2. 해설기사

LNG Carrier Propulsion by ME Engines and Reliquefaction

Ole Grøne

1. LNG Carrier Propulsion by ME Engines

LNG carriers represent the last stand for the in all other markets practically extinct marine steam turbines. With efficiencies of only about 30%, versus the diesel engines' more than 50%, and in combined systems even higher, diesel engines are the propulsion system of choice in the marine industry. This reason for the dominance of the diesel engines is clearly demonstrated in Fig. 1, showing the thermal efficiency of the various prime movers. As shown, steam turbine propulsion plants generally have a low efficiency and therefore need far more input energy than modern, fuel efficient diesel engines. With efficiency and CO_2 emission being largely inversely proportional, MAN B&W is proposing alternative propulsion concepts based on low speed diesel engines with electronic control for modern LNG tankers.

HFO burning fuel efficient Low Speed two-stroke diesel engines in single or twin propeller configuration, in combination with the reliquefaction of the Boil Off Gas (BOG), offer economic benefits for those trades where loss, i.e. consumption of cargo, is not accepted and the supply of the full amount of cargo is honoured. However, LNG carriers are expensive ships, and the contractual supply of cargo is usually tied by strict charterparty conditions. Therefore, the market has been hesitant to look at and accept other propulsion systems. Now this has changed. With the market launch of electronically controlled low speed diesels and reliable independent reliquefaction technology, all the traditional reasons not to leave the steam turbine have become invalid. It must also be realised that manning of steam driven commercial vessels will be increasingly difficult because of the phasing out of marine steam turbines.

The purpose of this paper is to demonstrate by comparison that the LNG transport industry can benefit greatly in terms of US$ savings by changing to electronically controlled low speed diesels

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while, at the same time, contributing to a better environment by significantly reducing CO_2 emission. The OVERALL conclusion is that more than US$ 3 million is lost every year through the funnel of every steam driven LNG carrier.

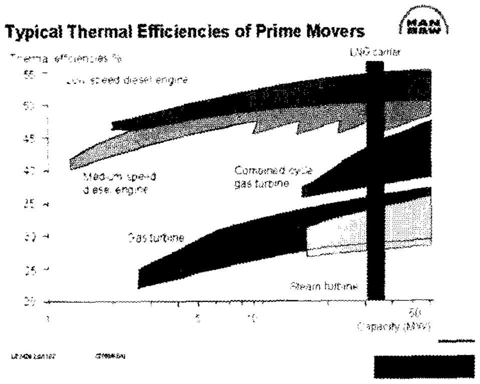


Fig. 1 Typical thermal efficiencies of prime movers.

2. Reliquefaction Technology

While reliquefaction is widely used in gas handling on land, it has been used on board ship so far only on LPG carriers. Recently, the technology for reliquefying LNG on board ship has been matured and commercialised. The present analysis is based on the Moss Reliquefaction, sold worldwide by Hamworthy KSE[1].

The patented system (Moss RS) for reliquefying boil-off gas, establishes a solution for pumping LNG back to the tanks and selling more LNG to the buyers of gas. The boil-off gas reliquefaction concept is based on a closed nitrogen cycle extracting heat from the boil-off gas. Several novel features such as separation and removal of incondensable omponents have resulted in a compact system with low power consumption.

The concept has the following technical merits:

- The nitrogen in the LNG boil-off gas (BOG) is not reliquefied; this results in reduced nitrogen in the tanks during the voyage, better control of tank pressure and lower power requirement for the RS system.
- The system uses only proven components with extensive references from air-separation and peak-shaving plants worldwide.
- The system is prefabricated on skids for easy installation and hook-up.
- The system has automatic capacity control.
- The system can be stopped when the cargo pumps are in operation. This eliminates the need for extra generator capacity.
- During ballast voyage, the cargo tank temperature can be maintained by spraying reliquefied LNG back into the cargo tanks.
- The system must be installed with 100% redundancy.
- No extra personnel are required for operation and maintenance.

The process can be described as follows:

The LNG boil-off is compressed by the low duty (LD) compressor (BOG compressor), and sent directly to the so-called cold box. The cold box in which the boil-off is reliquefied is cooled by a closed refrigeration loop(Brayton cycle). Nitrogen is the working medium. Fig. 2 shows the standard Moss RS reliquefaction system.

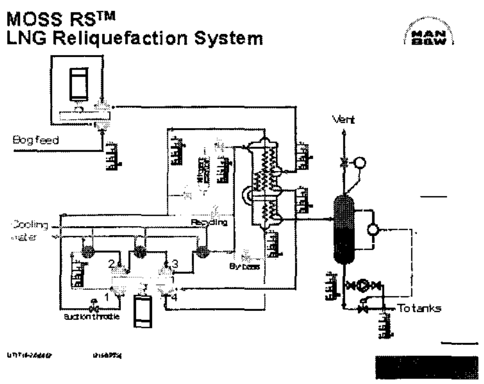


Fig. 2 N_2 compressor/expander.

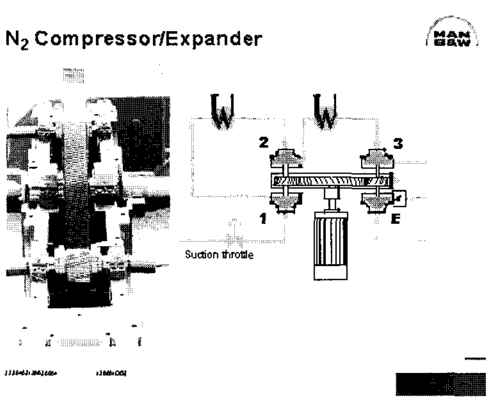


Fig. 3 N_2 compressor/expander.

2.1 Boil-off Cycle

The cargo cycle consists of an LD compressor, a plate-fin cryogenic exchanger, a separator and an LNG return pump. Boil-off is evacuated from the LNG tanks by means of a conventional centrifugal low duty compressor. The vapour is compressed to 4.5 bar and cooled at this pressure to approximately 160°C in a platefin cryogenic heat exchanger. This ensures condensation of hydrocarbons to LNG. The fraction of nitrogen present in the boil-off that cannot be condensed at this condition remains as gas bubbles in the LNG. Phase separation takes place in the liquid separator. From the separator, the LNG is dumped back to the storage tanks, while the nitrogen-rich gas phase is discharged (to atmosphere or burnt in an oxidizer).

2.2 Nitrogen Cycle

The cryogenic temperature inside the cold box is produced by means of a nitrogen compression-expansion cycle, shown in Fig. 3. Nitrogen gas at a pressure of 13.5 bar is compressed to 57 bar in a 3-stage centrifugal compressor. The gas is cooled by water (seawater or indirect) after each stage. After the last cooler, the gas is led to the warm part of the cryogenic heat exchanger where it is precooled to about 110°C and then expanded to a pressure of 14.5 bar in the expander. The gas leaves the expander at about 163°C and is then introduced into the cold part of the cryogenic heat exchanger where it cools and reliquefies the boil-off gas to LNG. The nitrogen then continues through the warm part of the cryogenic heat exchanger before it is returned to the suction side of the 3-stage compressor. The N_2-compressor/expander unit is a three-stage integrated gear centrifugal compressor with one expander stage. The unit has a gear with 4 pinions where each of the 4 wheels is coupled to a separate pinion. The result is that the expander work goes directly into the gearbox and relieves the electric motor.

The advantages of this solution are:
- More compact design

- Reduced cost
- Improved control of the refrigeration
- Reduced power consumption.

2.3 Control Systems

Generally, the temperature in the nitrogen loop decides the quantity of N2 in the coolant circuit. Increasing or decreasing the amount of nitrogen in the loop changes the cooling capacity. The amount is changed by injecting or withdrawing nitrogen from the receiver. If the cooling capacity is too high, the inlet expander temperature will decrease. The control valve to the receiver at the compressors discharge will open to withdraw the nitrogen from the main loop. Correspondingly, if the cooling capacity is too low, the inlet expander temperature will increase. The control valve from the receiver to the compressor suction side will open to inject nitrogen into the main loop. The relationship between cooling capacity and pressure changes is based on the fact that a turbo compressor is a constant volume flow machine. When the suction pressure is changing, the mass flow is changing and, correspondingly, the cooling capacity. The pressure ratio for the compressor is constant and independent of the suction pressure. Even if the cooling capacity is reduced, the outlet expander temperature will be nearly the same.

The BOG cycle is an independent loop. The cargo tank pressure is kept approximately constant by varying the mass flow through the compressor. The boil-off compressor will be a two-stage centrifugal compressor with diffuser guide vanes (DGV) for controlling the capacity. There is DGV on both stages, and they work in parallel, controlled by the same signal.

2.4 Redundancy

- Redundancy is required by the International Classification Society Association (IACS), as discussed later. The requirement is fulfilled if one of the following options is installed:
- Thermal oxidizer or flare system capable of burning the maximum boil-off rate.
- Two 100% reliquefaction plant with one cold box, comprising the following:

Two BOG-compressor units (two-stage centrifugal compressor)
Two N_2-compressor/expander units(three-stage integrated gear centrifugal compressor with one expander stage)
One cold box
One LNG phase separator
One LNG forced return pump
Auxiliary systems

Which one to operate of the two BOG-compressor units and N_2-compressor/expander units can be freely chosen by operating the applicable valves. Changeover of equipment is done manually, and must be done only when the machinery is shut down. Simultaneous parallel operation of the equipment will not be possible.

As the reliability of todays steam turbine driven LNG carriers is considered high, an alternative system must not deteriorate the availability of the LNG carriers. The reliquefaction system therefore only uses proven components first class, high quality with extensive references. The low-duty compressors in the RS system are the same as used on all LNG carriers today. The refrigeration cycle is in operation on the LNG carrier S/S LNG Jamal, and the 3-stage compressor with expander is operating on FPSOs and in onshore process plants.

The proposed cold box (plate fin heat exchanger) is widely used in onshore cryogenic installations. An availability analysis concludes 99.98 % availability, which is at the same level or better than ship machinery in general.

3. Demonstration Plants

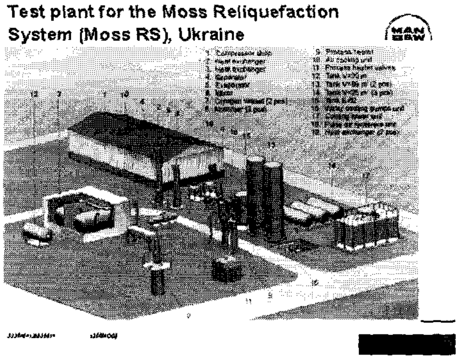


Fig. 4 Test plant for the Moss Reliquefaction System (Moss RS), Ukraine.

One test plant for the Moss Reliquefaction System (Moss RS) is located at the Ukrainian company Sumy Frunze located in Sumy, Ukraine, see Fig. 4. The Moss RS patent holder, Moss Maritime of Norway, is responsible for this plant. It comprises a pre-treatment plant and a liquefaction plant. Gas is supplied from the local gas grid, and a large pre-treatment plant is used for delivering gas over a wide specification range.

The purpose of the test plant is to verify all technical aspects of the reliquefaction concept and, in particular:

- Demonstrate the nitrogen Brayton cooling cycle
- Test the operating procedures
- Control system development

For another demo plant, Hamworthy KSE was awarded the contract by the Norwegian gas distribution company Gasnor in October 2001. The LNG production capacity is 60 ton/day (2500kg/hr), which corresponds to the boil-off rate on traditional size LNG carriers. This plant uses the same type of cooling cycle (Brayton) and control principles as the reliquefaction system for LNG carriers. The same 3-stage N_2 compressor with expander and the same type of cold box that will be used on LNG C are also installed. However, as the plant is onshore and the feed gas comes from the gas pipelines from the offshore fields in the North Sea, this plant needs additional equipment and systems. The plant shown in Fig. 5 thus consists of the following basic parts:

- Natural gas dehydration unit
- Natural gas CO_2 removal unit
- Nitrogen cooling circuit (same as proposed for LNG carriers)
- Main liquefier (cold box) with LNG receiver (similar type as proposed for LNG carriers)

- LNG storage tank and truck loading station.

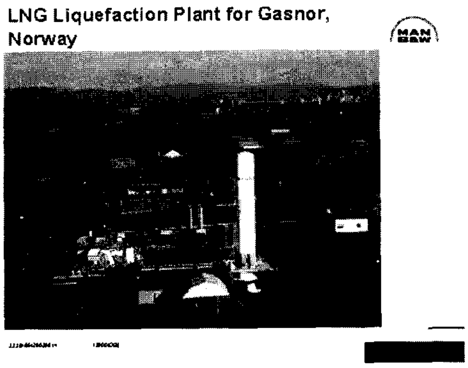


Fig. 5 LNG liquefaction plant for Gasnor, Norway.

Natural gas from the high-pressure feed line is reduced in pressure down to 120 barg and dehydrated down to a H_2O content of 1 ppm. The dry feed gas is further reduced in pressure down to 52 barg prior to removal of CO_2 down to a level of 50 ppm.

Liquefaction is accomplished at about 50 bar abs against cold nitrogen gas, which is cooled in a single-expansion cycle with three compressor stages and one expander stage.

The heaviest gas fractions are separated out and the gas liquefies in the lower-mid section of the cold box. The liquid is subcooled in the bottom section and led to the LNG flash drum via a valve, where the pressure is reduced to 0.5 barg, and the LNG is sent to a storage tank. The system is equipped to give a variable production rate by adjusting the mass flow of nitrogen. The first LNG was produced on this plant on March 15, 2003.

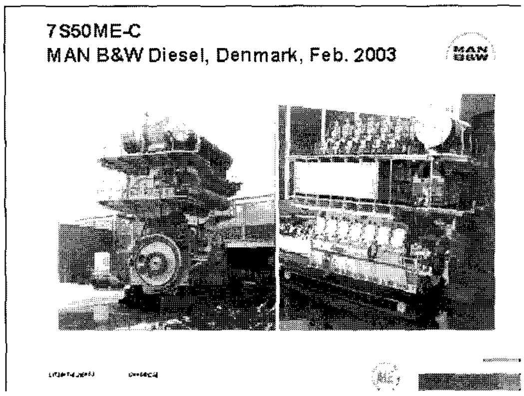
4. Diesel Engine Technology

MAN B&W offers a full programme of marine diesel engines for every conceivable application. The low speed engine programme is developed in Denmark and manufactured by a family of licensees at major shipbuilding centres of the world. Single unit powers range from 2,000 hp to well over 100,000 hp, all for direct coupled installation at propeller speeds from 250 rpm down to 60 rpm for the largest propellers. The power requirement for an LNG carrier calls for some 40,000 hp, typically two off 60 or 70 cm bore units.

MAN B&W low speed engines hold a world-wide market share of about 65 % in their segment.

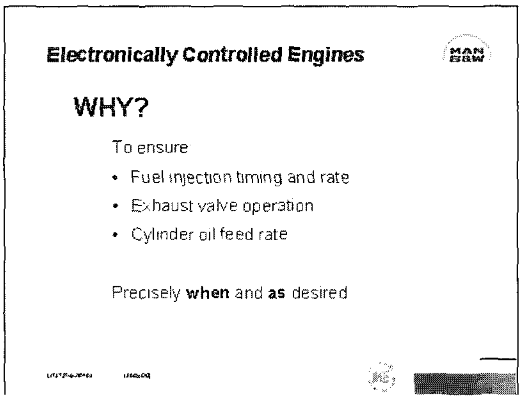
The introduction of electronically controlled camshaft-less low speed diesel engines[2] is now gaining momentum. MAN B&W has developed and refined this new technology in its ME-range of engines by combining traditional, proven technologies with enhanced electronic control so as to design engines which, while being both production-friendly and operationally easy to handle, yet will provide all benefits to the owner and operator of contemporary and future software achievements. Fig. 6 shows demonstration took place with the delivery of a 6S70ME-C engine at HSD in Korea in July 2003. The ME engines have the same speed and power as their MC counterparts.

Camshaft-controlled diesel engines have been the state of the art ever since the birth of reciprocating machinery and



7S50ME-C
MAN B&W Diesel, Denmark, Feb. 2003

Fig. 6 7S50ME-C, MAN B&W Diesel, Denmark, February 2003.



Electronically Controlled Engines

WHY?

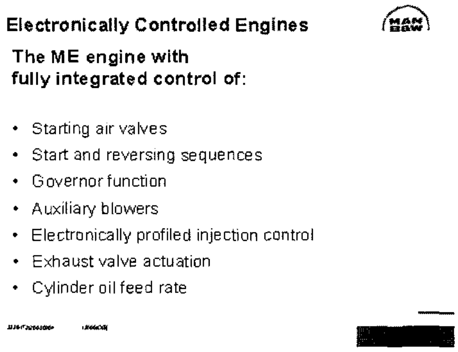
To ensure

- Fuel injection timing and rate
- Exhaust valve operation
- Cylinder oil feed rate

Precisely **when** and **as** desired

Fig. 7 Reasons to develop electronically controlled engines.

have been refined and developed ever since. However, a mechanical cam is fixed once made and, in spite of various mechanical and hydraulic add-on devices like VIT, etc., timing control possibilities are limited with mechanical cams. Not least fuel injection pressure control and variation over the load range have limitations with a cam-controlled engine. Therefore, the main purpose of changing to electronic control is to ensure fuel injection timing and rate, as well as the exhaust valve timing and operation, exactly when and as desired, see Fig. 7.



Electronically Controlled Engines

The ME engine with fully integrated control of:

- Starting air valves
- Start and reversing sequences
- Governor function
- Auxiliary blowers
- Electronically profiled injection control
- Exhaust valve actuation
- Cylinder oil feed rate

Fig. 8 Electronically controlled engines with fully intergrated electronic control.

Especially with respect to the fuel injection rate, the control system has been so de-signed that it is possible to maintain a rather high injection pressure also at low load, without the limitation of the camshaft-controlled engine, where this would result in too high pressure at high load. The cam angle, inclination and length are electronically variable. In addition, the ME engine features electronic control of the cylinder lube oil feed, by having our proprietary Alpha Lubricators integrated in the system. With the Alpha Lubrication system, about 0.3 g/bhph cyl. oil can be saved, compared with engines with mechanical lubricators.

The electronic control of the engine fuel injection and exhaust valves improves lowload operation, engine acceleration, and gives better engine balance and load control, leading to longer times between overhauls, also by implementation of enhanced diagnostics systems. It will give lower fuel consumption, lower cylinder oil consumption and, not least, better emission characteristics, particularly with regard to visible smoke and NO_x.

For the ME engines, the electronic control system has been made complete. Hence, the ME engine features fully integrated control of all functions like the governor, start and reversing, fuel, exhaust and starting valves, as well as cylinder oil feeding, as summarised in Fig. 8.

5. Elements of the ME-C Engine

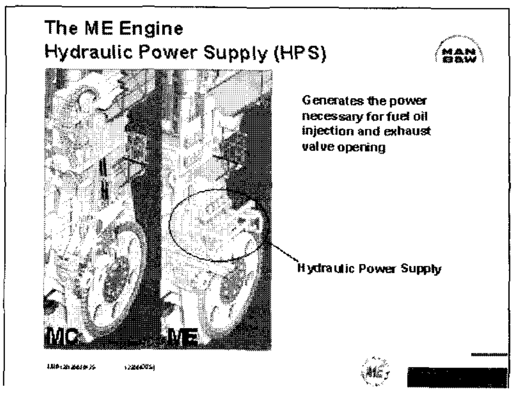
The mechanical difference between an MC-C engine and its electronically controlled counterpart, the ME-C engine, constitutes a number of mechanical parts that are made redundant and replaced by hydraulic and mechatronic parts with enhanced functions.

The following parts are omitted:

Chain drive
Chain wheel frame
Chain box on frame box
Camshaft with cams
Roller guides for fuel pumps and exhaust valves
Fuel injection pumps
Exhaust valve actuators
Starting air distributor
Governor
Regulating shaft
Mechanical cylinder lubricator
Local control stand

The above-mentioned parts are replaced by:

Hydraulic Power Supply (HPS)
Hydraulic Cylinder Units (HCU)
Engine Control System (ECS), controlling the following:
Electronically Profiled Injection (EPIC)
Exhaust valve actuation
Fuel oil pressure boosters
Start and reversing sequences
Governor function
Starting air valves
Auxiliary blowers
Crankshaft position sensing system
Electronically controlled Alpha Lubricator
Local Operating Panel (LOP)



The ME Engine
Hydraulic Power Supply (HPS)

Generates the power necessary for fuel oil injection and exhaust valve opening

Hydraulic Power Supply

Fig. 9 Hydraulic Power Supply (HPS).

Fig. 9 shows how the necessary power for fuel injection and exhaust valve operation previously provided via the chain drive is now provided from a Hydraulic Power Supply (HPS) unit located at the front of the engine at bedplate level. The main components of the Hydraulic Power Supply unit are the following:

Self cleaning filter with 10-micron filter mesh Redundancy filter with 25-micron filter mesh Start up pumps:
High-pressure pumps with supply pressure of 175 bar

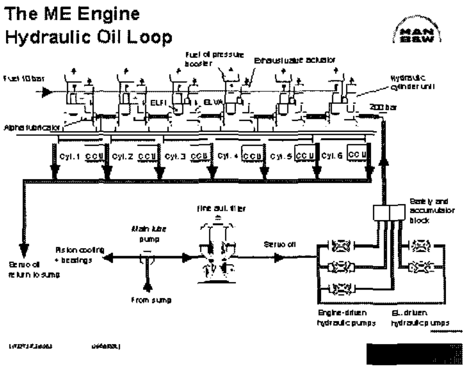


Fig. 10 ME engines, hydraulic oil loop

Low-pressure pumps for filling the exhaust valve push rod with supply pressure of 4 bar

Engine driven axial piston pumps supplying high pressure oil to the Hydraulic Cylinder Unit with oil pressures up to 250 bar

Before engine start, the hydraulic oil pressure used in the mechanical/ hydraulic system (for controlling the actuators) is generated by electrically driven start-up pumps. After start, the engine driven pump will take over the supply.

The engine driven pumps are gear or chain driven, depending on engine size. If so preferred, all pumps can also be electrically driven. The hydraulic pumps are axial piston pumps with flow controlled by the integrated control system. There are three engine driven pumps, but actually only two are needed for operation. Second-order moment compensators, where needed, can be integrated into the pump drive. Alternatively, electrically driven compensators can be used. If so preferred, the entire hydraulic oil system can be made as a separate, independent system.

Fig. 10 shows the entire hydraulic oil loop with the hydraulic power supply system and, as can be seen, the generated servo oil is fed via doublewalled piping to the Hydraulic Cylinder Units of which there is one per cylinder, mounted on a common base plate on the top gallery level on the engine. In this figure, also the important electronic control valves, i.e. the ELFI (a proportional ELectronic Fuel Injection control valve) and the ELVA (an on-off Electronic exhaust Valve Actuator) are shown.

The Hydraulic Cylinder Unit furthermore comprises a hydraulic oil distribution block with pressure accumulators, the exhaust valve actuator with ELVA, and a fuel oil pressure booster with ELFI, raising the fuel oil supply pressure during injection from the 10-bar supply pressure to the specified loaddependent injection pressure of 600-1000 bar. Permanent high pressure with preheated fuel oil on top of the engine is thereby avoided, without losing any advantage of highpressure injection.

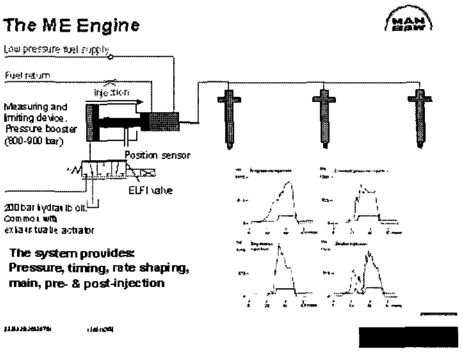


Fig. 11 Fuel injection system for ME engines.

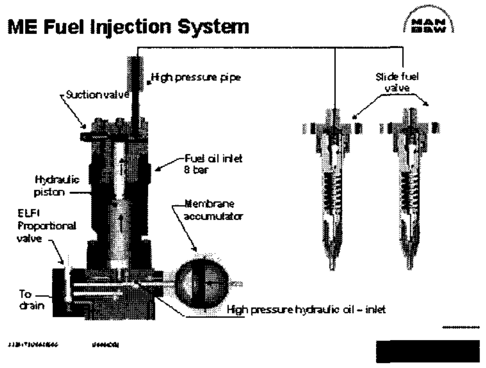


Fig. 12 Fuel oil pressure booster and control valve.

Fig. 11 shows the per cylinder fuel oil injection system, and Fig. 12 shows the individual components of the fuel oil pressure booster. As will appear, the fuel oil pressure booster is mechanically much simpler than the traditional fuel pump with roller, roller guide, VIT and cut-off wedges. About 15,000 hours have been logged on the first ME engine in service, and there has been virtually nothing to report. The fuel oil pressure booster is less exposed to wear than a traditional fuel oil pump and, with its significantly larger sealing length (compared with the conventional Bosch-type fuel pumps), a much longer lifetime can be expected.

Fig. 13 shows the actuator for the exhaust valve which responds to the electronic actuator signal from the engine control system.

Another system that benefits from mechanical simplification by being electronically rather than mechanically controlled on the ME engine is the starting air system, Fig. 14. The mechanical starting air distributor is past history.

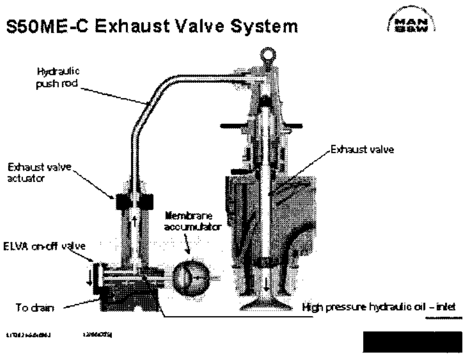


Fig. 13 Exhaust valve actuator and control valve.

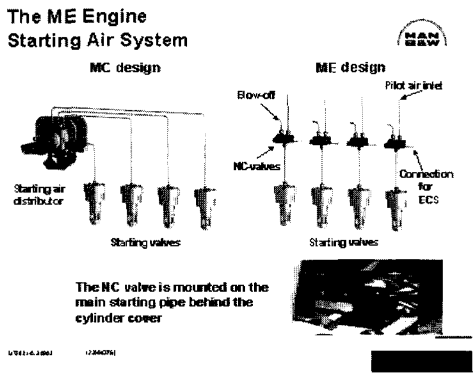


Fig. 14 ME engine starting air system.

The Alpha Lubricator system for cylinder oil feed rate control, already with more than 200 sets sold, benefits in the ME engine version by using the 200-bar servo oil pressure as the driving force rather than a separate pump station used in the stand-alone systems. On ME engines, the Alpha Lubricator is mounted on the hydraulic cylinder units, as shown in Fig. 10. The ME execution, therefore, as illustrated in Fig. 15, separates the cylinder oil from the servo oil.

The ME engine control system, simplified in Fig. 16, is designed with the principle that no single failure of anything

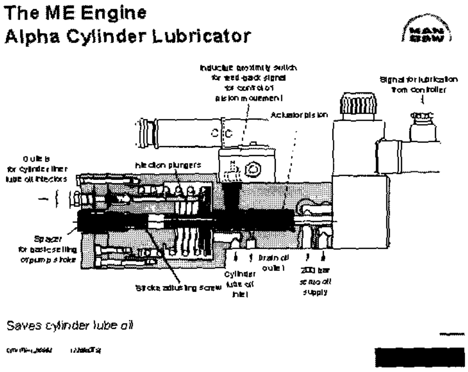


Fig. 15 Alpha cylinder lubricator for ME engine.

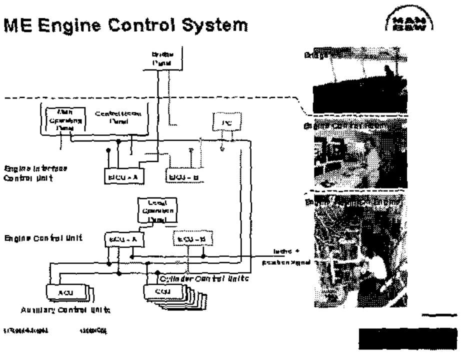


Fig. 16 ME engine control system.

shall make the engine inoperative. Therefore, all essential computers are with a hot stand-by.

All the computers in the system, referred to as Engine Interface Control Unit, Engine Control Units, Cylinder Control Units and Auxiliary Control Units, are of exactly the same design and can replace each other, in that they will adapt to the desired functionality of the particular location once installed, including if replaced by a spare. The computer, often referred to as a Multi-Purpose Controller, is a proprietary in-house development of MAN B&W Diesel. Thus, we can ensure spare part deliveries over the engines lifetime. The Local Operating Panel, incl. Cylinder Control and Auxiliary Control Units, is mounted on the middle gallery of the 7S50ME-C made in Denmark. The Control Units can, of course, also be located elsewhere.

As to installation aspects, an ME-C engine and an MC-C engine are, apart from the cabling of the control network, practically the same for a shipyard, as detailed below:

Overhaul height: same
Engine seating: same
Engine outline: modifications with no influence for yard
Engine weight: slightly reduced
Engine pipe connection: back flush from filter on engine added, other connections are unchanged
Gallery outline: slight modifications
Top bracing exhaust side: same
Capacity of auxiliary machinery: same
Lubricating oil system: slightly modified
Specification and installation of governor omitted
Other systems: same
Cabling: cables added for communication and network

6. Features of the ME-C Engine

As mentioned, the purpose of making electronic engines is focused around the virtues related to ensuring fuel injection and rate, as well as exhaust valve timing exactly when and as desired.

With respect to the exhaust valve movement, this means changing the 'cam length', as illustrated in Fig. 17, by

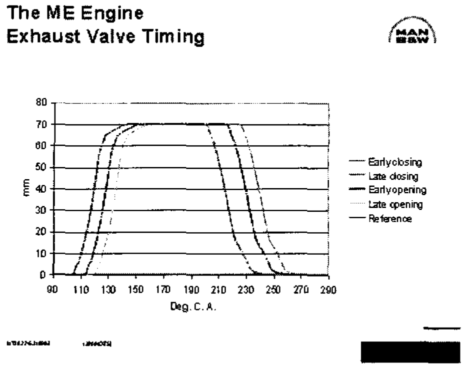


Fig. 17 Exhaust valve timing.

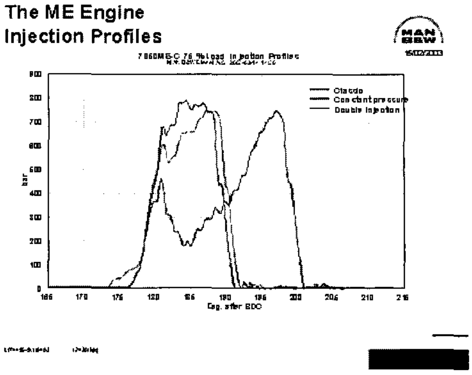


Fig. 18 Injection profiles.

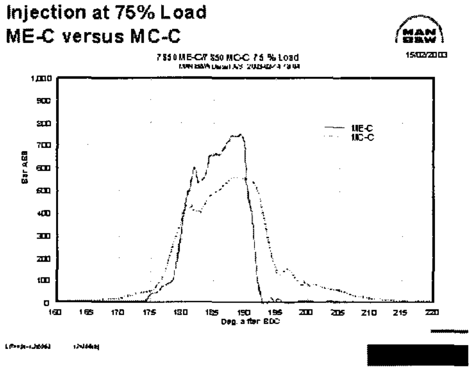


Fig. 19 Injection at 75% load, ME-C versus MC-C.

simply changing the point in time of activating the ELVA valve. This can be used to control the energy to the turbocharger, both during steady and transient load conditions. Smoke-free acceleration is a natural benefit apart from SFOC optimisation at any load.

Thanks to the multitude of possibilities with the ELFI, the proportional valve controlling the servo oil pressure to the fuel oil pressure booster, not only the fuel oil 'cam length', but also the 'cam inclination and angle' and even the number of activations per stroke can be varied for the fuel oil injection.

Fig. 18 illustrates different profiles demonstrated during testing of the 7S50ME-C. The double injection profile is specially tailored to provide a significant reduction of NO_x emissions.

Fig. 19 shows the selected injection rate on that engine at 75% load, compared with what it would have been with a fixed cam. The result is a more intensive heat release. A better heat release mirrors a better fuel consumption, also because the p_{max} can be kept high at low loads. At the low end of the load scale, the possibility for controlling the timing and rate of injection gives the possibility to demonstrate stable running down to 10% of MCR-rpm, i.e. 13 rpm against a water brake only. This could be even more stable against a propeller eliminating the need for stop-and-go operation through channels and canals and making ME engines particularly suitable for vessels with greatly varying load profiles.

General performance curves for the ME-C and MC-C engines are shown in Fig. 20. The lower part load fuel consumption is achieved by raising the p_{max} over the whole load range. In order

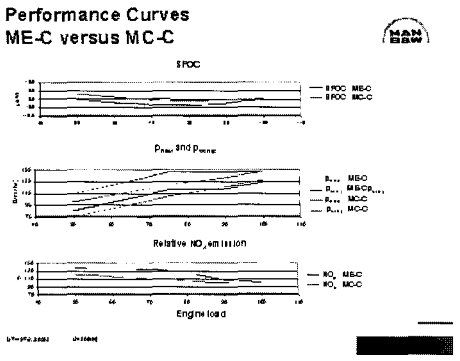


Fig. 20 Performance curves, ME-C versus MC-C.

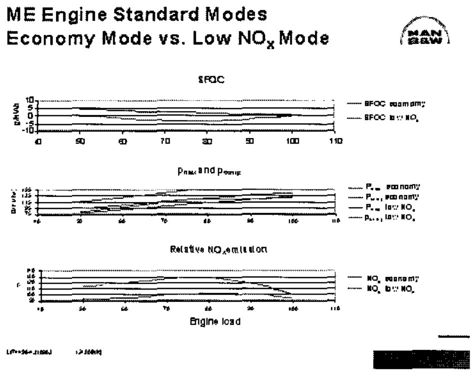


Fig. 21 Performance curves, economy versus low NO_x.

to avoid too much difference between p_{max} and p_{comp}, this pressure is also raised by exhaust valve timing control.

As also illustrated, the lower SFOC comes at a price in that the NO_x increases. For this reason, the first two modes to be incorporated in the control system of the ME engine, as standard, are the 'fuel economy mode' and the 'low-NO_x' mode. Fig. 21 illustrates the coagency between SFOC, NO_x, and pmax/pcomp for the two modes.

It goes without saying that an ME-C engine will comply with IMOs NO_x cap also in the fuel economy mode.

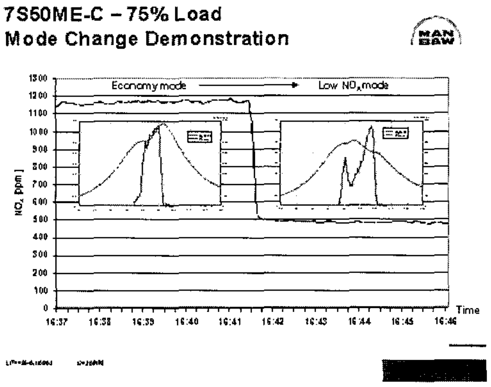


Fig. 22 Mode change demonstration.

The low-NO_x mode is intended for areas where lower than IMO NO_x limits do or will apply. Changing from one mode to the other is a matter of seconds only and, of course, is done while running, as illustrated in Fig. 22.

7. Advantages

The advantages of the ME-C range of engines are quite comprehensive, as seen below:

- Lower SFOC and better performance parameters thanks to electronically controlled variable timing of fuel injection and exhaust valves at any load
- Appropriate fuel injection pressure and rate shaping at any load
- Improved emission characteristics, with lower NO_x and smokeless operation
- Easy change of operating mode during operation
- Simplicity of mechanical system with well-proven traditional fuel injection technology familiar to any crew

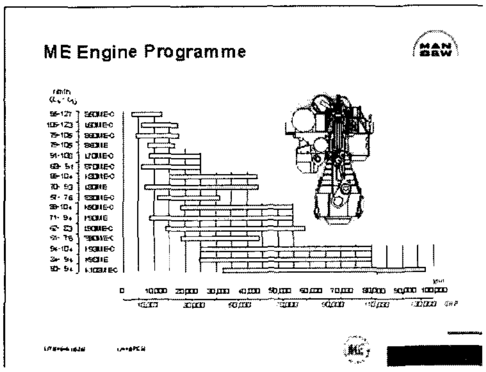


Fig. 23 ME engine program.

- Control system with more precise timing, giving better engine balance with equalized thermal load in and between cylinders
- System comprising performance, adequate monitoring and diagnostics of engine for longer time between overhauls
- Lower rpm possible for manoeuvring
- Better acceleration, astern and crash stop performance
- Integrated Alpha Cylinder Lubricators
- Up-gradable to software development over the lifetime of the engine

It is a natural consequence of the above that many more features and operating modes are feasible with our fully integrated control system and, as such, will be retrofittable and eventually offered to owners of ME-C engines.

Against this background, the ME-C engine programme, is shown in Fig. 23. The reference list now comprises nearly 40 engines of different sizes.

8. Propulsion Redundancy and Gas Handling

LNG carriers, like oil tankers, are not permitted to immobilize their propulsion machinery while in port and port areas. Hence, redundancy is required.

For the steam ship, redundancy is considered fulfilled by having two boilers, whereas no redundancy is required for the single steam turbine, propeller shaft and propeller. The two boilers will have a steam-dumping condenser to be used for surplus steam when the turbine is not operating.

For diesel engines, which require more maintenance on a routine basis than steam turbines, either a multiengine configuration or an alternative power supply possibility for a single engine configuration is required.

Immobilisation for carrying out maintenance work on a single configured two-stroke diesel engine has so far been considered an obstacle on LNG carriers.

Shuttle tankers in the North Sea were originally equipped with twin low speed engines and twin propellers. This ensured that approximately half of the propulsion power is always available, and that one of the diesel engines can be maintained without immobilising the vessel or compromising safety. However, now single engine ships are widely used for this trade, as for chemical carriers and LPG vessels, a virtual proof of the inherent 'self redundancy' of such engines.

The International Association of (marine) Classification Societies (IACS) redundancy considerations for a reliquefaction plant for LNG carriers are as stipulated in Fig. 24.

Redundant low speed engine propulsion

concepts, as outlined above, ensure that sufficient power is available for safe navigation and, for the twin engine concept with completely separated engine rooms, even an additional margin towards any damage is obtained.

For LNG carriers, a twin engine configuration is proposed to alleviate any possible doubt on reliability and redundancy.

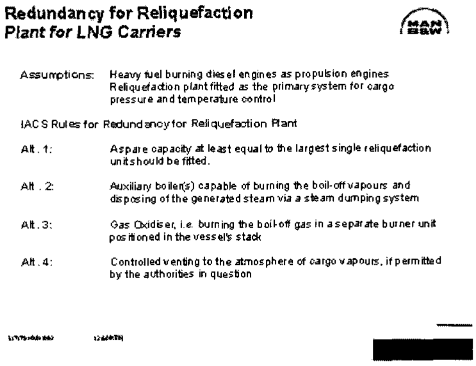
The average lifetime of commercial vessels is 25 years, by which time the vessels are usually scrapped for reasons of economy. Diesel engines could operate for decades beyond, as all wear parts are replaceable. Long living diesels are seen mainly in power plants. The low speed diesel engine has, though, a considerably longer lifetime, which makes it relevant for LNG carriers with an average lifetime of up to 40 years.

The latest series of electronically controlled engines, the ME series, are particularly suitable for the trade discussed, as the control system software can be updated routinely.

Maintenance requirements for diesels are predictable, and parts supplies over the engine lifetime are guaranteed by the manufacturer and/or designer.

Vibration levels are fully predictable and controllable. Hence, for 40 years use in LNG carriers, diesels are fully adequate.

Furthermore, the segregation of the gas cargo and heavy fuel for propulsion ensured with reliquefaction means that handling of gas in the engine room and surrounding areas is avoided, since the reliquefaction plant will be installed on deck.



Redundancy for Reliquefaction
Plant for LNG Carriers

Assumptions: Heavy fuel burning diesel engines as propulsion engines
Reliquefaction plant fitted as the primary system for cargo pressure and temperature control

IACS Rules for Redundancy for Reliquefaction Plant

Alt. 1: A spare capacity at least equal to the largest single reliquefaction unit should be fitted.

Alt. 2: Auxiliary boiler(s) capable of burning the boil-off vapours and disposing of the generated steam via a steam dumping system

Alt. 3: Gas Oxidiser, i.e. burning the boil-off gas in a separate burner unit positioned in the vessel's stack

Alt. 4: Controlled venting to the atmosphere of cargo vapours, if permitted by the authorities in question

Fig. 24 Redundancy for LNG carriers.

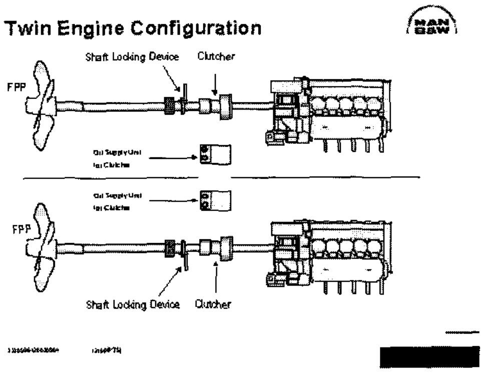


Fig. 25 Twin-engine configuration.

Based on the technology described in the foregoing, the machinery to replace the steam turbine and boilers is therefore the following:

- 2 x approx. 20,000 hp low speed fuel burning ME-type diesel engines

The twin-engine configuration is shown in Fig. 25.

The bridge and engine room control system shall be able to handle operation with both one (emergency) or two (normal) engines. The bridge and engine room control system shall, in the case of operation on two engines, be able to handle both individual control and simultaneous control of the engines. Simultaneous control consists of equality in power distribution, order for reversing, start of engines and stop of engines. The control system shall, in case of failure on one of the engines, be able to ensure continuous operation with only one engine without jeopardizing manoeuvrability or safety of the ship or engines.

Typical propulsion power requirements for LNG carriers of different sizes are shown in Fig. 26.

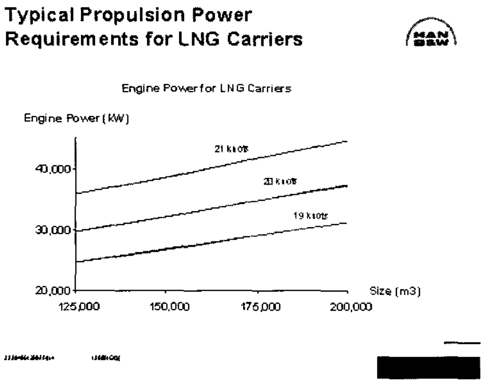
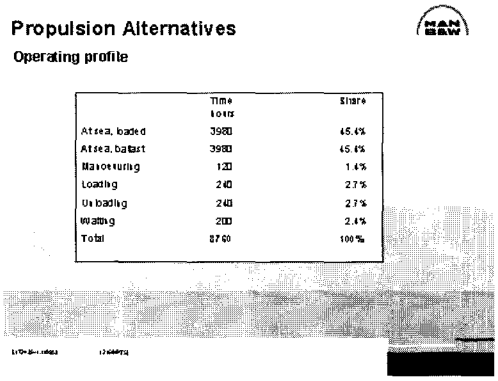


Fig. 26 Typical propulsion power requirments for LNG carriers.



	Time hours	Share
At sea, loaded	3980	45.4%
At sea, ballast	3980	45.4%
Manoeuvring	120	1.4%
Loading	240	2.7%
Unloading	240	2.7%
Waiting	200	2.4%
Total	8760	100%

Fig. 27 Voyage profile.

In the event of an emergency situation, with one engine out of service, the actual propeller curve for the working engine will be conceived as 'heavy' up to 5-10%.

In the case of FP propellers, it is presumed that, in most cases, the shaft is declutched from the engines and the propeller windmilling, while the engine can be repaired, alternatively that a shaft brake is applied.

In the case of CP propellers, it is presumed that the propeller is at zero pitch and the shaft brake is active. If engine repair or overhaul is to take place during sailing, declutching is necessary. In either case, the working engine will have to accept the 'heavy propeller', i. e. higher torque, which basically calls for a changed engine timing. With the ME engine concept, this can be done by push button only, activating "single engine running mode". This can be pre-programmed into the software just as the so-called "economy mode" and "low NO_x emission mode". Hence, the operating engine in case of non-availability of the other engine will be readily optimised for the purpose, and full mobility of the vessel ensured. As per calculation, a speed of 75% of the design speed of the vessel can be obtained with a single engine in operation.

9. Economical Evaluation

The operating costs and the additional in-come from the transport and sale of LNG for a 150,000 m^3 LNG carrier is analysed, including an analysis of the fuel oil, lubricating oil and maintenance costs for both propulsion and electricity production under various operating conditions, comparing steam turbines with the proposed configuration.

The analysis is based on state-of-the-art insulation of tanks, and thus BOG rate, and a traditional service speed of the vessel.

An evaluation of the operating costs and the additional income from selling reliquefied LNG shows that substantial economic benefits can be obtained. The outcome of the evaluation will depend on the actual project, i.e. voyage profile, service speed, economic factors, price of HFO and LNG, as well as of the Boil-Off rate.

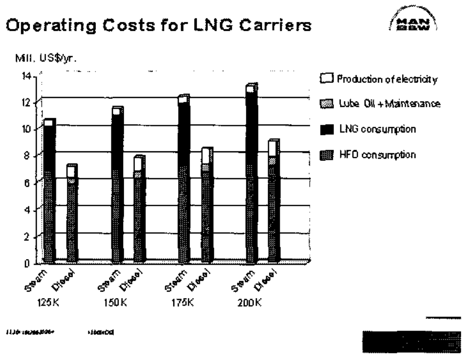


Fig. 28 Operating costs for LNG carriers.

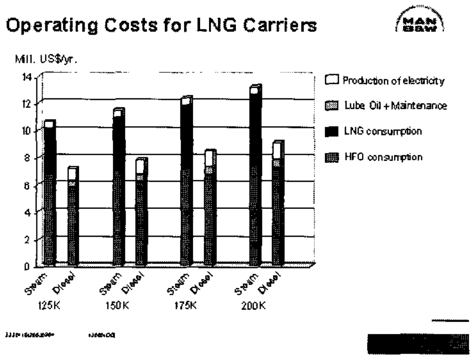


Fig. 29 Sensitivity to LNG price.

The efficiency of the steam propulsion system (i.e. combined efficiency of boiler, steam turbine and reduction gear) will vary depending on steam pressure, condensation temperature and boiler cleaning intervals. In any case, diesel engine propulsion offers significant economic savings for the operator.

The operating costs, calculated based on the Basic Data and a typical voyage profile as shown in Fig. 27, are indicated in the tables in the Appendix. Table 4 of this Appendix shows the final summary of the operating costs and indicates the potential for additional income.

In Fig. 28 the result of the summary in Table 4 is visualized. With the diesels plus reliquefaction plant, the BOG is replaced by HFO as the energy source and, of course, the same efficiency advantage of the diesels prevails. The additional sale of reliquefied BOG brings the large saving. This saving depends, of course, also on the price of LNG. Fig. 29 shows the optimum solution as a function of the LNG price and the Boil-off rate.

9.1 Investment cost

A system comprising the traditional steam plant is estimated to cost around US$ 20 million. The directcoupled diesel solution requires lower investment cost than the steam plant, as far as equipment is concerned. To this come installation costs, which are not considered.

Most shipyards (all) that today build LNG carriers have much more experience of installing diesel engines than steam turbines and boilers, which adds to the

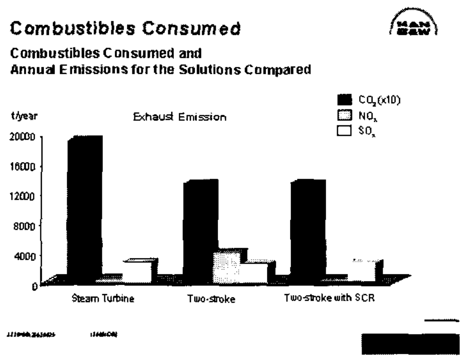


Fig. 30 Combustibles consumed and annual emissions for the solutions compared.

advantage of diesels. However, the twin-screw solution proposed does represent added cost on the hull side at some shipyards. This could be up to US$ 5 million, but the total cost is still comparable to that of the steam plant.

Against this background, the study does not include any payback calculation of the alternatives. MAN B&W will be pleased to recalculate the above analysis with any combination of data that any party may consider relevant for their project, including ships of different size than the 150,000 m3 carrier proposed.

10. Exhaust Emissions

The relative energy consumption for the two concepts is shown in Fig. 30, which also shows the expected annual exhaust emissions. The CO_2 emission is obviously largest for the steam plant due to its low efficiency. The SO_x from the fuel sulphur is about the same, as the same amount of fuel is used. This can be reduced by using fuel with low sulphur content. The proposed diesel solution complies with the IMO limits for NO_x emissions and is therefore without any NO_x abatement. However, the NO_x can, if needed, be reduced to any level by Selective Catalytic Reduction.

11. Conclusion

The benefit of diesel engine propulsion of LNG carriers is calculated to be up to approx. US$ 3.0 million per vessel per year. Especially the LNG selling price has a positive impact on the advantage of diesel engine propulsion. The benefit gained in operating costs and the additional income from the sale of LNG by diesel engine propulsion and reliquefaction will, in all cases, be sufficient to justify even large differences in investment costs, if such are called for at all.

Basically, diesel propulsion offers a CO_2 emission reduction of about 30% compared to the steam plant.

References

[1] Peter Skjoldager, Tore Lunde & Eirik Melaaen: *"Two-stroke Diesel Engines and Reliquefaction Systems for LNG Carriers"*, Motorship Conference, Hamburg, 2003.

[2] Ole Grøne: *"ME engines the New Generation of Diesel engines"*, Motorship Conference, Hamburg, 2003.

APPENDIX

Two-stroke Diesel Engines and
Reliquefaction Systems for LNG Carriers

Power Consumption		
Options Producers of propulsion power Producers of electrical power	Steam Turbine Turbo Generator	Two-Stroke Diesel Engine with Reliquefaction System
Engine power for propulsion		
Loaded conditions	29,808 kW	29,212 kW
Ballast conditions	29,808 kW	29,212 kW
Electrical power consumption		
Loaded conditions		
Electrical power consumption	1500 kWe	4743 kWe
Engine power	1563 kW	4941 kW
Ballast conditions		
Electrical power consumption	1500 kWe	3122 kWe
Engine power	1563 kW	3252 kW

Operating Costs at Loaded Conditions		
Options	Steam Turbine Turbo Generator	Two-Stroke Diesel Engine with Reliquefaction System
Main Engine(s)		
Delivered power (total)	29,808 kW	29,212 kW
Required energy	369.1 GJ/h	210.8 GJ/h
Available energy in BOG	176.3 GJ/h	0.0 GJ/h
Extra energy needed	192.9 GJ/h	210.8 GJ/h
HFO Consumption	4.8 t/h	5.3 t/h
LNG Consumption	3.5 t/h	0.0 t/h
Fuel oil costs	723.2 US$/h	790.5 US$/h
Cylinder oil costs	0.0 US$/h	35.1 US$/h
System oil costs	0.0 US$/h	4.7 US$/h
Maintenance costs	0.0 US$/h	29.2 US$/h
Auxiliary engines		
Delivered power	1563 kW	4941 kW
HFO consumption	0.5 t/h	1.0 t/h
Fuel oil costs	72.6 US$/h	150.3 US$/h
System oil costs	0.0 US$/h	3.5 US$/h
Maintenance costs	0.0 US$/h	12.4 US$/h
Operating costs per hour	795.8 US$/h	1025.6 US$/h

Operating Costs at Ballast Conditions		
Options	Steam Turbine Turbo Generator	Two-Stroke Diesel Engine with Reliquefaction System
Main Engine(s)		
Delivered power	29,808 kW	29,212 kW
Required energy	369.1 GJ/h	210.8 GJ/h
Available energy in boil-off gas	88.1 GJ/h	0.0 GJ/h
Extra energy needed	281.0 GJ/h	210.8 GJ/h
HFO Consumption	7.0 t/h	5.3 t/h
LNG Consumption	1.8 t/h	0.0 t/h
Fuel oil costs	1053.7 US$/h	790.5 US$/h
Cylinder oil costs	0.0 US$/h	35.1 US$/h
System oil costs	0.0 US$/h	4.7 US$/h
Maintenance costs	0.0 US$/h	29.2 US$/h
Auxiliary engines		
Delivered power	1563 kW	3252 kW
HFO cons. (LCV: 40,000 kJ/kg)	0.5 t/h	0.7 t/h
Fuel oil costs	72.6 US$/h	98.9 US$/h
System oil costs	0.0 US$/h	2.3 US$/h
Maintenance costs	0.0 US$/h	8.1 US$/h
Operating costs per hour	1126.3 US$/h	968.8 US$/h

Annual operating costs and value of lost LNG		
Options	Steam Turbine Turbo Generator	Two-Stroke Diesel Engine with Reliquefaction System
Operating costs during		
Loaded conditions	3,140,000 US$/yr	4,040,000 US$/yr
Ballast conditions	4,440,000 US$/yr	3,820,000 US$/yr
Operating costs/year (excl. LNG)	7,580,000 US$/yr	7,860,000 US$/yr
LNG account (per trip)		
Lost during loaded voyage	2438 m^3	- m^3
Lost during ballast voyage	1219 m^3	- m^3
Total Economy		
Operating costs	7,580,000 US$/yr	7,860,000 US$/yr
Value of lost LNG	3,910,000 US$/yr	- US$/yr
Total costs per year	**11,490,000 US$/yr**	**7,860,000 US$/yr**
Saving per year when using diesel engine propulsion		**3,630,000 US$/yr**

Basic Data for Economical Comparison	
Oil prices	
Heavy Fuel Oil	150 US$/ton
Lubricating oil for four-stroke engines	700 US$/ton
Cylinder L. O. for two-stroke engines	800 US$/ton
System oil for two-stroke engines	700 US$/ton
LNG prices	
LNG sales price	188 US$/ton
LNG sales price (LCV of methane = 50,000 KJ/kg)	4.0 US$/Mbtu
Voyage profile	
Distance (Pilot-Pilot)	6500 nm
Nominal Service Speed	20 knots
Loaded voyage	325 hours
Ballast voyage	325 hours
Reserve	24 hours
Time for unloading	24 hours
Time for loading	24 hours
Time per round-trip	722 hours
Round-trips per year	12.1
Propulsion power in loaded conditions	28,920 kW
Propulsion power in ballast conditions	28,920 kW

Size and boil-off rates LNG carrier	
Ship particulars	
Cargo capacity	150,000 m3
Boil-off rate in loaded conditions *	0.12% per day
Volume of methane	180.0 m3/day
Mass of methane (Density = 470 kg/m3)	84,600 kg/day
Energy in methane (LCV: 50,000 kJ/kg)	4230 GJ/day
Boil-off rate in ballast conditions	0.06% per day
Volume of methane	90.0 m3/day
Mass of methane (Density = 470 kg/m3)	42,300 kg/day
Energy in methane (LCV: 50,000 kJ/kg)	2115 GJ/day

Electrical power consumption	
Reliquefication plant	
Specific Power Consumption	920 W/kg/h
Loaded conditions	
Mass of methane	3525 kg/h
Power consumption	3243 kWe
Ballast conditions	
Mass of methane	1763 kg/h
Power consumption	1622 kWe
Other consumers of electrical power	
Steam turbine plant	1500 kWe
Two-stroke diesel engine plant	1500 kWe

*) Only methane is considered

Basic Data for Machinery

Boiler and Steam Turbine	
Specific fuel oil consumption (LCV: 42,700 kJ/kg)	290.0 g/kWh
Specific energy consumption	12,383 kJ/kWh
Specific maintenance costs	0.0 US$/MWh

Diesel Engines	
Typical data MAN B&W two-stroke MC/MC-C engine	
Specific fuel oil consumption (LCV: 42,700 kJ/kg)	169.0 g/kWh
Specific energy consumption	7216 kJ/kWh
Specific cylinder L.O. consumption	1.5 g/kWh
System oil consumption	80.0 kg/24h
Specific maintenance costs	1.0 US$/MWh
Typical data for small MAN B&W four-stroke HFO Gensets	
Specific fuel oil consumption (LCV: 42,700 kJ/kg)	190.0 g/kWh
Specific L.O. consumption	1.0 g/kWh
Specific maintenance costs	2.5 US$/MWh

Efficiency of Propulsion Plant

Propulsion plant	
Steam plant	
Loss in shaft line	1%
Loss in reduction gear	2%
Generator efficiency for turbogenerators	96%
Two-stroke diesel engine	
Loss in shaft line	1%
Generator efficiency for gensets	96%