Influence of Ambient Temperature Conditions on Main Engine Operation

Contents:

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Influence of Ambient Temperature Conditions on Main Engine Operation

Introduction

Diesel engines used as prime movers on ships are exposed to the varying climatic temperature conditions that prevail in different parts of the world, and must therefore be able to operate under all ambient conditions from winter to summer and from arctic to tropical areas.

As the temperature variations on the surface of the sea are rather limited, the diesel engine will not normally be exposed to really extreme temperatures. However, the changes that do occur in the ambient conditions will, among other things, cause a change in the specific fuel oil consumption, the exhaust gas amount and the exhaust gas temperature of the diesel engine. These changes are already described in our Project Guides and will therefore not be discussed in this paper.

Also the scavenge air, compression and maximum firing pressures of the diesel engine will change with climatic changes and, at very low ambient air temperatures, unrestricted engine operation requires adjustments of individual engine parameters.

This paper describes our recommendations on engine start-up, the supply of ventilation air to the engine room and engine operation under normal, high and extremely low ambient temperature conditions.

The paper is divided into three chapters which, in principle, may be read independently of each other, and all of which have the ambient air temperature as a common parameter.

The three chapters are entitled:

- Temperature Restrictions at Start of Engine
- Engine Room Ventilation
- Main Engine Operation under Normal, High and Extremely Low Ambient Temperature Conditions.

Chapter 1

Temperature Restrictions at Start of Engine

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

Normal start of engine

It is recommended that the engine jacket cooling water be heated to a minimum of 50°C before the engine is started and gradually run up to 90% of specified MCR (SMCR) speed for Fixed Pitch Propeller (FPP) plants and 80% pitch for Controllable Pitch Propeller (CPP) plants.

For further running up to 100% SMCR speed/pitch, it is recommended that the load be increased slowly over a period of 30 minutes or more.

Start of cold engine

Where it is not possible to comply with the above-mentioned recommendation, a minimum jacket cooling water temperature of 20°C can be accepted before the engine is started and run up slowly to 90% SMCR speed (FPP) or 80% pitch (CPP), respectively.

The engine must not be started before the jacket water has been preheated to at least 20°C.

Before exceeding 90% SMCR speed (FPP) or 80% pitch (CPP), a minimum engine jacket water temperature of 50°C should be obtained.

When the jacket cooling water temperature reaches a minimum of 50°C, the load is allowed to be increased slowly up to 100% SMCR speed (FPP) or 100% pitch (CPP) over a period of at least 30 minutes.

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

Note: The above recommendations are based on the assumption that the engine has already been well run-in.

Preheating during standstill periods

During short stays in port (i.e. less than 4-5 days), it is recommended that the engine is kept 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 starting-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.

Jacket cooling water systems with a built-in preheater

For two different jacket water preheater systems, A and B, the positioning of a preheater in the jacket cooling water system is shown schematically in Figs. 1 and 2, respectively.

For **system A**, the circulating water flow is divided into two branches, the one going through the engine and the other 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.

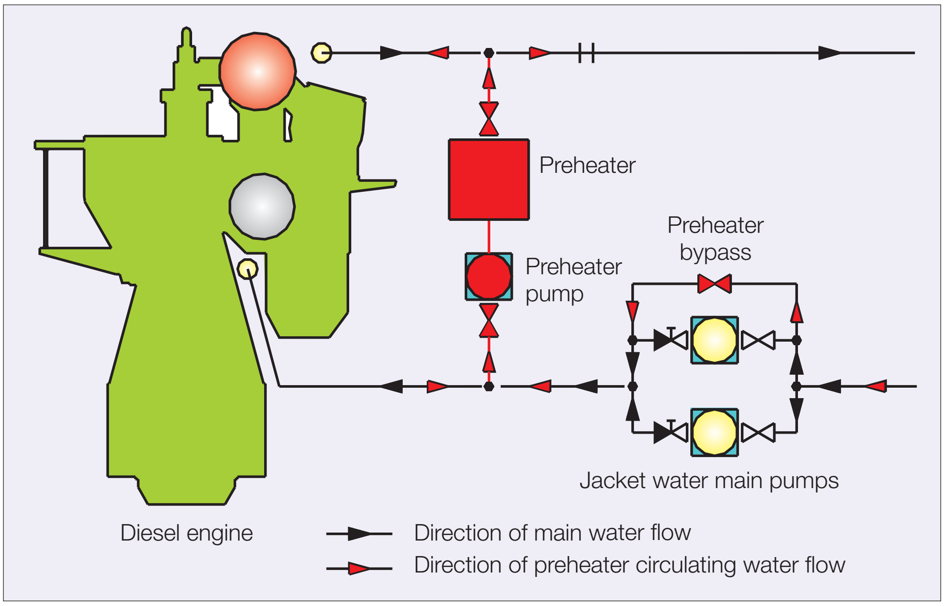


Fig. 1: Preheating of jacket cooling water system – System A

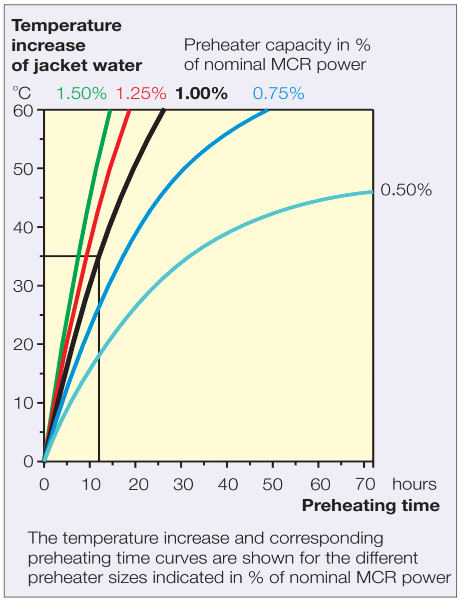


Fig. 3: Preheating of diesel engine

For **system B**, the preheater and circulating pump are placed in parallel with the jacket water main pumps, and the water flow direction is the same as for the jacket cooling water system.

In both cases, 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 Figs. 1 and 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 relationships are shown in Fig. 3. The relationships are 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 will be required. The curves in Fig. 3 are based on the assumption that, at the start of preheating, the engine and engine room are of equal temperature. 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.

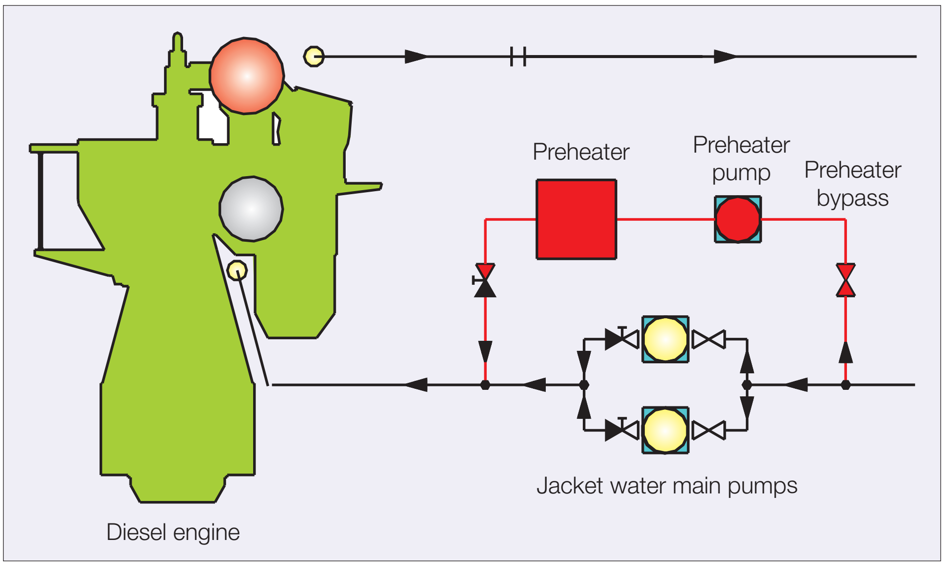


Fig. 2: Preheating of jacket cooling water system – System B

Chapter 2

Engine Room Ventilation

In addition to providing sufficient air for combustion purposes in the main engine, the auxiliary diesel engines, the 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.

An example of an engine room ventilation system, where ventilation fans blow air into the engine room via air ducts, is shown in Fig. 4.

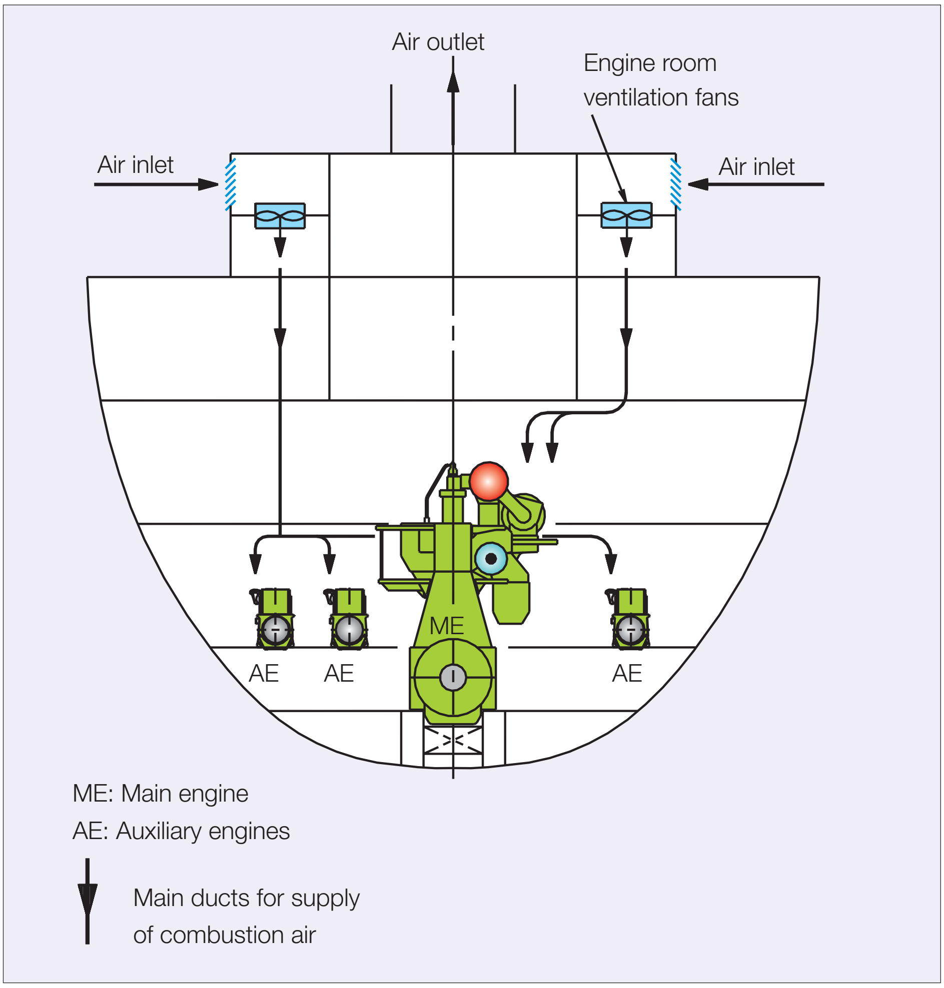


Fig. 4: Engine room ventilation system

Air temperature

Measurements show that the ambient air intake temperature (from deck) at sea will be within 1 to 3°C of the seawater temperature, i.e. max. 35°C for 32°C seawater, and max. 39°C for 36°C seawater.

Measurements also show that, in a normal ventilation air intake system, where combustion air is taken directly from the engine room of a ship, the engine room temperature is normally 10-12°C higher than the ambient outside air temperature. This temperature difference is even higher for winter ambient air temperatures, see Fig. 5. In general, the engine room temperature should never be below 5°C, which is ensured by stopping one or more of the air ventilation fans, thus reducing the air supply to and thereby the venting of the engine room.

This means that the average air temperature in a ventilated engine room will not be lower than 5°C and not higher than 39 + 12 = 51°C, say 55°C, as often used as maximum temperature for design of the engine room components.

Since the air ventilation ducts for a normal air intake system are placed near the turbochargers, the air inlet temperature to the turbochargers will be lower than the engine room temperature. Under normal air temperature conditions, the air inlet temperature to the turbocharger is only 1-5°C higher than the ambient outside air temperature.

This means that the turbocharger suction air temperature will not be higher than about 39 + 5 = 44°C (ref. 36°C S.W.), say 45°C.

For arctic running conditions, a ducted air intake system directly to the turbocharger can be an advantage, in order to maintain sufficiently high temperatures for the crew in the engine room. With a ducted air intake, the turbocharger's intake air temperature may be arranged to be approximately equal to the ambient outside air temperature.

Air supply

In the case of a low speed two-stroke diesel engine installed in a spacious engine room, the capacity of the ventilation system should be such that the ventilation air to the engine room is at least 1.5 times the total air consumption of the main engine, auxiliary engines, boiler, etc., all at specified maximum continuous rating (SMCR).

As a rule of thumb, the minimum engine room ventilation air amount corresponds to about 1.75 times the air consumption of the main engine at SMCR. Accordingly, 2.0 times the air consumption of the main engine at SMCR may be sufficient.

On the other hand, for a compact engine room with a small two-stroke diesel engine, the above factor of 1.5 is recommended to be higher, at least 2.0, because the radiation and convection heat losses from the engine are relatively greater than from large two-stroke engines, and because it may be difficult to achieve an optimum air distribution in a small engine room.

To obtain a correct supply of air for the main engine's combustion process, about 50% of the ventilation air should be blown in at the top of the main engine, near the air intake to the turbochargers, as shown in Fig. 4.

Otherwise, this can have a negative effect on the main engine performance. Thus, the maximum firing pressure will be reduced by 2.2% for every 10°C the turbocharger air intake temperature is raised, and the fuel consumption will go up by 0.7%.

Furthermore, a correct air supply near the turbochargers will reduce the deterioration of the turbocharger air filters (from oil fumes, etc., in the engine room air), and a too draughty engine room can be avoided.

Moreover, a sufficient amount of air should be supplied to areas with a high heat dissipation rate in order to ensure that all the heat is removed, for instance around auxiliary engines/generators and boilers. Ventilation ducts for these areas are not shown in Fig. 4.

In winter time, the amount of air needed to remove the radiation/convection heat from the engine room may be lower.

Air pressure

The air in the engine room should have a slightly positive pressure, but should not be more than about 5 mm WC (Water Column) above the outside pressure at the air outlets in the funnel.

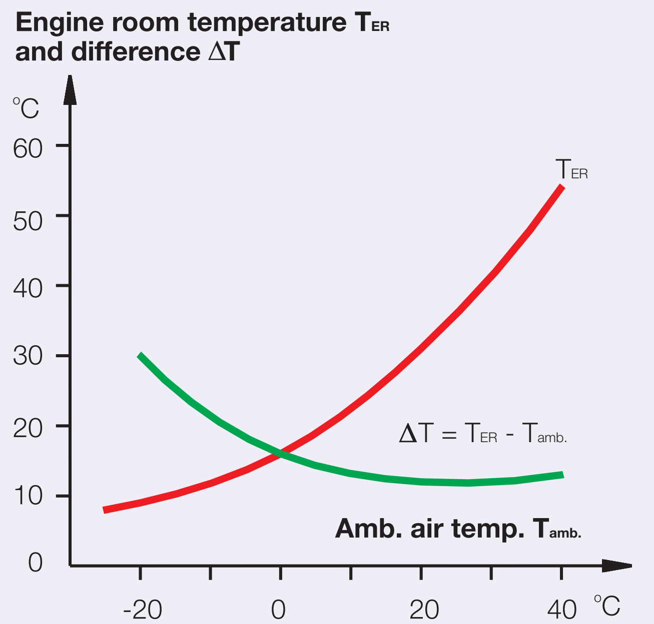
Accommodation quarters will normally have a somewhat higher over-pressure, so as to prevent oil fumes from the engine room penetrating through door(s) into the accommodation.

The ventilation air can be supplied, for example, by fans of the low-pressure axial and high-pressure centrifugal or axial types. The required pressure head of the supply fans depends on the resistance in the air ducts.

All ventilation air is normally delivered by low-pressure air supply fans which, to obtain sufficient air ventilation in all corners of the engine room, may require extensive ducting and a pressure head as stated below.

Low-pressure fans,
Δp = 60-100 mm WC

For further information, please consult engine room ventilation standard ISO 8861: 1998 (E).



The engine room temperature T_{ER} and the engine room/ambient air temperature difference ΔT are shown as functions of the ambient air temperature T_{amb}

Fig. 5: Engine room temperature

Chapter 3

Main Engine Operation under Normal, High and Extremely Low Ambient Temperature Conditions

Standard ambient design temperature conditions

For the purpose of determining a reference for fuel consumption and exhaust gas data of diesel engines, the following standard reference ambient conditions defined by ISO (International Standards Organisation) are to be used:

ISO 3046-1:2002(E) and
ISO 15550:2002 (E):
Barometric pressure .... 1000 mbar
Turbocharger air
intake temperature .... 25°C
Charge air coolant temperature .. 25°C
Relative air humidity.................... 30%

The corresponding ambient seawater temperature may be equal to or lower than the above-mentioned charge air coolant temperature, depending on and influenced by the design of the cooling water system and on the operation of the system (whether and to what extent the cooling water is recirculated).

The engine must be able to operate in unrestricted service under the special maximum ambient temperature conditions required by the ship.

According to IACS (International Association of Classification Societies) rule M28, these requirements (special maximum ambient temperature conditions) – normally referred to as tropical ambient reference conditions – are as stated in the box.

All MAN B&W two-stroke engines fulfil the above rule as standard.

The above standard values are normally used for ocean-going ships, whereas for stationary power plants, specific climatically determined conditions at site may require that different values be used for matching of the engine (site conditions instead of ISO) and unrestricted service.

IACS M28 ambient reference conditions (1978)
For the purpose of determining the power of main and auxiliary reciprocating internal combustion engines, the following ambient reference conditions apply for ships of unrestricted service:

Total barometric pressure.......1,000 mbar
Air temperature............................... +45°C
Relative humidity.............................. 60%
Seawater temperature...................... 32°C
(Charge air coolant-inlet)

Note:
The engine manufacturer shall not be expected to provide simulated ambient reference conditions on the test bed.

Depending on the cooling water system, the scavenge air coolant temperature may be equal to or somewhat higher than the above-mentioned seawater temperature. Thus, if a central cooling water system is used, the charge air coolant temperature will normally be about 36°C i.e. 4°C higher than the seawater temperature.

The standard layout data for the main engines are based on ISO ambient reference conditions, with the maximum allowable tropical ambient conditions of 45°C air and 32°C seawater/36°C cooling water, for which unrestricted service at 100% SMCR is still possible.

In some few cases, the owner requires operating conditions at tropical ambient temperature conditions higher than standard. This paper also describes such situations.

At the opposite end of the ambient temperature scale, too low a turbocharger air intake temperature may limit the engine operation in service if no special precautions have been taken. This paper also describes this situation.

Design recommendations for normal ambient temperature running conditions

Normal running conditions

As mentioned earlier, the main engine is, as standard matched/designed for ISO ambient reference conditions (25°C air/25°C cooling water) and max. tropical ambient conditions of 45°C at the turbocharger air inlet, and a 32°C seawater/36°C cooling water temperature to the scavenge air coolers. The lowest allowable air intake temperature for the above standard temperature matched engine is about –10°C.

Under these normal running conditions, i.e. at turbocharger air intake temperatures between +45°C and –10°C, and with a service power higher than 30% SMCR, it is recommended to keep both the turbocharger air inlet and scavenge air temperatures as low as possible, so as to reduce the specific fuel oil consumption of the diesel engine.

The scavenge air pressure will also be reduced when using a low scavenge air coolant temperature. Therefore, when operating at low ambient air temperatures giving a high scavenge air pressure, it is recommended to use as low a scavenge air coolant temperature as possible.

In general, MAN B&W Diesel recommends to operate the main engine with as low a scavenge air coolant temperature as possible.

Shipyards often specify a constant (maximum) central cooling water temperature of 36°C, not only for tropical

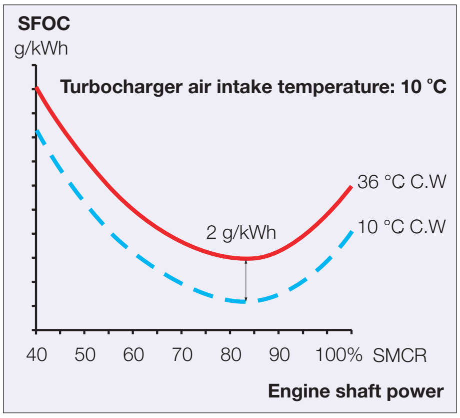


Fig. 6: Influence on SFOC of the cooling water (scavenge air coolant) temperature

ambient conditions, but also for winter ambient conditions. The purpose is to reduce the seawater pump flow rate when possible, and thereby to reduce the electric power consumption, and/or to reduce the water condensation in the air coolers.

However, when operating with 36°C cooling water instead of for example 10°C (to the scavenge air cooler), the specific fuel oil consumption (SFOC) will increase by approx. 2 g/kWh, see Fig. 6. Any obtained gain in reduced electric power consumption, therefore, will be more than lost in additional fuel costs of the main engine.

The cooling water temperature will normally be higher than 10°C – achieved by recirculating the cooling water – as this is the minimum permissible cooling water temperature for the lubricating oil cooler. This means that, in practice, the scavenge air temperature will never be lower than 10-12°C and, therefore, has no restrictive influence on the operation of the engine.

As a general rule, normal running of a diesel engine is possible, i.e. without any precautions being taken, at any turbocharger air inlet temperature below 45°C and above some –10°C, see Fig. 7a. Lower temperatures may result in a too high scavenge air pressure and higher temperatures may involve a too low scavenge air pressure. In both cases, special precautions have to be taken.

Operating at part load in special inland, bay and harbour areas

An increase of the seawater temperature and, thereby, the scavenge air temperature has a negative impact on the heat load conditions in the combustion chamber. Therefore, all two-stroke engines for marine applications have an alarm set point of 55°C for the scavenge air temperature for protection of the engine.

When operating at an increased seawater temperature existing in some inland, bay and harbour areas, the maximum power output of the engine should be reduced to an engine load resulting in a

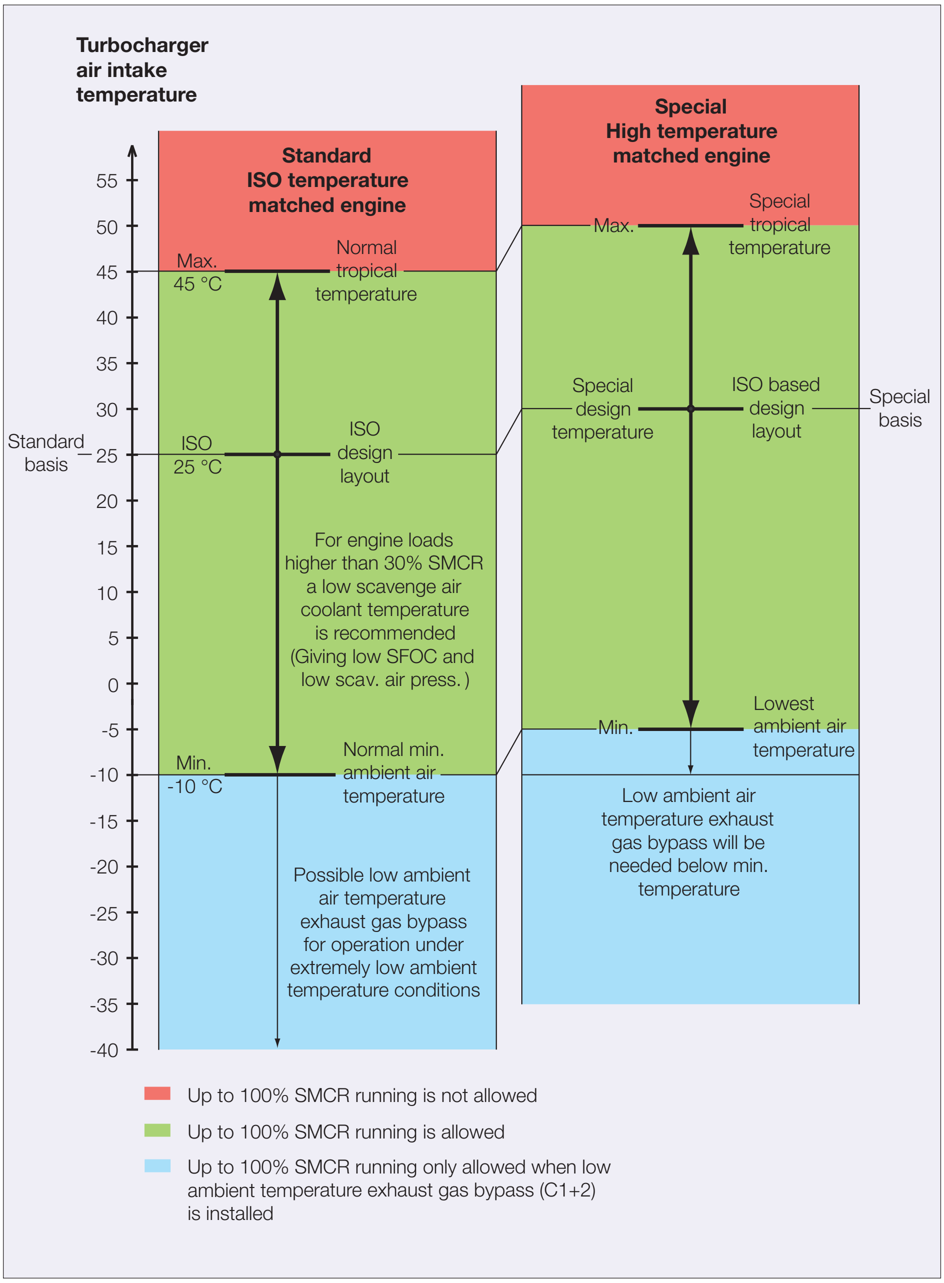


Fig. 7a: Principles for standard and special high ambient air temperature matched engine

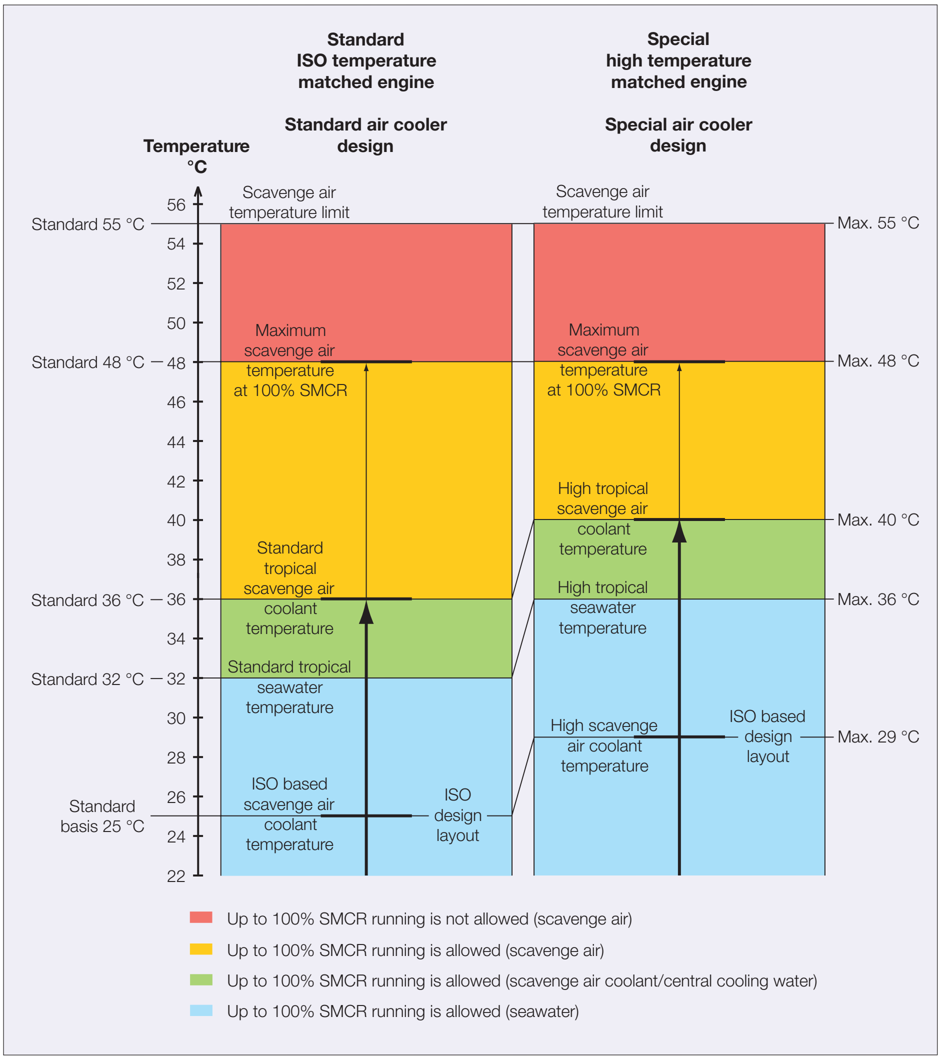


Fig. 7b: Principles for layout of scavenge air cooler for standard and special high scavenge air coolant temperature (illustrated for a central cooling water system)

scavenge air temperature below the level of the scavenge air temperature alarm.

The engine's obtainable load level will nevertheless in any case be much higher than required to secure safe manoeuvrability (4-6 knots) of the ship even at an extreme seawater temperature of for example 42°C.

When sailing in, for example, the harbour area during manoeuvring, the engine load will normally be relatively low (15-30% SMCR), and the corresponding scavenge air temperature will then only be slightly higher than the scavenge air coolant temperature. Therefore, a seawater temperature as high as for example 42°C in harbour areas is not considered a problem for the main engine.

In general, when sailing in areas with a high seawater temperature, it is possible to operate the main engine at part load as long as the scavenge air temperature alarm limit is not reached. If the alarm is activated, the engine load has to be reduced.

Design recommendations for specified high tropical temperature running conditions

As already mentioned, the standard layout data of the diesel engine is given at ISO ambient reference conditions, i.e. at 25°C air intake to turbocharger and 25°C cooling water temperature at inlet to the scavenge air cooler. The corresponding maximum allowable tropical temperature is 45°C air and 32°C seawater/36°C central cooling water.

Specified tropical air temperature higher than 45°C

An increase of for example 5°C of the maximum turbocharger air intake temperature from 45°C to 50°C will involve a reduction of the scavenge air pressure. The pressure reduction can be compensated by specifying a higher scavenge air pressure at ISO ambient conditions.

It has to be mentioned that part load optimised/matched engines are already specified with an increased scavenge air pressure. Therefore, part load optimisation, in combination with engines specified for high ambient air temperature operation, can result in a higher than normally acceptable scavenge air pressure.

In principle, when unrestricted operation at increased tropical ambient air temperature is required, the engine design layout has to be based on a higher design temperature than the ISO temperature, but now with the normal ISO based engine parameters (heat load and scavenge air pressure) valid for the higher design temperature. Thus, if the engine is required to be specified for a max. air intake temperature of for example 45 + 5 = 50°C, this involves that the engine instead of being matched to the ISO based air temperature of 25°C has to be matched to the 25 + 5 = 30°C air intake temperature and obtaining approximately the original ISO based engine heat load conditions for this higher ambient air temperature matching point.

In this example where a max. air intake temperature of 45 + 5 = 50°C is required, the allowable air temperature range will change from –10°C/45°C to –5°C/50°C, see Fig. 7a. If unrestricted operating at –10°C is still required for this high air temperature matched engine, a low ambient air temperature exhaust gas bypass (see later) is needed in order to avoid a too high scavenge air pressure at low ambient air temperatures.

Specified tropical cooling water temperatures higher than 32°C seawater/ 36°C central cooling water

The standard marine air cooler layout is specified with a ΔT of maximum 12°C from water inlet to air outlet of the scavenge air cooler, which gives a scavenge air temperature of 36 + 12 = 48°C and, accordingly, a margin of 7°C to the scavenge air temperature alarm limit of 55°C at 100% SMCR.

It is relevant to keep this temperature margin of 7°C as a fouling margin for all kinds of applications. This means that the scavenge air temperature of 55 – 7 = 48°C is the maximum temperature to be used for layout of the scavenge air cooler.

An increase of for example 3°C of the maximum scavenge air coolant temperature from 36°C C.W. to 39°C C.W. may involve a similar increase of the scavenge air temperature from 48°C to 51°C, which has a negative impact on the combustion chamber temperatures.

The increased maximum scavenge air coolant temperature can be compensated by specifying a scavenge air cooler with a reduced temperature difference, see Fig. 7b, illustrated for an engine with a central cooling water system. A reduction of the temperature difference can be obtained by a combination of increased water flow and/or a bigger scavenge air cooler. Approx. 15% increased coolant flow is necessary in order to obtain a 1°C lower scavenge air temperature at a given coolant temperature. The maximum allowed coolant flow velocity of the air cooler must, of course, not be exceeded.

As a ΔT of 8°C is considered to be the lowest possible temperature difference to be used for a realistic specification of a scavenge air cooler, accordingly, MAN B&W Diesel has 48 – 8 = 40°C as the maximum acceptable scavenge air coolant temperature for a central cooling water system, see Fig. 7b.

If the engine is required to be specified for a maximum central cooling water temperature of for example 36 + 3 = 39°C, this means that the engine, instead of being matched to the ISO based cooling water temperature of 25°C, has to be matched to the 25 + 3 = 28°C cooling water temperature and obtaining approximately the original ISO based engine heat load conditions and scavenge air temperature at the higher cooling water temperature matching point.

Engine design specification

If an engine has to be specified for operation in high ambient temperature conditions (i.e. higher than 45°C air and higher than 32°C seawater/36°C cooling water), this should be stated in the design specification order to the engine designer, in order to ensure that the correct engine and turbocharger are delivered.

Design recommendations for extremely low air temperature running conditions

The density of the air will be high, when the ship is operating in arctic conditions with a low turbocharger air intake temperature. As a result, the scavenge air pressure, the compression pressure and the maximum firing pressure will be high.

In order to prevent excessive pressures under such ambient air temperature conditions, the turbocharger air inlet temperature should be kept somewhat higher than the ambient air temperature (by preheating, if possible).

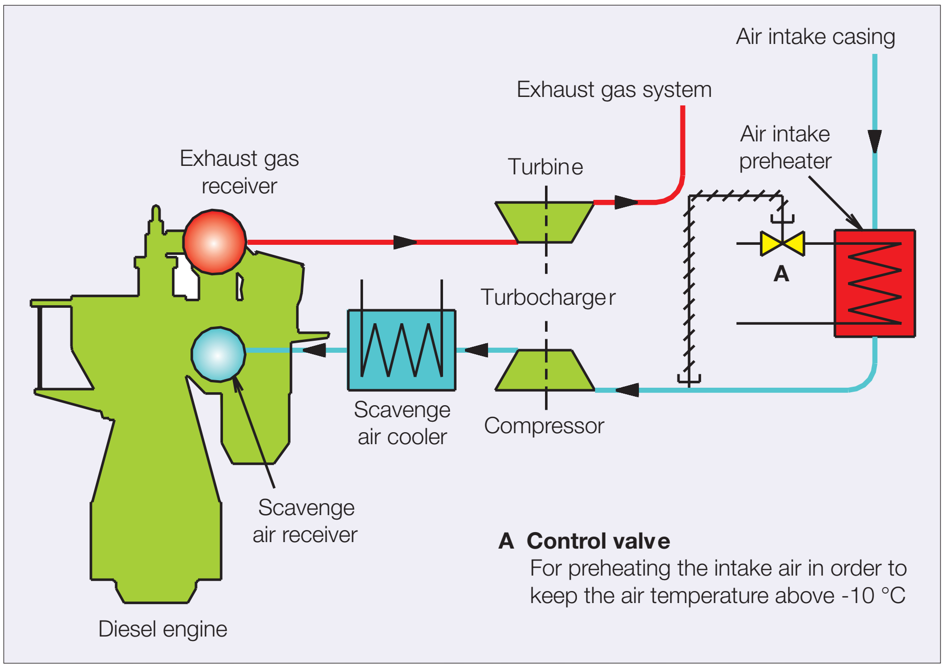


Fig. 8: Preheating of intake air

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

Possible different measures

For inlet air temperatures below approx. –10°C, the precautions recommended will depend very much on the actual operating profile of the ship. In this context, the following different measures may be mentioned:

1. Preheating of intake air (shipyard application)
2. Scavenge air bypass/blow-off system (not MAN B&W standard)
3. Exhaust gas bypass system (MAN B&W standard)

A reduced power output of the engine could also be a solution, if there is no demand for 100% SMCR power output under low ambient air temperature operation.

Preheating of intake air (shipyard application)
Considering that a low ambient temperature is the reason for the high density of the turbocharger intake air, the first idea that comes to mind is simply to heat the air. A diagram of such a system is shown in Fig. 8.

It should be mentioned, however, that preheating of the intake air is normally not acceptable, as the air preheating installation required is rather expensive. Alternative methods should therefore be considered.

If the ship sometimes operates at low (arctic) and sometimes high air temperatures, a bypass solution is recommended. With such a bypass, the engine is matched on the basis of ISO ambient conditions, while at low air temperature running conditions the scavenge air pressure may be controlled by opening the bypass.

The bypass arrangements, which in principle can be either a scavenge air (blow-off) or an exhaust gas bypass system, are shown schematically in Figs. 9 and 10 and are described below.

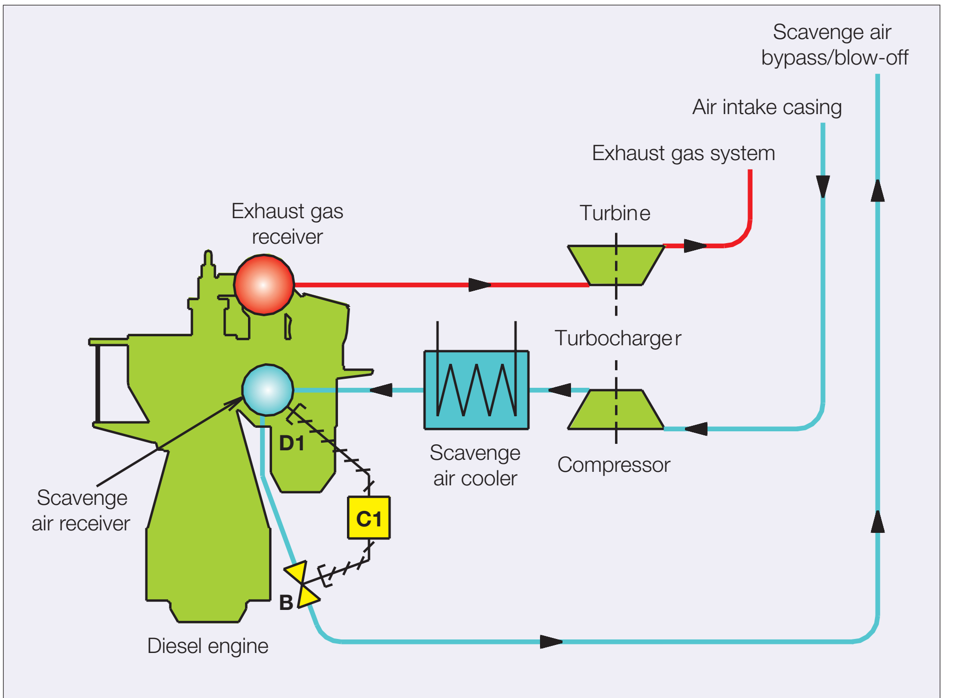
Scavenge air bypass/blow-off system (not MAN B&W standard)
In Fig. 9, excessive air supply to the engine is adjusted by blowing off part of the air, keeping the scavenge air pressure close to the pressure valid at SMCR/ISO operating conditions.

The air must not be blown off directly to the engine room as it contains oil fumes and creates noise.

The principle of controlling the scavenge air pressure, using control device C1, is shown in Fig. 11. The figure shows that in the upper power range of the engine, part of the air will blow off, thus reducing the scavenge air pressure, see arrow C1.

In fact, Fig. 11 also indicates that a reduction (limitation) of the maximum permissible engine power output in service could be a solution when sailing at low ambient air temperature, if – under this condition – there is no demand for operation at 100% SMCR.

Thus, when occasionally operating at, for example, –15°C air (and with 10°C cooling water), up to about 90% SMCR power may still be maintained for a normal engine without any adjustments being made.



B Blow-off valve
C1 Control device
Ensures that the scavenge air pressure does not exceed the SMCR/ISO value

D1 Electric measuring device
Scavenge air pressure

Fig. 9: Scavenge air bypass/blow-off system

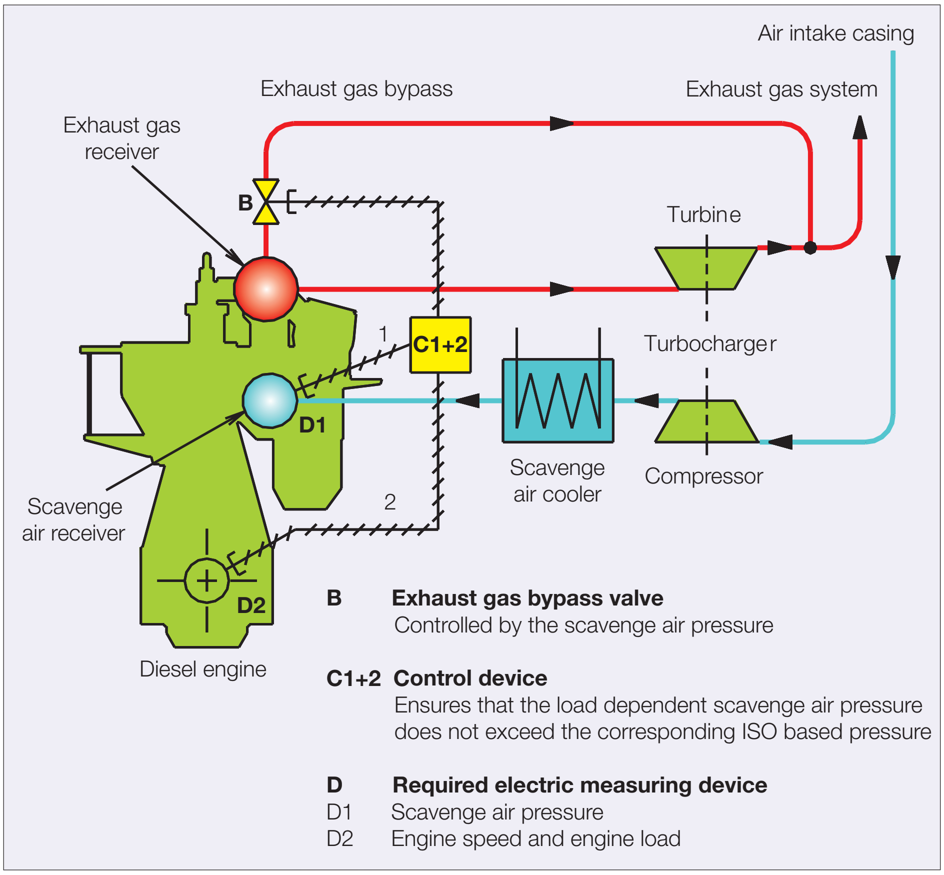


Fig. 10: Standard load-dependent low ambient air temperature exhaust gas bypass system

The C1 control system is a variable low ambient air temperature bypass system, which is electronically controlled, PLC-based (Programmable Logic Controller) and with an electrical or pneumatical actuator for variable adjustment of the bypass valve opening.

The C1 control system is only possible for low air temperatures not lower than about –15°C.

The scavenge air bypass system is not a MAN B&W Diesel standard system, and has so far not been installed.

Exhaust gas bypass system (MAN B&W standard)

With an exhaust gas bypass system (which is the MAN B&W Diesel standard recommendation), as shown in Fig. 10, part of the exhaust gas bypasses the turbocharger turbine, giving less energy to the compressor, and thus reducing the air supply to the engine.

For this bypass, a more applicable system with control device C1+2 is applied where the load-dependent scavenge air pressures are kept close to the corresponding ISO pressures.

The principle of controlling the scavenge air pressure by means of control device C1+2 is shown in Fig. 11, which shows that the exhaust gas bypass can be activated (variably open) over the entire load range of the engine if the air temperature is sufficiently low, i.e. if the scavenge air pressure is sufficiently high.

The standard low ambient air temperature exhaust gas bypass system C1+2 is based on an exhaust gas bypass valve of the butterfly type, with variable adjustment of the bypass opening. The opening of the bypass valve is activated by means of an electrical or pneumatical valve actuator, which is electronically controlled based on a PLC (Programmable Logic Controller).

As an option, the variable low ambient air temperature bypass system C1, described as for scavenge air bypass, can also be applied for the exhaust gas bypass system for moderate low ambient air temperatures. However, because the bypass flow area for low air temperatures is relatively large, the load dependent C1+2 system is recommended.

If an engine is to be specified for operation in special low ambient air temperature conditions, i.e. with a low ambient temperature exhaust gas bypass system, this should be stated in the design specification to the engine designer.

The installation of the low temperature dependent exhaust gas bypass system should only be introduced after consulting MAN B&W Diesel, Copenhagen.

Design features of the standard load-dependent exhaust gas bypass system C1+2

The installation of the adjustable load-dependent exhaust gas bypass system C1+2 ensures and maintains the optimal bypass flow area. This means that the bypass system, compared with C1, has a more advanced control device C1+2, which includes both scavenge air pressure and engine load parameters.

Besides the gauge for scavenge air pressure, the control system therefore requires a shaft power measuring device for measuring the shaft torque and engine speed (rpm), together with a fuel index transmitter.

For the electronically controlled ME engine, the bypass control C1+2 can be incorporated in the Engine Control System (ECS) as an add-on. 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.

This exhaust gas bypass system ensures that, when the engine is running at part load at low ambient 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, will ensure that the engine is not overloaded. At the same time, it will keep the exhaust gas temperature relatively high.

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 only fall by 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 low ambient temperature conditions.

Irrespective of whether a load-dependent exhaust gas bypass system is installed or not, the exhaust gas boiler steam production at ISO (25°C air/25°C C.W.) or higher ambient conditions will be the same, whereas in wintertime it may be different, as the scavenge air pressure is controlled by the bypass valve.

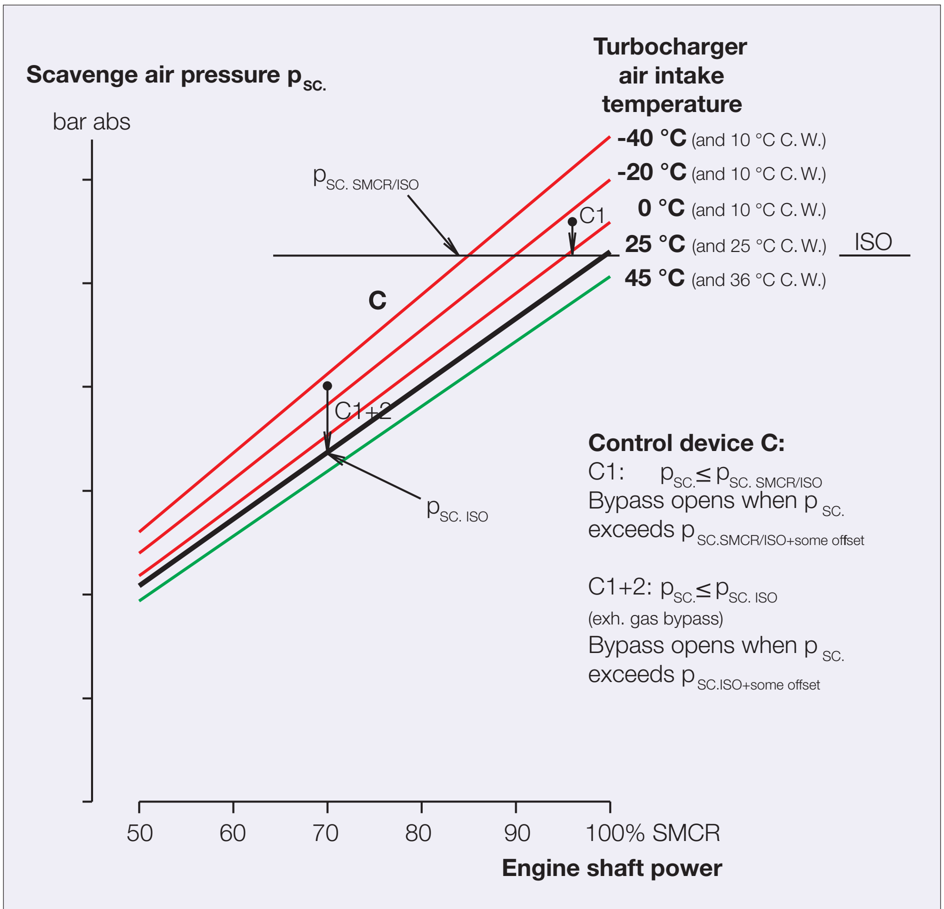


Fig. 11: Bypass valve controlling the scavenge air pressure p_{sc} (example)

As an example, Fig. 12 shows the influence of the load-dependent exhaust gas bypass system on the steam production when the engine is operated during winter, with an ambient air temperature of 0°C and a scavenge air cooling water temperature of 10°C.

The calculations have been made for a 6S60MC-C/ME-C engine equipped with a high-efficiency turbocharger. Fig. 12 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, C1+2, the engine can meet the steam demand.

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.

Special low temperature precautions in the diesel engine and auxiliaries

Lube oil viscosity at low ambient temperatures

Special recommendations for low seawater temperature conditions may be considered. 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.

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.

Other recommendations

Depending on the situation, one might consider introducing the following additional modifications of the standard design practice:

- 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
- 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, all MAN B&W two-stroke diesel engines meet the ice class demands, i.e. there will be no changes to the main engines.

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

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

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.

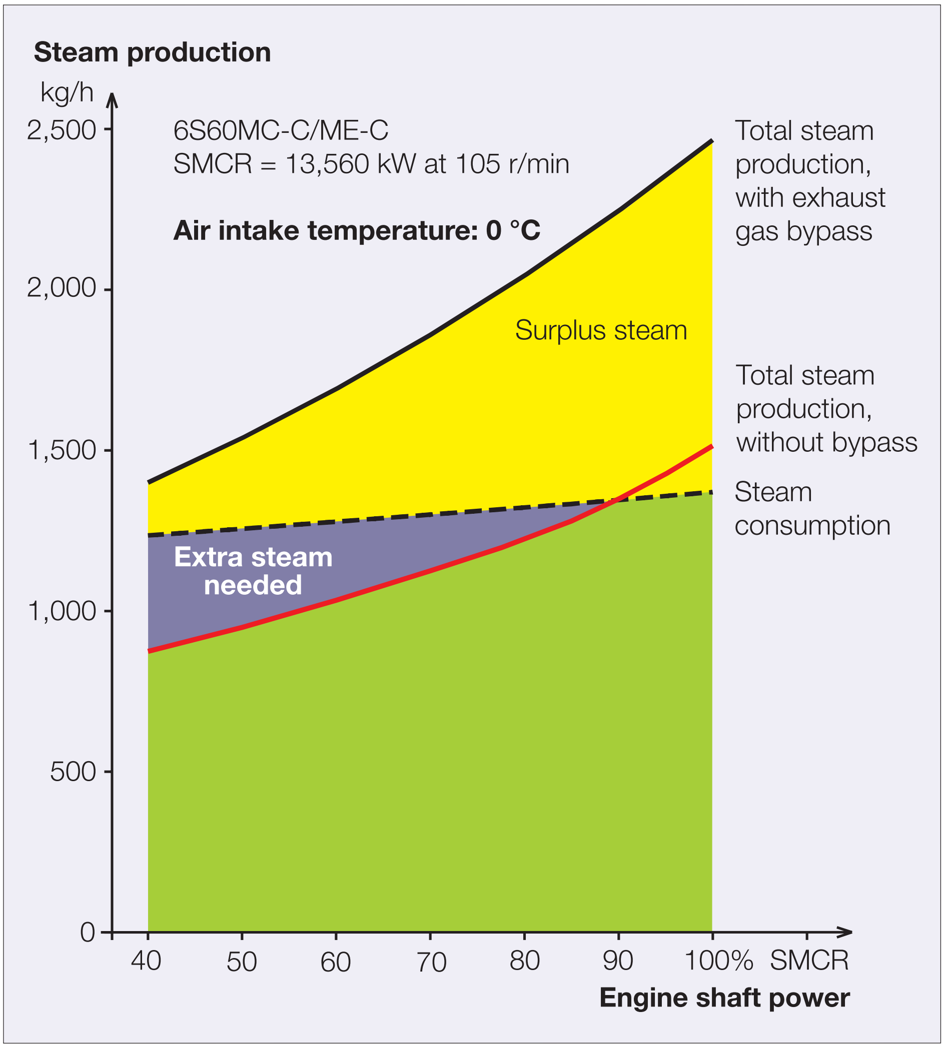


Fig. 12: Expected steam production by exhaust gas boiler at winter ambient conditions (0°C air) for main engine 6S60MC-C/ME-C with/without a load-dependent low air temperature exhaust gas bypass system

Closing Remarks

Diesel engines installed in ocean-going ships are often exposed to different climatic temperature conditions because of the ship's trading pattern, but as the temperature variations on the sea surface are normally relatively limited, the engines will normally be able to operate worldwide in unrestricted service without any precautions being taken.

Even if the ship has to sail in very cold areas, the MAN B&W engines can, as this paper illustrates, also operate under such conditions without any problems as long as special low temperature precautions are taken.

The use of the standard load-dependent low ambient air temperature exhaust gas bypass system may – as an additional benefit – also improve the exhaust gas heat utilisation when running at low ambient air temperatures.

Furthermore, at the other end of the temperature scale, if the ship should need to sail in unrestricted service in areas with very high ambient air temperatures, higher than 45°C, this will also be possible provided a high temperature matching of the engine is applied. Even when sailing should be needed at very high seawater temperatures, this will be possible provided a specially designed scavenge air cooler is installed on the diesel engine.