

PROPULSION (part 2)

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1 Gas turbines

1.1 Definition

Gas turbines are turbines extracting the necessary energy from combusting gases (chemical energy) and transform it into mechanical energy.

According to this definition even turbo compressors (present in diesel engines) can be seen as a type of gas turbine but this specific type of turbine is not taken into account in present chapter.

1.2 Description

1.2.1 Main parts

All parts of a gas turbine are mounted around one or more axis. The axis carries a compressor and a turbine and are connected with the consumer (Figure 1).

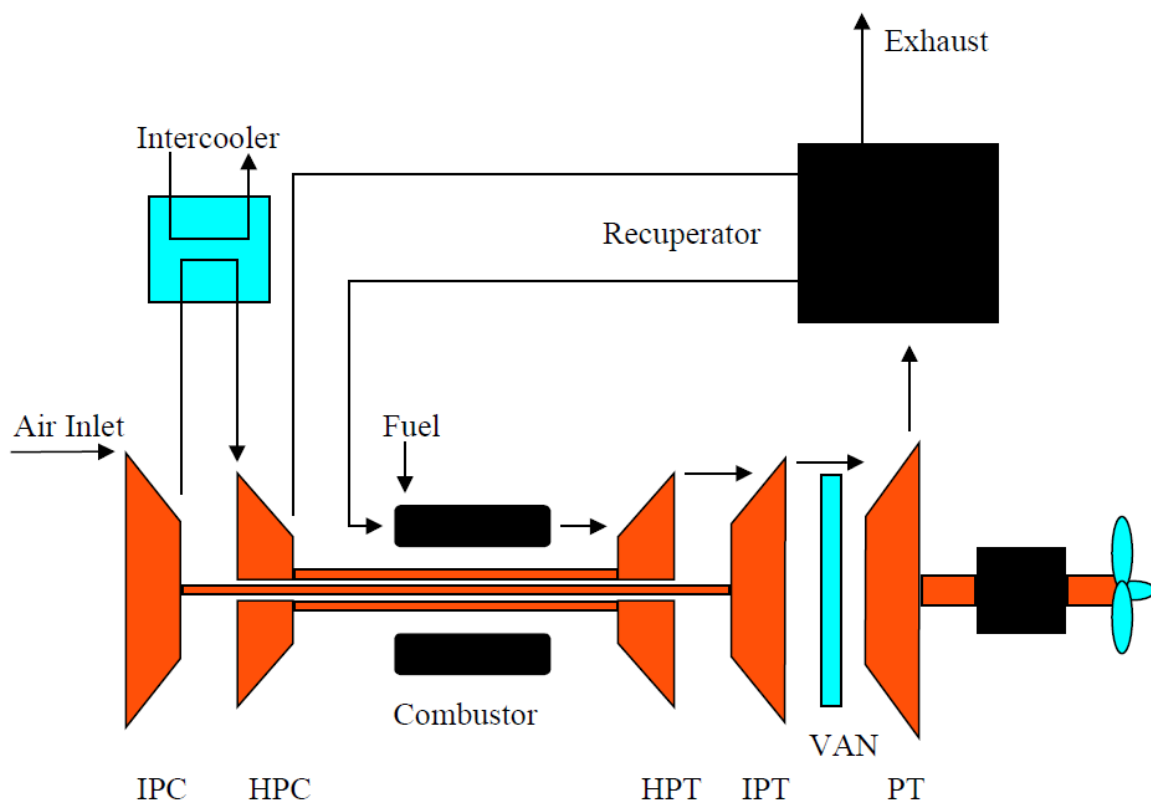


Figure 1 Inter-Cooled Recuperated Cycle (Colin,, 2003)

a. Compressor

The air, which is necessary for the combustion, is compressed up to a pressure of about 8 bar.

b. Combustor

The fuel is continuously injected and combusts in the combustor. The combustion increases the volume of gas. There are always a series of combustors (4 up to 16) which are lined up around the axis.

c. Turbine

The gases enter the turbine at a high speed because of the high pressure, temperature and increased volume. As a result the axis absorbs energy from this high energy gas flow. This energy is used for:

- The rotation of the compressor
- The rotation of the consumer such as an alternator or propeller

1.2.2 Auxiliary systems

a. Speed reduction

The rotational speed of a gas turbine is very high (3 000 up to 30 000 rpm). Therefore a gear box is necessary to decrease the rotational speed to have a rotational speed in the correct range for an alternator or a propeller (about 100 rpm).

b. Lub oil module

Every gas turbine has its own lubrication oil system which contains pumps, filters and intercoolers.

c. Control system

An active control system to control the amount of air that gets to the compressor. If the compressor generates more air than needed for the combustion, then a part of this compressed air will be directed immediately towards the turbine or the exhaust but first it will run through the combustor. This is to be able to control the temperature of these combustors (high enough to maintain a flame) and at the entrance of the turbine (low enough for the turbine vanes). The latter is more important because the temperature at the entrance of the turbine should be limited (for the construction) but as high as possible to increase the thermo dynamical efficiency (and to minimize the fuel cost). An intensive control of the amount of air for the combustion, independent of the required power, makes it more easy to fulfil the emission standards, especially for the NO_x emissions.

d. Start-up system

The start-up system can be electrical or pneumatic. The rotation of the compressor (and thus the entire turbine) has to be initiated before fuel can be injected. Then, the combustors can be enlightened electrically.

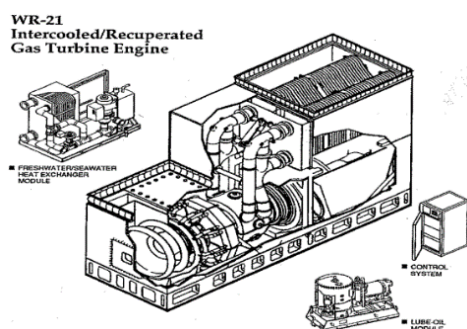


Figure 2

1.2.3 Optimisation

There are two options to increase the efficiency of the installation (Figure 2):

1. As valid for every compression the efficiency increases by compressing in different steps with intercooling in between.
2. A heat exchanger can extract energy (heat) from the exhaust gases and redistribute it for the heating of the air at the entrance of the combustors.

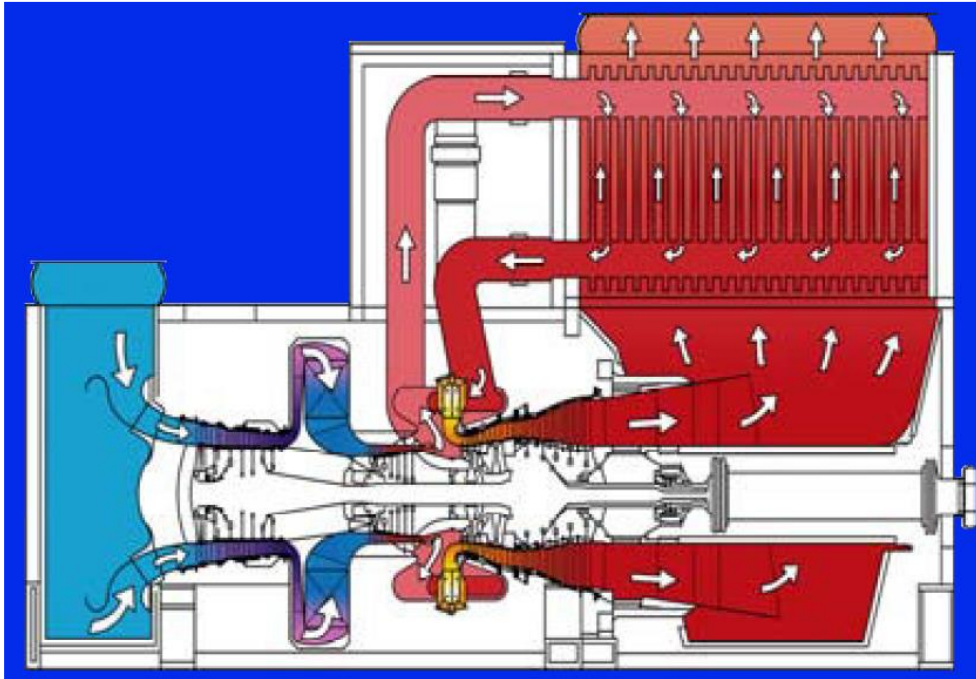


Figure 3 from *Proceedings of the International Gas Turbine Congress 2003 Tokyo*

1.3 Applications

A gas turbine is used to drive a propeller or an alternator when the total amount of mass is restricted but with the need for a high amount of power. The specific mass is the ratio between the mass of an installation and the maximal amount of power it can produce. A specific mass of 1-2 kg/kW is common for gas turbines.

The fuel consumption of a gas turbine is very high and the fuel has to be of high quality. The high quality is necessary to avoid small fuel particles to hit and demolish the turbine vanes.

Gasturbines can be found on planing boats, hovercraft, navy vessels and cruise ships.

For planing boats it is of high importance to have a wide range of power available at the moment the vessels get lifted out of the water without compromising the total weight of the ship.

The navy uses gas turbines in some of their frigates (high speed vessels without the need for long ships) and for mine sweepers. For the latter it is used for light

weight electrical generators located in an engine room above the water line. These generators provide the power needed for the electrical engines which are located in the propeller hub. In such way it creates a very quiet propulsion at low speed.

Gas turbines are also used on vessels which are subject to very strict environmental restrictions, especially concerning the NO_x emission. Some cruise ships are equipped with steam aggregates connected to diesel engines and with gas turbines. Depending on the priority (fuel consumption or emission norms) one or the other is used.

Gas turbines can also be found in mobile emergency pumps and emergency aggregates because of the easy transportation of a compact installation with high power capabilities.

2 Other Marine engine types

2.1 LNG Carriers

LNG carriers build before 2004 have a steam turbine as main engine, the steam is produced with boil-off gas, the boil-off gas rate (BOR) is typically 0.25 % of total cargo volume for the older vessels and 0.10 % ~ 0.15 % for the newer vessels.

More recent LNG carriers sometimes have a two stroke engine as main propulsion system. The boil of gas is then reliquefied (because it cannot be used for the propulsion) if the main engine is a classic HFO two stroke engine.

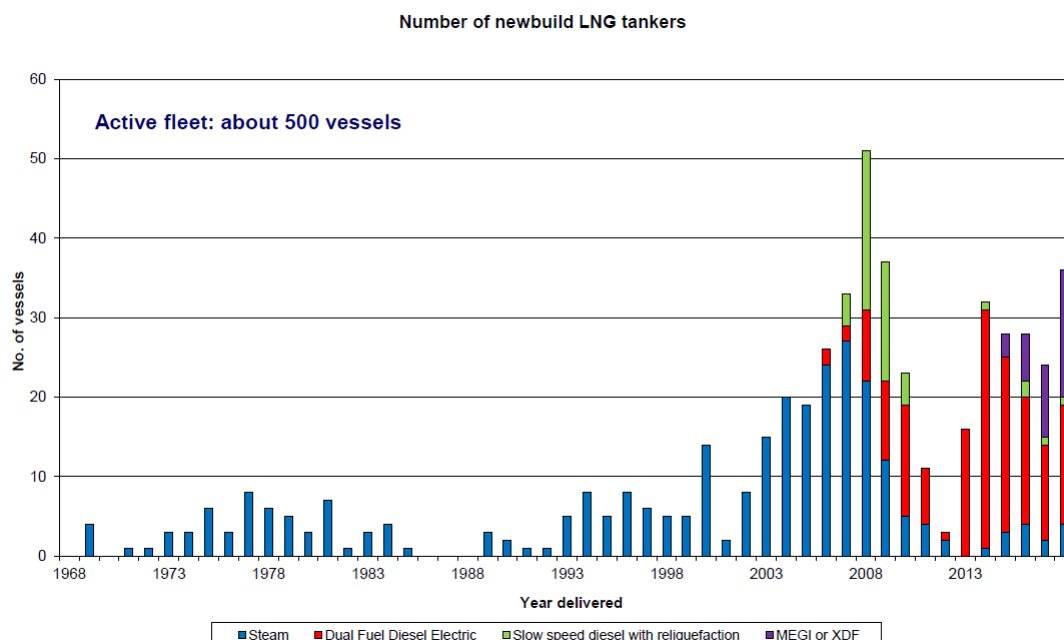


Figure 4 Source: Exmar

Some LNG-carriers however have another solution and inject the boil off gas into the two stroke engine (M-type, Electronically Controlled Gas Injection MEGI or XDF) and thus run (partly) on LNG (Dual fuel engines). The ratio between HFO and LNG

mainly depends on the price of HFO and LNG. In these marine LNG engines there is always a minimal amount of HFO (about 5%) necessary to secure the combustion since LNG does not self-combust and to avoid lubrication issues.

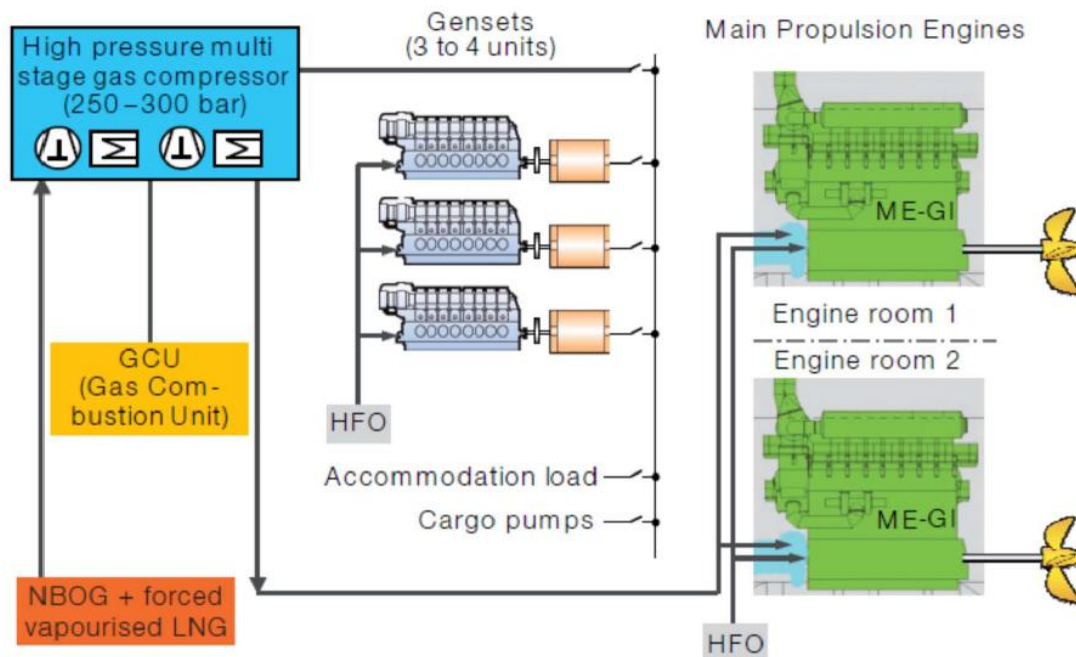


Figure 5 Principle of a two stroke dual fuel engine

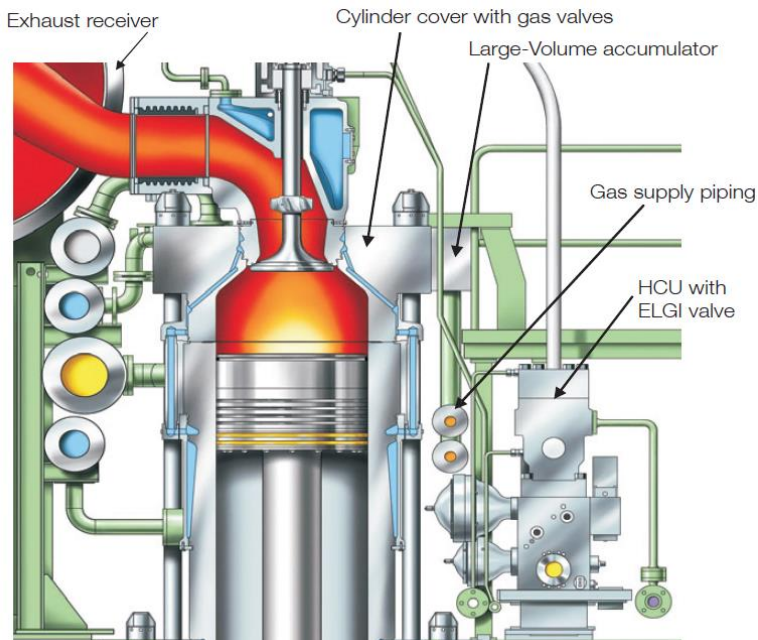


Figure 6 Dual fuel marine LNG engine www.mandieselturbo.com/

Sometimes the ship is driven with electric motors. Then the electricity is produced with four stroke engines running on diesel oil and natural gas (dual fuel four stroke)

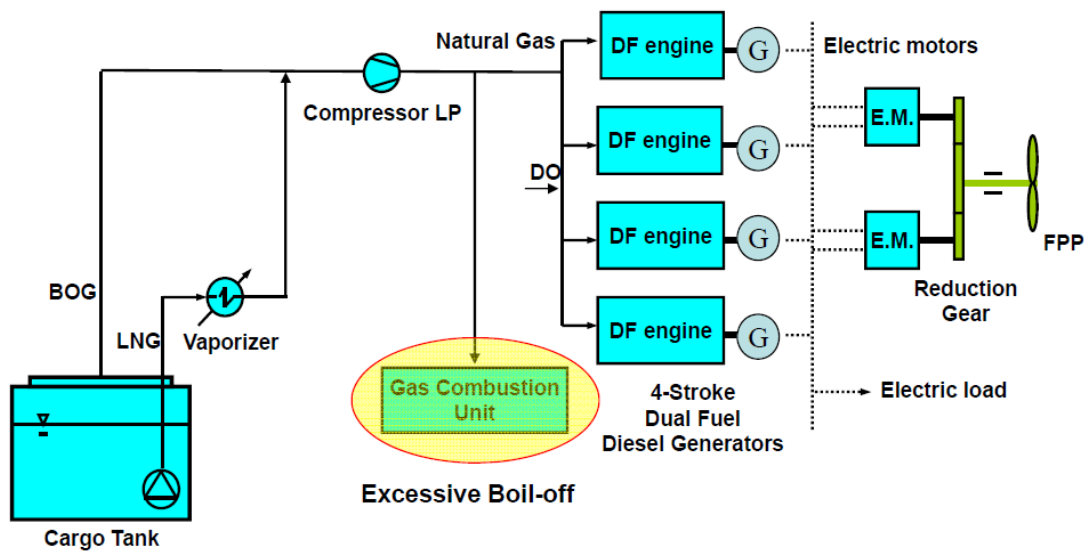


Figure 7 Dual fuel - electric propulsion system Source: Exmar

2.2 Nuclear powered engine

Once upon a time some people thought that a nuclear powered engine is a good idea to deliver power to commercial vessels. Nuclear power is used to heat water and transform to steam, this steam is then consumed in a (classic) steam turbine. The nuclear waste (plutonium) takes 240.000 years to be safe (to put this into perspective, mankind is here for 200.000 years).

Nowadays, the only modern nuclear powered non-naval ships are (big) icebreakers, the advantage is the enormous autonomy on fuel for propulsion. Submarines and aircraft carriers can be equipped with one or more nuclear powered engines, the first to be able to stay under water for a long period of time (only limited by the amount of food it can carry), the second for the almost unlimited amount of power available.

The 2022 energy crisis results in new ideas and concept on nuclear powered non-combat vessels (e.g. Ulstein).

2.3 Stirling engines

Stirling engines are air-independent (in the closed system) engines which are used in some (older) submarines. The main principle of a Stirling engine is based on the expansion of air (or another gas) when heated. This expansion is transformed to a translation of a piston and then, through a crankshaft, transformed to a rotation of an axis.

3 Comparing the different propulsion systems

3.1 Criteria

The three main propulsion systems (gas turbine, steam turbine and diesel engine) will be compared based upon these criteria:

1. The thermal efficiency or the specific fuel consumption
2. The quality of the fuel
3. Manoeuvrability or the ability to change the engine torque
4. The total mass of the installation
5. The cost to install
6. Maintenance and reliability
7. Oil and additives
8. Environmental issues
9. Other costs

3.2 Analysing the criteria

3.2.1 The thermal efficiency or specific fuel consumption

a. Definitions

1. The thermal efficiency is the ratio between the effective power (P_e) and the power to be delivered at the engine (P_t) (mostly as fuel). The difference between both powers is the lost power (P_v).

$$\eta = \frac{P_e}{P_t} = \frac{P_t - P_v}{P_t} = 1 - \frac{P_v}{P_t}$$

2. The lower heating value (H_o) of the fuel.

This indicates the amount of energy that can be extracted from 1kg of fuel. Common values are:

41.000 kJ/kg or 41 MJ/kg for residual fuels

42.7 MJ/kg for diesel oil

Up to 45 MJ/kg for petroleum gas and natural gas

If B indicates the fuel consumption (kg/s) then:

$$P_t = B \cdot H_o$$

3. The specific fuel consumption (b_e).

This indicates the amount of fuel necessary to create an effective power (P_e) of 1 kW or 1 MW.

$$b_e = \frac{B}{P_e} = \frac{\frac{P_t}{H_o}}{P_e} = \frac{1}{\eta \cdot H_o}$$

This expression shows the relation between the specific fuel consumption and the thermal efficiency.

The unit of specific fuel consumption is kg/s/kW, g/kWs=g/kJ, g/MJ, g/kWh or g/HPh

$$1 \frac{kg/s}{kW} = 1 \frac{kg}{kWs} = 1 \frac{kg}{kJ} = 1\,000 \frac{g}{kWs} = 1\,000\,000 \frac{g}{MJ}$$

$$1 \frac{g}{kWs} = 3\,600 \frac{g}{kWh}$$

$$1 \frac{g}{kWh} = 0.736 \frac{g}{HPh}$$

b. Efficiencies

According to Carnot (Figure 10) the thermal efficiency of an installation cannot exceed:

$$\eta \leq 1 - \frac{T_{out}}{T_{max}}$$

T_{max} is the peak temperature in the process

T_{out} is the temperature of the gases at the outlet (at the exhaust for gas turbines and diesel engines, at the entrance of the condenser for a steam installation).

The unit of both temperatures is always Kelvin.



Figure 8 Nicolas Léonard Sadi Carnot 1796-1832

In a diesel engine T_{max} is about 2273K (2000°C). This high temperature is possible because the temperature in the cylinder is not constantly high during the entire cycle. During the air inlet stroke the temperature drops to 373K (100°C) resulting in an acceptable (with cooling system) average temperature of about 1173K

(900°C). For a common temperature of exhaust gases of 723K (450°C), the maximal efficiency of a diesel engines becomes **0.68** ($1 - 723/2273$).

In a steam turbine the peak temperature will not exceed 773K (500°C) (not even in the most powerful installations). The steam pressures would become too high. The minimal temperatures can drop to 303K (30°C) which results in a maximal Carnot efficiency of **0.61**.

The max temperature in a gas turbine (which is constant) will be about 1273K (1000°C). The temperature at the outlet however is still high at 673K (400°C) and this results in a maximal efficiency of **0.47**.

The Carnot cycles is a theoretical approach and the real efficiencies are for all three systems much lower:

- The efficiency of a modern diesel engine is about 0.45
- Steam turbines have an efficiency of 0.30 (because of energy losses in the kettle among other losses)
- Gas turbines have an efficiency of about 0.25

The results obtained with the Carnot cycle preserve: A diesel engines has the highest efficiency and thus the lowest fuel consumption for the same delivered power while a gas turbine is the most consuming out of the three (again relative to the effective power).

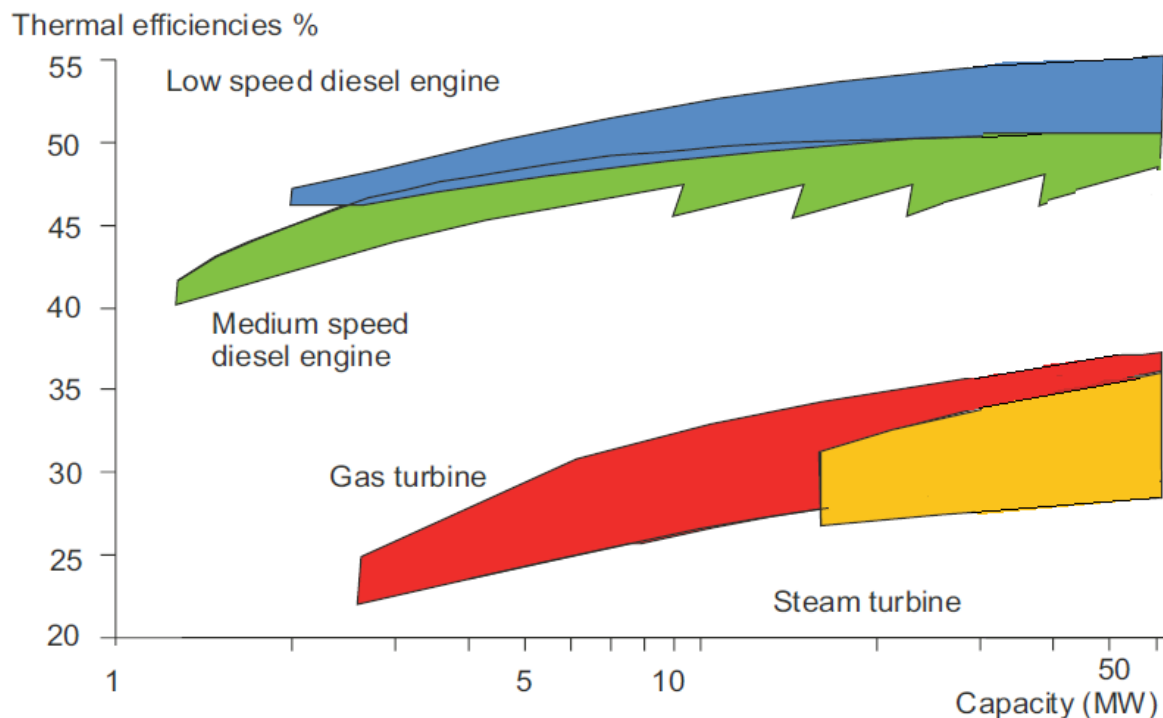


Figure 9 Thermal efficiencies of prime movers

3.2.2 The quality of the fuel

Because of its specific configuration a steam engine is not sensitive to the quality of the fuel.

The quality of the fuel becomes important when the combustion gases interfere with moving parts of the installation. This is even more true for a gas turbine. Cooling the turbine is difficult and the encounter speeds between combusted gases and the blades are very high. (gas, kerosene and dieseloil).

It is common to use residual fuels in marine diesel engines which do not differ much from the residual fuels used in kettles. In both installations the storage of the residual fuels become an important issue. The residual fuels have to be heated to be able to pump the fuel but it is not possible to maintain a temperature of about 30°C at all times taking into account the heat losses at a double bottom for example.

3.2.3 Manoeuvrability or the ability to change the engine torque

The blades of a turbine are designed for a specific rate of turn and velocity of the gases. At all other situations the layout of the turbines are not ideal. The efficiency and torque of the engine will decrease not running at or close to the design point.

This is not the case for a diesel engine where both the torque and efficiency remain high even at velocities far away from the design speed.

3.2.4 The total mass of the installation

A higher rate of turn will result in a more compact machine: the same volume will be used more frequent.

A turbine is for this reason much more compact than a diesel engine (for the same power). Gas turbines will result in the lightest engines (airplanes) while a steam turbine will lose a part of this advantage because of the presence of a kettle. For modern low speed diesel engines a ratio of about 30 kg/kW is common. For example a 15MW engine will weigh about 450 ton.

3.2.5 The cost to install

There is a close correlation between the cost to install and the total mass of the installation. High power slow marine diesel engines will be much more expensive to install than a light turbine. A gas turbine will be less expensive because of its lower weight but it uses very expensive materials.

3.2.6 Maintenance and reliability

Gas turbines are less reliable because: the turbine blades endure high temperatures and the impact at high speed of small particles (dust) coming from the fuel or air needed for the combustion.

Diesel engines are less fragile but important parts such as injectors and pistons have to be checked and renewed on a regular basis.

Steam engines are less demanding. The turbine is very reliable when the quality of the water in the kettle and the lubrication oil are at the required level. Most problems occur at the kettle when fuel of low quality is used (pipes may break).

3.2.7 Lub oil and additives

For diesel engines this is a significant cost. For cross head engines this cost is even increased because of the lubrication of the piston.

The cost for additives in closed circuits can be neglected. When additives are used in the fuel, the cost for additives will increase dramatically. These additives can be used in all three engine types.

3.2.8 Environmental issues

For the most stringent emission norms a turbine is the better solution because the combustion itself can be adapted (both temperature and air excess) without jeopardising the power of the turbine. Such an adaptation is not possible with a diesel engine.

3.2.9 Other costs

Each propulsion systems suffers from specific issues such as:

- To keep the installation under pressure the running cost (energy) for a steam turbine in standby modus is relative high.
- The limited reliability of a gas turbine or a high speed diesel engines makes it necessary to make the entire system redundant: two smaller units are chosen above one bigger unit. In this way the other unit can still produce power when the first is broken or being fixed.

3.3 Conclusions

1. A gas turbine is chosen when an installation with a high power but with a low total mass is wanted.
E.g.: Engines for the navy, some types of icebreakers, mobile emergency groups, propulsion of planing boats and hovercrafts.
2. When an installation with a high power is required and the total mass is unrestricted, the fuel consumption should be low and with no important sensitivity for different qualities of fuel, then a steam turbine is preferable.
E.g.: LPG and LNG-carriers use a part of its cargo for propulsion, although the more modern ones use dual fuel engines.
3. When the main limitation is the fuel price, a diesel engine will be the most favourable choice nevertheless the installation, lubrication and maintenance cost is high.
E.g.: About 99% of the merchant vessels have a diesel engine for its main propulsion.
4. A diesel engine cannot be used when the environmental issues are the most limiting factor

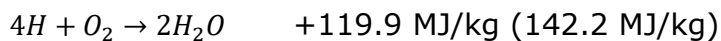
4 Special engine issues

4.1 Combustion

4.1.1 The chemical reaction

Fossil fuels consist mainly out of Carbon (C), Hydrogen (H) and Sulfur (S). During the combustion these elements react (exothermic) with oxygen (in the air) and thus consume air/oxygen and generate heat.

The three main chemical reactions are:



For every reactions the amount of energy is added. This energy is released (+) during the chemical reaction. For hydrogen not only the lower heating value but also the higher heating value is added in between brackets. The difference between both is the energy needed for the evaporation of the water.

All values are only valid for reactions that initiate and terminate at 0°C. This assumption is kept through this entire work. Although this is seldom the situation, the qualitative results and conclusions still stand when this assumption is not valid.

Other reactions:

1. Carbon monoxide CO



The creation of CO₂ starts from CO. When only a limited amount of oxygen (air) is available, then a part of the Carbon in the fuel will transform in CO instead of CO₂.

2. NO_x

Nitrogen oxide can be created during combustion. The nitrogen originates from the fuel (present in very limited amounts in the fuel but reacts easily) or from the air (air composes of about 70% N₂ but this nitrogen is very difficult to react) and reacts with oxygen. Different types of nitrogen oxides are created; NO, NO₂, N₂O₃,... common known as NO_x. This NO_x reacts in the atmosphere (with the humidity H₂O in the atmosphere) and hydrogen nitrate (HNO₃, Nitric acid). These aggressive acids are polluting and create the so called 'acid rain'.

The creation of these NO_x is more easily obtained at a combination of high temperature and high pressure. Both of these conditions are fulfilled in a diesel engine. Different solutions can be found to prevent the formation of acid rain:

- The pressure and temperature can be limited. This is obtained by suspending the exact time of fuel injection. Doing so, the expansion is already initiated at the end of the combustion (=moment of highest temperature). Most marine engines are capable of choosing the timing of the injection. With a “camshaft-less” engine (fully electronically controlled) this is even more convenient. Postponing the fuel injection has a negative influence on the efficiency of the engine.
- (Distilled) water can be injected together with the fuel. During the combustion this distilled water evaporates and the energy for evaporation introduces a temperature control system. The water needs energy to transform into steam and thus heat is extracted out of the system. There is a risk for the water to condense when the engine suddenly stops. This amount of water in the cylinders can cause dramatic issues when the engine is started again.
- A vaporizing base (e.g. Ammoniac) is injected in the exhaust gases. This is an immediate reactions and therefore used when a polluting emission is identified.

4.1.2 The heating value

For example, a specific carbon fuel consists out of the following chemical composition (expressed in percentage of the mass): 88%C, 9%H, 3%S. Or 1 kg of this fuel exist out of 0.88kg C; 0.09kg H and 0.03kg S.

The (lower) heating value is in this situation about:

$$0.88\text{kg } 33.87\text{MJ/kg} + 0.09\text{kg } 119.9\text{MJ/kg} + 0.03\text{kg } 9.25\text{MJ/kg} = 40.87\text{MJ}$$

This calculation does not take into account the energy necessary to dissolve the fuel molecules. The heating values of every component are idealized and are lower in reality.

Based upon this calculation the following conclusions can be drawn:

1. Fuels with a high dose of Hydrogen have a high heating value. This is the case for methane CH_4 , natural gas, propane and less for diesel oil. Heavy fuels have a low percentage of Hydrogen. A low amount of Hydrogen results in a high density, while a low density is related to a high heating value.
2. A relative high amount of sulfur results in a low heating value. The sulfur also has a good influence on the combustion.
3. The amount of heat produced will be lower when the Carbon does not entirely transforms into CO_2 .

The present theoretical calculation method has the sole ambition to ground the above statements.

Under normal working conditions the chemical composition of the fuel is unknown. The heating value can be obtained based upon published research. The chemical reaction of the fuel is 'estimated' based upon the measured input parameters: density of the fuel and percentage sulfur in the fuel. Diagrams (simplified) or formulae (dedicated software) are used.

For example according to the International Standard Organisation (ISO):

$$H_o = (46.704 - 8.802 d^3 + 3.167 d)(1 - (x + y + S')) + 9.420 S - 2.449x$$

With:

S = Sulfur in mass percentage

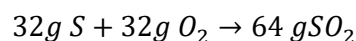
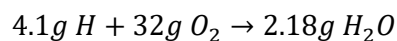
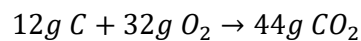
S' = Sulfur in mass percentage/100

x = water in mass percentage/100

y = ashes in mass percentage/100

4.1.3 Stoichiometric amount of air

The minimal amount of air needed (or stoichiometric amount of air) to combust 1kg fuel can be obtained from the chemical reactions. These reactions do not only show the amount of molecules but also the amount of mole (Avogadro's number. Because the mole masses are known (C 12g, O 16g, CO₂ 44g, H 1g, H₂O 18g, S 32g, O₂ 32 and SO₂ 64g) the reaction can be written as:



When 1kg of the fuel as composed in the previous paragraph is used:

$$880g C \rightarrow \frac{32}{12} 880g O_2 = \mathbf{2347g O_2}$$

$$90g H \rightarrow \frac{32}{4} 90g O_2 = \mathbf{720g O_2}$$

$$30g S \rightarrow \frac{30}{32} 30g O_2 = \mathbf{30g O_2}$$

$$TOTAL = \mathbf{3097g O_2}$$

The composition of air can be simplified by stating that it exists out of 76.7% N₂ and 23.2% O₂ (both mass percentage). When 3097 g O₂ is needed:

$$\frac{3097g O_2}{232g O_2/kg air} = 13.348 kg air$$

4.1.4 The air excess and oxygen in the exhaust

The air excess is controlled by measuring the amount of oxygen in the exhaust gases.

Usually the amount of oxygen in the exhaust gases is expressed in volume percentages (and not in mass percentage). The volume percentage is about the same as the mole percentage because a mole gas is about the same for all types of gas.

Not one specific example is used here but it is generalised with: a% C, b% H and c% S (mass percentages). a gram C implies b gram H and c gram S.

It is presumed that there is no presence of CO in the exhaust gases.

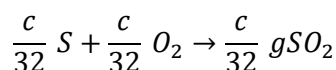
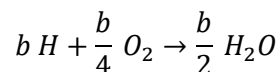
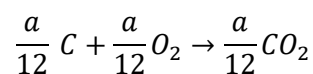
For every element the ratio between mole and volume is:

$$a \text{ grammes } C = \frac{a}{12} \text{ moles } C$$

$$b \text{ grammes } H = \frac{b}{1} \text{ moles } H$$

$$c \text{ grammes } S = \frac{c}{32} \text{ moles } S$$

The chemical reactions are:



The ratio of the actual amount of air in the combustion to the minimal amount of air in the combustion is by definition the air excess ratio λ .

a. $\lambda = 1$

Previous reactions result in an amount mole O_2 necessary for a+b+c gramme fuel:

$$\frac{a}{12} + \frac{b}{4} + \frac{c}{32}$$

Because air is about 79% N_2 and 21% O_2 (volume percentage) the needed amount of mole air is:

$$\left(\frac{a}{12} + \frac{b}{4} + \frac{c}{32} \right) \frac{100}{21}$$

And the amount of N_2 equals in this case:

$$\left(\frac{a}{12} + \frac{b}{4} + \frac{c}{32} \right) \frac{79}{21}$$

The exhaust gases are composed out of CO_2 , H_2O , SO_2 and N_2 all origination from the air. The total amount of mole air becomes (oxygen and nitrogen):

$$\left(\frac{a}{12} + \frac{b}{2} + \frac{c}{32}\right) + \left(\frac{a}{12} + \frac{b}{4} + \frac{c}{32}\right) \frac{79}{21}$$

When the amount of dry gases is desired (because of the type of measuring equipment) this results in (removing $\frac{b}{2}$ which stands for water in the equation):

$$\left(\frac{a}{12} + \frac{c}{32}\right) + \left(\frac{a}{12} + \frac{b}{4} + \frac{c}{32}\right) \frac{79}{21}$$

b. $\lambda > 1$

The amount of air available is increased:

$$\lambda \left(\frac{a}{12} + \frac{b}{4} + \frac{c}{32}\right) \frac{100}{21}$$

And because the amount of air is increased, the amount of oxygen is increased:

$$\lambda \left(\frac{a}{12} + \frac{b}{2} + \frac{c}{32}\right)$$

The volumes (in mole) not contributing in the chemical transformation are:
Air:

$$(\lambda - 1) \left(\frac{a}{12} + \frac{b}{4} + \frac{c}{32}\right) \frac{100}{21}$$

Oxygen:

$$(\lambda - 1) \left(\frac{a}{12} + \frac{b}{2} + \frac{c}{32}\right)$$

The exhaust gases contain the same gases as in $\lambda=1$ but the non-used air as well. The total amount (mole) becomes:

$$\left(\frac{a}{12} + \frac{c}{32}\right) + \left(\frac{a}{12} + \frac{b}{4} + \frac{c}{32}\right) \frac{79}{21} + (\lambda - 1) \left(\frac{a}{12} + \frac{b}{4} + \frac{c}{32}\right) \frac{100}{21}$$

This can be simplified to:

$$\frac{b}{4} + \left(\frac{a}{12} + \frac{b}{4} + \frac{c}{32}\right) \frac{100}{21} \lambda$$

The amount of oxygen in the exhaust gases is related to the ratio between the remained amount of oxygen and the volume of the exhaust gases:

$$\frac{(\lambda - 1) \left(\frac{a}{12} + \frac{b}{2} + \frac{c}{32}\right)}{\frac{b}{4} + \left(\frac{a}{12} + \frac{b}{4} + \frac{c}{32}\right) \frac{100}{21} \lambda}$$

Some devices measure the amount of oxygen relative to the dry gases. In such cases the formula becomes:

$$\frac{(\lambda - 1) \left(\frac{a}{12} + \frac{b}{2} + \frac{c}{32}\right)}{\frac{b}{4} + \left(\frac{a}{12} + \frac{b}{4} + \frac{c}{32}\right) \frac{100}{21} \lambda - \frac{b}{2}}$$

The derived relation can be plotted (and is done in Figure 10).

For an excess ratio lower than 1.4 this curve is not reliable because the presence of a significant amount of CO. Because this CO still has to consume O_2 to create CO_2 , the measured amount of oxygen will be higher than expected.

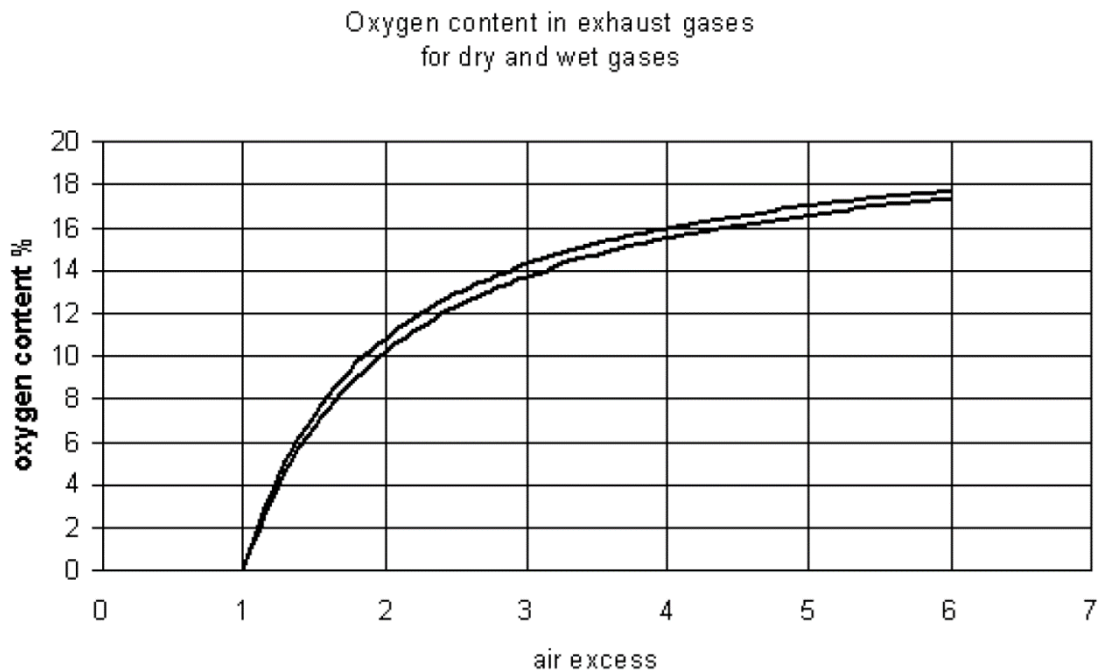


Figure 10

4.2 Hydrodynamic lubrication

4.2.1 Principle

A small layer of oil between two moving metal surfaces is necessary to avoid wear and tear due to friction. This layer should cover the entire surface. Therefore the pressure of this layer of oil has to overcome the pressure induced by the contact pressure of the metal surfaces. The pressure of this layer has to be as high as 200 bar or even 400 bar.

If such a high pressures is obtained with pumps, this is called hydrostatic lubrication. In most cases, the pressure is obtained with the moving parts of the engine itself, this is called hydrodynamic lubrication. The moving surfaces push the oil into areas which decrease in volume, as such increasing the oil pressure.

To obtain hydrodynamic lubrication the following conditions have to be fulfilled:

- a. The moving parts need to have the correct shape
The space available for the oil decreases progressively. The tolerance on the bearings has to be sufficient and the piston segments tapered (to avoid scratches on the inner walls).
- b. The motion of the parts has to be fast enough
Only in this situation the oil cannot escape. Problems may arise when:
 - The engine runs at low speed
 - At the upper and lower dead positions of the piston and rods.
- c. The viscosity of the lubrication oil may not be too low.
A viscosity of the oil which is too high can cause other issues:
 - The internal friction in the oil increases to much
 - The lub oil pump will have to provide high pressures to compensate losses in piping and filters
 - It will be more difficult to filter and clean (centrifugal) the oil
 - When the flow rate of the lost oil becomes too low, the oil will not manage its secondary functions (cleaning and cooling of inner bearings) any longer.

4.2.2 Mechanical deterioration of sliding metal surfaces

It is (economical) not possible to finish all parts of the construction with the correct tolerance. The final tolerance are obtained during the mechanical deterioration period of bearings and segments. The outer layer of these parts is finished with a material that easy tears and therefore shorten the wear period.

These parts should not be disassembled and reassembled often: a new initiation period will always be necessary, but will be harder every time because of the damaged top layer.

The initiation periods are specified by the manufacturer of the machine. The principle of an initiation period is to run the engine at a low load (the pressures on

the oil layer need to be limited) but a high velocity is wanted (for the hydrodynamic effect). This condition is easy to obtain with a generator engine or with a propulsion engine in combination with a controllable pitch propeller. This is hard to obtain with a propulsion engine with a fixed pitch propeller connected to it. In this situation there is a close relation between the velocity and the load. The max propeller speed should increase gradually. The prescribed initiation program of a cylinder liner can take up to 24 hours, the engine will have to run for 24 hours under initiation condition before it can run fully loaded.

4.2.3 Acceleration of the engine

An aggressive acceleration of the engine will cause problems for the hydrodynamic lubrication:

- The giant forces (necessary to obtain the acceleration) act also on the layer of oil. There is a high risk for a non-continuity of the layer.
- In most situations the velocities are low at the initiation of the acceleration.

These are some of the reasons to avoid aggressive accelerations.

4.3 Thermal stress

4.3.1 Description

Thermal stress is a typical issue on heat engines (e.g. diesel engines). The stress is caused by different thermal coefficients of expansion in one stiff construction. These differences are caused by different materials or different temperatures (or a combination of both). Different temperatures in a diesel engine are inevitable because of the need for cooling different parts. Specific regions with a high risk for thermal stress are the cylinder liner, cylinder cover and the piston crown.

Sometimes the stiffness of the construction is lowered to limit the stress. This is a common solution for steam kettles but cannot be used for diesel engines. The main solution becomes thus to limit the temperature gradient over the entire construction. The temperature in the inner surfaces can be lowered or the temperature of the cooled parts increased.

Lowering the temperature of the inner parts is not feasible because the gases would transfer too much heat to the walls and there is an increased risk for corrosion. The inner surfaces are maintained at working temperature and the peak temperature is only limited by the quality of the materials of the cylinder and the quality of the lubrication oil. A drastic change in temperature will be observed by monitoring the temperature of the cooling water and the temperature of the exhaust gases. The air excess is also an important parameter in this monitoring process.

Keeping the cooling water as high as possible is the most common solution for solving or avoiding problems with thermal stress. The flow rate of the cooling water will have to increase to keep the necessary cooling capacity. The maximal temperature of the cooling water is thus limited by the capacity of the pumps of the system (and of the risk for boiling. Boiling of cooling water should be avoided at all times).

4.3.2 Specific situations

a. Preheating the engine

Immediately after the engine is started the temperature of the inner surface of the cylinder reaches very high temperatures. For this reason it is important to have the temperature of the cooling water as high as possible before the engine is started. This preheating aids also the lubrication and ignition at the initial stage. Engines running on heavy fuel are preheated with a cooling temperature of 60°C while smaller engines are preheated by running the engine for about 5 minutes without loading it.

b. Cooling down

The engine still contains a lot of heat immediately after the engine was stopped. This energy has to be discharged or boiling symptoms will come into existence in the cooling circuit. This boiling will result in high temperature differences and thus thermal stress.

c. Tighten the bolts

The prescribed axial tightening forces are heavily influenced by thermal stress. The tightening forces on the cover bolts have to be restricted because the forces on these bolts increase a lot when the engine is running. The thermal expansion of the cover is even higher than the thermal expansion of the bolts. These bolts are almost impossible to pull out when the cover is not cooled down.

d. Thermal transformation and thermal crack

Sometimes the thermal stress reaches the elastic limit. This means that the part under consideration will not have its original shape when cooled down. This is called **thermal deformation**.

After thermal deformation some parts (the most overheated) will not end at their original length. This induces tensile stress (tension) and finally small thermal cracks over the entire surface are observed.

e. Welding

One of the most important problems while welding is thermal stress. Even more when very thick plates are welded (and these thick plates store a lot of heat and are very stiff) and cooled down.

4.4 Cold corrosion

4.4.1 Causes

Cold corrosion is caused by the formation of hydrogen carbonate CHO_3^- and carbonic acid H_2CO_3 (from CO_2 and H_2O) and the formation of hydrogen sulphate or sulphuric acid H_2SO_4 (from SO_3 and H_2O) during the combustion.

These two aggressive acids can cause huge damage in the engine when they condense on a cylinder wall or on the exhaust systems.

4.4.2 Solutions

a. Using fuels with a low percentage of sulfur

When there is only a very limited amount of sulfur, there will be only an insignificant amount of sulfuric acid created. The de-sulfurization of heavy fuels has become common but there is still about 1 to 3% sulfur present in the fuel.

Since January 2015 the amount of sulfur in emission controlled areas (Baltic Sea, North Sea and others) must be as low as 0.1%. (Before January 2015 this was 1%). To cope with these new regulations a scrubber can be installed. This scrubber washes the sulfur out of the exhaust gases and dumps it into the sea (whenever allowed by IMO regulations) or stores the sulfur into sludge tanks.

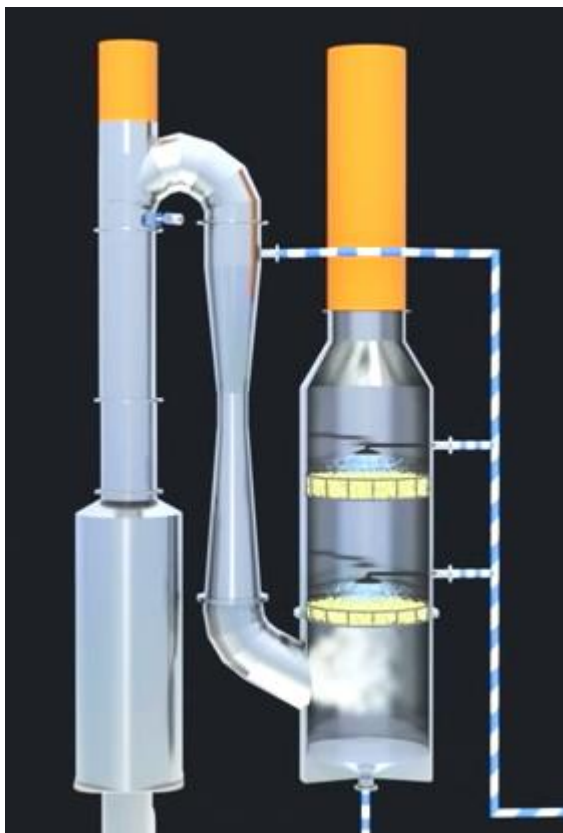


Figure 11 Wärtsilä scrubber principle

b. Prevent condensation

Keep the metal walls above the dew point. This is difficult, for sulfuric acid the dew point is about 140°C at atmospheric pressure and increases to 180°C at the common combustion pressures in a diesel engine.

Applications:

1. Keep the cooling water of a diesel engine as warm as possible (80°C). The upper part of the cylinder liner has to be warmer than 180°C (but lower than 200°C to protect the lubrication oil). It is more and more in practice to use oils which can cope with higher temperatures, the cooling water in these installations can be warmer.
2. The temperatures in the engine can be too cold especially at low loading condition. In this situation the cooling of the exhaust gas is limited. In some cases it is even better to heat the air (instead of extracting warmth from it). The possibility to accelerate the engine is very limited if the air is heated.
3. Cold corrosion will not take place at the vaporiser or overheater of a steam turbine (except during the start-up but that can be neglected) because the common steam pressures, the steam and pipe temperatures will be high above the dew point.
4. Not using the waste heat boiler at low engine loads, when the temperature of the gases is relative low. The condensation will take place in the economiser. This condensation does not only cause corrosion but causes also soot to stick and pollute. This soot can burn during an acceleration and result in a kettle fire.

c. Using materials resistant to corrosion

Chrome and titanium alloys are resistant to corrosion and are used in preheaters and exhaust systems of kettles. The lining walls of a diesel engines are made out of cast iron.

d. Additives and lubrication oil

The cylinder liner and segments are protected by the lubrication oil. This oil is alkaline and can neutralise the acids. The capability to neutralise the acids is expressed with the total base number or TBN.

5 Power and efficiency of a diesel engine

5.1 Indicated power, mean indicated pressure, indicated efficiency

5.1.1 Introduction

a. Air excess λ

Only based on the chemical reaction 1kg fuel needs 14-15 kg air for the combustion (page 19 or section 4.1.3). The necessary amount of air practically needed is higher because:

- To limit the temperatures
- To realise a quick and full combustion

The air excess is defined as:

$$\lambda = \frac{\text{available amount of air (mass)}}{\text{necessary amount of air (mass)}}$$

The numerator does not take into account the amount of air used during scavenging. The air excess of marine diesel engines varies between 1.6 and 5.

b. Mean piston speed c_m

The mean piston speed is the ratio between the stroke S [m] and the time of the piston to translate this distance $\frac{1}{2n}$ (n =rotations per second).

$$c_m = \frac{S}{\frac{1}{2n}} = 2Sn$$

This is a very significant speed of the engine. It is limited by the inertial forces of the piston and the resistance of the gasses at the valves. For a two stroke diesel engines this speed limit is about 7 m/s. This is a rather (s)low value because of the inertia of the heavy cross head. Four stroke diesel engines can have a mean piston speed of about 8.5-10 m/s (depending on the mass of the piston and the number of valves.)

If the crankshaft has a clean continuous rotational speed, the piston position, speed and acceleration over time are a sine function, its derivative and derivative of the derivative. As a consequence the ratio between the maximum speed and mean speed can be calculated.

c. Reference conditions

Power, efficiency and consumption (fuel and air) change with the working condition of the engine. An engine is tested in very stringent reference conditions to be able to compare the results with the properties delivered by the manufacturer. Common reference conditions are:

- Absolute atmospheric pressure 100kN/m²
- Atmospheric temperature 300K (27°C)
- Relative humidity of the intake air 60%
- Water temperature air cooling system 300K (27°C)

5.1.2 Indicated power

a. Definition

This is the mechanical power delivered to the piston during the combustion, compression and the expansion.

b. Determination

1) Power:

Power = Work per second

= Work per sequence multiplied by series per second

2) Number of series per second = nZ/a

With:

n = rounds per second [1/s]

Z = number of cylinders []

a = 1 for two stroke; 2 for four stroke[]

3) Work per sequence W_i

The work during the air inlet and exhaust stroke are not taken into account for the indicated power of a four stroke engine. In such a way is the calculation for two stroke and four stroke engines identical: W_i is the difference of the work of the expansion stroke and the work during the compression.

(a) Work during one "stroke"

This work is given by $\int_{V_s} p \cdot dV = A \int_s p \cdot ds$

p = pressure in the cylinder [Bar]

V = Volume above the piston [m^3]

V_s = Stroke volume of the cylinder

A = Area [m^2] of the cross section of the cylinder

$$= \frac{1}{4} \pi D^2$$

D = cylinder bore [m]

s = position of the piston [m]

S = stroke [m]

The development of the pressure p in function of the position of the piston s is given by the indicator diagram. The pressure is not constant during the sequence. Instead the mean pressure is used:

$$p_m = \frac{\int p \cdot dV}{V_s} = \frac{A \int p \cdot ds}{A \cdot S} = \frac{\int p \cdot ds}{S}$$

$\int p \cdot ds$ is the area under the curve p of the indicator diagram during a stroke (Figure 12), p_m is the mean height of this curve.

This can be done separately for the expansion and compression:

$$W_{id} = A \int_{s_d} p \cdot ds = A \cdot S \cdot p_{md} = V_s \cdot p_{md}$$

$$W_{ic} = A \int_{s_c} p \cdot ds = A \cdot S \cdot p_{mc} = V_s \cdot p_{mc}$$

(b) Work [J] per sequence, mean indicated pressure

The work during one sequence is:

$$W_i = W_{id} - W_{ic} = V_s \cdot p_{md} - V_s \cdot p_{mc} = V_s \cdot p_{mi}$$

Here p_{mi} is the mean indicated pressure

$$p_{mi} = p_{md} - p_{mc} = \frac{\int_{s_d} p \cdot ds}{S} - \frac{\int_{s_c} p \cdot ds}{S} = \frac{\int_{s_d} p \cdot ds - \int_{s_c} p \cdot ds}{S}$$

The numerator is the subtraction of two areas on the indicator diagram. This shows that the mean indicated pressure is the total average of the height of the indicator diagram.

The mean indicated pressure can be measured with specific instruments which are installed on the indicator valve, sometimes it is measured on the bolts of the cylinder head.

4) The indicated power [Watt; J/s] becomes:

$$P_i = W_i \frac{nZ}{a} = p_{mi} V_s \frac{nZ}{a} = p_{mi} A \cdot S \frac{nZ}{a} = p_{mi} \frac{\pi D^2}{4} \cdot S \frac{nZ}{a}$$

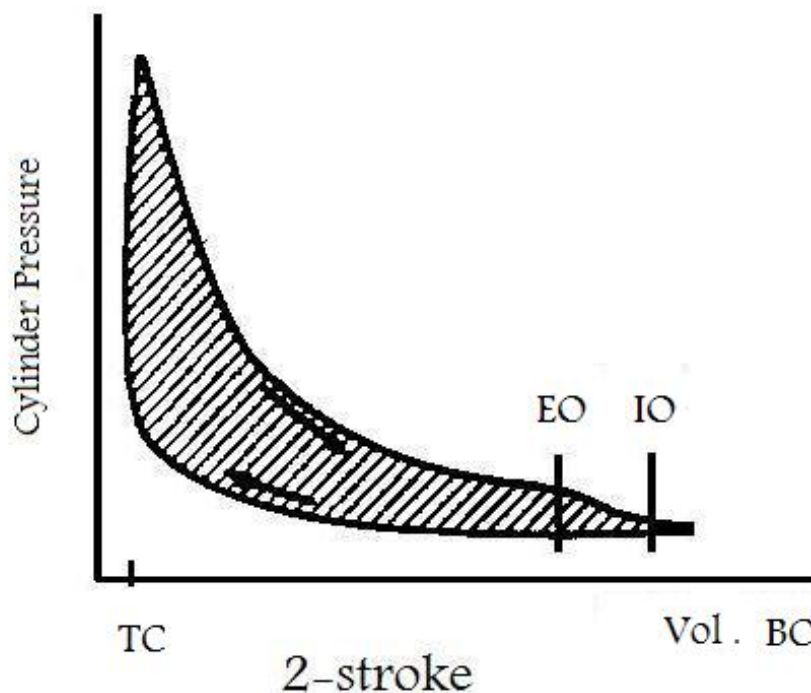


Figure 12 Indicator diagram

c. Cylinder constant and engine constant

The engine constant C_m combines different main parameters of an engine:

$$P_i = p_{mi} n C_m$$

$$\text{And } C_m = \frac{\pi D^2}{4} \cdot S \frac{Z}{a}$$

The manuals for engines with a varying amount of cylinders use the cylinder constant C_c (or cc) instead of the engine constant $C_m = \text{cc } Z$ or

$$P_i = p_{mi} n Z C_c$$

C_m and cc have the same unit [m^3] or $\left[100 \frac{\text{kW}}{\text{bar.rps}}\right]$

d. Power as a function of the piston speed

The indicated power can be expressed as:

$$P_i = p_{mi} A \cdot \frac{c_m}{2} \cdot \frac{Z}{a}$$

This indicates the importance of the mean piston speed c_m for the power.

5.1.3 Indicated efficiencya. Definition

The indicated efficiency is the ratio between the indicated Power and the “inserted” power (as fuel). If

B = fuel consumption [kg/s]

H_o = The lower heating value [kJ/kg]

Then the indicated efficiency is:

$$\eta_i = \frac{P_i}{B \cdot H_o}$$

b. Relation between indicated efficiency and motor regime

The indicated efficiency η_i changes with:

1) An incomplete combustion

This is not a common issue in diesel engine because diesel engines have a high air excess. This will only be an issue in very extreme situations. The losses due to an incomplete combustion are therefore not taken into account

2) Heat losses via the cooling system

Heat losses via the cooling of the cylinders is caused by the heat transfer on the cylinder walls and is dependant of:

- The temperature during combustion (or the air excess): In a heavy loaded engine the air excess and the temperature will be stable.
- Air turbulence: A heavy loaded engine will have more turbulence and as a consequence the heat losses will be larger.

Increasing the engine regime will increase the heat losses (because of the cylinder cooling) but the indicated power will increase even more. The efficiency will increase if the heat losses due to air cooling will increase less than the increasing power. It can be concluded that the heat losses via the cooling have only a minor influence on the indicated efficiency at a changing regime. These losses are about 20% of the indicated power, independent of the regime condition.

3) Heat losses via the exhaust system

These heat losses are the cause for a decreasing indicated power at a very low engine regime and close to the peak power:

- At low engine regime the rate of air flow (of combustion and scavenging) is too high in comparison to the need of air. This air flow takes away the heat (which is needed by the system) out of the system.
- If an engine runs close to its peak power, the last injected fuel particles will have difficulties 'finding' the last amount of air available on time. The available time between the end of the combustion and the opening of the outlet is so small that the last developed heat cannot be transferred into the motion of the piston. The exhaust temperatures will increase dramatically.

The optimum is at about 90% of the power peak. In this situation there is an air excess of about 2.2 (big engine on heavy fuel) and 1.8 (midspeed engine on dieseloil).

The relation between efficiency and air excess is given in Figure 13.

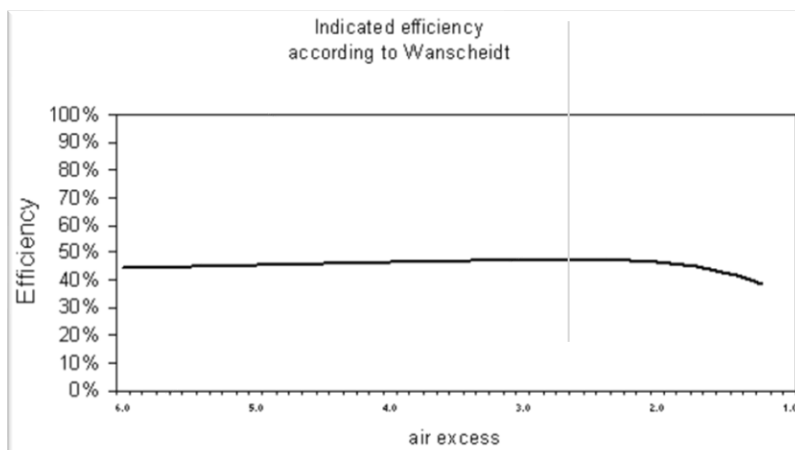


Figure 13 air excess – indicated efficiency

5.1.4 The relation between the mean indicated pressure and air excess

Decreasing the air excess is (relatively) similar as increasing the amount of fuel for a fix amount of air. The mean indicated pressure p_{mi} will thus increase.

5.2 Effective power, mean effective pressure, effective efficiency, mechanical efficiency

5.2.1 Effective power P_e

a. Definition

The effective power P_e is the power delivered at the outgoing axis of the engine. This is also called the brake power of an engine.

If M_e is the effective torque of an engine [Nm], then:

$$P_e = M_e \cdot \omega = M_e \cdot 2\pi n$$

b. Mean effective pressure p_{me}

Analogue as the expression for p_{mi} :

$$P_e = p_{me} \cdot n \cdot C_m \Rightarrow p_{me} = \frac{P_e}{n \cdot C_m}$$

It can be proven that the mean effective pressure is proportional to the torque of an engine:

$$P_e = M_e \cdot 2\pi n = p_{me} \cdot n \cdot C_m \Rightarrow M_e = p_{me} \frac{C_m}{2\pi}$$

5.2.2 Mechanical efficiency η_m

a. Definition

By definition the mechanical efficiency is the ratio between the effective power P_e delivered by the engine and the indicated power P_i delivered by the gasses:

$$\eta_m = \frac{P_e}{P_i} = \frac{p_{me} \cdot n \cdot C_m}{p_{mi} \cdot n \cdot C_m} = \frac{p_{me}}{p_{mi}}$$

The mechanical efficiency is thus also the ratio between the mean effective pressure and the mean indicated pressure.

b. Progress of the mechanical efficiency

The effective power could be calculated based upon the indicated power if the mechanical efficiency is known.

The indicated engine torque M_i is the torque an engine would produce if the mechanical efficiency is perfect (100%). The difference between the indicated torque M_i and effective torque M_e is the friction torque M_w . The friction torque is the torque necessary to overcome the internal mechanical resistance of an engine.

$$P_i = M_i \cdot 2\pi n$$

$$M_i \approx p_{mi}$$

$$M_i = M_e + M_w$$

The mechanical efficiency can be written as:

$$\eta_m = \frac{P_e}{P_i} = \frac{M_e}{M_i} = \frac{M_i - M_w}{M_i} = 1 - \frac{M_w}{M_i}$$

M_i is proportional to p_{mi} . It will be proven that M_w is about constant. Therefore it can be concluded that:

$$\eta_m = 1 - \frac{cte}{p_{mi}}$$

The mechanical efficiency is always smaller than 1. The transformation from indicated power on the piston into effective power on the crankshaft introduces losses:

- Friction, mainly between the piston and cylinder liner. The necessary force decreases with an increasing speed. This is caused by the hydrodynamic lubrication which is more easy to maintain at higher speeds of the piston (C_m).
- Direct driven pumps (fuel pumps, cool water pumps, lubrication pumps) consume energy (and therefore a part of the indicated power). The power needed increases for fuel injected pumps but remains about the same for all other types of pumps. The output pressure is controlled to remain constant.
- The power needed to drive the camshaft and all other moving parts: valves, gear boxes, controllers... The required forces increase with increasing engine speed.
- The power needed for a mechanical scavenger (two stroke) or intake stroke and exhaust stroke (four stroke).
 - For a four stroke engine with an efficient turbo compressor this is different. The compressed air is blew into the cylinder. This compressed air pushes the piston down. In this situation the piston does not require power, the power is even delivered by the compressed air to the piston
 - This component increases for a two stroke engine. Only a limited amount of two stroke engines are equipped with these pumps and/or are capable to shortcut it.

A limited increase of the required forces can be expected. This increase is however not proportional to the engine speed. The error is even smaller if it is expected that the sum of all forces remains constant. This is even more true for engines with a constant rotation speed.

The effective forces increase (with increasing engine speed) and therefore will η_m increase with increasing engine load (Figure 14).

Power losses induced by friction is more important for engines running at a high propeller rate. The losses are not only dependant on the forces which have to be overcome but also on the engine's rotation speed. Running a generator set at a low load is therefore not favourable.

5.2.3 Effective efficiency η_e

a. Definition

This efficiency is the ratio between the effective power on the axis and the "inserted" power as fuel. If

B = fuel consumption

H = lower heating value

then

$$\eta_e = \frac{P_e}{B \cdot H_o} = \frac{P_e}{P_i} \frac{P_i}{B \cdot H_o} = \eta_m \cdot \eta_i$$

b. Progress of the effective efficiency

According to the definition the progress is determined by the combination of the mechanical and indicated efficiencies.

5.2.4 Specific fuel consumption b

As mentioned before this indicates the amount of fuel necessary to create an effective power (P_e) of 1 kW or 1 MW or effective work of 1kJ, 1MJ.

The consumption can be unrealistically low by using fuels with an extreme high heating value. This value should be mentioned in the test results. The specific fuel consumption is furthermore defined by the effective efficiency.

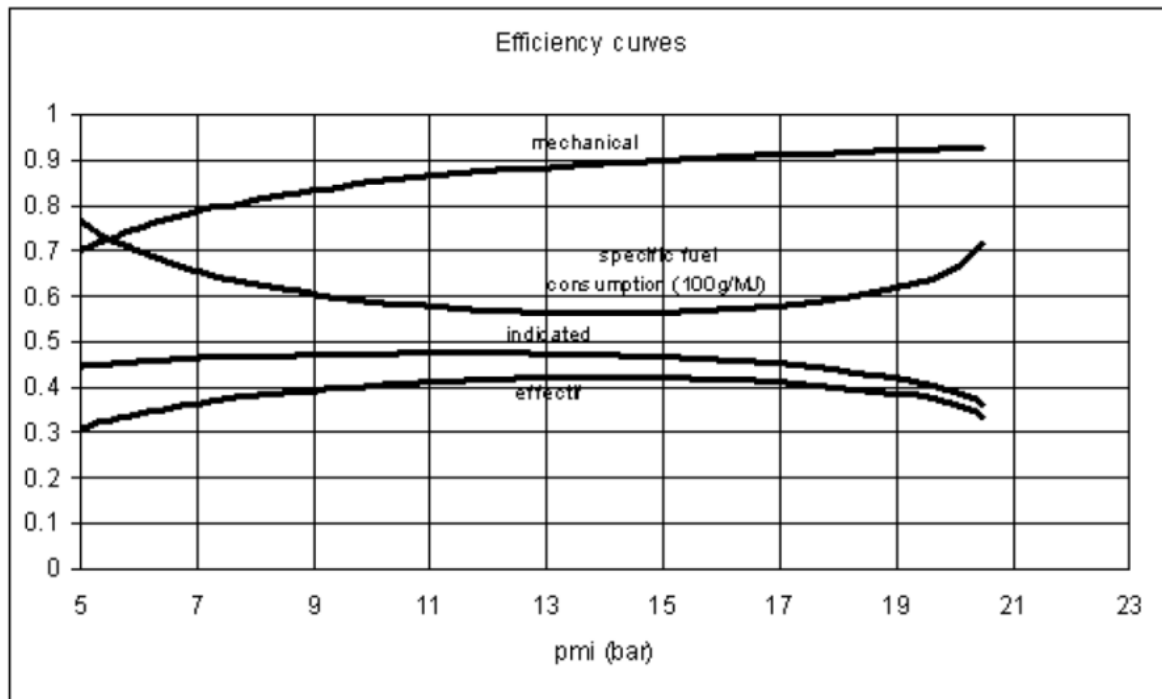


Figure 14 Efficiency curves

5.3 Performance chart

5.3.1 General properties

This diagram (Figure 15) summarizes the main properties of an engine. It is created at specific test conditions. Some remarks are important to analyse a performance chart correctly:

- 1) What are the atmospheric conditions during the performance test?
What is the environmental pressure and temperature? What is the temperature of the cooling water? If the test is not carried out at the standard atmospheric condition a transformation of the results to the standard conditions is necessary.
- 2) What was the quality of the fuel?
Not only the heating value but also other properties such as viscosity and carbon sediment are important
- 3) Is the test carried out according to a generator (constant rotational speed) or according to a propulsion unit ($p_{me} = cte \cdot n^2$).

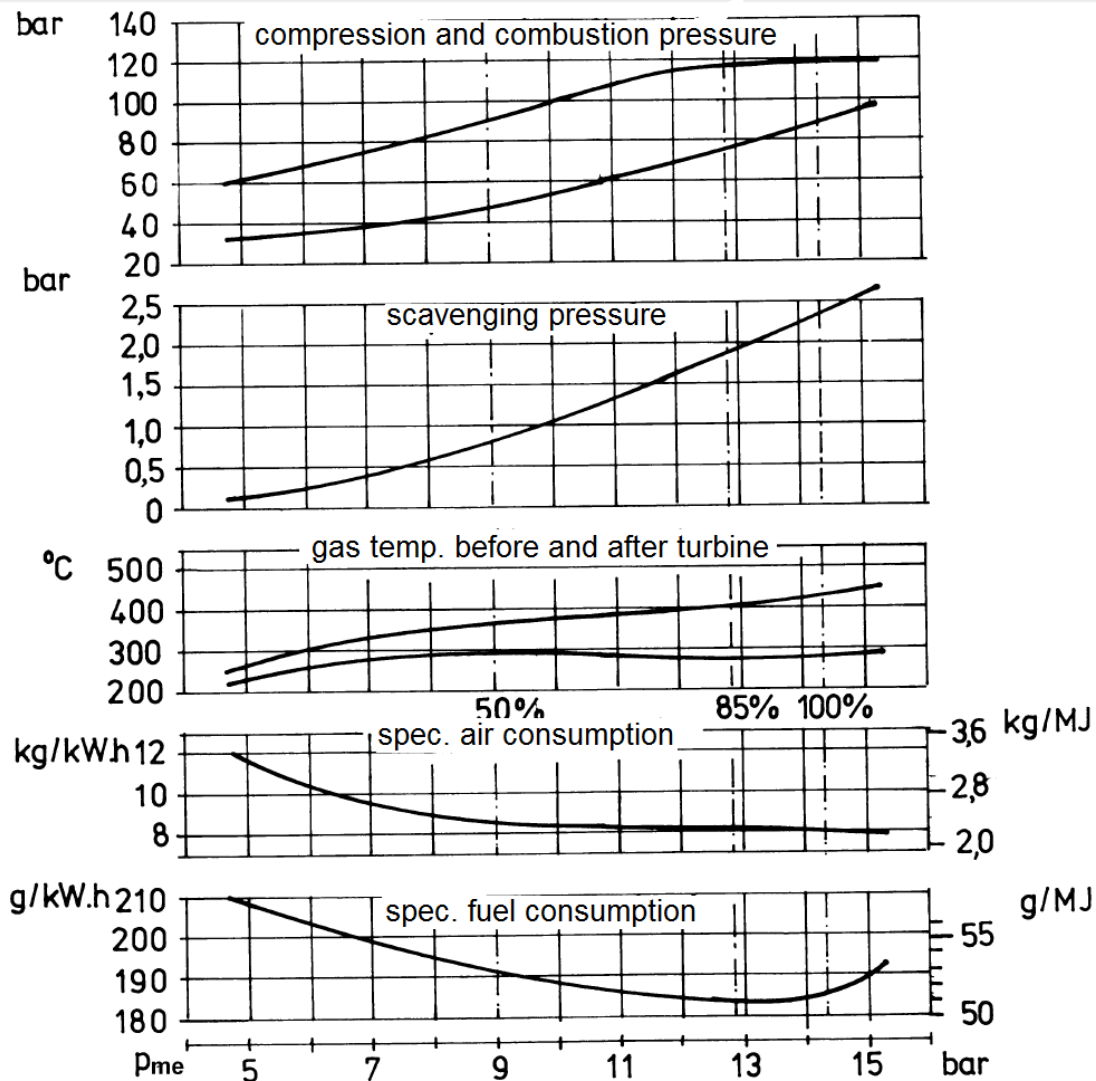


Figure 15 Performance chart

5.3.2 The different curves of a performance chart

a. Specific fuel consumption

§ 3.2.1 and § 5.2.4

b. Specific air consumption

The specific air consumption should increase together with the specific fuel consumption to realize a constant air excess. In reality a higher air consumption is observed at low p_{me} and p_{mi} values and vice versa.

The higher air consumption at low mean pressures in a two stroke engine is because of the necessity for scavenging: the minimal pressure is higher than the pressure needed for the combustion. Even the atmospheric pressure can be too high for low loaded four stroke engines.

At very high mean pressures it is difficult for the amount of air to get into the cylinder during the short time available.

c. Inlet pressure or scavenging pressure

When the efficiency of the scavenging and the inlet air temperature are constant then the amount of air, in the cylinder, should be proportional to the scavenging pressure. To realize a constant air excess the amount of fuel (and p_{me}) should also be proportional to this pressure. In other words this curve should be a straight line. At low and high pressures this straight line will overestimate the pressure (air resistance and initial scavenging pressure).

d. Compression pressure

This pressure should be proportional to the scavenging pressure. A relative loss of 1% can be observed at high loading conditions. This is due to non-perfect seals of the piston segments and because of the cooling of the cylinder. This loss may not exceed 1%.

e. Exhaust temperature

The exhaust temperature is an indication for the heat losses of the engine. A correct exhaust temperature is of high importance for the turbo compressor.

f. Amount of (black) smoke

When the exhaust gasses contain black smoke this is mostly because of an incomplete combustion. An incomplete combustion is easy to realize at very low and very high loading conditions. This emission is also of high importance for stationary engines because of emission and environmental regulations.

5.4 Power balance

5.4.1 Balance

This balance is the accounting of all the involved powers. Not all parts of the balance are possible to measure. For example the heat discharged via an oil circuit can be measured while the frictional losses are not; a part of the oil also cools down the system. Comparing different types of power balance of different engines is therefore not straightforward.

In the next examples the numbers are relative to the added power or in this specific case: fuel.

A four stroke engine:

Effective power:	40%
Exhaust:	35%
Fresh water cooling:	13%
Oil cooling:	6%
Air cooling:	4%
Other losses:	2%

For a two stroke engine:

Effective power:	40%
Exhaust:	32%
Air cooling:	12%
Cylinder cooling (water):	10%
Piston cooling:	3%
Other losses:	2%
Oil cooling:	1%

“Other losses” are all losses which cannot be measured or because of the inaccuracies of the measurements. These are for example the thermal radiation of the engine (0.5-1%).

Concluding remarks:

To develop the same power, the amount of air needed is higher for a two stroke engine. This is the reason for the higher losses because of the air cooling.

Contra intuitive a two stroke engine does not have significant higher exhaust losses although they consume more air. This is because of a combustion closer to the top dead point or a higher compression ratio.

When the losses due to air cooling are excluded then the losses due to cooling are lower for a two stroke engine: The dimensions of the cylinder are higher and therefore the combustion takes place at a higher distance from the cylinder walls.

5.4.2 Sankey diagram

All powers are represented as a flow in a Sankey-diagram. Losses are presented as a horizontal flow while the initial power and the effective power are represented as vertically oriented flows.

This type of representation is not only expressive but power lines existing in the engine can be presented as well (for example the exhaust gasses before entering the turbo compressor). The diagram can be extended with other machinery such as the evaporator and the waste heat boilers or even with the entire engine room. The electrical power (for oil or water pumps, scavenging fans) can also easily be added to the diagram (in Figure 16 this equals 1%)

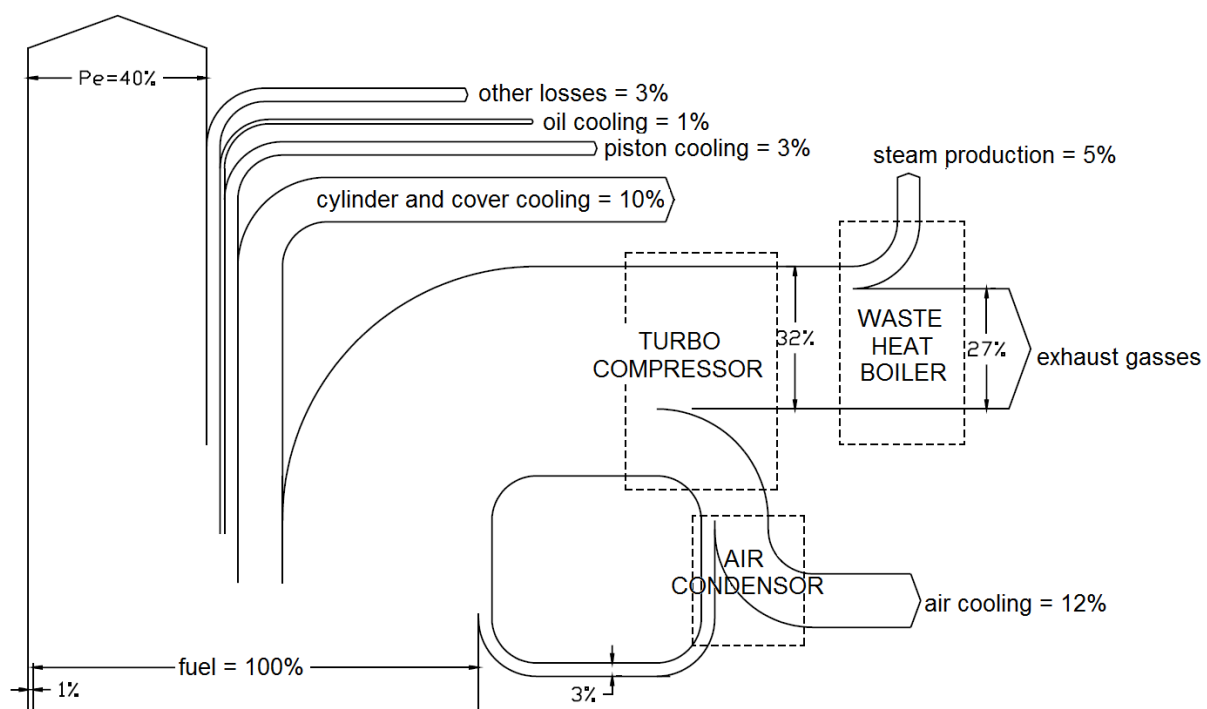


Figure 16 Sankey diagram

6 Ship Resistance

6.1 Different types of resistance

The hull of a sailing ship is influenced by different types of forces. These forces can be split upon two classes:

1. normal forces, perpendicular to the hull plating
2. tangential forces, directed along the hull plating

The normal forces are caused by the pressure of the surrounding water (both static and dynamic) while the tangential forces are caused by friction of the water along the hull .

All calculations are based upon the procedures as described by ITTC (International Towing Tank Conference). This is a well-established organisation with contribution from towing tanks all over the world.

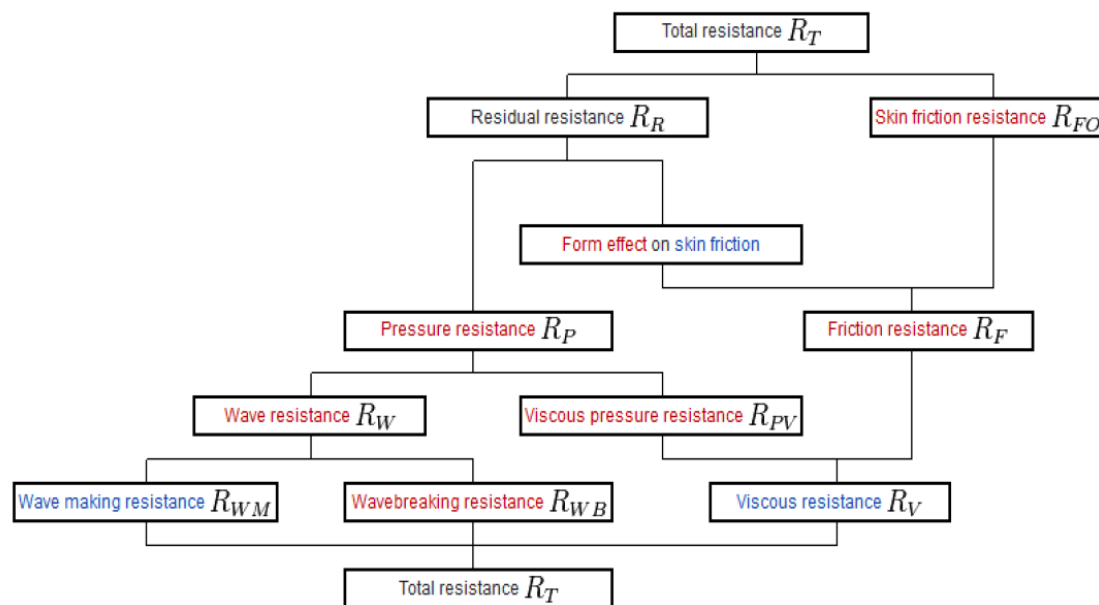


Figure 17 decomposition of different resistance components

6.1.1 Normal forces

Resistance to normal pressure exist out of two parts:

6.1.1.1 *Form resistance, viscous pressure drag*

Assume a plate is towed perpendicular to its plane, then this plate will experience a force opposite to the direction of motion. This force results from the pressures on the plate (overpressure at the front, under pressure at the back side). The same happens on a sailing vessel:

A sailing displacement vessel moves water around. This amount of water moves in the opposite direction of the vessel. For a forward sailing vessel, this water will be moved backwards (from bow to stern). The water particle that was initially in position A will be moved to position B. Based on Bernoulli's principle, an increase in speed will result in a pressure drop (the static water pressure will decrease).

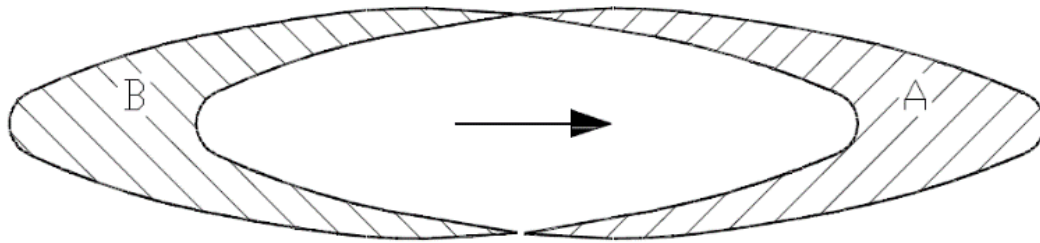


Figure 18

$$p + \frac{1}{2} \rho v^2 = C^{te}$$

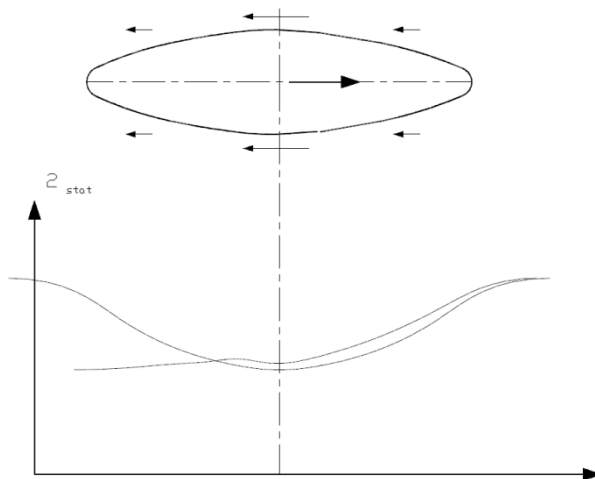


Figure 19

The static pressure will decrease from the bow towards the parallel midship section. As a consequence the static pressure should increase from the parallel midship section towards the stern of the vessel, at the end it should be back at the initial

value. The latter is in reality not true. A part of the velocity is 'lost' because of the friction *in* the water. As a result the pressure around the stern of the vessel will be lower than the pressure around the stem of the vessel.

The friction of the water generates a boundary layer. This boundary layer results in a mass of water 'stuck' to the hull of the vessel and travels at the same speed through the water. In shallow water this amount of water can be multiple times the mass of the vessel. This amount of added mass consumes a drastic amount of energy from the main machine of the vessel.

The boundary layer will detach from the hull when it becomes too wide. A wide wake will be the result. This is a high turbulent region induced by the interaction of the wake and the surrounding water. This turbulent region will prohibit the reconversion of the static pressure at the stern (an even lower static pressure is the result). This phenomenon is more prominent for towed vessels. When a vessel has a rotating propeller, the boundary layer will be attached longer to the hull but the suction force of the propeller will come into existence.

6.1.1.2 Wave resistance

A moving vessel will generate a combination of two wave patterns: transverse waves (primary wave system or Bernoulli wave system) and a Kelvin wave pattern (secondary wave system). Both patterns are created at the bow and at the stern.

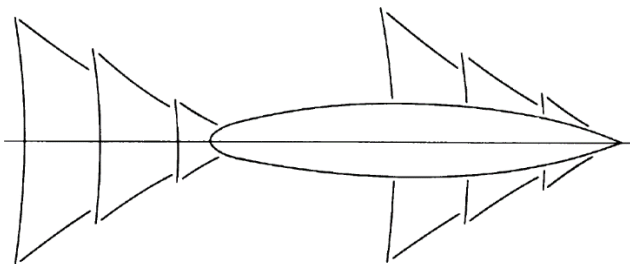


Figure 20

The waves at the free surface will cause a changing pressure along the hull and thus a changing resistance of the vessel.

The transverse waves travel at the same speed of the vessel (the transverse waves 'stand still' looking from the ship). In deep water for ships sailing at a low Froude number most energy is lost in the transverse waves and not in the Kelvin pattern.

The wave length of a wave travelling in deep water at a speed V is:

$$\lambda = \frac{2\pi V^2}{g}$$

For some speeds the bow wave and stern wave crest and troughs reinforce each other. To avoid this it is important that the transverse wave length equals: $\lambda(k + 0.5) = L_{pp}$

With:

k = an even number

L_{pp} = length between perpendiculars of the vessel

Sometimes not L_{pp} but the fictive length Z is chosen:

$$Z = C_p \cdot L_{pp} + \frac{\lambda}{4}$$

$$C_p = \frac{\frac{\nabla}{\rho}}{A_m L_{pp}}$$

C_p = prismatic coefficient []

A_m = midship area [m²]

This formula is based upon the principle that the first wave top is not located at the stem but a little bit more towards the stern, especially for narrow vessels.

An increased resistance comes into existence when $Z = (k + 0.5)\lambda$ or

$$C_p \cdot L_{pp} + \frac{2\pi V^2}{4g} = (k + 0.5) \frac{2\pi V^2}{g}$$

This will occur when the vessels speed equals

$$\frac{V}{\sqrt{L_{pp}}} \sqrt{\frac{C_p g}{2\pi(k + 0.25)}} = 1.25 \sqrt{\frac{C_p}{(k + 0.25)}}$$

The resistance curve will peak at these values.

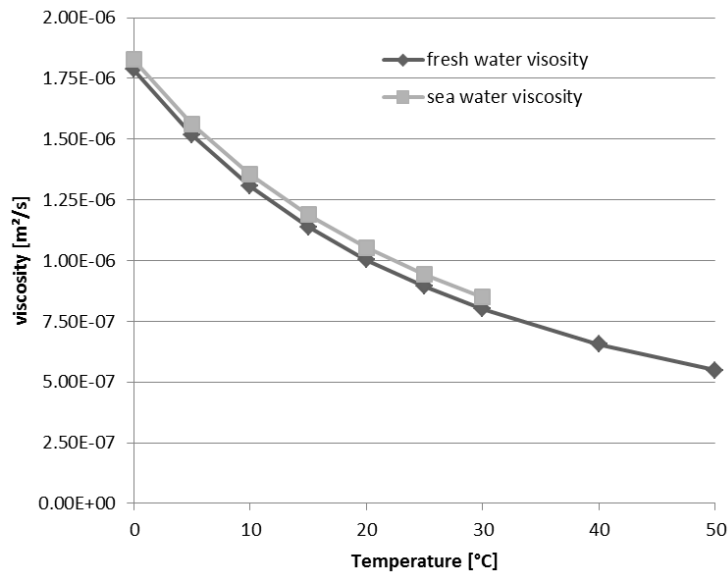
Practically the forward speeds according to k -values 1 and 2 are the more stringent. Slow ships with a long length will never have this negative wave interaction. Shorter and fast ships will endure this interaction. A design length and speed combination should be found to avoid this situation.

6.1.2 Tangential forces: frictional forces

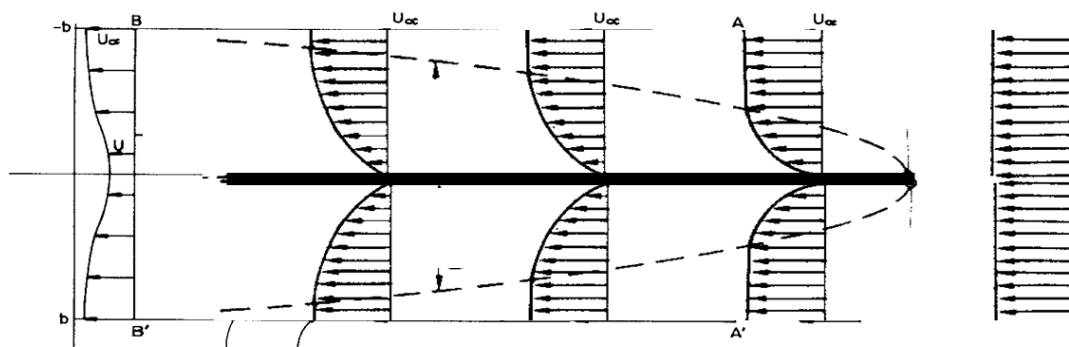
The skin friction resistance is caused by the viscosity of the seawater.

a. Temperature and density

The following figure shows the kinematic viscosity for fresh water and sea water at different temperatures:

**Figure 21**

When there is a high amount of sediment in the water (as at the harbour of Zeebrugge) the viscosity has to be increased according to predefined factors.

**Figure 22 Boundary layer on a flat plate**

b. Calculating the frictional resistance

The frictional resistance of a ship can be calculated based upon a generic function. This function is based upon the resistance of a smooth painted (infinitesimal) small plate and updated with some correction terms. The correction terms change with corrosion, surface roughness...

An (old) formula as proposed by William Froude for the frictional resistance is:

$$R_f = \left(1.366 + \frac{2.53}{2.68 + L_{pp}}\right) S \cdot V^{1.852}$$

With

R_f = frictional resistance [N]

S = wetted surface [m²]

The wetted surface can be predicted by

$$S = 2.6 \sqrt{\frac{\nabla}{\rho}} L_{pp}$$

The frictional resistance is made dimensionless C_f :

$$C_f = \frac{R_f}{\frac{1}{2} \rho V^2 S}$$

The dimensionless frictional coefficient C_f can be calculated according to the 1957 ITTC formula:

$$C_{f,ITTC} = \frac{0.075}{(\log_{10} Re - 2)^2}$$

Re is the Reynolds number (and together with the Froude number the most important dimensionless number in hydrodynamics). The Reynolds number is defined as:

$$Re = \frac{V \cdot L_{pp}}{\nu}$$

ΔC_f is added to C_f to take into account the roughness of the surface. Also for this ΔC_f there exists an empirical formula. A formula is proposed by Bowden-Davison:

$$\Delta C_f = \left[105 \left(\frac{k_s}{L_{pp}} \right)^{1/3} - 0.64 \right] \cdot 10^{-3}$$

k_s is the mean hull roughness (MHR). This is the peak to peak value measured over a distance of 50mm. For newly painted vessels a value between 95 and 125 μm and increasing with 20 $\mu\text{m}/\text{year}$ until a roughness of 225 μm is reached.

The formula as proposed by Townsin takes into account the average hull roughness which is the average of about 15 measurements of MHR.

$$\Delta C_f = \left[\left[44 \left(\frac{AHR}{L_{pp}} \right)^{1/3} - 10 \cdot Re^{1/3} \right] + 0.125 \right] \cdot 10^{-3}$$

Both formula are based upon hulls painted with TBT SPC paints (tributyltin self polishing copolymer paints).

The total frictional resistance becomes

$$C_f = C_{f,ITTC} + \Delta C_f$$

6.1.3 Total resistance

The total resistance is the summation of the form resistance (viscous, normal force), the wave resistance (non-viscous, normal force) and the frictional resistance (viscous, tangential force) augmented with some (minor) corrections.

Excluding these corrections the total resistance can be written as

$$R_T = R_D + R_w + R_f$$

Or in dimensionless form:

$$C_T = \frac{R_T}{\frac{1}{2} \rho V^2 S}$$

$$C_T = C_D + C_w + C_f$$

According to the ITTC 1978 procedures the form resistance can be written as a fixed percentage (k) of the frictional resistance of a bare hull (no roughness). This percentage is derived according to the Prohaska plot.

$$C_D = k \cdot C_{f,ITTC}$$

$$C_T = k \cdot C_{f,ITTC} + C_w + C_{f,ITTC} + \Delta C_f$$

$$C_T = (1 + k) C_{f,ITTC} + C_w + \Delta C_f$$

At slow speed the frictional resistance is more dominant than the wave resistance while the wave resistance becomes the most important resistance component at higher speeds.

6.2 Theoretical approximation of the resistance

Via a dimension analyses it can be proven that the resistance coefficient (as defined) is only dependant of the Froude number and the Reynolds number. For two geometrical identical ships it can be written that

$$C_T = F(Re, Fr)$$

F(): a function dependant on the hull geometry

Re: Reynolds number []

$$Re = \frac{V \cdot L_{pp}}{\nu}$$

Fr: Froude number []

$$Fr = \frac{V}{\sqrt{g \cdot L_{pp}}}$$

Measurements on a ship model tested at the same Froude number and Reynolds number as the full scale vessel will result in the same dimensionless total

resistance coefficient C_T . The only scale factor α when both numbers are equal at model scale and full scale is for $\alpha=1$. In other words when the full scale vessel is tested (index m stands for the model):

$$Re = \frac{V \cdot L_{pp}}{\nu} = \frac{V_m \cdot L_{pp_m}}{\nu_m}$$

$\nu = \nu_m$ (kinematic viscosity of water)

$$V \cdot L_{pp} = V_m \cdot L_{pp_m}$$

$$Fr = \frac{V}{\sqrt{g \cdot L_{pp}}} = \frac{V_m}{\sqrt{g \cdot L_{pp_m}}}$$

$$\frac{V}{\sqrt{L_{pp}}} = \frac{V_m}{\sqrt{L_{pp_m}}}$$

$$\Rightarrow L_{pp_m} = L_{pp}$$

$$\frac{L_{pp_m}}{L_{pp}} = \alpha = 1$$

To avoid this issue empirical solutions are proposed.

6.3 Practical solutions to predict the resistance

6.3.1 Concept



Figure 23 William Froude
1810-1879

In the 19th century William Froude built the first towing tank. To overcome the contradiction as explained in 6.2 he assumed that the frictional resistance is only dependant of the Reynolds number while all the other resistances are dependant of the Froude number:

$$C_T = (1 + k)C_{f,ITTC}(Re) + C_w(Fr) + \Delta C_f$$

The frictional resistance is calculated based upon the Reynolds number and the formula as proposed by the ITTC in 1957 while C_w and k (Prohaska number) are defined during model tests carried out in a towing tank with (small) ship models (geometrical identical) and scaled according to the Froude law.

6.3.2 ITTC 1978 calculation method

Only the method for model tests for ships without a propeller are explained. Often models are equipped with a scaled propeller to test the influence of the propeller on the hydrodynamics of the model. This effect will be explained in the next chapter of this syllabus.

Model tests are carried out in a towing tank. The model is rigidly attached to the main carriage of the towing tank and towed in the tank at a wide range of forward speeds. During these tests the longitudinal force is measured.

- $C_{f,ITTC}$ is calculated
- R_T is measured and C_T calculated for all speeds
- ΔC_f is calculated and subtracted from C_T the result is C_R
- For very low forward speeds R_T is measured and the wave resistance is neglected, the only remaining force results in the calculation for k
- C_w can be calculated
- All dimensionless coefficients are Froude dependant and are equal for the ship model as well as the full scale ship.
- C_f can be calculated for the full scale ship
- C_T can be calculated for the full scale ship
- Corrections for the full scale ship are added (wind, waves etc.)

As such, the total resistance of the full scale vessel is known before the ship is built or full scale trials are carried out.

6.4 Corrections

6.4.1 Bulbous bow

A bulbous bow influences both the frictional resistance as well as the wave resistance.

a. Wave resistance

A bulbous bow generates a wave pattern that (partly) compensates the wave pattern generated by the stem. Therefore the total wave resistance will decrease because of the presence of a bulbous bow.

The increased buoyancy at the bow section will decrease the draft at the forward perpendicular and so decreases the Kelvin waves.

b. Frictional resistance

A more smooth flow at the stem facilitates a more easy recuperation of the energy at the stern. The suction effect at the stern will therefore decrease.

For high speed vessels $\left(\frac{V}{\sqrt{Lpp}} = 1.8\right)$ a bulbous bow decreases the total resistance at the design speed with 15%.

For slower vessels $\left(\frac{V}{\sqrt{Lpp}} = 1\right)$ the total resistance at design speed will decrease between 5 and 20% depending on the loading condition of the vessel.

6.4.2 Appendages

The appendages (rudder, struts,...) will generate frictional resistance. Model tests are carried out with a bare hull (no appendages) and with all appendages. Based

upon the difference between both tests the influence of the appendages on the total resistance of the vessel can be calculated. It is supposed that the resistance of the appendages R_a is linear with the square of the forward speed:

$$R_a = c_a \cdot S \cdot V^2$$

With

c_a = a coefficient derived with model tests

s = surface of attack [m²]

6.4.3 Air resistance

According to ITTC the air resistance can be defined as

$$R_{AA} = \frac{1}{2} \rho_{air} V_{air}^2 A_x \cdot C_{AA}$$

R_{AA} : the air resistance

ρ_{air} : density of the air

V_{air} : relative speed between ship and air (combination of wind and forward speed of the vessel)

A_x : lateral area above the water line

C_{AA} : coefficient depending on the geometry above the water line and the direction between ship and wind. This coefficient is defined in a wind tunnel.

6.4.4 Sea state

The influence of the sea state on the resistance of a vessel is expressed with the wave height.

The formula of Kreitner for head waves is:

$$\Delta R_0 = 0.64 H_s^2 C_b \frac{B^2}{L_{pp}} g \rho$$

With

H_s : significant wave height [m]

When the waves encounter the vessel at an angle α

$$\Delta R = \Delta R_0 \left(\frac{2}{3} + \frac{\cos \alpha}{3} \right)$$

6.4.5 Bilge keel

The main function of a bilge keel is to limit the roll motion of a vessel. The bilge keel will result in a higher wetted surface and therefore in a higher frictional resistance.

Because the thickness of a bilge keel is difficult to realize on model scale sometimes the bilge keel is not attached to the scale model. In this situation the wetted surface of the bare hull S is augmented with the wetted surface of the bilge keel S_{BK} and the summation $(S+S_{BK})$ is used to calculate the frictional resistance of the vessel.

6.4.6 The total resistance coefficient at full scale

The total resistance coefficient at full scale (without the resistance caused by the appendages and sea state) becomes

$$C_{TS} = \frac{S + S_{BK}}{S} \left((1 + k) C_{f,ITTC,S} + \Delta C_f \right) + C_R + C_{AA}$$

6.5 Exercises

6.5.1 A capsized passenger ship

The length of a capsized passenger ship is 81 times larger than the length of the geometrical identical model. How fast should the model (in upright position) be towed in a towing tank to be able to calculate the resistance according to 17.5 knots at full scale?

6.5.2 A container carrier

A container carrier has at model scale a waterline surface of 2.0m². At full scale this surface is 9800m². What is the forward speed of the model in the towing tank which corresponds with 20 knots full scale?

6.5.3 A bulk carrier

The displacement of a bulk carrier is at full scale 15625 times heavier than the geometrical identical ship model. How fast should the model be towed in a towing tank to be able to calculate the resistance according to 17.5 knots at full scale?

7 The propeller

7.1 Quantity of motion

7.1.1 The thrust

In this initial stage of the explanation of the working process of a propeller the effect of the wake is neglected. The velocity of the vessel equals, with this assumption, the velocity of the water at the propeller.

The propeller accelerates the velocity of the water (and thus the quantity of motion Q) ($Q = \text{mass} \cdot V$). Based upon the classic theory the force T (Thrust) is defined as:

$$T = \frac{dQ}{dt} = \frac{d(mV)}{dt} = m \frac{dV}{dt} = \frac{m}{dt} dV = \dot{m} dV$$

To know the thrust generated by a propeller only the quantity of motion has to be known.

If

$\dot{m}$ = the mass of water which runs through the propeller per second [kg/s]

V = forward speed of the vessel (and the propeller) [m/s]

a = a dimensionless value which indicates the absolute speed of the amount of water that interacts with the propeller (on the propeller) []

b = a dimensionless value which indicates the absolute speed of the water leaving the propeller (behind the propeller) []

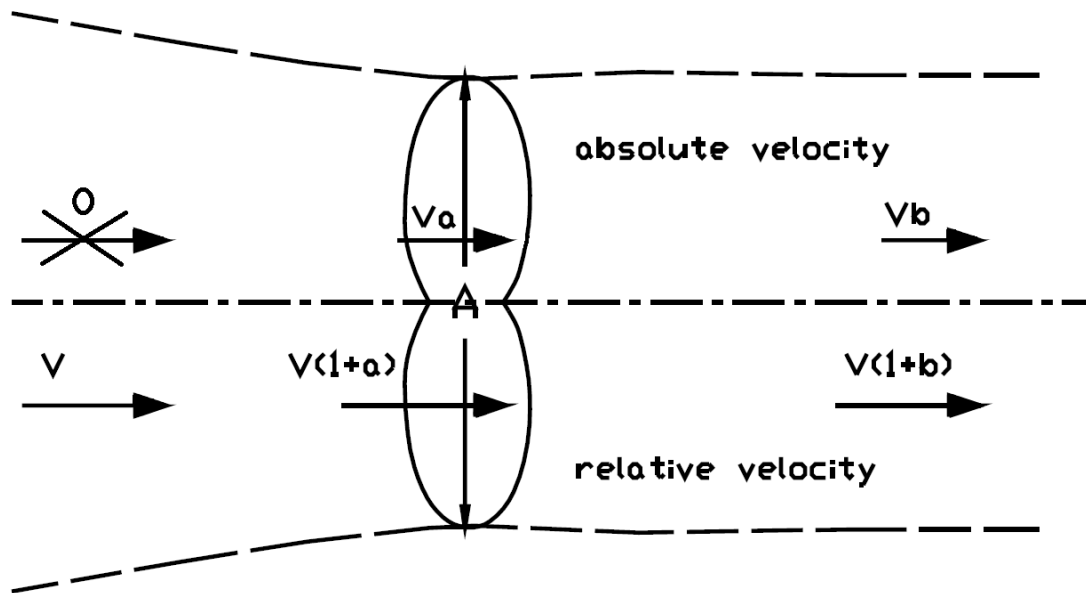
A = area of the propeller disc [m²] (with D the propeller diameter [m])

$$A = \pi \left(\frac{D}{2} \right)^2$$

From here on it is assumed that all water particles move without turbulence and only in the direction opposite to the forward speed of the vessel (first quadrant).

The mass of water that flows through the propeller per time step equals

$$\dot{m} = \rho A V (1 + a)$$

**Figure 24**

The thrust T is

$$T = \frac{dQ}{dt} = \frac{m \Delta V}{\Delta t} = \dot{m} \Delta V = \dot{m} V b$$

T is not only the force generated by the propeller to push the water backwards but also the force of the water on the propeller (and the ship).

7.1.2 The effective propulsion power on the propeller shaft

The effective propulsion power on the propeller shaft is the product of the thrust and the absolute value of the velocity of the water.

$$P_u = T \cdot V = \dot{m} \cdot V \cdot b \cdot V = \dot{m} V^2 b$$

7.1.3 The power to be delivered at the water

a. Based upon the thrust T

The thrust T also acts on the water, this water translates at a speed a times V at the point of application of this force. The necessary work per second is

$$P_w = T \cdot V \cdot a = \dot{m} \cdot b \cdot V \cdot V \cdot a = \dot{m} \cdot V^2 \cdot a \cdot b$$

b. Based upon the kinetic energy

The water is initially at rest, every second an amount of kinetic energy is added to this water. This energy equals the work per second on the water ($=P_w$).

$$P_w = \Delta E = \frac{\dot{m}(Vb)^2}{2}$$

c. Conclusion

The combination of the deduction from thrust and kinetic energy (and all previous assumptions) results in:

$$\frac{b}{2} = a$$

7.1.4 Propeller efficiency

The propeller efficiency η_o is the ratio between the effective propulsion power on the propeller shaft and the power to be delivered. This efficiency is the highest efficiency which theoretically can be obtained because the latter power is still based on the absence of turbulence.

$$\eta_o = \frac{P_u}{P_u + P_w} = \frac{\dot{m}V^2b}{\dot{m}V^2b + \frac{\dot{m}(Vb)^2}{2}} = \frac{1}{1 + \frac{b}{2}}$$

To optimize the propeller efficiency the b -value should be as low as possible. The rate of water mass ($\dot{m}$) should be large in order to have more thrust ($T = \dot{m}Vb$). This can be obtained easiest with a large propeller surface A . Or the bigger the propeller diameter the better. The propeller diameter however is limited by practical boundary conditions such as the draft of the vessel and the minimal distance between the hull and propeller to avoid cavitation.

7.2 Propeller geometry

7.2.1 Propeller concept

The initial aim of the propeller is to add a uniform axial velocity to the surrounding water. This is obtained (for a classic propeller) by designing the propeller blades as a helicoidally curved surface, the helicoid.

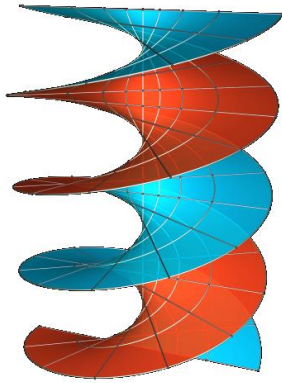


Figure 25 www.indiana.edu/~minimal/archive/Classical/Classical/Helicoid

7.2.2 Development of the propeller

A cylindrical cross section of a helicoidally curved surface results in a helix. A translation p_i (pitch) is the result of the development of this cylindrical surface over one turn. β is the angle between the helix and the axis.

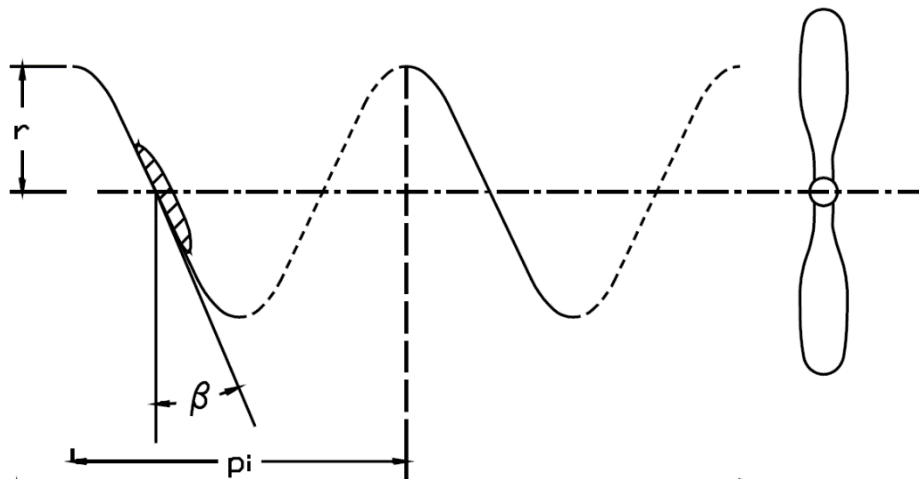
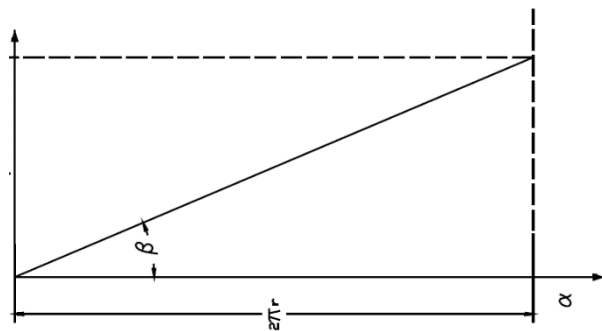


Figure 26 pitch and the angle between helix and axis

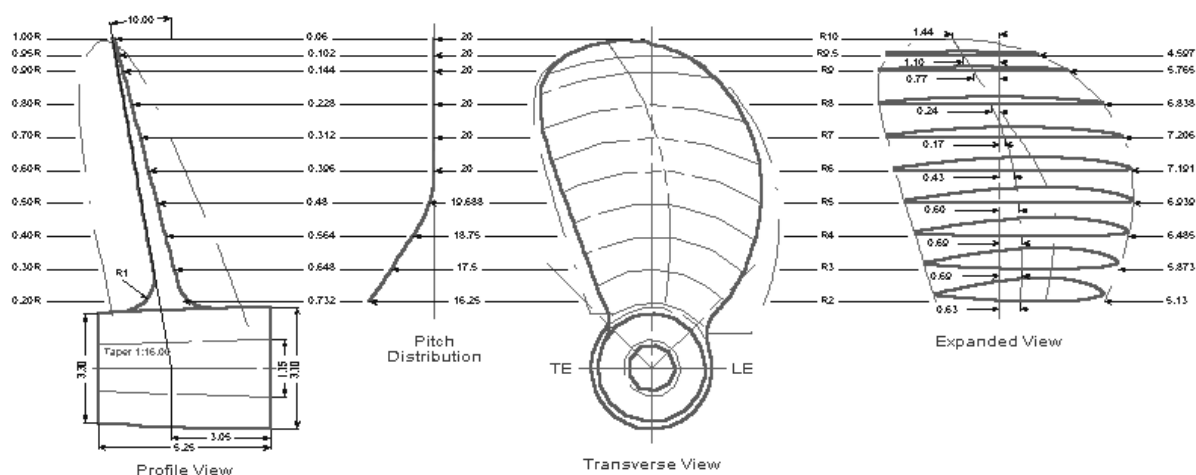
The pitch should be constant at every position r from the axis on the propeller when a relative velocity $V(1+b)$ is desired to be added to the water. Therefore the angle β is dependant of the position from the propeller shaft.

$$\tan \beta = \frac{p_i n}{2\pi r n} = \frac{V(1+b)}{2\pi r n}$$

**Figure 27**

For a controllable pitch propeller the pitch is changed with a constant value over the entire blade. As a result a constant pitch is only obtained for one position of the blade on the shaft.

The pitch is constant for a fixed pitch propeller from the tip towards the propeller root but at the last 10% the pitch decreases. The geometrical pitch is defined as the mean of the total pitch at every position.

**Figure 28 Technical drawing one propeller blade**

Sometimes the blades are not perpendicular positioned on the axis but are bent backwards. This angle is known as the rake. An advantage of rake is that the propeller can be positioned closer to the hull, then the propeller shaft can be shorter and will bent less.

The projection of a blade on a surface perpendicular to the propeller shaft can be symmetrical or not. This deviation angle is known as skew.

The thickness of a propeller blade decreases from root to tip (the loading of the propeller also decreases from root to the propeller tip).

7.3 Foil theory

7.3.1 The angle of attack

The cross section of a propeller blade is a (hydro) foil (as seen on the right side of Figure 28). The angle of attack i is known as the angle between the (longitudinal) axis of the foil and the relative direction of the flow of the water (Figure 29).

Based upon the classic theory the angle of attack can be calculated.

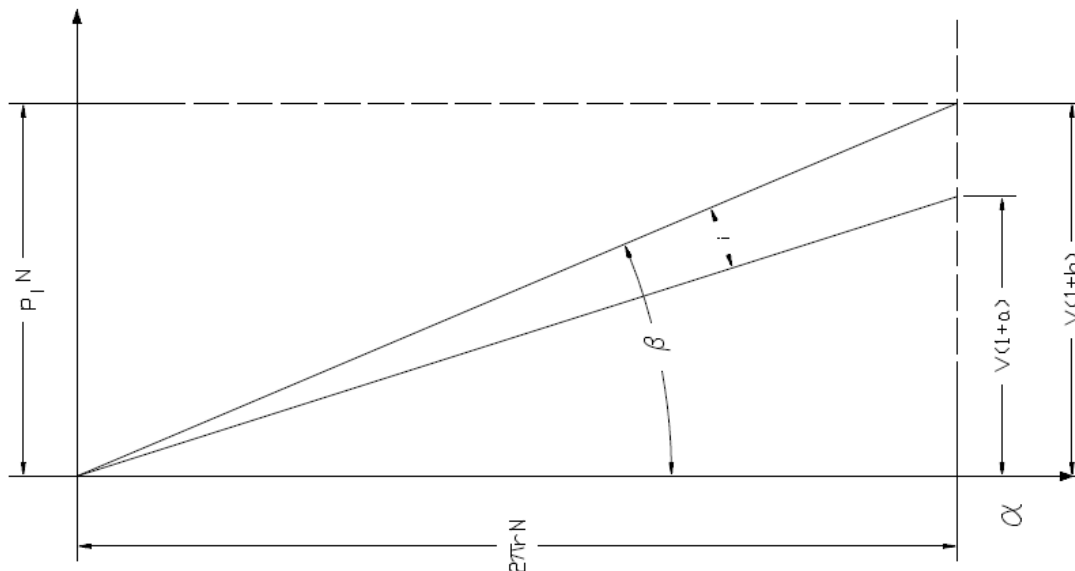


Figure 29 angle of attack (i)

The water travelling along the hydrofoil generates thrust. With an increasing angle of attack the lift will increase. Three physical phenomena generating lift can be distinguished:

- Change in the velocity of the flow
- Change of the direction of the flow
- Creation of turbulence

a. Change in the velocity of the flow

A pressure difference is generated between both sides of a hydrofoil because of its specific geometry. A water particle which travels at the suction side of a hydrofoil (upper part in Figure 30) will have travelled in the same amount of time a longer distance than a neighbouring water particle which has travelled the foil at the pressure side. Based upon the Bernoulli principle a pressure difference is generated. (This is also the principle why not every plane drops out of the sky).

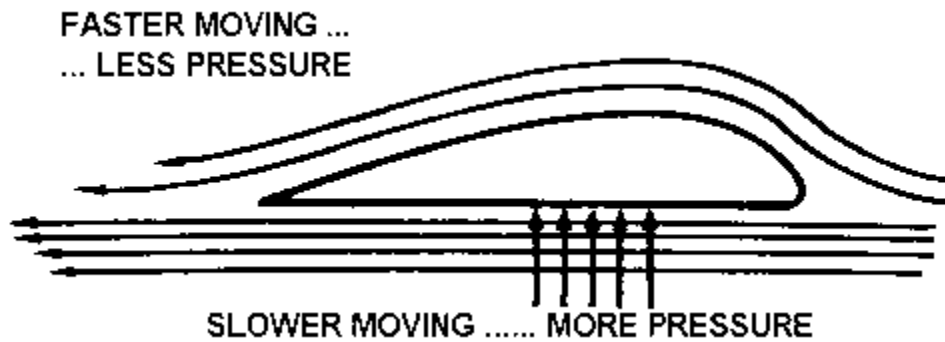


Figure 30 pressure difference because of different travelled distances

b. Change in the direction of the flow

The flow will change its direction more with an increasing angle of attack of the profile (until stall occurs). The kinetic energy delivered by the flow on the profile will transform into pressure on the profile.

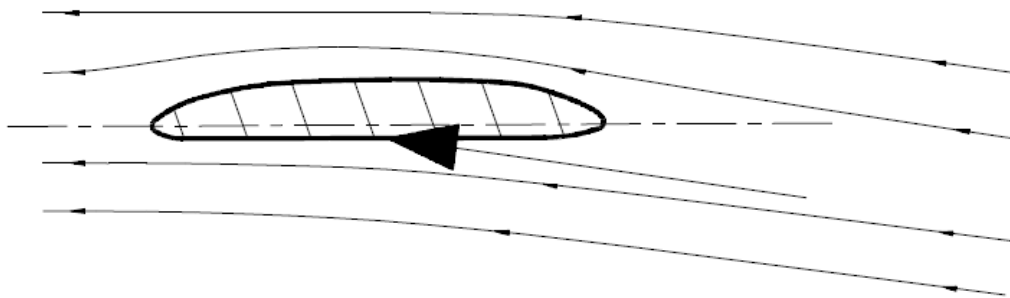


Figure 31 change of flow direction

7.3.2 Cavitation

The profile has a suction side (low pressure) and pressure side (high pressure). When the pressure at the suction side is lower than the damp pressure of the fluid, cavitation will come into existence. The water on the suction side will transform into water vapour (not air!), bubbles come into existence. This phenomenon is known as cavitation (similar as in a pump) and has a negative influence on the efficiency and durability of the propeller.

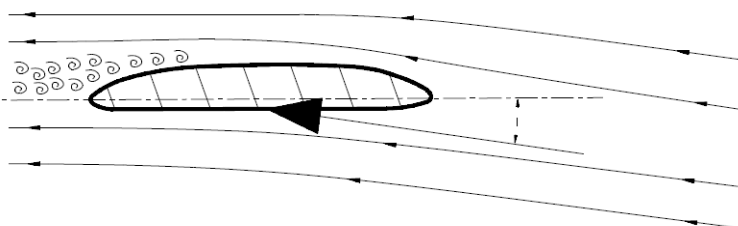


Figure 32 principle of cavitation



Figure 33 Cavitation on a CPP as seen in a cavitation tunnel

The impact on the propeller when the water vapour retransforms into water (with an implosion) may cause severe damage.

Some remarks:

- Figures 30 to 32 are a simplification of a three dimensional flow. The flow takes place not only in the cylindrical surface but there is also a motion towards the propeller tips. Therefore there is a higher risk for cavitation at the propeller tips, therefore, at these tips the pitch is sometimes reduced.
- The pressure difference between both sides remains up to the edge of the blade. This will introduce vortices leaving the propeller blade. These vortices introduce new losses and may cause vibrations/noise on the hull. At the tip of the blades these vortices are even more prominent. CLT-propellers (Contracted and Loaded Tip) are designed to minimize these losses.



Figure 34 CLT-propeller

7.3.3 Hydrofoil theory for a blade section

A propeller translates with velocity V through an amount of water at rest (no wake). The water will accelerate due to the suction effect of the propeller. Therefore the water impacts the blade with a velocity $U=(1+a)V$. The rotation of the water induced by the propeller is neglected.

The tangential acceleration at a position on the propeller with a distance r from the rotation axis and propeller rate n equals $2\pi n r$.

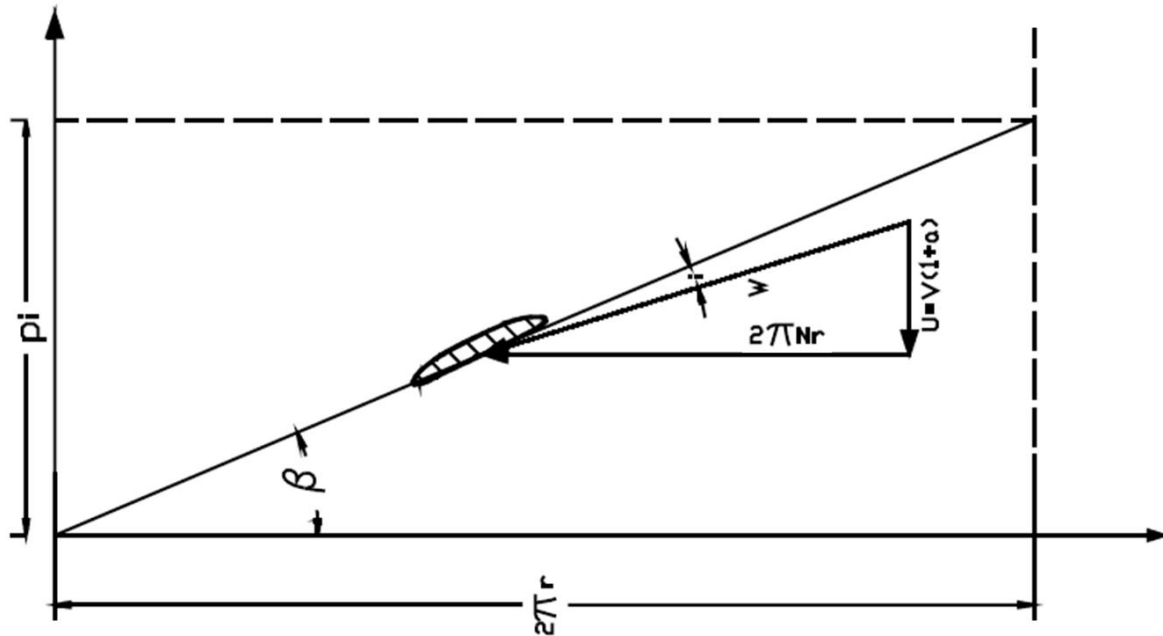


Figure 35

W is the relative velocity of the water on the propeller blade. W is the combination of the previous two velocities. This velocity W enters the blade at an angle i . This angle i is an indication for the pressure and forces on the propeller blades.

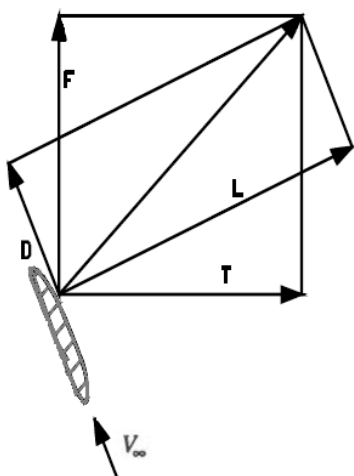


Figure 36

The resultant force R can be split according to two principles:

- into a drag force D (parallel to the undisturbed flow) and a lifting force L (perpendicular to the undisturbed flow)
- into a force T according to the direction of the axis (pushing the ship forward) and a component F according to the motion of rotation of the propeller.

The summation of all the forces T for all (parts) of the propeller blades results in the total thrust T . The forces F can be combined in the total torque needed. This torque will define the power needed for the propeller (not the Thrust!).

This power and delivered thrust T is defined by the relative velocity W (and thus by n and V for each propeller). When the first assumption does not stand the velocity V has to be replaced by V' .

In a towing tank or cavitation tunnel the following functions are defined for each propeller:

$$T = f(n, V')$$

$$P_h = f(n, V')$$

$$\eta_h = f(n, V')$$

7.4 Graphical representation of the propeller properties

7.4.1 The advance ratio

Based upon a dimension analyses the thrust can be written as:

$$T = \rho D^2 V'^2 \cdot f\left(\frac{Dn}{V'}\right) \cdot f\left(\frac{\nu}{DV'}\right) \cdot f\left(\frac{p}{\rho V'^2}\right) \cdot f\left(\frac{gD}{V'^2}\right)$$

With

D = propeller diameter

V' = the velocity of the water before the propeller

n = propeller rate

ρ = density of the water

ν = viscosity of the water

p = static pressure

When the thrust is measured (in a towing tank, cavitation tunnel or at full scale) these four (dimensionless) functions should be taken into account. These four functions can be rewritten:

$$f\left(\frac{\nu}{DV'}\right) = f\left(\frac{1}{Re}\right) \text{ with } Re \text{ the dimensionless Reynolds number}$$

$$f\left(\frac{gD}{V'^2}\right) = f\left(\frac{1}{Fr^2}\right) \text{ with } Fr \text{ the dimensionless Froude number}$$

$$f\left(\frac{Dn}{V'}\right) = f\left(\frac{1}{J}\right) \text{ with the advance ratio } J$$

J is the advance ratio and the most defining parameter for a propeller. Different propellers will be compared at the same advance ratio J .

7.4.2 The propeller efficiency

a. The propeller efficiency and advance ratio

The thrust can be written as:

$$T = \rho D^2 V'^2 \cdot f(J)$$

$$T = \rho (JnD)^2 D^2 \cdot f(J)$$

$$T = \rho n^2 D^4 \cdot f(J)$$

$$T = \rho n^2 D^4 \cdot K_T$$

For the torque a similar expression is found:

$$M = \rho n^2 D^5 \cdot f(J)$$

$$M = \rho n^2 D^5 \cdot K_Q$$

and for the open water efficiency of the propeller η_h :

$$\eta_h = \frac{1}{2\pi} \frac{K_T}{K_Q} J$$

All the important properties of a propeller are visualised by plotting the functions K_T , K_Q and to the advance ratio J with:

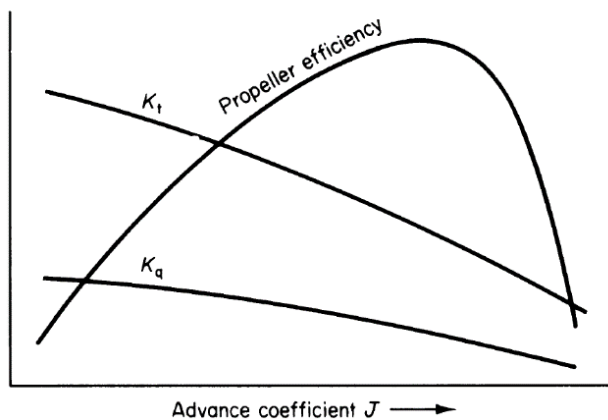


Figure 37 Propeller diagram

b. Representation according to Taylor (outdated)

Taylor introduced a graph, based upon a systematic series of model tests, to choose the properties of a propeller when the required forward speed and power to be delivered is known.

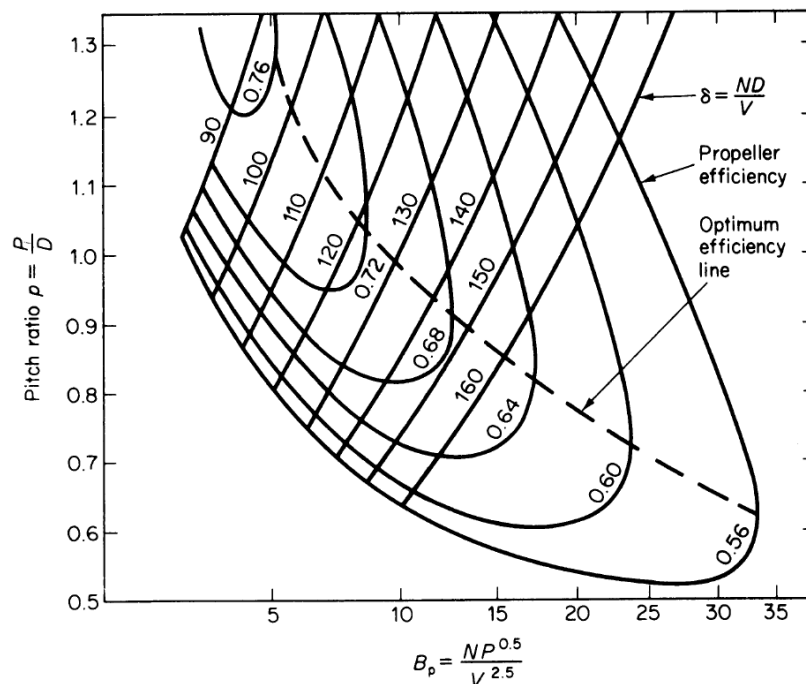


Figure 38 Taylor propeller graph

First B_p is calculated based upon the required propeller rate n , power P and forward speed V . The calculated value B_p crosses the optimum efficiency line. The advance ratio J can be found and the propeller diameter D calculated. On the vertical axis the optimal pitch ratio can be found. Remark that Figure 35 is not in SI units (but in rpm, knots, feet and horse power).

7.5 Power and thrust of a propeller

Tests are carried out in a towing tank with a ship model with a controlled propeller.

7.5.1 Froude number

The propeller at model scale is geometrical identical (but scaled) as the full scale propeller. For standard tests carried out in the towing tank the Froude number at model scale and full scale is the same and equals:

$$Fr = \frac{V}{\sqrt{gD}} \approx \frac{nD}{\sqrt{gD}} = n \sqrt{\frac{D}{g}}$$

D is the diameter of the propeller or sometimes 75% of the cord length of the propeller blade. (The ratio of Froude numbers based upon the ship length will also be equal).

7.5.2 Reynolds number

When a propeller is scaled only according to the Froude number the thrust at model scale will be (relatively) lower compared to full scale. The Reynolds number at model scale will be much lower than the full scale Reynolds number because the friction is at model scale (relatively) much higher. To compensate this lower thrust force at model scale, an additional force F_D is sometimes used:

$$F_D = \frac{1}{2}(C_{FM} - C_F)\rho S_M V_M^2$$

7.5.3 Test procedure for the open water diagram of a propeller

The propeller has to deliver a thrust T to the vessel. This force T must be higher than the total resistance R_T because of the suction effect of the propeller.

The K_T - K_Q diagram is delivered together with the propeller (model tests are carried out in a cavitation tunnel by the manufacturer of the propeller and according to the Reynolds number).

For every propeller rate n_{TM} , the thrust T_{TM} is measured. Now the value K_{TM} can be calculated as well as J .

$$K_T = \frac{T}{\rho n^2 D^4}$$

$$J = \frac{V'}{Dn}$$

When the advance ratio J is known, the velocity V' can be calculated (because n_{TM} and D are also known).

The torque on the propeller M can be measured during the model test and K_{QM} calculated.

$$K_Q = \frac{M}{\rho n^2 D^5}$$

From the table K_{QTM} is known (based upon the advance ratio J)

Based upon K_{TM} and K_{QM} the values K_{TS} and K_{QS} of the full scale vessel can be calculated (ITTC Rec. Proc. 7.5-0203-01.4). The required power is derived from K_{QS}

Comparing K_{QM} (from measurement at model scale) and K_Q (from table) shows the influence of the hull on the propeller.

7.6 Sea trials

Remember:

P_e = effective power, this power is communicated by the manufacturer of the main engine [kW]

T = Thrust delivered by the propeller [N]

n = Propeller rate [rpm]

V' = the axial velocity of the water in front of the propeller and relative to the propeller [m/s]

R = ship resistance [N]

It is assumed that the required power P_h is delivered with the data sheets of the propeller as well as the thrust:

$$P_h = f(V', n)$$

$$T = f(V', n)$$

Between the effective power of the main engine and the thrust some losses are introduced. The global efficiency η_g can be written as:

$$\eta_g = \frac{R \cdot V}{P_e}$$

Sea trials are important for the comparison between the measured parameters at full scale and these parameters predicted based upon model scale measurements and calculations.

- Losses due to appendages
- The suction effect of the propeller
- The wake effect
- The propeller efficiency
- The efficiency of the propeller shaft transmission
- The interaction of the propeller and a specific hull

7.6.1 Losses due to appendages

Sometimes, the resistance is only known for a bare hull without appendages such as stabilisers and bilge keel. R is the bare hull resistance while R' is the resistance with all appendages (rudder, propeller, stabilizers...). The appendage efficiency is:

$$\eta_a = \frac{R}{R'}$$

η_a is under normal conditions more than 0.9

7.6.2 The suction effect of the propeller

a. Suction effect of the propeller

The resistance of a ship is measured in a towing tank with an appended hull (with propeller(s) and rudder(s) and without appendages (bare hull)). The suction effect of a propeller creates a decreasing pressure region around the stern. This pressure drop 'pulls the vessel back'. This is why the ratio R'/T will be lower than 1. A part of the thrust is used to pull the vessel back. This phenomenon is referred to as the thrust deduction t . The thrust deduction t can be measured on a ship model with appendages.

$$t = \frac{T - R'}{T}$$

t is mostly between 0.07 and 0.20 (the lower the better)

b. Wake

The relative velocity at the propeller would be V if there was no wake. Because of the wake this velocity becomes V' .

The wake fraction w can also be derived from model tests with

$$w = \frac{V - V'}{V}$$

w is mostly between 0.30 and 0.05 and is beneficial for the hull efficiency (the higher the better).

c. Hull efficiency

The hull efficiency combines the effect of the wake fraction and thrust deduction. The hull efficiency is the ratio between the needed power by the hull and the needed power to be delivered by the propeller.

$$\eta_c = \frac{R' \cdot V}{T \cdot V'}$$

$$\eta_c = \frac{1 - t}{1 - w}$$

η_c is a value between 0.85 and 0.95

7.6.3 Open water propeller efficiency

The propeller efficiency in open water (delivered by the manufacturer of the propeller) is the ratio between the effective power of the propeller (in calm water) and the power to be delivered on the propeller shaft.

$$\eta_h = \frac{T \cdot V'}{P_h}$$

η_h is a value between 0.50 and 0.75 and depends on the friction and turbulence created by the propeller.

7.6.4 The interaction between the propeller and hull

The power to be delivered to the propeller may differ from the theoretical value (because of the complex turbulences at the location of a propeller in the wake of a vessel). The adaptation efficiency is the ratio between the real and theoretical power;

$$\eta_A = \frac{P_h}{P_{hr}}$$

η_A varies between 0.95 and 1.05.

7.6.5 The efficiency of the propeller shaft transmission

This is the ratio between the power on the propeller and the power delivered by the engine P_e .

$$\eta_l = \frac{P_{hr}}{P_e}$$

common range for η_l is between 0.95 and 0.98.

7.6.6 Overall propulsion efficiency

The global propulsion efficiency becomes:

$$\eta_g = \frac{R \cdot V}{P_e}$$

Or

$$\eta_g = \eta_a \cdot \eta_c \cdot \eta_H \cdot \eta_A \cdot \eta_l$$

Common values for η_g are between 35 and 65%

7.7 Other propeller types

7.7.1 Grimm wheel

The Grimm wheel recuperates energy from the wake.

A second free rotating propeller (Grimm wheel) is added behind the conventional propeller. The blades of the Grimm wheel have two functions:

- The section of the blades closest to the shaft work as a turbine driven by the wake produced by the propeller
- The top section of the blades have a different pitch and work as a propeller.

The total diameter of the propulsion system increases because the Grimm wheel has to be larger than the conventional propeller.

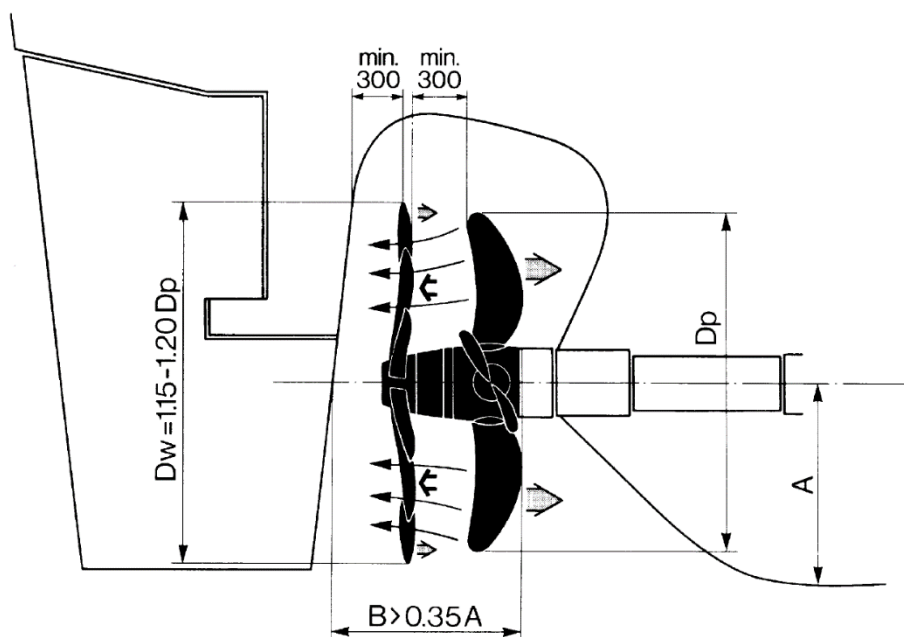


Figure 39 Grimm Wheel

7.7.2 Azimuth thruster

This system combines a propeller and rudder. A propeller is attached under the hull with the propeller sometimes in front of the driving system. The entire system can rotate around a vertical axis making it possible to deliver thrust in all directions. This creates a very manoeuvrable vessel, also at low speed. Therefore Azimuth thrusters can be found on cruise vessels and ships with dynamic positioning.

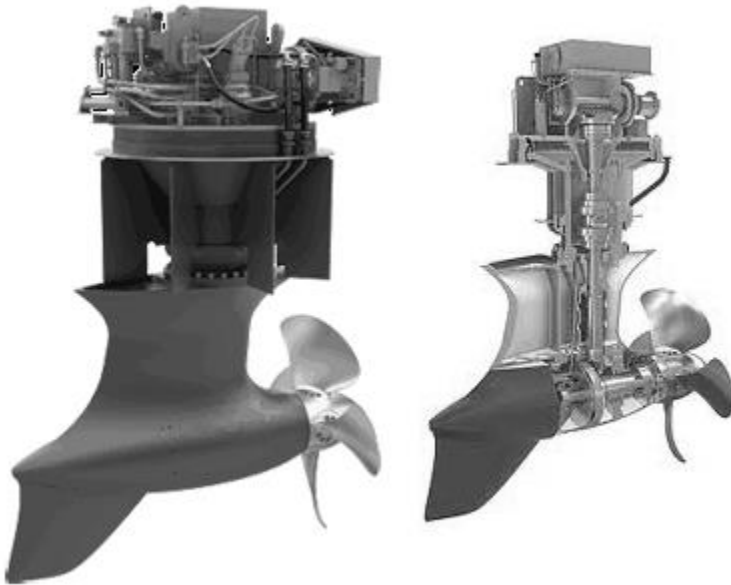


Figure 40 http://www.rolls-royce.com/marine/products/propulsors/azimuth_thrusters/



Figure 41 contra rotating
http://www.steerprop.com/userfiles/image/sp_eco.jpg

Azimuth Thruster

7. 7.3 Contra rotating propeller

Two propellers are attached and have a different direction of rotation. Both propellers are driven by the main machine (different than Grimm wheel). Contra rotating propellers can also be used in an azimuth configuration (Figure 32).



Figure 42 http://www.rolls-royce.com/marine/products/propulsors/azimuth_thrusters

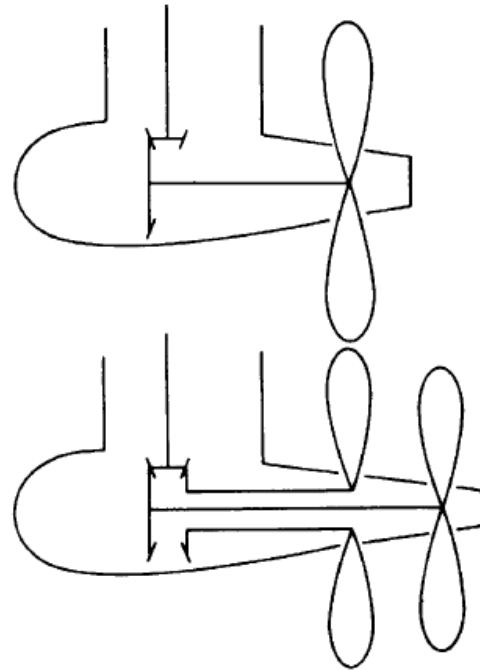


Figure 43

7. 7.4 Voith Schneider propeller

A Voith Schneider propeller (VSP) creates thrust by rotating vertical positioned blades. The rotational speed and angle of the blades can be varied to control the thrust and direction of the thrust. VSP's are used on tug boats and (small) vessels with the need for high precision positioning.

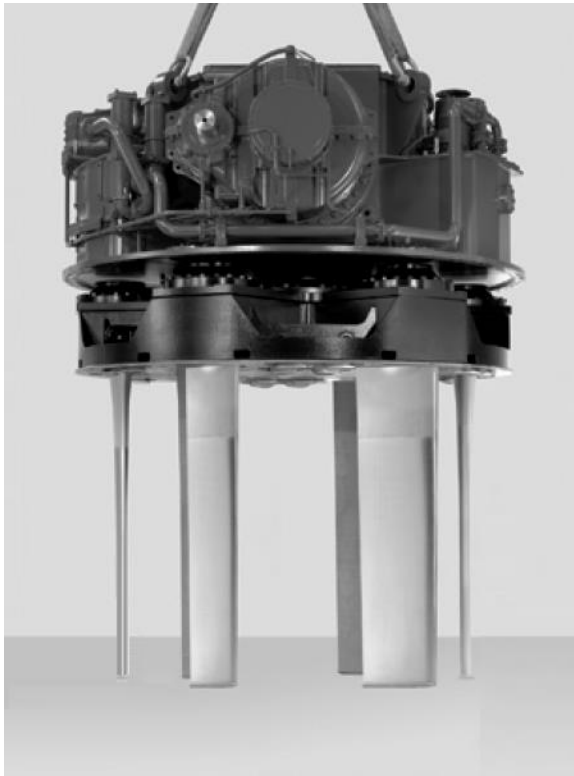


Figure **44**
<http://www.voith.com/en/products-services/power-transmission>

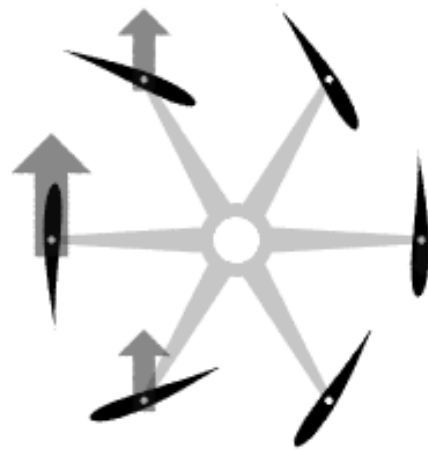


Figure 45

8 The behaviour of the main engine

8.1 Propulsion principles

This chapter will describe the behaviour of a vessel in different working conditions (draft, roughness, acceleration). The geometry of the vessel under consideration will not change during this analysis.

8.1.1 The ship resistance

The ship resistance can be written as (Chapter 6):

$$R = (C_f + C_D) \frac{1}{2} \rho V^2 S$$

Or simplified as:

$$R = aV^x$$

A series of factors is replaced by the term a . This coefficient will be dependant of:

- The wetted surface S . Sometimes this is even more simplified by assuming the wetted surface to be proportional to $\nabla^{2/3}$. The coefficient a will change with the draft.
- The roughness of the hull can change the factor a with 25%
- C_f and C_D change with the Froude number and Reynolds number (or the speed).

A relation between the factor x and the Froude number Fr is sought for. Model tests resulted in the following statistic results:

Fr	0.10	0.15	0.20	0.25	0.30
Factor “x”	2.0	2.2	3.0	3.1	3.2

A common bulkcarrier sails at a Froude number of about 0.10, the resistance of the bulkcarrier will be proportional to the square of the speed. For faster vessels such as passenger ships and (older) container carriers, the resistance can be proportional to the cube of the speed.

8.1.2 Propulsion power and forward speed

The net power required for the propulsion is:

$$R.V = aV^x.V = aV^{x+1}$$

The power to be delivered by the engine in this situation is:

$$P_e = \frac{1}{\eta_g} aV^{x+1}$$

η_g is the global propulsion efficiency as shown in previous chapter.

The admiralty coefficient:

By definition the admiralty coefficient A_c is

$$A_c = \frac{\nabla^{2/3} \cdot V^3}{P_e} = \frac{\nabla^{2/3} \cdot V^3}{\frac{1}{\eta_g} a V^{x+1}}$$

This expression is constant if

η_g is constant

a is proportional to $\nabla^{2/3}$

$x = 2$

8.1.3 Propulsion power and engine speed

The relation between the forward speed V and the engine speed depends of the reduction (if present), the pitch of the propeller (controllable pitch) and the propeller slip.

The propeller slip s [], pitch p [m], forward speed V [m/s] and propeller rate n [1/s] relate as:

$$s = 1 - \frac{V}{n \cdot p} \Rightarrow V = n \cdot p \cdot (1 - s)$$

A constant ratio b between the forward speed and propeller rate can be assumed

$$V = b \cdot n$$

The factor b decreases when the pitch decreases or if the propeller slip increases. “ b ” can be simplified as being constant for a vessel with a fixed pitch propeller.

The power to be delivered by the main engine for the propulsion is

$$P_e = \frac{1}{\eta_g} a (b \cdot n)^{x+1} = \frac{1}{\eta_g} a \cdot b^{x+1} \cdot n^{x+1} = c \cdot n^{x+1}$$

Common merchant vessels have a speed factor (x) of 2 thus

$$P_e = c \cdot n^3$$

This formula is known as the **propeller law**. The propeller law gives the power delivered to a propeller to keep the vessel at a constant speed.

$$c = \frac{1}{\eta_g} a \cdot b^{x+1}$$

The factor c will decrease if

- The draft of the vessel decreases
- The pitch of the propeller decreases
- The efficiency of the propulsion increases
- The roughness of the hull decreases

The diagram $P_e - n$ is plotted on logarithmic axes, the propulsion law are straight lines on the diagram, one line for one value for c .

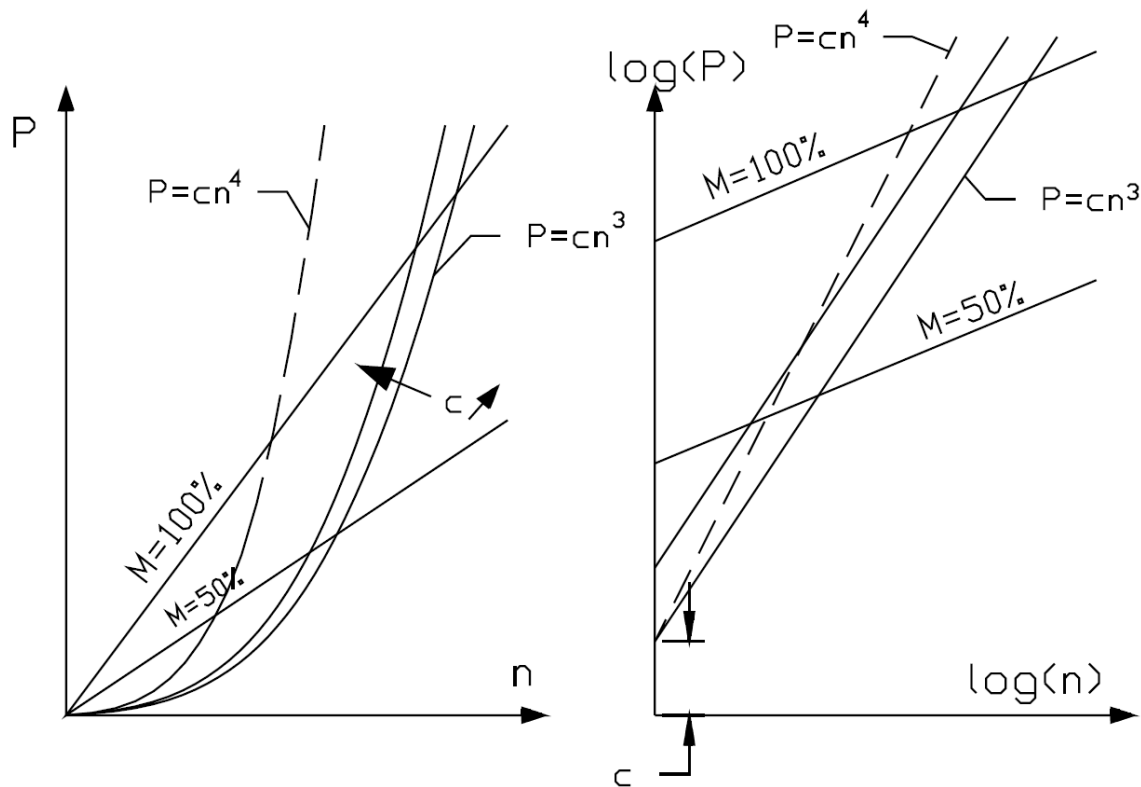


Figure 46

8.1.4 Engine torque

The angular velocity ω is the ratio between the power and torque of an engine:

$$\omega = \frac{P}{M} \Rightarrow M = \frac{P}{\omega} = \frac{P}{2\pi n}$$

With the effective power P_e

$$M_e = \frac{P_e}{2\pi n}$$

At a constant speed this formula can be written as

$$M_e = \frac{c \cdot n^{x+1}}{2\pi n} = \frac{c}{2\pi} \cdot n^x$$

This is also a representation of the propeller law and (similar as for the effective power P_e) this function is a straight line when plotted in a logarithmic $M - n$ diagram.

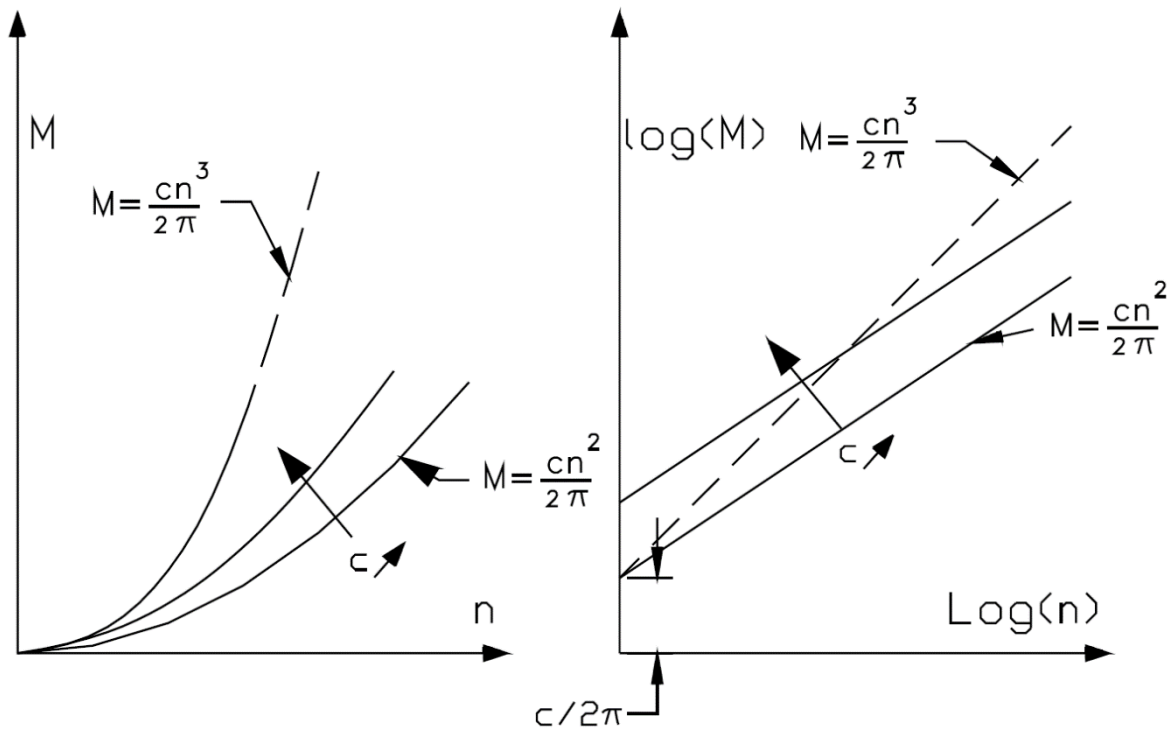


Figure 47

8.1.5 Mean effective pressure

The mean effective pressure p_{me} can be expressed as

$$p_{me} = \frac{P_e}{n \cdot C_m}$$

At a constant speed the effective power can be written proportional to a higher degree of the propeller rate

$$p_{me} = \frac{c \cdot n^{x+1}}{n \cdot C_m}$$

$$p_{me} = \frac{c}{C_m} \cdot n^x$$

The effective torque M_e and the mean effective pressure p_{me} are both proportional to n^x and as a consequence proportional to each other. The straight lines indicating a constant effective torque M_e are also lines indicating a constant mean effective pressure p_{me} and vice versa.

8.1.6 Fuel consumption

a. Fuel consumption per hour

The fuel consumption B [kg/s] is defined as

$$B = \frac{P_e}{\eta_e H_o}$$

With η_e the engine efficiency (section 5.2.3) and H_o the lower heating value of the fuel.

The fuel consumption per unit time (hour, day, month, year) can be expressed with the forward speed of the vessel V or with the rotational speed of the engine n .

a.1 Forward speed of the vessel V

The effective power P_e in the previous expression can be expressed as a function of the forward speed V

$$B = f(V) = \frac{\frac{1}{\eta_g} a V^{x+1}}{\eta_e H_o} = \frac{a V^{x+1}}{\eta_g \eta_e H_o}$$

If the factor x equals 2 (as for most common vessels) than the fuel consumption will be 8 times higher when the speed doubles.

The fuel coefficient F_c is defined as

$$F_c = \frac{\nabla^{\frac{2}{3}} \cdot V^3}{B} = \frac{\nabla^{\frac{2}{3}} \cdot V^3}{\frac{a V^{x+1}}{\eta_g \eta_e H_o}}$$

When the same boundary conditions as for A_c are valid and η_e is constant then

$$\frac{B_1}{B_2} = \left(\frac{\nabla_1}{\nabla_2} \right)^{\frac{2}{3}} \left(\frac{V_1}{V_2} \right)^3$$

a.2 The engine speed n

The fuel consumption B can also be expressed as a function of the engine speed n .

$$B = f(n) = \frac{P_e}{\eta_e H_o}$$

$$B = \frac{c \cdot n^{x+1}}{\eta_e H_o}$$

If the propeller rate doubles, the fuel consumption will be 2^3 or 8 times more.

b. Fuel consumption per cycle b_c

The fuel consumption per cycle b_c is obtained with the ratio between the fuel consumption B and the number of cycle per second $n.Z/a$

$$b_c = \frac{B}{\frac{n.Z}{a}} = \frac{\frac{P_e}{\eta_e H_o}}{\frac{n.Z}{a}} = \frac{\frac{c.n^{x+1}}{\eta_g \eta_e H_o}}{\frac{n.Z}{a}} = \frac{c.a.n^{x+1}}{\eta_e H_o n.Z} = Cte.n^x$$

Z indicates the number of cylinders

a is 1 for a two stroke engine and 2 for a four stroke engine.

If the efficiency is constant then the fuel consumption per cycle b_c is also proportional to the mean effective pressure p_{me} and the engine torque M_e .

c. Fuel consumption per mile covered b_m

The fuel consumption per mile covered b_m is obtained with the ratio between the fuel consumption and the forward speed.

$$b_m = \frac{B}{V} = \frac{\frac{P_e}{\eta_e H_o}}{V} = \frac{\frac{a.V^{x+1}}{\eta_g \eta_e H_o}}{V} = \frac{a.V^x}{\eta_g \eta_e H_o}$$

c.1 *Forward speed of the vessel V*

The fuel consumption during one voyage b'_m is obtained by multiplying b_m with the length of the voyage d .

$$b'_m = b_m \cdot d$$

If the fuel coefficient F_c can be assumed to be constant then

$$\frac{b'_{m1}}{b'_{m2}} = \left(\frac{V_1}{V_2}\right)^{\frac{2}{3}} \left(\frac{V_1}{V_2}\right)^2 \frac{d_1}{d_2}$$

If the same voyage is carried out at twice the original speed then the fuel consumption will be 4 times the original fuel consumption (if the factor x equals 2).

c.2 *The engine speed*

The influence of the engine speed on the fuel consumption per mile b_m is

$$b_m = \frac{B}{V} = \frac{\frac{P_e}{\eta_e H_o}}{b.n} = \frac{\frac{c.n^{x+1}}{\eta_e H_o}}{b.n} = \frac{c}{b.\eta_e H_o} \cdot n^x$$

d. The economic speed

The previous formulae are used to calculate the economic speed. The total cost per travelled mile is (simplified) the summation of

- The fuel cost (see above)
- The costs constant in time (€/hour) These are the costs which are constant and return after a predefined period of time. Therefore these costs, when expressed per mile (€/mile) decrease for an increasing speed (mile/hour). These are for example the amortisation, crew cost, insurance, support by the shipowner, maintenance etc.
- The costs which are proportional to the amount of voyages. These are the administration costs in a harbour for example.

The sum of these three costs (relative to the forward speed) have a minimal value. The forward speed according to this minimum is the economic speed.

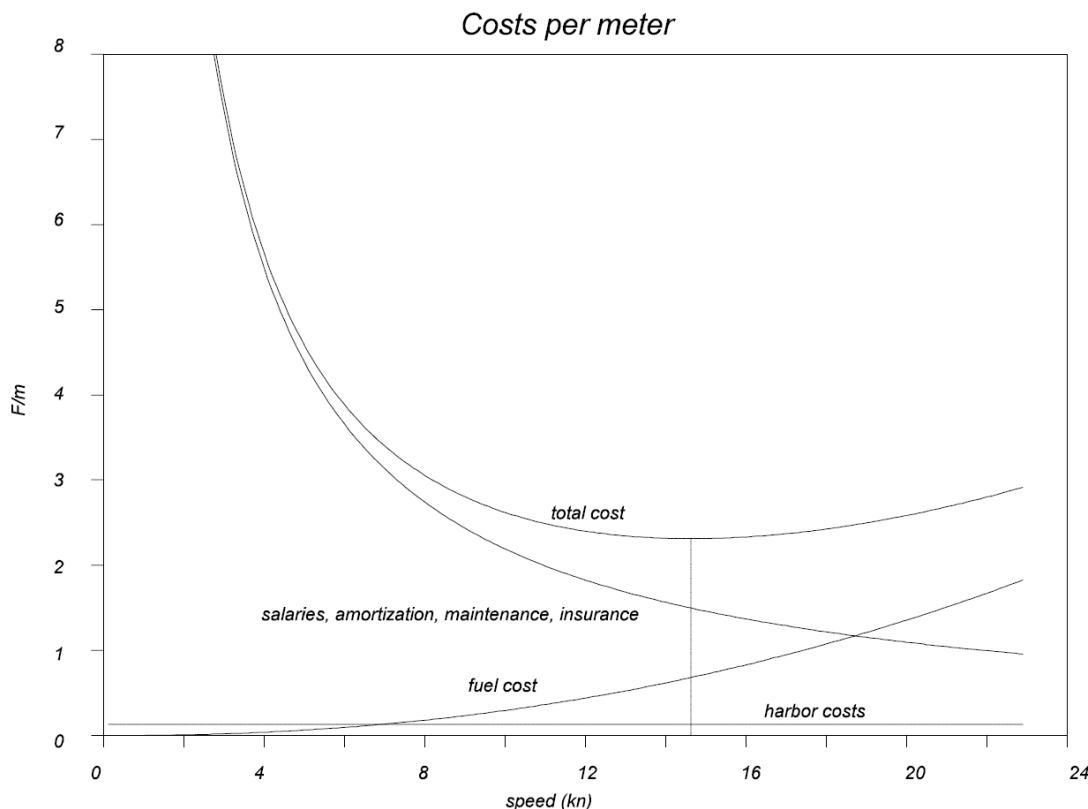


Figure 48

The economic speed does not remain constant in time. If the freight rate increases then the desired forward speed will also increase. The more freight transported the higher the profits.

The current has a specific influence on the desired forward speed. The desired speed has to increase to minimize the time to sail against the current while the desired speed will decrease to profit a longer time a favourable direction of the current.

8.2 Operating envelope of an engine

8.2.1 The maximum engine speed

Most engines may not run at speeds exceeding 100%. Sometimes engines can run upto 103% but never in combination with full power. The maximum engine speed is also limited because of critical speeds. The critical speed is explained in next chapter.

8.2.2 Power limit for continuous running

The power limit for continuous running is 100% of the peak value. The power delivered is checked by monitoring the temperature in the exhaust gasses. Remark that this is a check only and not an accurate measurement of the power.

8.2.3 The maximum fuel injection per cycle b_c

There cannot be more fuel injected than the 100% value. This is even mechanically limited with a stop on the lever of the fuel injection pumps.

8.2.4 The overload limit

At low engine speeds the maximal allowable fuel injection decreases because

1. The hydrodynamic lubrication of the bearings is not optimal and incomplete at the combination of high pressures in the cylinder and the low piston speed.
2. A two stroke engine will **overheat** under these circumstances. The low engine speed will result in a lower speed of the turbo compressor and as a result a lower scavenging pressure. Less air is available for the combustion (lower air excess) and the temperatures in the cylinder will be too high. The exhaust gasses (which are monitored) sometimes do not increase in temperature. The exhaust gasses are cooled before they are monitored because of the longer period of scavenging.
3. An overheated four stroke engine can result in the **instability** of the turbo compressor. The engine speed reduction results in a lower air consumption and a high scavenging pressure. The engine overloads when the speed of the turbo engine compressor decreases.

Except during an "emergency run" these limits are automatically respected to avoid damage to the engine.

8.2.5 Running without scavenging pressure

When the turbo compressor is not used to create scavenging air under pressure then the maximal fuel injection is limited only by the volume of the cylinder (naturally aspirated engine). The maximal allowable fuel injection is constant in this situation.

8.2.6 Temporary overload

Most engine manufacturers allow an engine to be overloaded (more power than the prescribed maximal power) for a limited amount of time (1-2 hours). The power can be more than 10% of the peak value but the turbo charger may not become instable. Running an engine overloaded during a longer period of time than prescribed may jeopardize the warranty of the engine.

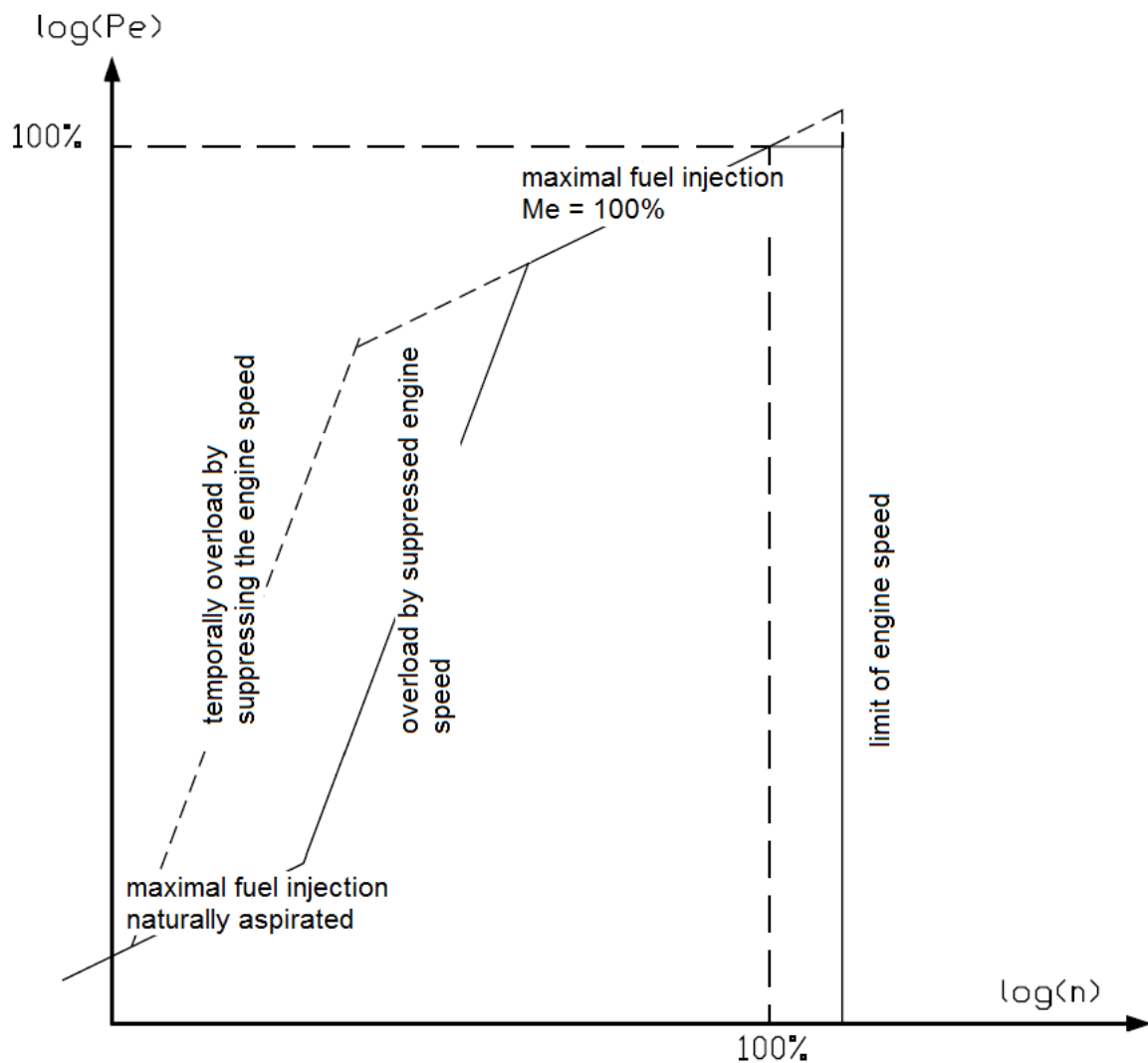


Figure 49

8.3 Matching the engine and ship type

8.3.1 Constant forward ship speed

For a fully loaded vessel at design speed and with a low roughness of the hull (recent anti-fouling) the main engine runs at 85-90% of peak power. There is a margin to avoid the engine to overload when the roughness of the hull increases. This margin changes with the expected time between two dry dockings (and renewing of the anti-fouling), the type of anti-fouling used and the desired capability of acceleration of the vessel.

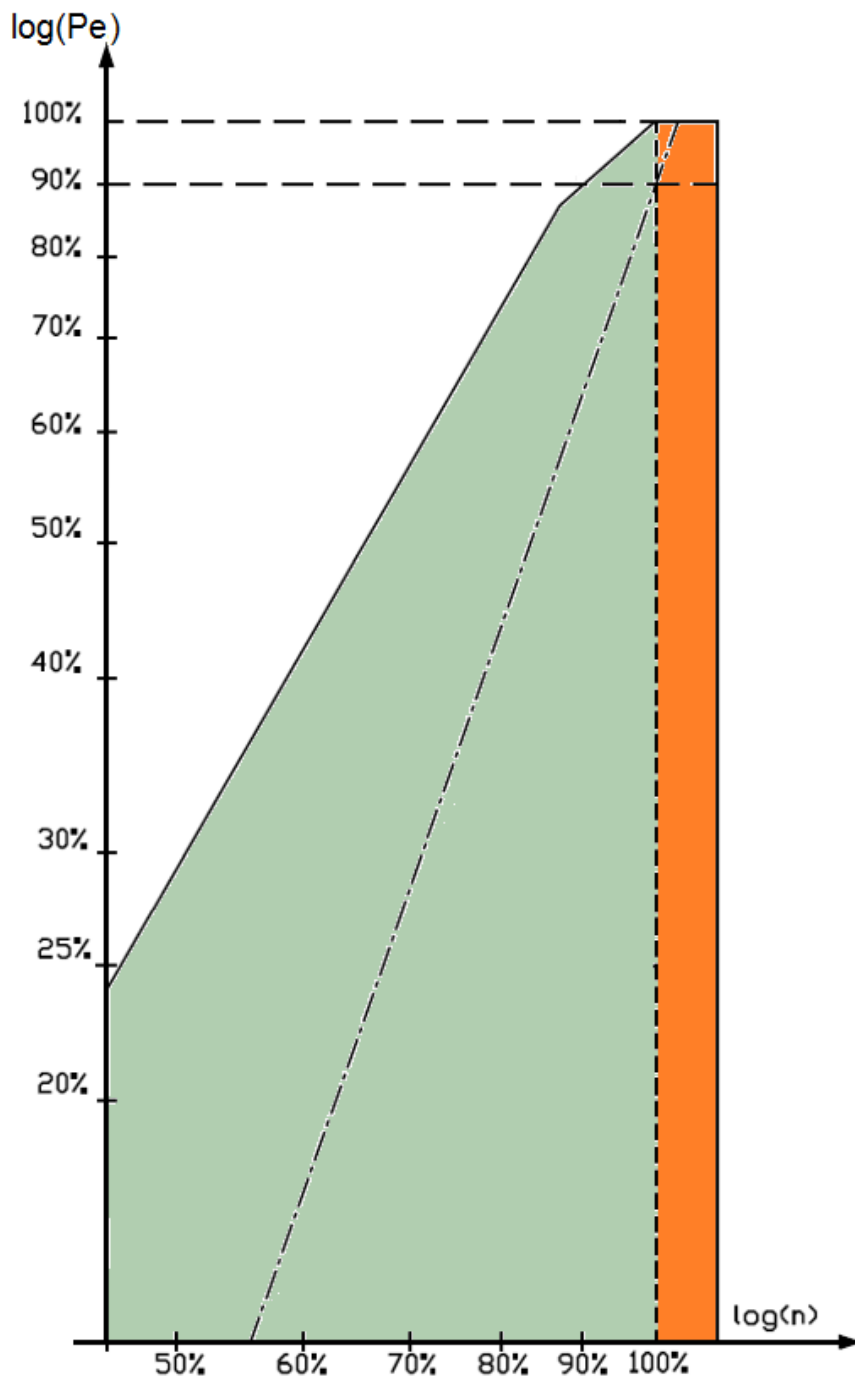


Figure 50 Operating envelope of one engine with the propeller law of one propeller

8.3.2 Acceleration of the vessel

- a) With a fixed pitch propeller it is important to limit the fuel injection until the maximal engine speed is reached. The engine may overload and therefore the injection is limited according to predefined diagrams
- b) With a controllable pitch propeller the acceleration can be obtained according to two different concepts:
 1. The ship can run with maximal pitch. The maximal pitch results in a higher propeller efficiency and lower fuel consumption. In this situation there is risk for the engine to get overloaded. The fuel injection will be limited and the maximal power of the engine will not be reached. The acceleration capability is limited.
 2. The engine can run at a higher speed when the pitch of the propeller is (temporally) limited. This higher speed allows the engine to run at full power and full fuel injection. Powerful accelerations are possible but only at a high fuel consumption.
 3. The combinator is a common solution for controllable pitch propellers. The combinator combines both previous concepts. Changing the position of the lever of the combinator will result in both a new propeller rate and pitch. The relation between the position of the lever and the combination of propeller rate and pitch is predefined but can be changed by the consumer. Some possible settings for the combinator are:
 - i. At low speeds the pitch will be low. The low pitch makes higher propeller rates (and accelerations) possible.
 - ii. The pitch will increase gently with increasing forward speed. At slow speeds (and low pitch) there is a high manoeuvrability while at the higher speeds the propeller efficiency increases.
 - iii. Running the engine at constant speed and only changing the pitch to manoeuvre. This results in a high manoeuvrability of the vessel.

The engine can always be started at the minimal pitch. Mostly the engine runs at a minimal speed before the pitch is increased.

8.3.3 Some remarks

All engines are optimized by the manufacturer during the sea trials to work optimal within a predefined speed range. The engines can be widely adapted but the manufacturer gives a chart with the 'economical feasible operating range' within the technical operating range.

8.4 Specific conditions

8.4.1 Two engines connected to one propeller

When the vessel is designed to run with two engines (connected to one propeller shaft) but only one engine is available then the maximal ship speed, engine speed and propeller speed will not be obtained. There is a high risk of overloading the engine. If the engine is connected to a controllable pitch propeller then the engine can be protected by limiting the pitch.

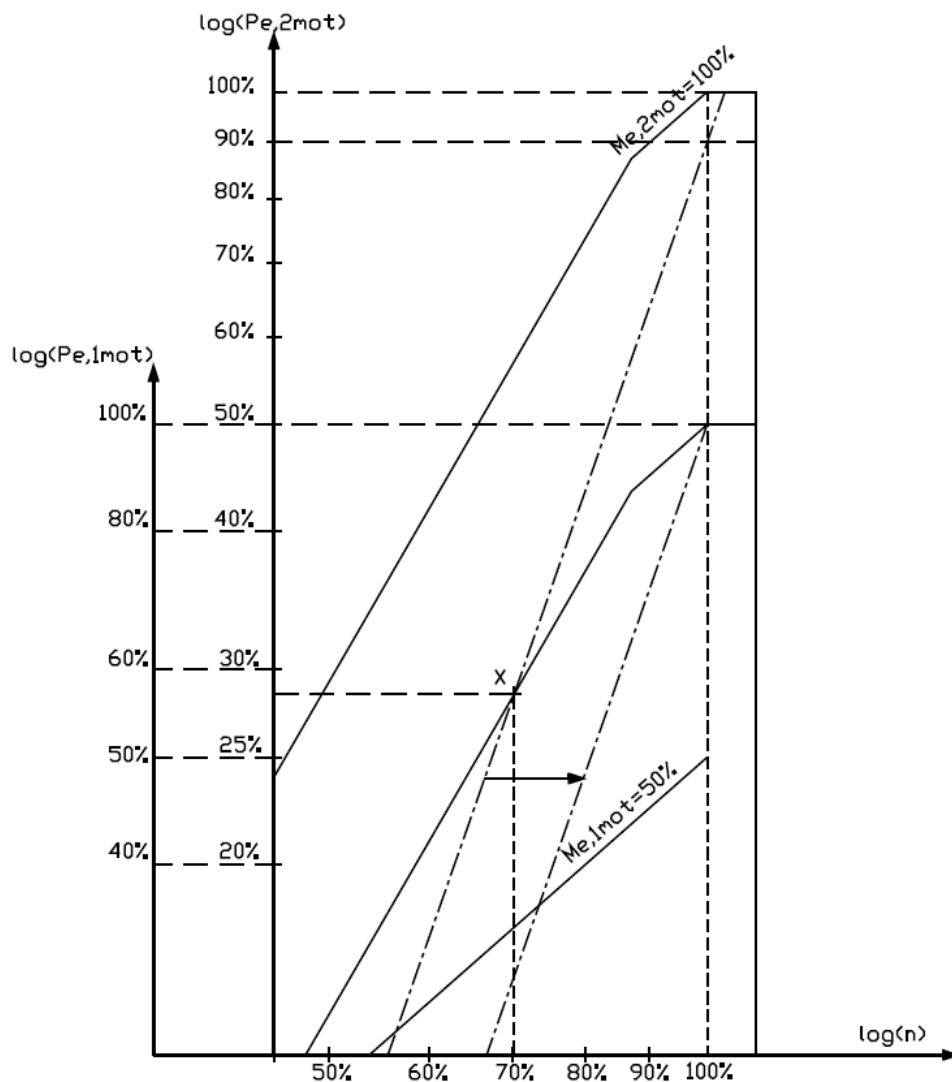


Figure 51

8.4.2 Two engines connected to one propeller each

When one engine fails the other engine may overload. The risk for overload is not as high as in previous case (8.4.1) because the propeller diameter will be lower. Normally, in this configuration the propellers are FPP.

8.4.3 An engine with shaft generator

For propulsion units with a high power (two stroke) the outgoing axis may be connected to a shaft generator. This power take off (PTO) pushes the engine towards its limits. At full sea the required power from the PTO is about constant. At lower engine speeds the part of the PTO becomes more dominant. In some situations (engine speeds below 70%) the PTO must be switch off to avoid overload of the engine.

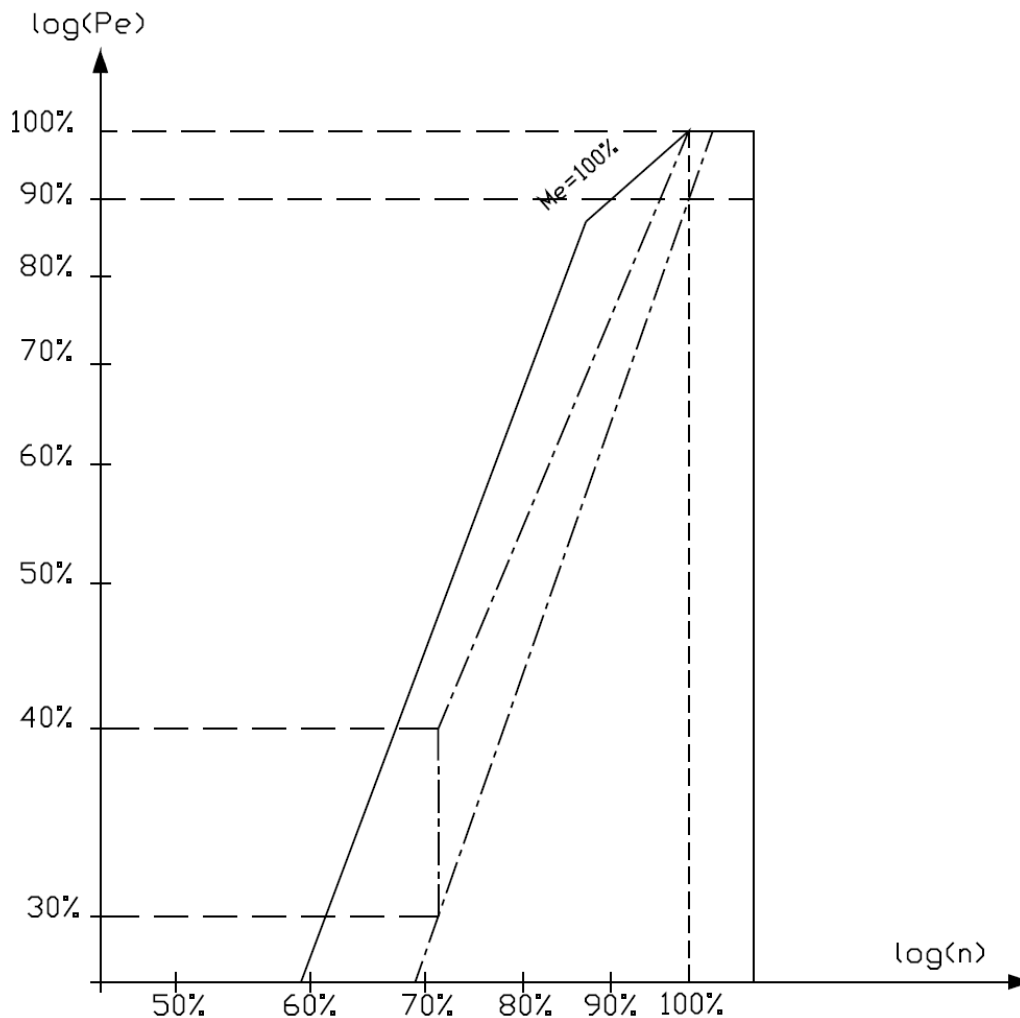


Figure 52

9 Ship vibration

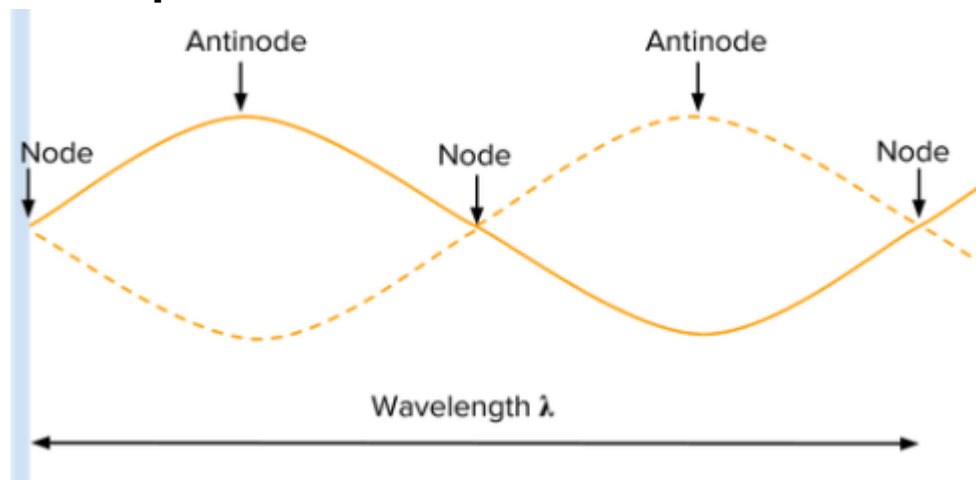


Figure 53

9.1 Theoretical considerations

9.1.1 The natural frequency

All objects will vibrate after a free deformation within its elastic range. This vibration is a natural vibration if there is no exchange of energy with its environment and if the vibration is unrestricted.

The natural vibration takes place at a specific frequency: the natural frequency of the object. The natural frequency is independent of the original deformation of the object but changes with:

1. The type of vibration
2. The stiffness of the body. The stiffer the object the higher the natural frequency.
3. The mass of the object and the mass distribution or moment of inertia of the object. More heavy objects will have a lower natural frequency.

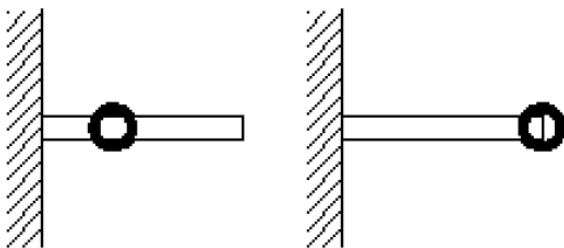


Figure 54

The natural frequency can be calculated if these three parameters are known.

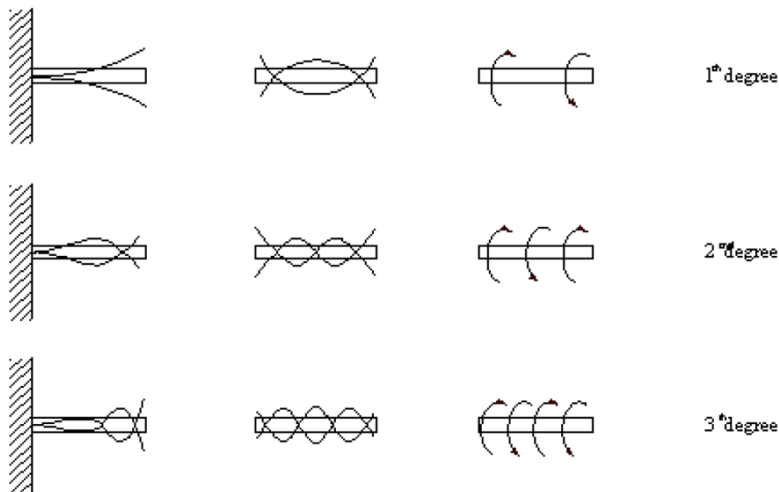


Figure 55 three types of vibration at three different degrees

The type of vibration with the smallest amount of nodes corresponds with the lowest natural frequency. This is also the vibration which results in the highest deformation and as a consequence is the most dangerous. This is a vibration in the first degree. A vibration with one more node is of the 2nd degree and is less dangerous and results in less deformation.

9.1.2 Vibrational excitation

The vibrational excitation is a non-constant force or torque active on the object. For a ships this variable force or torque is mainly coming from the main engine or propeller. The frequency of the excitation always is proportional to the rate of turn of the machine (engine, propeller):

$$f = kn$$

f = frequency of the excitation

n = the rotational speed of the machine (engine speed, propeller rate)

k = the proportional factor / the order of the excitation

Example:

A four stroke engine with six cylinders runs at an engine speed of 900rpm (or 15 rps). Three combustions take place each rotation. As a result there is an excitation of the order 3 (among others). The frequency of the excitation of the third order is $f = 3 \cdot 15 = 45 \text{ Hz}$

9.1.3 Resonance

All vibrational complications originate from a natural vibration amplified by an excitation. This is known as resonance.

Resonance occurs when the frequency of the excitation is about the same as the natural frequency of the body. In this situation the vibrational amplitude will increase more and more. This situation can result into the collapse of the object (e.g. Tacoma Narrow Bridge).

For a ship this situation occurs when the main engine runs at the critical speed. The rotational speed of the main engine at critical speed is shown in red on the tachometer. Although not shown on the tachometer the rotational speeds above the highest velocity is mostly also a critical speed. In normal conditions this speed cannot be reached. When the propulsion system is changed (a new propeller, engine, crankshaft...) this critical speed can be reached and become dangerous.

All possible critical speeds can be found in Figure 56. In this figure the frequency is plotted to the rotational speed.

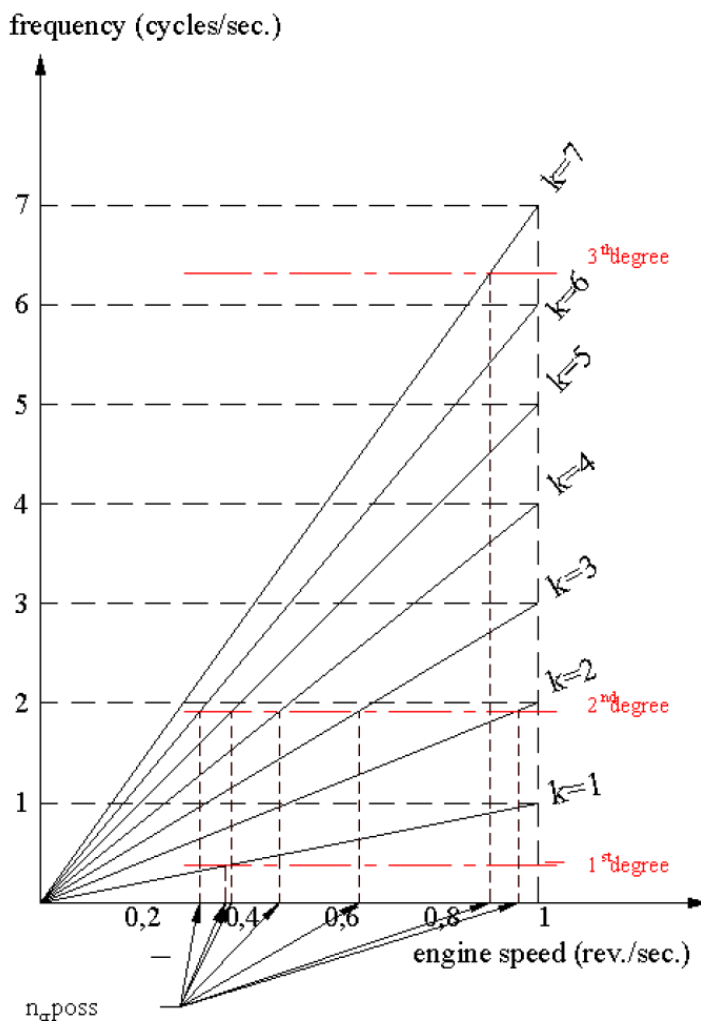


Figure 56

The natural frequencies do not change with the engine speed and are therefore horizontal lines. The frequency of the excitation are straight lines through the origin. Every order of the excitation is a line with a different slope. Resonance occurs at the intersection of the horizontal line of the natural frequency with the line of the frequency of the excitation. The resonance disappears when the rotational speed of the engine is changed.

9.2 Torsional vibration of the entire propulsion system

9.2.1 The full vibration

If a crankshaft introduces a vibration then this vibration acts on all parts fixed to the crankshaft:

- a) If the propeller is directly attached to the engine then the crankshaft, propeller shaft and propeller will endure the same vibration. The natural frequency will not only depend of the crankshaft but also of the properties of propeller and propeller shaft.
 - The installation of a new (bigger) propeller (or alternator) will decrease the natural frequency.
 - The installation of a propeller shaft with a larger diameter (more stiff) will result in a higher natural frequency.
- b) The mass of the alternator in a diesel-electric configuration will influence the natural frequency.
- c) If the propeller is connected to the main engine via a gear box then the gear box (tooth of the gear) have to be protected from the vibrations coming from the propeller and main engine. This can be obtained via a flexible coupling.

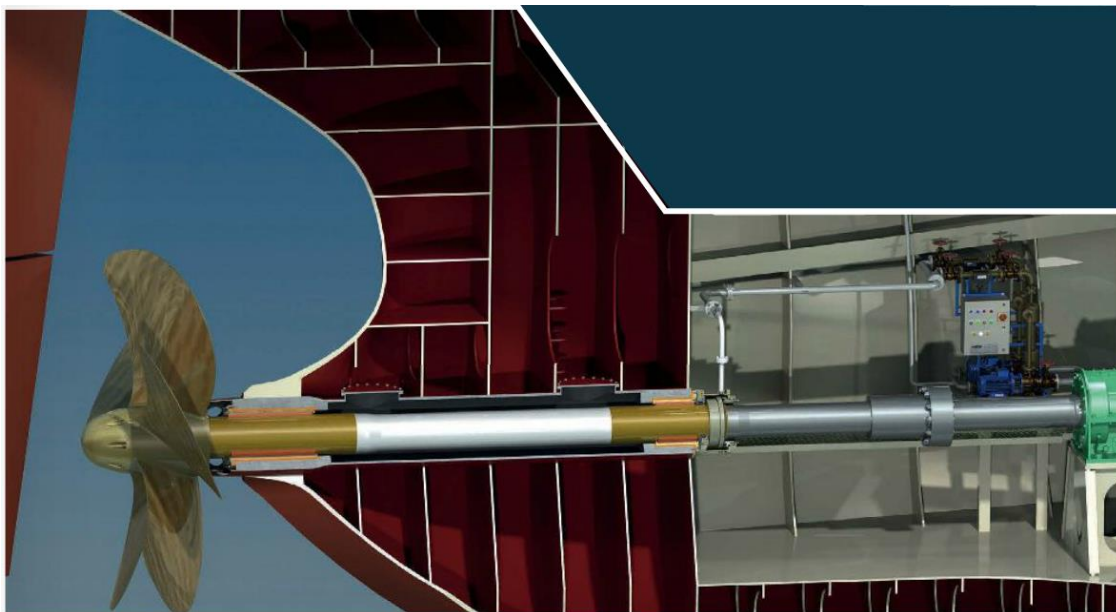


Figure 57

9.2.2 The changing torque

a) Origin

The torque of an engine always is irregular. For example a six cylinder four stroke engine bumps 6 times per cycle or 3 times every rotation.

b) The order of the excitation

Every cylinder in a four stroke engine combusts every two rotations. This results in an excitation of the order 0.5. Such an excitation can be split into a series of harmonic excitations. The harmonic excitations are always a multiple of the base excitation. Harmonic excitations are found of the order: $k = 0.5; 1.0; 1.5; 2.0; 2.5; 3.0; 3.5; \dots$

The harmonic excitation in a two stroke engine (one combustion every cycle) are:

$k = 1.0; 2.0; 3.0; 4.0; \dots$

The harmonic excitation interact and some (partly) compensate each other. Therefore some harmonic excitations are more dominant than others. In a 6 cylinder four stroke engine the orders 3; 6; 9 are more dominant than 1.5; 4.5; 7.5.

The amplitude of the frequency of the excitation becomes negligible when the frequency is large (30Hz). In this situation the frequency of the excitation is also very high.

c) One broken cylinder

When a cylinder is out of order, the absence of a combustion will result in a dominant excitation (at every rotation for a two stroke and every two rotations for a four stroke engine). The order of the excitation will be $k = 1$ (2 stroke) and $k = 0.5$ (4 stroke). The absence of the expected bump results in the energy which is about four times smaller than the expected bump. The engine torque must be divided by 4. Because the torque is proportional to the square of the engine speed n^2 , the engine speed should be halved to obtain this.

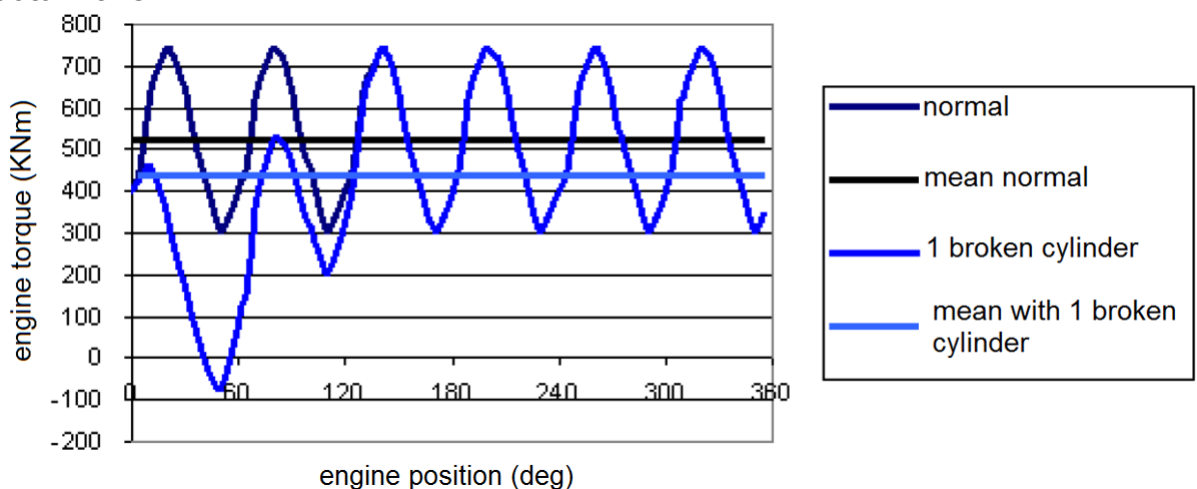


Figure 58

Depending of the origin of the defect (injection, valve, electronically) the engine manufacturer will prescribe a limit for the engine speed of about half of the maximal engine speed.

9.2.3 The critical speed

For different reasons the number of critical speeds is less than suggested in figure 41.

- a) Some vibrations are damped because of internal friction or because of the presence of a specific damper.
- b) Some critical speeds are out of the range of the engine speed.
- c) At very low engine speeds the combustion and excitation is not powerful.
- d) High order frequencies can be neglected because of the low amplitudes.
- e) Natural frequencies of the first and second degree are dominant for an engine while the 4th degree order is not taken into account.

Different resonance frequencies will result in torsional vibrations but acceptable for the object and will not result into damage.

9.2.4 The torsional damper

A torsional damper can be installed to stop the transition of a vibration. A torsional damper exists out of a heavy ring mounted around the crankshaft. The connection between this heavy ring and the crankshaft is via a dry friction, blade springs or heavy oil. The ring will pass on the normal rotation but will absorb the higher frequency vibrations.

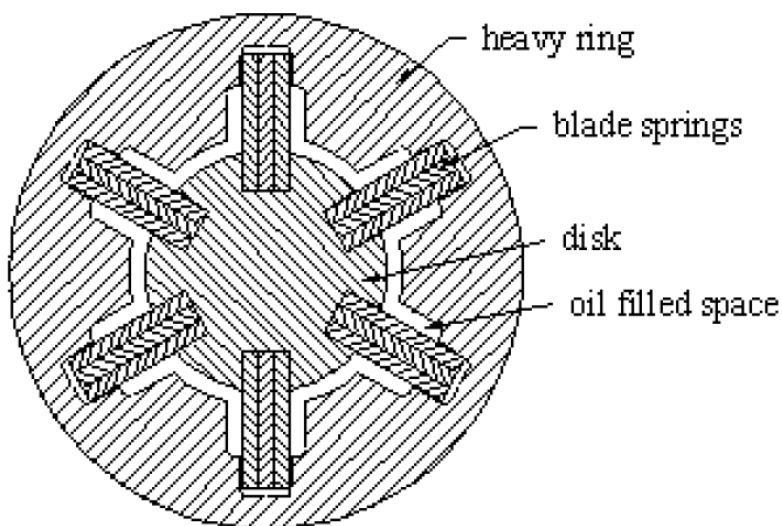


Figure 59 <https://voith.com/corp-en/vibration-dampers.html>

Sometimes this damper is installed in the elastic coupling between the engine and gearbox. The total mass of all components at the gear box' side will act as heavy ring in such a configuration.

9.3 Vibration of the hull

9.3.1 Types of vibration

There are three types of vibration of the entire hull (the hull is seen as one single object):

- Vibrations parallel to the vertical plane, around y axis
- Vibrations parallel to the horizontal plane, around z axis
- Torsional vibrations, around x-axis

All three types of vibrations have a specific set of natural frequencies

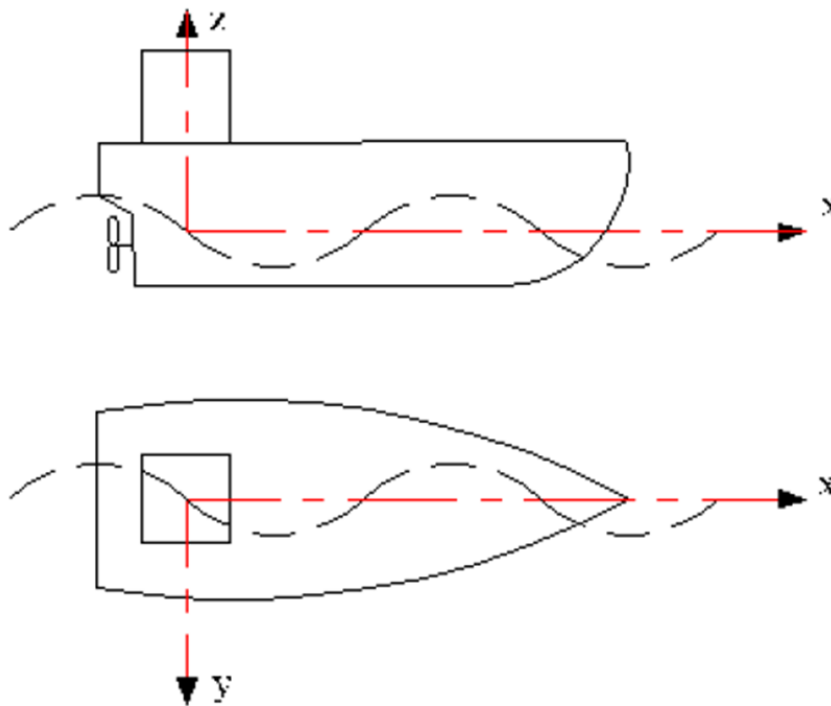


Figure 60

9.3.2 Inertia of the main engine

Two types of inertial forces coming from the main engine can be distinguished:

- Centrifugal forces coming from different cranks and parts of the connection rod. These centrifugal forces can be compensated with specific counter masses on the cranks.
- Inertial forces coming from the translation (up and down) of the piston, cross head, connection rod... These forces are vertical for an inline engine.

The motion of the piston originates from the slope of the connection rod and crank. The dominant influence of the motion comes from the crank. These are the primary forces. These forces result in one bump per rotation.

The slope/position of the connection rod generates a secondary piston motion. This slope is generated two times per cycle.

9.3.3 Aligning the inertia of the piston

Most inertial forces can be neutralized with a smart choice of the combustion sequence. For example the forces of a piston going up can be compensated with a piston going down. However, there are 'left over' forces and moments which have to be compensated by means of alignment: counter masses driven by gear, chains or mounted on an extra axis.

9.3.4 Other causes of vibration

- a) The cross head or trunk will generate high indirect forces. These forces can generate torsional vibrations on the engine. Very large cross head engines have vibrational dampers mounted between the upper part of the engine and the walls of the engine room.
- b) The thrust is not a constant force but a pulsating force. This pulsating force acts on the thrust bearing of the propeller.
- c) Waves generated by the propeller hit the stern section of the hull. These waves cause vibrations on the hull. The frequency is proportional to the propeller rate and the number of propeller blades.
- d) Vortices occur at the tip of the propeller blades. These vortices can generate a high frequency vibration when hitting the hull. A ducted propeller, a different (more) number of propeller blades or a different geometry of the propeller can avoid this type of damage.

