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**LNG Carriers with ME-GI Engine and High Pressure Gas Supply System**

**Introduction**

The latest introduction to the marine market of ship designs with the dual-fuel low speed ME-GI engine has been very much supported by the Korean shipyards and engine builders, Doosan, Hyundai, Samsung and Daewoo.

Thanks to this cooperation it has been possible to introduce the ME-GI engines into the latest design of LNG carriers and get full acceptance from the Classification Societies involved.

This paper describes the innovative design and installation features of the fuel gas supply system for an LNG carrier, comprising multi-stage low temperature boil-off fuel gas compressor with driver and auxiliary systems, high-pressure piping system and safety features, controls and instrumentation. The paper also extensively describes the operational control system required to provide full engine availability over the entire transport cycle.

The demand for larger and more energy efficient LNG carriers has resulted in rapidly increasing use of the diesel engine as the prime mover, replacing traditional steam turbine propulsion plants.

Two alternative propulsion solutions have established themselves to date on the market:

- low speed, heavy fuel oil burning diesel engine combined with a liquefaction system for BOG recovery
- medium speed, dual-fuel engines with electric propulsion.

A further low speed direct propulsion alternative, using a dual-fuel two-stroke engine, is now also available:

- high thermal efficiency, flexible fuel/gas ratio, low operational and installation costs are the major benefits of this alternative engine version
- the engine utilises a high-pressure gas system to supply boil-off gas at pressures of 250-300 bar for injection into the cylinders.

Apart from the description of the fuel gas supply system, this paper also discusses related issues such as requirements for classification, hazardous identification procedures, main engine room safety, maintenance requirements and availability.

It will be demonstrated that the ME-GI based solution has operational and economic benefits over other low speed based solutions, irrespective of vessel size, when the predicted criteria for relative energy prices prevail.
Propulsion Requirements for LNG Carriers with Dual-Fuel Gas Injection

In 2004, the first diesel engine order was placed for an LNG carrier, equipped with two MAN B&W low speed 6S70ME-C engines. Today, the order backlog comprises more than 90 engines for various owners, mainly oil companies, all for Qatar gas distribution projects.

While the HFO burning engine is a well known and recognised prime mover, the low speed dual-fuel electronically controlled ME-GI (gas injection) engine has not yet been ordered by the market.

Although the GI engine, as a mechanically operated engine, has been available for many years, it is not until now that there is real potential. Cost, fuel flexibility and efficiency are the driving factors.

The task of implementing the two-stroke ME-GI engine in the market has focused on the gas supply system, from the LNG storage tanks to the high-pressure gas compressor and further to the engine. A cooperation between the shipyard HHI, the compressor manufacturer Burckhardt Compression, AG (BCA), the classification society and MAN Diesel has been mandatory to ensure a proper and safe design of the complete gas distribution system, including the engine. This has been achieved through a common Hazid / Hazop study.

Configuration of LNG carriers utilising the boil-off gas

The superior efficiency of the two-stroke diesel engines, especially with a directly coupled propeller, has gained increasing attention. On LNG carriers, the desired power for propulsion can be generated by a single engine with a single propeller combined with a power take home system, or a double engine installation with direct drive on two propellers. This paper concentrates on the double engine installations

2 x 50 %, which is the most attractive solution for an LNG carrier of the size 145 kcum and larger. By selecting a twin propeller solution for this LNG carrier, which normally has a high Beam/draft ratio, a substantial gain in propeller efficiency of some 5 % for 145 kcum and larger, and up to 9 % or even more for larger carriers is possible.

Redundancy in terms of propulsion is not required by the classification societies, but it is required by all operators on the LNG market. The selection of the double engine ME-GI solution results not only in redundancy of propulsion, but also of redundancy in the choice of fuel supply. If the fuel gas supply fails, it is possible to operate the ME-GI as an ME engine, fuelled solely with HFO.

For many years, the LNG market has not really valued the boil-off gas, as this has been considered a natural loss not accounted for.

Today, the fuel oil price has been at a high level, which again has led to considerations by operators on whether to burn the boil-off gas instead of utilising 100 % HFO, DO or gas oil. Various factors determine the rate of the boil-off gas evaporation, however, it is estimated that boil-off gas equals about 80-90 % in laden voyage, and in ballast voyage 40-50 % of the energy needed for the LNG vessel at full power. Therefore, some additional fuel oil is required or alternative forced boil-off gas must be generated. Full power is defined as a voyage speed of 19-21 knots. This speed has been accepted in the market as the most optimal speed for LNG

<table>
<thead>
<tr>
<th>LNG carrier size (cum)</th>
<th>Recommended two-stroke solution</th>
<th>Propulsion power (kW)</th>
<th>Propulsion speed (knots)</th>
<th>Beam/draft ratio</th>
<th>Estimated gain in efficiency compared to a single propeller</th>
</tr>
</thead>
<tbody>
<tr>
<td>145,000-150,000</td>
<td>2 x 6S60ME-GI 2 x 5S65ME-GI</td>
<td>2 x 14,280 2 x 14,350</td>
<td>19-21</td>
<td>3.8</td>
<td>5%</td>
</tr>
<tr>
<td>160,000-170,000</td>
<td>2 x 5S70ME-GI 2 x 7S60ME-GI</td>
<td>2 x 16,350 2 x 16,660</td>
<td>19-21</td>
<td>4.0</td>
<td>&gt; 5%</td>
</tr>
<tr>
<td>200,000-220,000</td>
<td>2 x 6S65ME-GI 2 x 6S70ME-GI</td>
<td>2 x 17,220 2 x 19,620</td>
<td>19-21</td>
<td>4.2</td>
<td>9%</td>
</tr>
<tr>
<td>240,000-270,000</td>
<td>2 x 7S65ME-GI 2 x 7S70ME-GI</td>
<td>2 x 20,090 2 x 21,770</td>
<td>19-21</td>
<td>4.5</td>
<td>&gt; 9%</td>
</tr>
</tbody>
</table>
carriers when both first cost investment and loss of cargo is considered.

To achieve this service speed, a two-stroke solution for the power requirement for different LNG carrier sizes is suggested in Table I.

With the high-pressure gas injection ME-GI engine, the virtues of the two-stroke diesel principle are prevailing. The thermal efficiency and output remain equivalent to that obtained when burning conventional heavy fuel oil. The high-pressure gas injection system offers the advantage of being almost independent of gas/oil fuel mixture, as long as a small amount of pilot oil fuel is injected for ignition.

In order for the ME-GI to achieve this superior efficiency of 50 % (+/− 5 % fuel tolerances) during gas running, the gas fuel requires a boost to a pressure of maximum 250 bars at 100 % load.

At lower loads the pressure required decreases linearly to 30 % load, where a boost pressure of 150 bars is required. To boost this pressure, a high-pressure compressor solution has been developed by BCA, which is presented in this paper.

Fig. 1 shows an example of an LNG carrier with the recommended ME-GI application.

**Fuel Gas Supply System – Design Concept**

The basic design concept of the fuel gas supply system presented in this paper considers the installation of two 100 % fuel gas compressors. Full redundancy of the fuel gas compressor has been considered as a priority to satisfy classification requirements (see Fig. 2).

Each compressor is designed to deliver the boil-off gas at a variable discharge pressure in the range of 150 to 265 bar g (15–26.5 MPa g), according to required engine load to two 50 % installed ME-GI engines A and B. The selected compressor runs continuously, and the standby compressor is started manually only in the event of malfunction of the compressor selected.

The amount of boil-off gas (BOG), and hence the tank pressure, varies considerably during the ship operating cycle. The design concept therefore requires that the compressors be able to operate under a number of demanding conditions, i.e. with:

- a wide variation of BOG flow, as experienced during loaded and ballast voyage,
- a variation in suction pressure according to storage tank pressure,
- a very wide range of suction temperatures, as experienced between warm start-up and ultra cold loaded operation, and
- a variable gas composition.

The compressor is therefore fitted with a capacity control system to ensure gas delivery at the required pressure to the ME-GI engine, and tank pressure con-
LNG Carriers with ME-GI Engine and High Pressure Gas Supply System

trol within strictly defined limits. These duty variables are to be handled both simply and efficiently without compromising overall plant reliability and safety.

The compressor is designed to efficiently deliver both natural boil-off gas (nBOG) and, if required, forced (fBOG) during the ballast voyage.

Finally, the basic design concept also considers compressor operation in alternative running mode to deliver low pressure gas to the gas combustion unit (GCU). Operation with gas delivery simultaneously to both GCU and ME-GI is also possible.

Alternative fuel gas supply system concepts, employing either 2 x 50 % installed compressors and a separate supply line for the GCU, or 1 x 100 % compressor in combination with a BOG liquefaction plant, are currently being considered by the market.

These alternative concepts are not described further in this paper.

Fuel Gas Supply System – Key Components

Fuel gas compressor 6LP250-5S_1

The compression of cryogenic LNG boil-off gas up to discharge pressures in the range of 10-50 barg (1.0 to 5.0 MPa g) is now common practice in many LNG production and receiving terminals installed world wide today.

Compressor designs employing the highly reliable labyrinth sealing principle have been extensively used for such applications. The challenge for the compressor designer of the ME-GI application is to extend the delivery pressure reliably and efficiently by adding additional compression stages to achieve the required engine injection pressure. In doing so, the compressor’s physical dimensions must consider the restricted space available within the deck-mounted machinery room.

The fuel gas compressor with the designation 6LP250-5S_1 is designed to deliver low-temperature natural or forced boil-off gas from atmospheric tank pressure at an inlet temperature as low as −160°C, up to a gas injection pressure in the range of 150 to 265 bar. A total of five compression stages are provided and arranged in a single vertical compressor casing directly driven by a conventional electric motor. The guiding principles of the compressor design are similar to those of API 618 for continuous operating process compression applications.

The compressor designation is as follows:

6LP250-5S_1

6 number of cranks
L labyrinth sealing piston, stages 1 to 3
P ring sealing piston, stages 4 to 5
250 stroke in mm
5 number of stages
S cylinder size reference
1 valve design

A unique compressor construction allows the selection of the best applicable cylinder sealing system according to the individual stage operating temperature and pressure. In this way, a very high reliability and availability, with low maintenance, can be achieved.

Oil-free compression, required for the very cold low pressure stages 1 to 3, employs the labyrinth sealing principle, which is well proven over many years on LPG carriers and at LNG receiving terminals. The avoidance of mechanical friction in the contactless labyrinth cylinder results in extremely long lifetimes of sealing components (see Appendix 1).
The high-pressure stages 4 and 5 employ a conventional API 618 lubricated cylinder ring sealed compressor technology (see Fig. 3).

Six cylinders are mounted on top of a vertical arranged crankcase. The double acting labyrinth compression stages 1 to 3 are typical of those employed at an LNG receiving terminal.

The single acting stages 4 and 5 are a design commonly used for compression of high-pressure hydrocarbon process gases in a refinery application (Fig. 4).

The two first-stage labyrinth cylinders, which are exposed to very low temperatures, are cast in the material GGGNi35 (Fig. 5). This is a nodular cast iron material containing 35 % nickel, also known under the trade name of Ni-Resist D5.

Fig. 3: Highly reliable cylinder sealing applied for each compression stage

Fig. 4: Main constructional features of the 6LP250-5S compressor
This alloy simultaneously exhibits remarkable ductility at low temperatures and one of the lowest thermal expansion coefficients known in metals.

The corresponding pistons are made of nickel alloyed cast iron with laminar graphite. Careful selection of cylinder materials allows the compressor to be started at ambient temperature condition and cooled down to BOG temperature without any special procedures.

Second and third stage labyrinth cylinders operate over a higher temperature range and are therefore provided with a cooling jacket. Cylinder materials are nodular cast iron and grey cast iron respectively.

The oil lubricated high-pressure 4th and 5th stage cylinders are made from forged steel and are provided with a coolant jacket to remove heat of compression.

In view of the smaller compression volumes and high pressure, the piston and piston rod for stages 4 and 5 are integral and manufactured from a single forged steel material stock. Compression is single acting with the 4th stage arranged at the upper end and the 5th stage at the lower end and arranged in step design. Piston rod gas leakage of the 5th stage is recovered to the suction of the 4th stage (see Fig. 6).

**Motion work – 6LP250-5S**

The 6-crank, 250 mm stoke compressor frame is a conventional low speed, crosshead design typically employed for continuous operating process duties. The industry design standard for this compressor type is the American
Petroleum Industry Standard API 618 for refinery process application.

The forged steel crankshaft and connecting rods are supported by heavy tri-metal, force lubricated main bearings. Oil is supplied by a crankshaft driven main oil pump. A single distance piece arranged in the upper frame section provides separation between the lubricated motion work and the non-lubricated compressor cylinders.

The passage of the crankshaft through the wall of the crankcase is sealed off by a rotating double-sided ring seal immersed in oil. Thus, the entire inside of the frame is integrated into the gas containing system with no gas leakage to the environment (see Fig. 7).

Capacity Control – Valve Unloading

Capacity control by valve unloading is extensively employed at LNG terminals where very large variations in BOG flows are experienced during LNG transfer from ship to storage tank.

The capacity of the compressor may be simply and efficiently reduced to 50 % in one step by the use of valve unloaders. The nitrogen actuated unloaders (see Fig. 8) are installed on the lower cylinder suction valves and act to unload one half of the double-acting cylinders.

Additional stepless regulation, required to control a compressor capacity corresponding to the rate of boil-off and the demand of the engine, is provided by returning gas from the discharge to compressor suction by the use of bypass valves. The compressor control system is described in detail later in this paper.
Compressor System Engineering – 6LP250-5S

A compressor cannot function correctly and reliably without a well-designed and engineered external gas system. Static and dynamic mechanical analysis, thermal stress analysis, pulsation analysis of the compressor and auxiliary system consisting of gas piping, pulsation vessels, gas intercoolers, etc., are standard parts of the compressor supplier’s responsibility.

A pulsation analysis considers upstream and downstream piping components in order to determine the correct sizing of pulsation dampening devices and their adequate supporting structure.

The compressor plant is designed to operate over a wide range of gas suction temperatures from ambient start-up at +30°C down to −160°C without any special intervention.

Each compressor stage is provided with an intercooler to control the gas inlet temperature into the following stage. The intercooler design is of the conventional shell and tube type. The first-stage intercooler is bypassed when the suction temperature falls below set limits (approx. −80°C).

The P&I diagram for the compressor gas system is shown in Appendix III.

Bypass valves are provided over stage 1, stages 2 to 3, and stages 4 to 5. These valves function to regulate the flow of the compressor according to the engine set pressure within defined system limits. Non-return valves are provided on the suction, side to prevent gas back-flow to the storage tanks, between stages 3 and 4, to maintain adequate separation between the oil-free and the oil lubrication compressor stages, and at the final discharge from the compressor.

Compressor safety

Safety relief valves are provided at the discharge of each compression stage to protect the cylinders and gas system against overpressure. Stage differential relief valves, where applicable, are installed to prevent compressor excessive loading.

Pressure and temperature instrumentation for each stage is provided to ensure adequate system monitoring alarm and shutdown. Emergency procedures allow a safe shutdown, isolation and venting of the compressor gas system.

The design of the gas system comprising piping, pulsation vessels, gas intercoolers, safety relief valves and accessory components follows industry practices for hydrocarbon process oil and gas installations.

Process duty – compressor rating

The sizing of the fuel gas compressor is directly related to the “design” amount of nBOG and, therefore, to the capacity of the LNG carrier.

The fuel gas system design concept considers compressor operation not only for supplying gas to the ME-GI engine, but also to deliver gas to the gas combustion unit (GCU) in the event that the engine cannot accept any gas.

The design nBOG rates are typically in the range of 0.135 to 0.15 % per day of tanker liquid capacity. Design nBOG rates are 0.10 to 0.12 % may be expected.

The compressors are therefore rated to handle the maximum amount of natural BOG defined by the tank system supplier and consistent with the design rating of GCU.

Design nBOG rates are typically in the range of 0.135 to 0.15 % per day of tanker liquid capacity. During steady-state loaded voyage, a BOG rate of 0.10 to 0.12 % may be expected.

Carrier capacities in the range 145 to 260 kcum have been considered, resulting in the definition of 3 alternative

| Volume LNG tanker | 210,000 |
| Max. BOG rate LNG tanker | % | 0.15 |
| Density of methane liquid at 1.06 bar a | kg/m3 | 427 |
| BOG mass flow | kg/h | 5,600 |
| LNG tank pressure low / high | bar a | 1.06/1.20 |
| Temperature BOG low | °C | −140 |
| Temperature BOG high | °C | −40 |
| Temperature BOG start up | °C | +30 |
| Delivery P to ME-GI pressure low / high | bar a | 150/265 |
| Temperature NG delivery to ME-GI | °C | +45 |
| Compressor shaft power | kW | 1,600 |
| Delivery P to GCU | bar a | 4.0 to 6.5 |

Table II: Rated process design data for a 210 kcum carrier
compressor designs which differ according to frame rating and compressor speed.

Rated process design data for a carrier capacity of 210 kcum are as shown in Table II.

The rating for the electric motor driver is determined by the maximum compressor power required when considering the full operating range of suction temperatures from +30 to −140°C and suction pressures from 1.03 to 1.2 bar a.

ME-GI Gas System Engineering

The ME-GI engine series, in terms of engine performance (output, speed, thermal efficiency, exhaust gas amount and temperature, etc.) is identical to the well-established, type approved ME engine series. The application potential for the ME engine series therefore also applies to the ME-GI engine, provided that gas is available as a main fuel. All ME engines can be offered as ME-GI engines.

Since the ME system is well known, the following description of the ME-GI engine design only deals with new or modified engine components.

Fig. 9 shows one cylinder unit of a S70ME-GI, with detail of the new modified parts. These comprise gas supply double-wall piping, gas valve control block with internal accumulator on the (slightly modified) cylinder cover, gas injection valves and ELGI valve for control of the injected gas amount. In addition, there are small modifications to the exhaust gas receiver, and the control and manoeuvring system.

Apart from these systems on the engine, the engine and auxiliaries will comprise some new units. The most important ones, apart from the gas supply system, are listed below, and the full system is shown in schematic form in Appendix IV

The new units are:

- Ventilation system, for venting the space between the inner and outer pipe of the double-wall piping.
- Sealing oil system, delivering sealing oil to the gas valves separating the control oil and the gas
- Inert gas system, which enables purging of the gas system on the engine with inert gas.

Fig. 9: Two-stroke MAN B&W S70ME-GI
The GI system also includes:

- Control and safety system, comprising a hydrocarbon analyser for checking the hydrocarbon content of the air in the double-wall gas pipes.

The GI control and safety system is designed to “fail to safe condition”. All failures detected during gas fuel running including failures of the control system itself, will result in a gas fuel Stop/Shut Down, and a change-over to HFO fuel operation. Blow-out and gas-freeing purging of the high-pressure gas pipes and the complete gas supply system follows. The change-over to fuel oil mode is always done without any power loss on the engine.

The high-pressure gas from the compressor-unit flows through the main pipe via narrow and flexible branch pipes to each cylinder’s gas valve block and accumulator. These branch pipes perform two important tasks:

- They separate each cylinder unit from the rest in terms of gas dynamics, utilising the well-proven design philosophy of the ME engine’s fuel oil system.
- They act as flexible connections between the stiff main pipe system and the engine structure, safeguarding against extra-stresses in the main and branch pipes caused by the inevitable differences in thermal expansion of the gas pipe system and the engine structure.

The buffer tank, containing about 20 times the injection amount per stroke at MCR, also performs two important tasks:

- It supplies the gas amount for injection at a slight, but predetermined, pressure drop.
- It forms an important part of the safety system.

Since the gas supply piping is of common rail design, the gas injection valve must be controlled by an auxiliary control oil system. This, in principle, consists of the ME hydraulic control (system) oil system and an ELGI valve, supplying high-pressure control oil to the gas injection valve, thereby controlling the timing and opening of the gas valve.

The buffer tank, containing about 20 times the injection amount per stroke at MCR, also performs two important tasks:

- It supplies the gas amount for injection at a slight, but predetermined, pressure drop.
- It forms an important part of the safety system.

The ME-GI Injection System

Dual fuel operation requires the injection of both pilot fuel and gas fuel into the combustion chamber.

Different types of valves are used for this purpose. Two are fitted for gas injection and two for pilot fuel. The auxiliary media required for both fuel and gas operation are as follows:

- High-pressure gas supply
- Fuel oil supply (pilot oil)
- Control oil supply for activation of gas injection valves
- Sealing oil supply.

The gas injection valve design is shown in Fig. 10. This valve complies with traditional design principles of compact design. Gas is admitted to the gas injection valve through bores in the cylinder cover. To prevent gas leakage between cylinder cover/gas injection valve and valve housing/spindle guide, sealing rings made of temperature and gas resistant material are installed. Any gas leakage through the gas sealing rings will be led through bores in the gas injection valve and further to space between the inner and the outer shield pipe of the double-wall gas piping system. This leakage will be detected by HC sensors.

The gas acts continuously on the valve spindle at a max. pressure of about 250 bar. To prevent gas from entering the control oil activating system via the clearance around the spindle, the spindle is sealed by sealing oil at a pressure higher than the gas pressure (25-50 bar higher).
The pilot oil valve is a standard ME fuel oil valve without any changes, except for the nozzle. The fuel oil pressure is constantly monitored by the GI safety system, in order to detect any malfunctioning of the valve.

The designs of oil valve will allow operation solely on fuel oil up to MCR. If the customer’s demand is for the gas engine to run at any time at 100% load on fuel oil, without stopping the engine, this can be done. If the demand is prolonged operation on fuel oil, it is recommended to change the nozzles and gain an increase in efficiency of around 1% when running at full engine load.

As can be seen in Fig. 11 (GI injection system), the ME-GI injection system consists of two fuel oil valves, two fuel gas valves, ELGI for opening and closing of the fuel gas valves, and a FIVA valve to control (via the fuel oil valve) the injected fuel oil profile. Furthermore, it consists of the conventional fuel oil pressure booster, which supplies pilot oil in the dual fuel operation mode. This fuel oil pressure booster is equipped with a pressure sensor to measure the pilot oil on the high pressure side. As
mentioned earlier, this sensor monitors the functioning of the fuel oil valve. If any deviation from a normal injection is found, the GI safety system will not allow opening for the control oil via the ELGi valve. In this event no gas injection will take place.

Under normal operation where no malfunctioning of the fuel oil valve is found, the fuel gas valve is opened at the correct crank angle position, and gas is injected. The gas is supplied directly into an ongoing combustion. Consequently the chance of having unburnt gas eventually slipping past the piston rings and into the scavenge air receiver is considered to be very low. Monitoring the scavenge air receiver pressure safeguards against such a situation. In the event of high pressure, the gas mode is stopped and the engine returns to burning fuel oil only.

The gas flow to each cylinder during one cycle will be detected by measuring the pressure drop in the accumulator. By this system, any abnormal gas flow, whether due to seized gas injection valves or blocked gas valves, will be detected immediately. The gas supply will be discontinued and the gas lines purged with inert gas. Also in this event, the engine will continue running on fuel oil only without any power loss.

**High-Pressure Double-Wall Piping**

A common rail (constant pressure) gas supply system is to be fitted for high-pressure gas distribution to each valve block. Gas pipes are designed with double-walls, with the outer shielding pipe designed so as to prevent gas outflow to the machinery spaces in the event of rupture of the inner gas pipe. The intervening space, including also the space around valves, flanges, etc., is equipped with separate mechanical ventilation with a capacity of approx. 30 air changes per hour. The pressure in the intervening space is below that of the engine room with the (extractor) fan motors placed outside the ventilation ducts. The ventilation inlet air is taken from a non-hazardous area.

Gas pipes are arranged in such a way, see Fig. 12 and Fig. 13, that air is sucked into the double-wall piping system from around the pipe inlet, from there into the branch pipes to the individual gas valve control blocks, via the branch supply pipes to the main supply pipe, and via the suction blower into the atmosphere.

Ventilation air is exhausted to a fire-safe place. The double-wall piping system is designed so that every part is ventilated. All joints connected with sealings to a high-pressure gas volume are being ventilated. Any gas leakage will therefore be led to the ventilated part of the double-wall piping system and be detected by the HC sensors.

The gas pipes on the engine are designed for 50% higher pressure than the normal working pressure, and are supported so as to avoid mechanical vibrations. The gas pipes are furthermore shielded against heavy items falling down, and on the engine side they are placed below the top-gallery. The pipes are pressure tested at 1.5 times

![Fig. 12: Branching of gas piping system](image)
the working pressure. The design is to be all-welded, as far as it is practicable, using flange connections only to the extent necessary for servicing purposes. The branch piping to the individual cylinders is designed with adequate flexibility to cope with the thermal expansion of the engine from cold to hot condition. The gas pipe system is also designed so as to avoid excessive gas pressure fluctuations during operation.

For the purpose of purging the system after gas use, the gas pipes are connected to an inert gas system with an inert gas pressure of 4-8 bar. In the event of a gas failure, the high-pressure pipe system is depressurised before automatic purging. During a normal gas stop, the automatic purging will be started after a period of 30 min. Time is therefore available for a quick re-start in gas mode.
Fuel Gas System - Control Requirements

The primary function of the compressor control system is to ensure that the required discharge pressure is always available to match the demand of the main propulsion diesel engines. In doing so, the control system must adequately handle the gas supply variables such as tank pressure, BOG rate (laden and ballast voyage), gas composition and gas suction temperature.

If the amount of nBOG decreases, the compressor must be operated on part load to ensure a stable tank pressure, or forced boil-off gas (fBOG) added to the gas supply. If the amount of nBOG increases, resulting in a higher than acceptable tank pressure, the control system must act to send excess gas to the gas combustion unit (GCU).

Tank pressure changes take place over a relatively long period of time due to the large storage volumes involved.

A fast reaction time of the control system is therefore not required for this control variable.

The main control variable for compressor operation is the feed pressure to the ME-GI engine, which may be subject to controlled or instantaneous change. An adequate control system must be able to handle such events as part of the “normal” operating procedure.

The required gas delivery pressure varies between 150-265 bar, depending on the engine load (see Fig. 14 below).

The compressor must also be able to operate continuously in full recycle mode with 100 % of delivered gas returned to the suction side of the compressor. In addition, simultaneous delivery of gas to the ME-GI engine and GCU must be possible.

When considering compressor control, an important difference between centrifugal and reciprocating compressors should be understood. A reciprocating compressor will always deliver the pressure demanded by the down-stream user, independent of any suction conditions such as temperature, pressure, gas composition, etc. Centrifugal compressors are designed to deliver a certain head of gas for a given flow. The discharge pressure of these compressors will therefore vary according to the gas suction condition.

This aspect is very important when considering transient starting conditions such as suction temperature and pressure. The 6LP250-5S_1 reciprocating compressor has a simple and fast start-up procedure.

Compressor control – 6LP250-5S_1

Overall control concept

Fig. 15 shows a simplified view of the compressor process flow sheet. The system may be effectively divided into a low-pressure section (LP) consisting of the cold compression stage 1, and a high-pressure section (HP) consisting of stages 2 to 5.

The main control input for compressor control is the feed pressure Pset required by the ME-GI engine. The feed pressure may be set in the range of 150 to 265 bar according to the desired engine load. If the two ME-GI engines are operating at different loads, the higher set pressure is valid for the compressor control unit.

If the amount of nBOG is insufficient to satisfy the engine load requirement, and make-up with fBOG is not foreseen, the compressor will operate on part load to ensure that the tank pressure remains within specified limits. The ME-GI engine will act independently to increase the supply of HFO to the engine. Primary regulation of the compressor capacity is made with the 1st stage bypass valve, followed by cylinder valve unloading and if required bypass over stages 2 to 5. With this sequence, the compressor is able to operate flexibly over the full capacity range from 100 to 0 %.

Fig. 14: Gas supply station, guiding specification
If the amount of nBOG is higher than can be burnt in the engine (for example during early part of the laden voyage) resulting in higher than acceptable suction pressure (tank pressure), the control system will send excess gas to the GCU via the side stream of the 1st compression stage.

In the event of engine shutdown or sudden change in engine load, the compressor delivery line must be protected against overpressure by opening bypass valves over the HP section of the compressor.

During start-up of the compressor with warm nBOG, the temperature control valves will operate to direct a flow through an additional gas intercooler after the 1st compression stage.

The control concept for the compressor is based on one main control mode which is called “power saving mode”. This mode of running, which minimises the use of gas bypass as the primary method of regulation, operates within various well defined control limits.

The system pressure control limits are as follows:

- $P_{\text{min}}$ suction: Prevents sub-pressure in compressor inlet manifold - tank vacuum.
- $P_{\text{high}}$ suction: Suction manifold high-pressure - system safety (GCU) on standby.
- $P_{\text{max}}$ suction: Initiates action to reduce inlet manifold pressure.
- $P_{\text{max}}$: Prevents overpressure of ME-GI feed compressor discharge manifold.

A detailed description of operation within these control limits is given below. Power saving mode

Economical regulation of a multi-stage compressor is most efficiently executed using gas recycle around the 1st stage of compression. The ME-GI required set pressure $P_{\text{set}}$ is therefore taken as control input directly to the compressor 1st stage bypass valve, which will open or close until the actual compressor discharge pressure is equal to the $P_{\text{set}}$. With this method of control, BOG delivery to the ME-GI is regulated without any direct measurement and control of the delivered mass flow. If none of the above control limits are active, the controller is able to regulate the mass flow in the range from 0 to 100%.

The following control limits act to overrule the ME-GI controller setting and initiate bypass valve operation:

- $P_{\text{min}}$ suction: (tank pressure below set-level)
The control scenario is falling suction pressure. If the Pmin limit is active, the 1st stage recycle valve will not be permitted to close further, thereby preventing further reduction in suction pressure. If the pressure in the suction line continues to decrease, the recycle valve will open governed by the Pmin limiter.

**Action of Pressure will fall at the ME-GI control compressor discharge system:**

- **Pmin suction**
  - If the Pmin limit is active, the 1st stage recycle valve will not be permitted to close further, thereby preventing further reduction in suction pressure.
  - If the pressure in the suction line continues to decrease, the recycle valve will open governed by the Pmin limiter.

**fBOG:**

- If a spray cooling or forced vaporizer is installed, it may be used for stabilising the suction pressure and thereby increase the gas mass flow to the engine. Such a system could be activated by the Pmin suction pressure limit.

**P_high suction** (tank pressure above set level)

- The control scenario is increasing suction pressure due to either reduced engine load (e.g., approaching port, manoeuvring) or excess nBOG due to liquid impurities (e.g., N2).

The control limiter initiates a manual start of the GCU (the GCU is assumed not to be on standby mode during normal voyage).

There is no action on the compressor control or the ME-GI control system.

**P_max suction** (tank pressure too high)

- The control scenario is the same as described above, however, it has resulted in even higher suction pressure. Action must now be taken to reduce suction pressure by sending gas to the GCU.

- The high pressure alarm initiates a manual sequence whereby the 1st stage bypass valve PCV01 is closed and the bypass valve PCV02 to the GCU is opened. When the changeover is completed, automatic Pset control is transferred to the GCU control valve PCV02. The gas amount which cannot be accepted by the ME-GI will be burned simultaneously in the GCU.

- No action is taken in the ME-GI control system.

**GCU-only operating mode**

The control scenario considers a situation where gas injection to the ME-GI is not required and tank gas pressure is at the level of Phigh.

- The nBOG is compressed and delivered to the GCU by means of a gas take-off after the 1st stage.

- The following actions are initiated:
  - manual start of the GCU
  - closing of the bypass valve around 1st stage
  - fully opening of the bypass valves around stages 2-5.

- In this mode, the compressor is operating with stages 2-5 in full recycle at a reduced discharge pressure of approximately 80 bar. The pressure setting of the GCU feed valve is set directly by the GCU in the range 3 to 6 bar a.

- There is no action on the ME-GI controller.
Machinery Room Installation – 6LP250-5S

The layout of the cargo handling equipment and the design of their supporting structure presents quite a challenge to the shipbuilder where space on deck is always at a premium. In conjunction with HHI and the compressor maker, an optimised layout of the fuel gas compressor has been developed.

There are many factors which influence the compressor plant layout apart from limited space availability. (See Fig. 16.) External piping connections, adequate access for operation and maintenance, equipment design and manufacturing codes, plant lifting and installation are just a few.

The compressor together with accessory items comprising motor drive, auxiliary oil system, vessels, gas coolers, interconnecting piping, etc., are manufactured as modules requiring minimum assembly work on the ship deck. Separate auxiliary systems provide coolant for the compressor frame and gas coolers.

If required, a dividing bulkhead may separate the main motor drive from the hazardous area in the compressor room. A compact driveshaft arrangement without bulkhead, using a suitably designed ex motor, is however preferred. Platforms and stairways provide access to the compressor cylinders for valve maintenance. Piston assemblies are withdrawn vertically through manholes in the roof of the machinery house (see Fig. 17).
Requirements for Cargo Machinery Room Support Structure

Fig. 18 shows details of the compressor base frame footprint and requirement for support by the ship structure.

Reciprocating compressors, by nature of movement of their rotating parts, exhibit out-of-balance forces and moments which must be considered in the design of the supporting structure for acceptable machinery vibration levels.

As a boundary condition, the structure underneath the cargo machinery room must have adequate weight and stiffness to provide a topside vibration level of (approximately) 1.2 - 1.5 mm/s. Satisfactory vibration levels for compressor frame and cylinders are 8 and 15 mm/s respectively (values given are rms – root mean square).

Foundation deflection due to ship movement must, furthermore, be considered in the design of the compressor plant to ensure stress-free piping terminations. Maintenance requirements - availability/reliability

The low speed, crosshead type compressor design 6LP250-5S, like the ME-GI diesel engine, is designed for the life time of the LNG carrier (25 to 30 years or longer). Routine maintenance is limited purely to periodic checking in the machinery room.

Maintenance intervention for dismantling, checking and eventual part replacement is recommended after each 8,000 hours of operation. Annual maintenance interventions will normally require 50-70 hours work for checking and possible replacing of wearing parts.

Major intervention for dismantling and bearing inspection is recommended every 2-3 years.

Average availability per compressor unit is estimated to be 98.5 % with best availability approximately 99.5 %. With an installed redundant unit, the compressor plant availability will be in the region of 99.25 %.

Any unscheduled stoppage of the 6LP250-5S compressor will most likely be attributable to a mal-function of a cylinder valve. With the correct valve design and material selection (Burckhardt uses its own design and manufacture plate valves) these events will be very seldom, however a valve failure in operation cannot be entirely ruled out.

LNG boil-off gas is an ideal gas to compress. The gas is relatively pure and uncontaminated, the gas components are well defined, and the operating temperatures are stable once “cool-down” is completed.

These conditions are excellent for long lifetime of the compressor valves where an average lifetime expectancy for valve plates is 16,000 hours. Therefore, we do not expect any unscheduled intervention per year for valve maintenance. Such a maintenance intervention will take approx. 7-9 hours for compressor shutdown, isolation and valve replacement.

A total unscheduled maintenance intervention time of 25 hours, assuming 8,000 operating hours per year, may be used for statistical comparison. On this basis compressor reliability is estimated at 99.7 %.
Our experience in many installations shows that no hours are lost for unscheduled maintenance. The reliability of these compressors is therefore comparable to that of centrifugal compressor types.

Requirements for Classification

When entering the LNG market with the combined two-stroke and reliquefaction solution, it was discovered that there is a big difference in the requirements from operators and classification societies.

Being used to cooperating with the classification societies on other commercial ships, the rules and design recommendations for the various applications in the LNG market are new when it comes to diesel engine propulsion. In regard to safety, the high availability and reliability offered when using the two-stroke engines generally fulfill the requirements, but as the delivery and pick up of gas in the terminals is carried out within a very narrow time window, redundancy is therefore essential to the operators.

As such, a two-engine ME-GI solution is the new choice, with its high efficiency, availability and reliability, as the traditional HFO burning engines.

Compared with traditional diesel operated ships, the operators and shipowners in the LNG industry generally have different goals and demands to their LNG tankers, and they often apply more strict design criteria than applied so far by the classification societies.

A Hazid investigation was therefore found to be the only way to secure that all situations are taken into account when using gas for propulsion, and that all necessary precautions have been taken to minimize any risk involved.

In 2005, HHI shipyard, HHI engine builder, BCA and MAN Diesel therefore worked out a hazard identification study that was conducted by Det Norske Veritas (DNV), see Appendix V.
Actual Test and Analysis of Safety when Operating on Gas

The use of gas on a diesel engine calls for careful attention with regard to safety. For this reason, ventilated double-walled piping is a minimum requirement to the transportation of gas to the engine.

In addition to hazard considerations and calculations, it has been necessary to carry out tests, two of which were carried out some years ago before the installation and operation of the Chiba power plant 12K80MC-GI engine in 1994.

A crack in the double-wall inner pipe

The first test was performed by introducing a crack in the inner pipe to see if the outer pipe would stay intact. The test showed no penetration of the outer pipe, thus it could be concluded that the double-wall concept lived up to the expectations.

Pressure fluctuation

The second test was carried out to investigate the pressure fluctuations in the relatively long piping from the gas compressor to the engine.

By estimation of the necessary buffer volume in the piping system, the stroke and injection of gas was calculated to see when safe pressure fluctuations are achieved within given limits for optimal performance of the engines. The piping system has been designed on the basis of these calculations.

Main Engine Room Safety

The latest investigation, which was recently finished, was initiated by a number of players in the LNG market questioning the use of 250 bar gas in the engine room, which is also located under the wheel house where the crew is working and living.

Even though the risk of full breakage happening is considered close to negligible and, in spite of the precautions introduced in the system design, MAN Diesel found it necessary to investigate the effect of such an accident, as the question still remains in part of the industry: what if a double-wall pipe breaks in two and gas is released from a full opening and is ignited?

As specialists in the offshore industry, DNV were commissioned to simulate such a worst case situation, study the consequences, and point to the appropriate countermeasures. DNV’s work comprised a CFD (computational fluid dynamics) simulation of the hazard of an explosion and subsequent fire, and an investigation of the risk of this situation ever occurring and at what scale.

As input for the simulation, the volume of the engine room space, the position of major components, the air ventilation rate, and the location of the gas pipe and control room were the key input parameters.

Realistic gas leakage scenarios were defined, assuming a full breakage of the outer pipe and a large or small hole in the inner fuel pipe. Actions from the closure of the gas shutdown valves, the ventilation system and the ventilation conditions prior to and after detection are included in the analysis. The amount of gas in the fuel pipe limits the duration of the leak. Ignition of a leak causing an explosion or a fire is furthermore factored in, due to possible hot spots or electrical equipment that can give sparks in the engine room.

Calculations of the leak rate as a function of time, and the ventilation flow rates were performed and applied as input to the explosion and fire analyses.
**Simulation Results**

The probability of this hazard happening is based on experience from the offshore industry.

Even calculated in the worst case, no structural damage will occur in the HHI LNG engine room if designed for 1.1 bar over pressure.

- No areas outside the engine room will be affected by an explosion.

If this situation is considered to represent too high a risk, unattended machinery space during gas operation can be introduced. Today, most engines and equipment are already approved by the classification societies for this type of operation.

By insulation, the switchboard room floor can be protected against heat from any jet fires.

- No failure of the fuel oil tank structure, consequently no escalation of fire.

The above conclusion is made on the assumption that the GI safety system is fully working.

In addition, DNV has arrived at a different result based on the assumption that the safety system is not working. On the basic in these results DNV have put up failure frequencies and developed a set of requirements to be followed in case a higher level of safety is required.

After these conclusion made by DNV, HHI has developed a level for their engine room safety that satisfies the requirements from the classification societies, and also the requirements that are expected from the shipowners.

This new engine room design is based on the experience achieved by HHI with their first orders for LNG carriers equipped with 2 x 6S70ME-C and liquefaction plant. The extra safety that will be included is listed below:

Double-wall piping is located as far away as possible from critical walls such as the fuel tank walls and switchboard room walls.

In case of an engine room fire alarm, a gas shutdown signal is sent out, the engine room ventilation fans stops, and the air inlet canals are blocked.

During gas running it is not possible to perform any heavy lifting with the engine room crane.

A failure of the engine room ventilation will result in a gas shutdown.

HC sensors are placed in the engine room, and their position will be based on a dispersion analysis made for the purpose of finding the best location for the sensors.

The double-wall piping is designed with lyres, so that variation in temperatures from pipes to surroundings can be absorbed in the piping.

In fact, any level of safety can be achieved on request of the shipowner. The safety level request will be achieved in a co-operation between the yard HHI, the engine builder HHI, the classification society and MAN Diesel A/S.

The report “Dual fuel Concept: Analysis of fires and explosions in engine room” was made by DNV consulting and can be ordered by contacting MAN Diesel A/S, in Copenhagen.
Engine Operating Modes

One of the advantages of the ME-GI engine is its fuel flexibility, from which an LNG carrier can certainly benefit. Burning the boil-off gas with a variation in the heat value is perfect for the diesel working principle. At the start of a laden voyage, the natural boil-off gas holds a large amount of nitrogen and the heat value is low. If the boil-off gas is being forced, it can consist of both ethane and propane, and the heat value could be high. A two-stroke, high-pressure gas injection engine is able to burn those different fuels and also without a drop in the thermal efficiency of the engine. The control concept comprises two different fuel modes, see also Fig. 19.

- fuel-oil-only mode
- minimum-fuel mode

The fuel-oil-only mode is well known from the ME engine. Operating the engine in this mode can only be done on fuel oil. In this mode, the engine is considered “gas safe”. If a failure in the gas system occurs it will result in a gas shutdown and a return to the fuel-oil-only mode and the engine is “gas safe”.

The minimum-fuel mode is developed for gas operation, and it can only be started manually by an operator on the Gas Main Operating Panel in the control room. In this mode, the control system will allow any ratio between fuel oil and gas fuel, with a minimum preset amount of fuel oil to be used.

The preset minimum amount of fuel oil, hereafter named pilot oil, to be used is in between 5-8% depending on the fuel oil quality. Both heavy fuel oil and marine diesel oil can be used as pilot oil. The min. pilot oil percentage is calculated from 100% engine load, and is constant in the load range from 30-100%. Below 30% load MAN Diesel is not able to guarantee a stable gas and pilot oil combustion, when the engine reach this lower limit the engine returns to Fuel-oil-only mode.

Gas fuels correspond to low-sulphur fuels, and for this type of fuel we recommend the cylinder lube oil TBN40 to be used. Very good cylinder condition with this lube oil was achieved from the gas engine on the Chiba power plant. A heavy fuel oil with a high sulphur content requires the cylinder lube oil TBN 70. Shipowners intending to run their engine on high-sulphur fuels for longer periods of time are recommended to install two lube oil tanks. When changing to minimum-fuel mode, the change of lube oil should be carried out as well.

Players in the market have been focused on reducing the exhaust emissions during harbour manoeuvring. When testing the ME-GI at the MAN Diesel research centre in Copenhagen, the 30% limit for minimum-fuel mode will be challenged taking advantage of the increased possibilities of the ME fuel valves system to change its injection profile, MAN Diesel expects to lower this 30% load limit for gas use, but for now no guaranties can be given.

Fig. 19: Fuel type modes for the ME-GI engines for LNG carriers
Launching the ME-GI

As a licensor, MAN Diesel expects a time frame of two years from order to delivery of the first ME-GI on the testbed.

In the course of this time, depending on the ME-Gi engine size chosen, the engine builder will make the detailed designs and a final commissioning test on a research engine. This type approval test (TAT) is to be presented to the classification society and ship owner in question to show that the compressor and the ME-Gi engine is working in all the operation modes and conditions.

In cooperation with the classification society and engine builder, the most optimum solution, i.e. to test the compressor and ME-GI engine before delivery to the operator has been considered and discussed. One solution is to test the gas engine on the testbed, but this is a costly method. Alternatively, and recommended by MAN Diesel, the compressor and ME-GI operation test could be made in continuation of the gas trial. Today, there are different opinions among the classification societies, and both solutions are possible depending on the choice of classification society and arrangement between ship owners, yard and engine builder.

MAN Diesel A/S has developed a test philosophy especially for approval of the ME-GI application to LNG carriers, this philosophy has so far been approved by DNV, GL, LR and ABS, see Table III. The idea is that the FAT (Factory Acceptance Test) is being performed for the ME system like normal, and for the GI system it is performed on board the LNG carrier as a part of the Gas Trial Test. Thereby, the GI system is tested in combination with the tailor-made gas compressor system for the specific LNG carrier. Only in this combination it will be possible to get a valid test.

Prior to the gas trial test, the GI system has been tested to ensure that everything is working satisfactory.

<table>
<thead>
<tr>
<th>Volume LNG tanker</th>
<th>MAN Diesel Copenhagen</th>
<th>Engine builder testbed</th>
<th>Yard Quay trial</th>
<th>Yard Sea trial</th>
<th>Gas trial</th>
</tr>
</thead>
<tbody>
<tr>
<td>MAN B&amp;W research engine – 4T50MX or similar suitable location</td>
<td>TAT of ME-GI control system and of gas components. Test according to MBD test program. Subject to Class approval.</td>
<td>Test according to: • IACS UR M51 MBD Factory Acceptance Test program (FAT) for ME engines. - do -</td>
<td>Test according to: • Yard and Engine Builder test program approved by Class - do -</td>
<td>Test according to: • Yard and Engine Builder test program approved by Class - do -</td>
<td>After loading gas, the following tests are to be carried out: • Acceptance test of the complete gas system including the main engine. • Test of the ME-GI control system according to MBD test program approved by Class - do -</td>
</tr>
<tr>
<td>First ME-GI production engine</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>Marine diesel oil and gas</td>
</tr>
<tr>
<td>Second and following ME-GI engines</td>
<td>Gas and marine diesel oil</td>
<td>Marine diesel oil</td>
<td>Marine diesel oil and/or heavy fuel oil</td>
<td>Marine diesel oil and/or heavy fuel oil</td>
<td>Marine diesel Heavy fuel oil and gas</td>
</tr>
</tbody>
</table>

Table III: MAN B&W ME-GI engines – test and class approval philosophy
Machinery Concepts Comparison

In this chapter, the ME-C and ME-GI engines in the various configurations will be compared. The comparison will show the most suitable propulsion solution for a modern LNG carrier.

The study is made as objective as possible, however, only MAN Diesel supported systems are compared.

Both the ME-C engine with reliquefaction and the ME-GI engine with gas compressor can be used either in twin engine arrangements, coupled to two fixed pitch (FPP) or two controllable pitch propellers (CPP), or as a single main engine coupled to one FPP.

For LNG carriers, the total electricity consumption of the machinery on board is higher than usual compared with most other merchant ship types.

Therefore, the electrical power generation is included in the comparison.

Thus, the various main propulsion machinery solutions may be coupled with various electricity producers, such as diesel generators (DG), the MAN Diesel waste heat recovery system, called the Thermo Efficiency System (TES), or a shaft generator system (PTO).

Applying the propulsion data listed in Tables IV and V, the estimated data for the electrical power consumption in Tables VI and VII, MAN Diesel has calculated the investment and operational costs of all the alternative configurations illustrated in Fig. 20.

The investment and operational costs have been analysed and the results have been compared using the Net Present Value (NPV) method, see Fig. 21.

In order to quantify the effect of the machinery chosen on the total exhaust gas emissions, and thereby bring it directly into the comparison, costs for the various emission pollutants have been assumed and used in some of the calculations, thereby visualising a possible future economic impact of the emissions.

The following emission fees have been used in the calculations:

- CO₂: 17.3 USD/tonne
- NOx: 2,000 USD/tonne
- SO₂: 2,000 USD/tonne

It has been assumed that the CO₂ fee is to be paid for the complete CO₂ emission, whereas the NOx and SO₂ fees are to be paid for only 20% of the total NOx and SO₂ emissions, since the two latter pollutants are mostly a problem when the ship operates close to the coast line.

Calculations have been made, taking different HFO and LNG prices and different time horizons (10, 20 and 30 years) into account, and with and without the incorporation of the estimated emission fees.

The calculations have been made for three different sizes of LNG carriers; 150,000, 210,000 and 250,000 m³.

Finally, the Net Present Value results, for each LNG carrier size, have been scaled towards each other in such a way that the highest Net Present Value, which represents the alternative with

![Fig. 20: Alternative two-stroke propulsion and power generation machinery systems](image)

The NPV formula

Each cash inflow/outflow is discounted back to its Present Value. Then they are summed. Therefore:

\[
NPV = \sum_{t=0}^{n} \frac{C_t}{(1+r)^t} - C_0
\]

Where

- \( t \) - the time of the cash flow
- \( n \) - the total time of the project
- \( r \) - the discount rate
- \( C_t \) - the net cash flow (the amount of cash) at that point in time.
- \( C_0 \) - the capital outlay at the beginning of the investment time (\( t = 0 \))

![Fig. 21: NPV definition](image)
the highest cost for each combination of fuel prices, time horizon and emission scenario has been nominated to equal 100% cost, whereas the remaining Net Present Values within the same category have been listed in percentages of the above most expensive configuration.

### Table IV: Results of propulsion power prediction calculations for LNG carriers of the membrane type

<table>
<thead>
<tr>
<th>Case</th>
<th>Unit</th>
<th>A</th>
<th>B</th>
<th>C</th>
</tr>
</thead>
<tbody>
<tr>
<td>Ship capacity m³</td>
<td></td>
<td>150,000</td>
<td>210,000</td>
<td>250,000</td>
</tr>
<tr>
<td>Design draught m</td>
<td></td>
<td>11.6</td>
<td>12.0</td>
<td>12.0</td>
</tr>
<tr>
<td>1. Single propeller</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Propeller diameter m</td>
<td>m</td>
<td>1 x 8.60</td>
<td>1 x 8.80</td>
<td>1 x 9.00</td>
</tr>
<tr>
<td>SMCR power kW</td>
<td></td>
<td>1 x 31,361</td>
<td>1 x 39,268</td>
<td>1 x 45,152</td>
</tr>
<tr>
<td>SMCR speed rpm</td>
<td></td>
<td>92.8</td>
<td>91.8</td>
<td>93.8</td>
</tr>
<tr>
<td>Main engine (without PTO)</td>
<td></td>
<td>1 x 7K90ME-Mk 6</td>
<td>1 x 7K98ME-Mk 7</td>
<td>1 x 8K98ME-Mk 6</td>
</tr>
<tr>
<td>2. Twin-skeg and Twin-propulsion</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Propeller diameter m</td>
<td>m</td>
<td>2 x 8.10</td>
<td>2 x 8.40</td>
<td>2 x 8.70</td>
</tr>
<tr>
<td>SMCR power kW</td>
<td></td>
<td>2 x 14,898</td>
<td>2 x 18,301</td>
<td>2 x 20,780</td>
</tr>
<tr>
<td>SMCR speed rpm</td>
<td></td>
<td>88.1</td>
<td>90.5</td>
<td>88.0</td>
</tr>
<tr>
<td>Main engine (without PTO)</td>
<td></td>
<td>2 x 5S70ME-C-Mk 7</td>
<td>2 x 6S70ME-C-Mk 7</td>
<td>2 x 7S70ME-C-Mk 7</td>
</tr>
<tr>
<td>Ballast draught m</td>
<td>m</td>
<td>9.7</td>
<td>9.9</td>
<td>10.3</td>
</tr>
<tr>
<td>Average engine load in SMCR</td>
<td>%</td>
<td>68</td>
<td>68</td>
<td>68</td>
</tr>
</tbody>
</table>

### Table V: Average ship particulars used for propulsion power prediction calculations for LNG carriers of the membrane type

<table>
<thead>
<tr>
<th>Case</th>
<th>Unit</th>
<th>A</th>
<th>B</th>
<th>C</th>
</tr>
</thead>
<tbody>
<tr>
<td>Ship capacity m³</td>
<td></td>
<td>150,000</td>
<td>210,000</td>
<td>250,000</td>
</tr>
<tr>
<td>Scantling deadweight dwt</td>
<td></td>
<td>80,000</td>
<td>108,000</td>
<td>129,000</td>
</tr>
<tr>
<td>Scantling draught m</td>
<td>12.3</td>
<td>12.7</td>
<td>12.7</td>
<td></td>
</tr>
<tr>
<td>Average design ship speed</td>
<td>knot</td>
<td>20.0</td>
<td>20.0</td>
<td>20.0</td>
</tr>
<tr>
<td>Design deadweight dwt</td>
<td></td>
<td>74,000</td>
<td>98,500</td>
<td>118,000</td>
</tr>
<tr>
<td>Light weight of ship t</td>
<td>30,000</td>
<td>40,000</td>
<td>48,000</td>
<td></td>
</tr>
<tr>
<td>Design displacement of ship t</td>
<td></td>
<td>104,000</td>
<td>138,500</td>
<td>166,000</td>
</tr>
<tr>
<td>Design draught m</td>
<td>11.6</td>
<td>12.0</td>
<td>12.0</td>
<td></td>
</tr>
<tr>
<td>Length overall m</td>
<td>288</td>
<td>315</td>
<td>345</td>
<td></td>
</tr>
<tr>
<td>Length between perpendiculars m</td>
<td></td>
<td>275</td>
<td>303</td>
<td>332</td>
</tr>
<tr>
<td>Breadth m</td>
<td>44.2</td>
<td>50.0</td>
<td>54.0</td>
<td></td>
</tr>
<tr>
<td>Breadth/design draught ratio</td>
<td></td>
<td>3.81</td>
<td>4.17</td>
<td>4.50</td>
</tr>
<tr>
<td>Block coefficient, perpendicular</td>
<td></td>
<td>0.720</td>
<td>0.743</td>
<td>0.753</td>
</tr>
<tr>
<td>Sea margin %</td>
<td>15</td>
<td>15</td>
<td>15</td>
<td></td>
</tr>
<tr>
<td>Engine margin %</td>
<td>10</td>
<td>10</td>
<td>10</td>
<td></td>
</tr>
<tr>
<td>Light running margin %</td>
<td>5</td>
<td>5</td>
<td>5</td>
<td></td>
</tr>
</tbody>
</table>

Table IV: Results of propulsion power prediction calculations for LNG carriers of the membrane type
Table V: Average ship particulars used for propulsion power prediction calculations for LNG carriers of the membrane type
### Table VI: Electrical power consumption for reliquefaction

<table>
<thead>
<tr>
<th>Load scenario</th>
<th>Reliquefaction (kW)</th>
<th>Other consumers (kW)</th>
<th>Total electricity consumption (kW)</th>
</tr>
</thead>
<tbody>
<tr>
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Table VI: Electrical power consumption for reliquefaction
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<th>Gas compressor (kW)</th>
<th>Other consumers (kW)</th>
<th>Total electricity consumption (kW)</th>
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<td>2100</td>
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Table VII: Electrical power consumption for ME-GI
### Table VII: Relative NPV expenses of the 210,000 m³ LNG carrier over a 20-year time horizon

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<tr>
<th>Fuel Price 1</th>
<th>ME + DG</th>
<th>ME + TES + DG</th>
<th>ME + PTO + DG</th>
<th>ME + TES + PTO + DG</th>
<th>ME + DG</th>
<th>ME + TES + DG</th>
<th>ME + PTO + DG</th>
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<tbody>
<tr>
<td>HFO (US$/ton)</td>
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<td>317.5</td>
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<td>297.5</td>
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</table>

TES = Thermo Efficiency System, PTO = Power Take Off, DG = Diesel Gas, NPV = Net Present Value (see Fig. 21)
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