 0.192415 | 0.208628 | 0.225265 | 0.268448 | 0.313454 |
| 13 | 0.148151 | 0.155677 | 0.163350 | 0.171164 | 0.179110 | 0.187184 | 0.203686 | 0.220620 | 0.264543 | 0.310243 |
| 14 | 0.143228 | 0.150871 | 0.158667 | 0.166609 | 0.174688 | 0.182898 | 0.199678 | 0.216893 | 0.261501 | 0.307818 |
| 15 | 0.139065 | 0.146824 | 0.154742 | 0.162809 | 0.171017 | 0.179358 | 0.196403 | 0.213882 | 0.259117 | 0.305978 |
| 16 | 0.135517 | 0.143390 | 0.151426 | 0.159615 | 0.167948 | 0.176414 | 0.193710 | 0.211436 | 0.257241 | 0.304577 |
| 17 | 0.132471 | 0.140457 | 0.148608 | 0.156915 | 0.165367 | 0.173952 | 0.191485 | 0.209440 | 0.255759 | 0.303509 |
| 18 | 0.129843 | 0.137937 | 0.146201 | 0.154621 | 0.163186 | 0.171885 | 0.189639 | 0.207805 | 0.254586 | 0.302692 |
| 19 | 0.127563 | 0.135763 | 0.144134 | 0.152663 | 0.161336 | 0.170142 | 0.188103 | 0.206462 | 0.253656 | 0.302066 |
| 20 | 0.125576 | 0.133879 | 0.142354 | 0.150986 | 0.159761 | 0.168667 | 0.186820 | 0.205357 | 0.252916 | 0.301587 |
| 21 | 0.123838 | 0.132240 | 0.140814 | 0.149545 | 0.158417 | 0.167416 | 0.185746 | 0.204444 | 0.252327 | 0.301219 |
| 22 | 0.122313 | 0.130811 | 0.139479 | 0.148303 | 0.157266 | 0.166353 | 0.184846 | 0.203690 | 0.251858 | 0.300937 |
| 23 | 0.120971 | 0.129560 | 0.138319 | 0.147231 | 0.156278 | 0.165447 | 0.184090 | 0.203065 | 0.251485 | 0.300720 |
| 24 | 0.119787 | 0.128463 | 0.137308 | 0.146303 | 0.155430 | 0.164673 | 0.183454 | 0.202548 | 0.251186 | 0.300554 |
| 25 | 0.118740 | 0.127500 | 0.136426 | 0.145498 | 0.154699 | 0.164013 | 0.182919 | 0.202119 | 0.250948 | 0.300426 |
| 30 | 0.115025 | 0.124144 | 0.133411 | 0.142803 | 0.152300 | 0.161886 | 0.181264 | 0.200846 | 0.250310 | 0.300115 |

TABLE 6.1

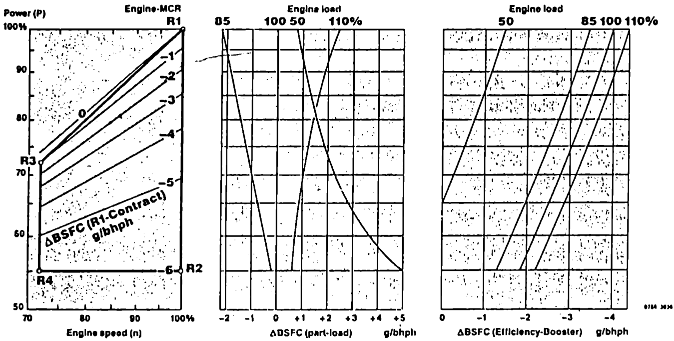

Power (P)
Engine-MCR
R1
100%
90
80
70
60
50
R3
R4
R2
0
-1
-2
-3
-4
-5
-6
ΔBSFC (R1-Contract) g/bhph
70
80
90
100%
Engine speed (n)
Engine load
85
100 50
110%
-2
-1
0
+1
+2
+3
+4
+5
ΔBSFC (part-load)
g/bhph
Engine load
50
85 100 110%
0
-1
-2
-3
-4
ΔBSFC (Efficiency-Booster)
g/bhph

FIGURE 6.2

# GENERAL CONCLUSIONS AND RECOMMENDATIONS

1. Because of the intense competition with the medium speed engine, manufacturers of two stroke slow speed engines are now forced to look for even greater improvements in engine efficiency which is a difficult task as now thermal efficiency values are well over 50 percent.

2. There is still potential for higher cylinder maximum pressures after a period of consolidation at today's already high levels of around 130 bar.

3. There is now little scope for further increases in stroke/bore ratios and turbochanger efficiency, but the most interesting potential for specific fuel consumption reduction is by the turbo-compound system.

4. There may also be gains from better insulation of combustion chambers and the use of ceramics and waste heat recovery systems.

5. The slow speed two-stroke engine is the most well established of all marine prime movers and the development in terms of output and fuel consumption rate over a very short period has been tremendous. This trend is likely to continue such that the two-stroke engine will retain its market lead for some time to come.

6. The aim of all four-stroke medium speed engine manufacturers is particularly to develop their products to compete more effectively with two-stroke engines on operating cost terms while exploiting fully the built-in first cost advantage.

7. The 1980s was one in which the fortunes of the four-stroke medium speed engine improved as the market share for this type has increased slightly and technical developments with some

leading manufacturers have produced significant reductions in specific fuel consumption.

8. Particularly noticeable improvements in medium speed engines were the introduction of more economical long-stroke versions and the application of power turbines in a similar manner to that offered for two-stroke engines.

9. The medium speed engine is economically seen as a very attractive alternative due to its low first cost for a given output.

10. The turbo-compound system offered for two-stroke engines is in itself a positive development but it must have a lower purchase price to be an attractive proposition for the shipowner.

11. The shaft generator (power take-off) is economically a positive development compared with one diesel generator.

12. The trend towards large bore and a very small number of cylinders is in itself a positive development from an economical point of view.

13. The ideal propulsive installation for medium power ships could be:

    One slow speed diesel engine with the lowest number of cylinders and the lowest RPM directly connected to a fixed pitch propeller through a conventional tail shaft avoiding all couplings, reduction gears, etc.

It is hoped that these descriptions will define the subject of choosing main and auxiliary engines for merchant ships. However, the choice of the right engine plant in a correct ship design is still dependent upon the given constraints and personal opinions or preferences. But then the decision maker should at least know what his choice means in economic terms.

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