Ice Classed Ships
Main Engines
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Introduction

Many merchant ship types are built for a given ice class notation which depends on the classification society and on the ice form and thickness during winter operation.

Building a ship for an ice class for winterisation means for example that the hull has to be thicker with stronger girders, beams and bulkheads which, of course again, depend on the degree of ice class.

In general, for a normal ship, the installed propulsion power of the main engine needed to obtain the required ship speed in service might also be sufficient for propulsion conditions during wintertime.

However, sometimes, depending on the form and thickness of the ice, and thereby the ice class required, together with the lowest appearing ambient air temperature at winter operation, some increased demands for the main engine have to be met. The extra demands for the main engine are described in this paper.

The application of ice classed ships might probably also be intensified because the global warming is shrinking the Arctic ice pack which, in the future, may open new sea routes, as for example the Northeast and Northwest passages, see Fig. 1, Ref. [1].

This paper does not describe ice class requirements for special ice-going ships like icebreakers, but only merchant ships, and it does not go into detail with hull design, etc.

As the focal point for the description of the ice classed ships, the Finnish-Swedish Ice Classes have been used.

Fig. 1: Possible sailing routes of the northwest and northeast passages (28 August 2012)
Ice Classes and Requirements

Ships with an ice class have a strengthened hull to enable them to navigate through sea ice. Depending on the class, sea chests, i.e. the openings in the hull for seawater intake, have to be properly arranged in order to avoid blocking up with ice. Most of the stronger classes require several forms of rudder and propeller protection, and strengthened propeller tips are often required.

Different ice classes and types exist, depending on the classification societies, but the most often ice class referred to is the Finnish-Swedish ice class.

Finnish-Swedish ice class designation

These ice class rules are divided into four types of ice classes based on ice strengthening hull structural design required of ships for navigating in ice, see Ref. [2].

The design requirements for ice classes are a minimum speed of 5 knots in the below stated brash ice channels:

<table>
<thead>
<tr>
<th>Ice class</th>
<th>Ice thickness</th>
</tr>
</thead>
<tbody>
<tr>
<td>1A Super</td>
<td>1.0 m and a 0.1 m thick consolidated layer of ice</td>
</tr>
<tr>
<td>1A</td>
<td>1.0 m</td>
</tr>
<tr>
<td>1B</td>
<td>0.8 m</td>
</tr>
<tr>
<td>1C</td>
<td>0.6 m</td>
</tr>
</tbody>
</table>

It is anticipated that ice ramming does not take place. Most ice-classed ships have the 1C notation. Classes 1A, 1B and 1C are assumed to rely on ice breaker assistance but normally not for 1A Super. International classification societies have incorporated the Finnish-Swedish ice class rules to their own rule books and offer equivalent ice class notations that are recognised by the Finnish-Swedish authorities, see for example Tables 1 and 2.

DNV ice class Ice-1A is for example accepted by the authorities as being equivalent to the Finnish-Swedish ice class 1A.

Other ice classes

However, the ice class designations and requirements applied by the different classification societies may sometimes be different from each other, see Ref. [3]. Thus, when ordering a 1A Super newbuilding you may have:

- Finnish-Swedish rules
  5 knots in 1.0 m brash ice with 0.1 m frozen top layer
- Lloyd's Register
  5 knots in 0.3 m solid ice with snow on top
- American Bureau of Shipping
  Only brash ice
- Det Norske Veritas
  Ship specific.

Some examples of ice class rules and equivalents are shown in Table 1, Ref. [3], and Table 2, Ref. [4]. In Table 1, the ice classes mentioned are anticipated to be without ice ramming. In Table 2, the old DNV (Det Norske Veritas) POLAR rules for very heavy ice conditions are based on occasional ice ramming occurring.

Vessels operating in very tough arctic ice conditions with operation in multi year ice may also be constructed to meet the Russian Register of Shipping ice class rules, for example when sailing to the new/coming oil and gas area in the arctic region in northwest Siberia.

New polar ice class rules

In addition to the above, in 2008 IACS (International Association of Classification Societies) issued unified rules for polar class vessels, PC1 through PC7, where PC1 is the highest one.

In effect since January 2012, DNV has issued their similar polar classes PC-1 through PC-7.

The new polar class rules, see Table 3, are divided into seven classes and is intended to replace the existing DNV arctic rules. PC-1 is the highest ice class valid for a ship operating anywhere in the polar oceans at any time of the year, whereas the lowest classes PC-7 and PC-6 are
intended for summer/autumn operation in the first year ice and are comparable with the Finnish-Swedish ice classes 1A and 1A Super (1A*), respectively. For information, PC-6 and PC-7 are also shown in parenthesis in Table 2.

<table>
<thead>
<tr>
<th>DNV Class Notations</th>
<th>Equivalent Baltic (Finnish-Swedish) Ice class</th>
<th>Vessel Type</th>
<th>Ice Condition</th>
<th>Impact Limits</th>
</tr>
</thead>
<tbody>
<tr>
<td>ICE-C</td>
<td></td>
<td></td>
<td>Very light ice condition</td>
<td></td>
</tr>
<tr>
<td>ICE-1C</td>
<td>1C</td>
<td>All ship types</td>
<td>First year ice and Broken channel:</td>
<td>No ramming</td>
</tr>
<tr>
<td>ICE-1B</td>
<td>1B</td>
<td></td>
<td>0.4 m ice thickness</td>
<td></td>
</tr>
<tr>
<td>ICE-1A (PC-7)</td>
<td>1A</td>
<td></td>
<td>0.8 m ice thickness</td>
<td></td>
</tr>
<tr>
<td>ICE-1A* (PC-6)</td>
<td>1A Super</td>
<td></td>
<td>1.0 m ice thickness</td>
<td></td>
</tr>
<tr>
<td>ICE-1A* F</td>
<td></td>
<td></td>
<td>1.0 m ice thickness</td>
<td></td>
</tr>
<tr>
<td>ICE-05</td>
<td></td>
<td></td>
<td>First year ice with pressure ridges</td>
<td></td>
</tr>
<tr>
<td>ICE-10</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>ICE-15</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>POLAR-10</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>POLAR-20</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>POLAR-30</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>ICEBREAKER</td>
<td></td>
<td></td>
<td>Icebreaking is main purpose</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>First year ice with pressure ridges</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

Table 2: DNV rules and other requirements – hull

<table>
<thead>
<tr>
<th>Baltic (Finnish-Swedish) Ice class</th>
<th>Polar Ice Class</th>
<th>Operating conditions</th>
<th>Winterised</th>
</tr>
</thead>
<tbody>
<tr>
<td>PC-1</td>
<td>All ice-covered waters year-round</td>
<td>ARCTIC</td>
<td></td>
</tr>
<tr>
<td>PC-2</td>
<td>Moderate multi-year ice-year-round</td>
<td></td>
<td></td>
</tr>
<tr>
<td>PC-3</td>
<td>2nd year ice year-round</td>
<td></td>
<td></td>
</tr>
<tr>
<td>PC-4</td>
<td>Thick 1st year ice-year-round</td>
<td>COLD</td>
<td></td>
</tr>
<tr>
<td>PC-5</td>
<td>Medium 1st year ice-year-round</td>
<td></td>
<td></td>
</tr>
<tr>
<td>ICE-1A*</td>
<td>PC-6</td>
<td>Medium 1st year ice summer/autumn</td>
<td></td>
</tr>
<tr>
<td>ICE-1A</td>
<td>PC-7</td>
<td>Thin 1st year ice summer/autumn</td>
<td>BASIC</td>
</tr>
<tr>
<td>ICE-1B</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>ICE-1C</td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

Table 3: DNV new polar classes PC-1 through PC-7
Temperature Restrictions and Load-up Procedures at Start of MAN B&W Two-stroke Main Engine [5]

In order to protect the engine against cold corrosion attacks on the cylinder liners, some minimum temperature restrictions and load-up procedures have to be considered before starting the engine.

Load-up procedures described below are valid for MAN B&W two-stroke engines with a cylinder bore greater than or equal to 80 cm, and may with benefit also be applied on engines with a smaller bore. However, if needed, the existing load-up programme recommendation (from 90% to 100% in 30 minutes) is still valid for engines with bore sizes from 70 cm and down.

Note: the recommendations described below are based on the assumption that the engine has already been well run in.

Start of warm engine – normal load-up procedures

As a summary, the load-up procedures recommended for normal start of engine are shown in Table 4a.

Fixed pitch propellers

Normally, a minimum engine jacket water temperature of 50°C is recommended before the engine may be started and run up gradually up to 80%, and slowly from 80% to 90% of specified MCR load (SMCR power) during 30 minutes.

Controllable Pitch Propellers

For running-up between 90% and 100% of SMCR rpm, it is recommended that the speed be increased slowly over a period of 60 minutes.

Recommended start of engine at normal very low engine load operation

For engines normally running at 10% to 40% engine low load operation, an extra slow load-up procedure is recommended compared with load-up procedures described above, and is also shown in Table 4a.

Start of cold engine – exceptional load-up procedures

As a summary, the load-up procedures recommended for exceptional start of a cold engine are shown in Table 4b.

Fixed pitch propellers

In exceptional circumstances where it is not possible to comply with the above-

<table>
<thead>
<tr>
<th>Required jacket water temperature at normal start of engine: minimum 50°C</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>FPP:</strong> Fixed Pitch Propeller</td>
</tr>
<tr>
<td><strong>CPP:</strong> Controllable Pitch Propeller</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>1. at normal engine load operation</th>
</tr>
</thead>
<tbody>
<tr>
<td>A. Run up slowly</td>
</tr>
<tr>
<td>Minimum temp. 50°C</td>
</tr>
<tr>
<td>FPP – From 0% up to 80% SMCR speed</td>
</tr>
<tr>
<td>CPP – From 0% up to 50% SMCR power</td>
</tr>
<tr>
<td>B. Run up slowly (minimum 30 min.)</td>
</tr>
<tr>
<td>FPP – From 80% up to 90% SMCR speed</td>
</tr>
<tr>
<td>CPP – From 50% up to 75% SMCR power</td>
</tr>
<tr>
<td>C. Run up slowly (minimum 60 min.)</td>
</tr>
<tr>
<td>FPP – From 90% up to 100% SMCR speed</td>
</tr>
<tr>
<td>CPP – From 75% up to 100% SMCR power</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>2. at normal very low engine load operation</th>
</tr>
</thead>
<tbody>
<tr>
<td>A. Run up slowly</td>
</tr>
<tr>
<td>If normally 10% to 40% engine low load operation (slide fuel valves needed) extra slowly load-up procedure is recommended: minimum 30 min. from 10% to 40% load and minimum 60 min. from 40% to 75% load</td>
</tr>
</tbody>
</table>

Table 4a: Temperature restrictions and load-up procedures at normal start of engine
mentioned normal recommendations, a minimum of 20°C can be accepted before the engine is started and run up slowly to 80% of SMCR rpm.

Before exceeding 80% SMCR rpm, a minimum jacket water temperature of 50°C should be obtained before the normal start load-up procedure described above may be continued.

### Controllable Pitch Propellers

In exceptional circumstances where it is not possible to comply with the above-mentioned normal recommendations, a minimum of 20°C can be accepted before the engine is started and run up slowly to 50% of SMCR power.

Before exceeding 50% SMCR power, a minimum jacket water temperature of 50°C should be obtained before the above-mentioned normal start load-up procedure may be continued.

The time period required for increasing the jacket water temperature from 20°C to 50°C depends on the amount of water in the jacket cooling water system, and on the engine load.

### Preheating during standstill periods

During short stays in ports (i.e. less than 4-5 days), it is recommended to keep the engine preheated, the purpose being to prevent temperature variations in the engine structure and corresponding variations in thermal expansions, and thus the risk of leakages.

The jacket cooling water outlet temperature should be kept as high as possible (max. 75-80°C), and should – before start-up – be increased to at least 50°C, either by means of the auxiliary engine cooling water, or by means of a built-in preheater in the jacket cooling water system, or a combination of both.

A standard preheater system with a built-in preheater is shown in Fig. 2.

The circulating water flow is divided into two branches, one going through the engine and one going through the cooling water system outside the engine. As the arrows indicate, the preheater water flows in the opposite direction through the engine, compared with the main jacket water flow. As the water inlet is at the top of the engine, the engine preheating is more effective in this way.

![Fig. 2: Preheating of jacket cooling water system](image)

**Start of cold engine (exceptional load-up procedures)**

**Required jacket water temperature at start of cold engine: minimum 20°C**

<table>
<thead>
<tr>
<th>Engine Type</th>
<th>Load-Up Procedure</th>
<th>Temp. Requirement</th>
</tr>
</thead>
<tbody>
<tr>
<td>FPP</td>
<td>Run up slowly</td>
<td>Minimum 20°C</td>
</tr>
<tr>
<td>CPP</td>
<td>Run up slowly</td>
<td>Minimum 50°C</td>
</tr>
<tr>
<td>CPP</td>
<td>Run up slowly</td>
<td>Minimum 50°C</td>
</tr>
</tbody>
</table>

**Recommended start of engine at normal engine load operation**

<table>
<thead>
<tr>
<th>Engine Type</th>
<th>Load-Up Procedure</th>
<th>Temp. Requirement</th>
</tr>
</thead>
<tbody>
<tr>
<td>FPP</td>
<td>From 0% up to 80% SMCR speed</td>
<td></td>
</tr>
<tr>
<td>CPP</td>
<td>From 0% up to 50% SMCR power</td>
<td></td>
</tr>
<tr>
<td>FPP</td>
<td>From 80% up to 90% SMCR speed</td>
<td></td>
</tr>
<tr>
<td>CPP</td>
<td>From 50% up to 75% SMCR power</td>
<td></td>
</tr>
<tr>
<td>FPP</td>
<td>From 90% up to 100% SMCR power</td>
<td></td>
</tr>
<tr>
<td>CPP</td>
<td>From 75% up to 100% SMCR power</td>
<td></td>
</tr>
</tbody>
</table>

**Table 4b: Temperature restrictions and load-up procedures at start of cold engine in exceptional cases**

---

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The preheater operation is controlled by a temperature sensor after the preheater.

**Preheater capacity**

When a preheater is installed in the jacket cooling water system, as shown in Fig. 2, the preheater pump capacity should be about 10% of the jacket water main pump capacity. Based on experience, it is recommended that the pressure drop across the preheater should be approx. 0.2 bar. The preheater pump and the jacket water main pump should be electrically interlocked to avoid the risk of simultaneous operation.

The preheater capacity depends on the required preheating time and the required temperature increase of the engine jacket water. The temperature and time relationship is shown in Fig. 3. The relationship is almost the same for all engine types.

If a temperature increase of for example 35°C (from 15°C to 50°C) is required, a preheater capacity of about 1% of the engine’s nominal MCR power is required to obtain a preheating time of 12 hours.

When sailing in arctic areas, the required temperature increase may be higher, possibly 45°C or even higher, and therefore a larger preheater capacity is required, as shown with dotted lines in Fig. 3.

The curves are based on the assumption that, at the start of preheating, the engine and engine room are of equal temperatures. It is assumed that the temperature will increase uniformly all over the engine structure during preheating, for which reason steel masses and engine surfaces in the lower part of the engine are also included in the calculation.

The results of the preheating calculations may therefore be somewhat conservative.

**Engine Room Ventilation [5]**

In addition to providing sufficient air for combustion purposes in the main engine, auxiliary diesel engines, fuel fired boiler, etc., the engine room ventilation system should be designed to remove the radiation and convection heat from the main engine, auxiliary engines, boilers and other components.

A sufficient amount of ventilation air should be supplied and exhausted through suitably protected openings arranged in such a way that these openings can be used in all weather conditions. Care should be taken to ensure that no seawater can be drawn into the ventilation air intakes.

Furthermore, the ventilation air inlet should be placed at an appropriate distance from the exhaust gas funnel in order to avoid the suction of exhaust gas into the engine room.

Major dust and dirt particles can foul air coolers and increase the wear of combustion chamber components. Accordingly, the air supplied to the engine must be cleaned by appropriate filters. The size of particles passing through the air intake filter should not exceed 5 µm.

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![Fig. 3: Preheating of MAN B&W two-stroke diesel engine.](image-url)

The temperature increase and corresponding preheating time curves are shown for the different preheater sizes indicated in % of nominal MCR power.
An example of a standard engine room ventilation system, where ventilation fans blow air into the engine room via air ducts, is shown in Fig. 4.

However, the capacity of the ventilation system according to ISO 8861:1998 (E) has to be at least 150% of the air consumption of the main engine, auxiliary engines, etc. at 100% SMCR.

Therefore, for arctic operation with very low ambient air temperatures, the application of a direct inlet air suction system for the combustion air to the main engine itself, could be an advantage in order to avoid too low air temperature in the engine room. This means that the engine room itself is fitted with its own separate air ventilation system and fans.

**Design Recommendations of MAN B&W Two-stroke Main Engine for Operation at Extremely Low Air Temperature [5]**

When a standard ambient temperature matched main engine on a ship operates under arctic conditions with low turbocharger air intake temperatures, the density of the air will be too high. As a result, the scavenge air pressure, the compression pressure and the maximum firing pressure will be too high. In order to prevent such excessive pressures under low ambient air temperature conditions, the turbocharger air inlet temperature should be kept as high as possible (by heating, if possible).

Furthermore, the scavenge air coolant (cooling water) temperature should be kept as low as possible and/or the engine power in service should be reduced.

However, for an inlet air temperature below approx. -10°C, some engine design precautions have to be taken.

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Fig. 4: Standard engine room ventilation system
Main precautions for extreme low air temperature operation, arctic exhaust gas bypass

With a load-dependent arctic exhaust gas bypass system (standard MAN Diesel & Turbo recommendation for extreme low air temperature operation), as shown in Fig. 5, part of the exhaust gas bypasses the turbocharger turbine, giving less energy to the compressor, thus reducing the air supply and scavenge air pressure to the engine.

For the electronically controlled ME engine (ME/ME-C/ME-B), the load-dependent bypass control will be incorporated in the Engine Control System (ECS).

Engine load, fuel index and scavenge air pressure signals are already available for the ME software and, therefore, additional measuring devices are not needed for ME engines.

In general, a turbocharger with a normal layout can be used in connection with an exhaust gas bypass. However, in a few cases a turbocharger modification may be needed.

The exhaust gas bypass system ensures that when the engine is running at part load at low ambient air temperatures, the load-dependent scavenge air pressure is close to the corresponding pressure on the scavenge air pressure curve which is valid for ISO ambient conditions. When the scavenge air pressure exceeds the read-in ISO-based scavenge air pressure curve, the bypass valve will variably open and, irrespective of the ambient conditions, ensure that the engine is not overloaded. At the same time, it will keep the exhaust gas temperature relatively high.

Other low temperature precautions

Low ambient air temperature and low seawater temperature conditions come together. The cooling water inlet temperature to the lube oil cooler should not be lower than 10°C, as otherwise the viscosity of the oil in the cooler will be too high, and the heat transfer inadequate. This means that some of the cooling water should be recirculated to keep up the temperature.

Furthermore, to keep the lube oil viscosity low enough to ensure proper suction conditions in the lube oil pump, it may be advisable to install heating coils near the suction pipe in the lube oil bottom tank.

The following additional modifications of the standard design practice should be considered as well:

- Larger electric heaters for the cylinder lubricators or other cylinder oil ancillary equipment
- Cylinder oil pipes to be further heat traced/insulated
- Upgraded steam tracing of fuel oil pipes

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**Fig. 5: Standard load dependent low ambient air temperature arctic exhaust gas bypass system**
Increased preheater capacity for jacket water during standstill
- Different grades of lubricating oil for turbochargers
- Space heaters for electric motors
- Sea chests must be arranged so that blocking with ice is avoided.

**Ships with ice class notation**

For ships with the Finnish-Swedish ice class notation 1C, 1B, 1A and even 1A super or similar, most MAN B&W two-stroke diesel engines meet the ice class demands, i.e. there will normally be no changes to the main engines. This again means that the standard thrust bearings for most of the MAN B&W two-stroke engines are sufficient.

If the ship is with ice class notation 1A super and the main engine has to be reversed for going astern (FP propeller), the starting air compressors must be able to charge the starting air receivers within half an hour, instead of one hour, i.e. the compressors must be the double in size compared to normal.

For other special ice class notations, the engines need to be checked individually.

The exhaust gas bypass system to be applied is independent of the ice classes, and only depends on how low the specified ambient air temperature is expected to be. However, if the ship is specified with a high ice class like 1A super, it is advisable to make preparations for, or install, an exhaust gas bypass system.

**Increased steam production in wintertime**

During normal operation at low ambient temperatures, the exhaust gas temperature after the turbochargers will decrease by about 1.6°C for each 1.0°C reduction of the intake air temperature. The load-dependent exhaust gas bypass system will ensure that the exhaust gas temperature after the turbochargers will fall by only about 0.3°C per 1.0°C drop in the intake air temperature, thus enabling the exhaust gas boiler to produce more steam under cold ambient temperature conditions.

Irrespective of whether a bypass system is installed or not, the exhaust gas boiler steam production at ISO ambient conditions (25°C air and 25°C cooling water) or higher ambient temperature conditions, will be the same, whereas in wintertime the steam production may be relatively increased, as the scavenger air pressure is controlled by the bypass valve.

As an example, Fig. 6 shows the influence of the load-dependent exhaust gas by-pass system on the steam production when the engine is operated in wintertime, with an ambient air temperature of 0°C and a scavenger air cooling water temperature of 10°C.

The calculations have been made for a 6S60MC-C7/ME-C7 engine equipped with a high-efficiency turbocharger, i.e. having an exhaust gas temperature of 245°C at SMCR and ISO ambient conditions.

Fig. 6 shows that in wintertime, it is questionable whether an engine without a bypass will meet the ship’s steam demand for heating purposes (indicated for bulk carrier or tanker), whereas with a load-dependent exhaust gas bypass system, the engine can meet the steam demand.

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**Fig. 6: Expected steam production by exhaust gas boiler at winter ambient conditions (0°C) for main engine with and without a load-dependent low air temperature arctic exhaust gas bypass system**

**Steam production kg/h**

- **SMCR = 13,560 kW at 105 r/min**
- **Air intake temperature: 0°C**
- **Cooling water temperature: 10°C**

**Total steam production, with arctic exhaust gas bypass**

**Surplus steam**

**Total steam production, without bypass**

**Extra steam needed**

**Steam consumption**

**Engine shaft power**

0 50 100 150 200 250

40 50 60 70 80 90 100 % SMCR
Extended Main Engine Load Diagram

As described later, a Controllable Pitch propeller (CP propeller) may, with advantage, be applied for high ice classed ships.

However, because of the high efficiency and simplicity, a Fixed Pitch Propeller (FP propeller) may often be preferred for low ice classes.

When a ship with fixed pitch propeller is operating in normal sea service, it will in general be operating around the design propeller curve 6, as shown in the standard load diagram in Fig. 7.

Sometimes, when operating in heavy weather, the fixed pitch propeller performance will be more heavy running, i.e. for equal power absorption of the propeller, the propeller speed will be lower and the propeller curve will move to the left.

As the two-stroke main engines are directly coupled to the propeller, the engine has to follow the propeller performance, i.e. also in heavy running propeller situations. For this type of operation, there is normally enough margin in the load area between line 6 and the normal torque/speed limitation line 4, see Fig. 7. To the left of line 4 in torque-rich operation, the engine will lack air from the turbocharger to the combustion process, i.e. the heat load limits may be exceeded and bearing loads might also become too high.

Measurements show that the propeller curve at bollard pull (zero ship speed) will be approximately 15-20% heavy running, but of course depending on the propeller arrangement and ship type. This indicates the maximum size of heavy running operation when sailing in thick ice involving a very high torque on the propeller.

FP propeller and no ice ramming

For ships with special operating conditions, like occasionally operating in thick ice, it would be an advantage during normal operation conditions to be able to operate the propeller/main engine as much as possible close to line 6, but in ice situations with heavy running propeller inside the torque/speed limit, line 4.

For ships occasionally operating in heavy ice, the increase of the operating speed range between line 6 and line 4 of the standard load diagram may be carried out as shown in Fig. 8 for the extended load diagram for speed derated engine with increased light running.

The maximum speed limit (line 3) of the engines is normally 105% of the SMCR speed, as shown in Fig. 7.

However, for speed and, thereby, power derated engines it is possible to ex-

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**Fig. 7: Standard MAN B&W two-stroke engine load diagram**

- Line 1: Propeller curve through SMCR point (M)
  - layout curve for engine
- Line 2: Heavy propeller curve
  - fouled hull and heavy seas
- Line 3: Speed limit
- Line 4: Torque/speed limit
- Line 5: Mean effective pressure limit
- Line 6: Light propeller curve
  - clean hull and calm weather
  - layout curve for propeller
- Line 7: Power limit for continuous running
- Line 8: Overload limit
- Line 9: Sea trial speed limit
- Line 10: Constant mean effective pressure (mep) lines
tend the maximum speed limit to 105% of the engine's nominal L1 speed, line 3', see Fig. 8, but only provided that the torsional vibration conditions permit this. Thus, the shafting, with regard to torsional vibrations, has to be approved by the classification society in question, based on the extended maximum speed limit.

When choosing an increased light running to be used for the design of the propeller, the load diagram area may be extended from line 3 to line 3', as shown in Fig. 8, and the propeller/main engine operating curve 6 may have a correspondingly increased heavy running margin before exceeding the torque/speed limit, line 4.

A corresponding slight reduction of the propeller efficiency may be the result, due to the higher propeller design speed used.

**CP propeller and ice ramming**

When a ship with CP propeller is operating under ice ramming conditions, the running point on the combinatory curve of the CP propeller (could be on line 6) will suddenly change because of the ice ramming and move to the left in the load diagram. The reason is that there is some reaction time in changing the CP propeller pitch.

For such running conditions, the extended load diagram shown in Fig. 8 may also be useful for the main engine operation.

**Ice Class Demands for Propeller Type and Main Engine Power Output**

**Propulsion advantages with CP propeller**

Normally, and also valid for low ice classes, FP propellers are installed in merchant ships because of its simplicity and high efficiency. The propellers are cast in one block, and therefore the position of the blades, and hence the propeller pitch, is once and for all fixed with a given pitch that cannot be changed in operation. This means that when operating in, for example heavy weather and ice, the propeller performance curve will be very heavy (reduced speed for same power).

Compared to the FP propeller, for the CP propeller, the position of the blades, and thereby the propeller pitch, can be controlled to avoid heavy running and overload of the main engine. Therefore, CP propellers can, with advantage, be applied both for moderate ice classes as well as for very strong ice classes.

Ships designed for high ice classes with ice ramming are often based on diesel-electric propulsion with CP propellers.

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**Fig. 8: Extended load diagram for MAN B&W two-stroke speed derated engine with increased light running**
However, as long as the extent of the ice ramming operation is in a favourable proportion to the total operational time of the vessel, which will be the case for most commercial ships, a simple propulsion system consisting of a single low speed two-stroke main engine directly coupled to a ducted CP propeller will be an extremely reliable and cost-efficient propulsion system.

**Required minimum propulsion power output**

When sailing in ice with a bulk carrier or a tanker, the ship has to be ice classed for the given operating need of trading in coastal states with seasonal or year-round ice-covered seas.

Besides the safety of the hull structure under operation in ice, the minimum required propulsion power for breaking the ice has to be met.

Depending on the ice class rules and specific ice classes required for a ship, the minimum ice class required propulsion power demand may be higher or lower than the normal SMCR power used for an average bulk carrier or tanker without ice class notation.

The ice class rules most often used and referred to for navigation in ice are, as previously mentioned, the “Finnish-Swedish Ice Class Rules” valid for operation without ice ramming, see Ref. [2]. These rules are issued by the Finnish Maritime Administration and apply to all classification societies via IACS (International Association of Classification Societies).

**Existing ships with conventional main engines**

Based on the average bulk carriers and tankers before 2007, the minimum power demand, according to the formulae of the Finnish-Swedish ice classed ships, class 1A Super, 1A, 1B and 1C, has been estimated, see Fig. 9, Ref. [7] and Fig. 10, Ref. [8]. In general, the lowest ice classes, 1B and 1C can – power-wise – almost always be met.

However, the strongest classes, 1A Super and 1A, will require a higher propulsion power than the normally needed average SMCR power for bulk carrier and tankers without ice class notation.

Model tests have shown that the power found when using the above ice class

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**Fig. 9: Minimum required propulsion SMCR power demand (CP propeller) for existing average-size bulk carriers with Finnish-Swedish ice class notation (for FP propeller add +11%)**
formulae is often in excess of the real power needed for propulsion of the ship in ice. Furthermore, it has been concluded that the formulae can only be used within certain limitations of ship particulars and therefore Annex 1, listing the restrictions to the validity of the formulae, has been added to the rules.

Ships outside the limitations stipulated in Annex 1 have to be model tested individually, e.g. Capesize bulk carriers and Suezmax tankers longer than the max. limitation for overall ship length stated in Annex 1 (65.0 m < Loa < 250.0 m).

It is to be expected that many owners may choose to use model test in any case, and independent of the ship length, because the model test may show that a smaller engine can be installed than what can be calculated using the formulae.

**Future ships with modern main engines**

In the Finnish-Swedish ice class formula, the needed installed propulsion power is inverse proportional with the propeller diameter, i.e. the larger the propeller is, the lower the needed power will be.

Modern ships of the future may be installed with a high efficient propeller, i.e. with a 10-12% larger propeller diameter. This means a lower optimum propeller speed, and an ultra long stroke MAN B&W two-stroke main engine of the G-type to be installed, involving an about 5-8% higher total efficiency.

The larger propeller means that in the future, the ice class power demands in Figs. 9 and 10 will be 10-12% lower and easier to be met with the modern MAN B&W two-stroke G-engine types.

**EEDI corrections for ice classed ships**

The IMO (International Maritime Organisation) based Energy Efficiency Design Index (EEDI) is a mandatory index required on all new ships contracted after 1 January 2013. The index is used as an instrument to fulfill international requirements regarding \( \text{CO}_2 \) emissions on ships. EEDI represents the amount

![Fig. 10: Minimum required propulsion SMCR power demand (CP propeller) for existing average-size tanker with Finnish-Swedish ice class notation (for FP propeller add +11%)](image)
of CO₂ emitted by a ship in relation to the transported cargo and is measured in gram CO₂ per dwt per nautical mile.

The EEDI value is calculated on the basis of maximum cargo capacity (70% for container ships), propulsion power, ship speed, SFOC (Specific Fuel Oil Consumption) and fuel type. Depending on the date of contract, the EEDI is required to be a certain percentage lower than an IMO defined reference value depending on the type and capacity of the ship.

The main engine’s 75% SMCR (Specified Maximum Continuous Rating) figure is as standard applied in the calculation of the EEDI figure, in which also the CO₂ emission from the auxiliary engines of the ship is included.

However, for ice classed ships – particularly for ships with a high ice class notation – the EEDI found according to the above standard described method would give too high figures compared to reality. The main reason is that most of the year main engines are sailing in normal weather conditions without ice and at relatively low part load conditions compared to a standard ship.

Therefore, at present, the IMO is preparing some correction factors to the EEDI calculations valid for ice classed ships. The corrections will, depending on the ice class in question, among others, compensate for high main engine power and loss of the ship capacity (dwt).

The corrections are made for the Finnish-Swedish ice class rules, 1A Super, 1A, 1B and 1C.

Low Load Operation and Service
Optimisation of MAN B&W Two-stroke Main Engines

An ice classed ship with high ice class has a relatively high demand to the SMCR power (max. power) caused by the extra power margin needed for sailing in ice.

However, most of the time, the ship will normally be operating in ice-free areas involving that the main engine in normal sea service may operate at low load. Therefore, in such cases a low load optimisation of the main engine might be a good idea.

Fuel consumption and optimisation possibilities

The current economic scenario has placed more emphasis on operational flexibility in terms of demand for improved part-load and low-load SFOC. As described below, different optimisation possibilities for the MAN B&W two-stroke engines have been developed.

NOₓ regulations place a limit on the SFOC on two-stroke engines. In general, NOₓ emissions will increase if SFOC is decreased and vice versa. In the standard configuration, the engines are optimised close to the IMO NOₓ limit and, therefore, NOₓ emissions may not be further increased.

The IMO NOₓ limit is given as a weighted average of the NOₓ emission at 25, 50, 75 and 100% load. This relationship can be utilised to tilt the SFOC profile over the load range. This means that SFOC can be reduced at part load or low load at the expense of a higher SFOC in the high-load range without exceeding the IMO NOₓ limit.

Only high-load optimisation is available for engines with conventional efficiency turbochargers (64% instead of 67%) and non-adjustable maximum firing pressure at part load (MC engines without VIT).

Optimisation of SFOC in the part-load (50-85%) or low-load (25-70%) range requires selection of a tuning method:

- ECT: Engine Control Tuning (only available on ME/ME-C engines)
- VT: Variable Turbine Area
- EGB: Exhaust Gas Bypass.

Furthermore, a turbocharger cut-out method is available for SFOC reduction at part/low load operation.

Particularly, SFOC optimisation in the low-load range for below two tuning methods are recommended and explained in the following:

- Exhaust gas bypass (EGB)
- Turbocharger (TC) cut-out

Exhaust gas bypass (EGB) tuning

Particularly, the above-mentioned EGB tuning method for ‘low load’ operation in normal sea service is applicable for ships installed with relatively high main engine power caused by the power demands to a high ice classed ship.

This method requires installation of EGB technology. The turbocharger(s) on the engine is/are matched at 100% load with fully open EGB. At approximately 85% load, the EGB starts to close and is fully closed below 70% load. With this technology, SFOC is decreased at low load, at the expense of a higher SFOC at high load, see Fig. 11a.
An SFOC reduction of 5 g/kWh for low load optimisation makes it possible to obtain a fuel cost reduction of up to approx. 3% of the specific consumption. The daily consumption will of course be reduced further due to the low load.

**Turbocharger (TC) cut-out**

Besides the above-mentioned EGB low load method, cut-out of one turbocharger can be applied on MAN B&W two-stroke engines with more than one turbocharger. The cut-out can be effected either by means of blind plates or pneumatically actuated valves.

During cut-out, the maximum acceptable engine load is limited to 65% and 75% of SMCR power for engines with 3 or 4 turbochargers, respectively. TC cut-out cannot, as standard, be combined with other methods of low or part load SFOC optimisations. Examples of SFOC reductions on low load operation with one TC cut-out is shown in Fig. 11b. Turbochargers of equal sizes have been anticipated in the examples. TC cut-out with 1 of 2 TCs is also possible, and also with different TC sizes.

The cut-out will enhance the performance of the working turbochargers and, thereby, lead to higher scaveng, compression and maximum combustion pressures, ultimately resulting in lower SFOC and lower exhaust gas temperatures and amount.

Data for changes in SFOC, exhaust gas temperature and amount can be supplied on request for the actual project. Depending on the specific engine layout, the heat load can increase significantly when running close to the reduced limit for maximum acceptable engine load.
Propulsion Systems Applied and Example

Ice classes without ramming
For ice classed ships anticipated to be without ice ramming, the standard diesel-mechanical propulsion systems for merchant ships can be applied, i.e. with Controllable Pitch propeller (CP propeller) or with Fixed Pitch propeller (FP propeller) directly coupled to an MAN B&W two-stroke main engine, see Fig. 12a.

Depending on the ice class in question, the main engine thrust bearing has to be designed accordingly. For most of the MAN B&W two-stroke engines, the standard thrust bearing is sufficient.

Ice classes with ramming
The ramming on ice may involve occasional high torque on the propulsion system and, therefore, the diesel-electric system with CP propeller may often be preferred, as the electric motor is suitable for high torque deviations, see Fig. 12b.

However, such a propulsion system has a lower efficiency (11-12%) compared with a propulsion system with CP propeller directly coupled to an MAN B&W two-stroke engine. Therefore, as the major time in ship operation is often in normal sea service without ice, alternative to the conventional diesel-electric propulsion system might be preferred.

Example of a bulk carrier with ice class DNV POLAR-15 propelled by an MAN B&W two-stroke main engine
A 31,500 dwt bulk carrier, delivered in 2006, has been designed to comply with DNV POLAR-15 (comparable with the new DNV PC-4) for sailing with nickel concentrate from a mine in Labrador, Canada.

The high ice class means that the ship is able to sail unsupported through 1.5 m thick ice which is a necessary ability for sailing to Labrador in the wintertime.

Where a normal bulk carrier of this size had an installed 6S50ME-C7 main engine with SMCR = 9,480 kW × 127 r/
min, the POLAR-15 classed bulk carrier was delivered with a 7S70ME-C7 with SMCR = 21,770 kW × 91 r/min. The larger bore caused by the extra power demand of the ice class, and the lower speed because of a larger propeller diameter.

The main special feature of ships classified for POLAR-15 is the capability of ice ramming. In short, the ramming procedure consists of sailing with a specified speed through the ice, until the ship is stopped by the resistance of the ice.

The ship is sailed astern to come free of the packed ice, and then is sailed full ahead into the ice, to break through the ice until the ship stops again by the resistance of the ice. The procedure is used for thick ice and the ice ridges, which put some very unusual demands on the main engine.

For the ship to be able to sail ahead and astern within a short time cycle, the ship is equipped with a CP propeller. With this, it is not necessary to reverse the main engine, which can be a time-consuming task. The CP propeller is furthermore enclosed in a nozzle, both for protection of the propeller against blocks of ice and for extra thrust.

Because the ship is occasionally used for ramming ice, the load of the main engine will cycle up and down. The engine will be highly loaded when breaking the ice, and low loaded through the pitch reversal periods for the propeller. The engine will also be high loaded when the ship is sailed astern and once again sailing ahead to ram the ice. The engine, therefore, has an extended load diagram.

The procedure for ramming the ice can occur up to 10 times per hour, which will set extra demand on the turbochargers and auxiliary blowers. Because the load will come below 25% SMCR every time, the sailing direction of the ship is changed, the auxiliary blowers will have to start up and make sure that there is a sufficient pressure on the scavenge air. The auxiliary blowers are therefore designed to cope with up to 20 starts per hour.

Ice ramming also poses some other stresses on the propulsion system, as the dynamic loading will be different from normal propulsion mode. The thrust bearing on the main engine will have to handle the propeller thrust including the dynamic loading from the ice operation, which has led to a modification in the thrust bearing including the engine structure around the thrust bearing, the journal bearing support and the thrust collar.
The crankshaft thrust cam has been modified compared to the normal 7S70ME-C7. These modifications have led to a better distribution of the bearing load, so an increase of the maximum load has been avoided compared to the standard thrust bearing configuration in a standard ship operating in open water. The crankshaft design has also been checked against the torsional load plus the impact from propeller blades hitting the ice.

The ship described above went on sea trial in 2006 and good performance has been logged through intensive ice trials.

**Closing Remarks**

As this paper shows, MAN B&W two-stroke engines, installed as main engines in ocean-going ships, are able to operate both in open waters and in very cold and even in arctic areas in ice conditions (no ice ramming) without any problems, as long as special low temperature and heavy running precautions have been taken.

Even in case of partly ice ramming conditions valid for the high ice classes, an MAN B&W two-stroke engine directly coupled to a CP propeller, can be applied as prime mover.

Furthermore, special fuel consumption optimisation methods valid for the MAN B&W two-stroke engines like ‘exhaust gas bypass low load tuning’ and ‘turbocharger cut-out’ is available. These methods may with advantages be applied when operating at normal sea service conditions (without ice) where the required engine load is relatively low, and thereby may reduce the specific fuel oil consumption of the main engine.

Table 5 on page 23 describes as a short executive summary, the installation recommendations mentioned in this paper of different ice class notations.
# Installation recommendations for ice classed ships

Ok = Standard is ok  
R = Recommended  
N = Not recommended  
C = Check  

<table>
<thead>
<tr>
<th>Propeller types:</th>
<th>Normally used ice classes</th>
<th>High ice classes</th>
</tr>
</thead>
<tbody>
<tr>
<td>FP = Fixed pitch</td>
<td>Equivalent Finnish-Swedish ice classes 1C, 1B, 1A</td>
<td>Equivalent Finnish-Swedish ice class 1A Super</td>
</tr>
<tr>
<td>CP = Controllable pitch</td>
<td>Normally used ice ramming</td>
<td>Higher ice classes than equivalent Finnish-Swedish</td>
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<td>Without ice breaking</td>
<td>With ice breaking</td>
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<td>R</td>
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<tr>
<td>Engine torque</td>
<td>Ok/C</td>
<td>Ok/C</td>
</tr>
<tr>
<td>Crankshaft</td>
<td>Ok/C</td>
<td>Ok/C</td>
</tr>
<tr>
<td>Thrust bearing</td>
<td>Ok/C</td>
<td>Ok/C</td>
</tr>
<tr>
<td>Extended load diagram</td>
<td>N/C</td>
<td>N/C</td>
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<td>Ok</td>
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<tr>
<td>Starting air system</td>
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<td>Ok</td>
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<tr>
<td>Part flow load tuning</td>
<td>(R/R)</td>
<td>(R/R)</td>
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<tr>
<td>or turbocharger cut-out</td>
<td>N</td>
<td>N</td>
</tr>
<tr>
<td>For air temp &lt; −10°C</td>
<td>Arctic exhaust gas bypass</td>
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<td></td>
<td>Jacket water preheater increased</td>
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<td>Direct air suction (air ventilation)</td>
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<td>Lube oil systems</td>
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<td></td>
<td>Steam tracing of fuel oil pipes</td>
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<td>Diesel-electric propulsion system</td>
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<td>Propeller type recommended</td>
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<td>Four-stroke main engine - geared</td>
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<td>Seawater chests</td>
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</table>

Table 5: Installation recommendations for ice classed ships
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