
CHAPTER 1

INTRODUCTION

1.1 The Marine Diesel Propulsion System

1.1.1 Historical Note

Propulsion of the vast majority of modern merchant ships (*e.g.* containerhips and VLCCs) utilises the marine Diesel engine as propeller prime mover. Typical marine propulsion plants include a single, long-stroke, slow-speed, turbocharged, two-stroke Diesel engine directly coupled to the vessel's single large-diameter, fixed-pitch propeller. This configuration can reach quite large power outputs (up to 30–40 MW from a single unit) and yet is characterised by operational reliability due to its conceptual simplicity.

Since mechanisation of propulsion was first introduced in shipping in the mid-19th century various eras can be clearly distinguished. Early motor ships were propelled by side wheels or screw propellers and powered by reciprocating steam engines appropriately arranged in the vessel's hull. Later, transition to steam turbine powerplants was slowly effectuated and was completed by the end of World War II.

However, today the Diesel engine dominates over marine propulsion [1]. There are three major reasons for this fact [2,3]: (a) the superior (thermal) efficiency of Diesel engines over the other propulsion prime movers, (b) following the use of alkaline cylinder lubrication oils, large Diesel engines can burn heavy fuel oil (HFO) and (c) slow-speed Diesel engines can be directly connected to the propeller without the need of gearbox and/or clutch and are reversible. On the other hand, Diesel engines require a larger engine room compared to gas turbines, their major rival nowadays. Indeed, Diesel engines have lower specific power per unit volume and weight. This can be a problem when extremely large power outputs are required, *e.g.* for aircraft carriers or some projected large high-speed vessels.

1.1.2 Marine Engine Configuration and Operation

The propulsion demands of large merchant vessels can be covered using a single slow-speed, direct-drive Diesel engine. This type of engine can burn very low quality fuel, such as HFO, more easily than medium-speed Diesel engines because the physical space and time available to combustion are significantly larger. Slow-speed engines are usually built with a smaller number of cylinders and, in consequence, a smaller number of moving parts, increasing thus the reliability of

the propulsion system. A section of a large marine, two-stroke, turbocharged engine used for ship propulsion is shown in Figure 1.1.

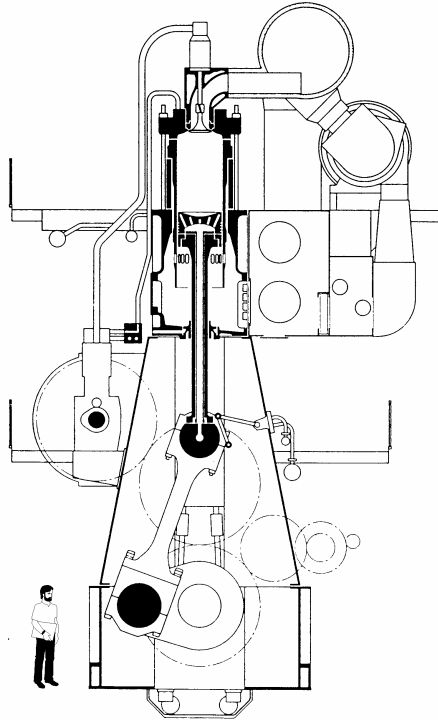


Figure 1.1 Section of large marine Diesel engine

The main parts of the engine are:

- **Bedplate and crankcase:** The bedplate supports the engine and ensures alignment of the shaft. The crankcase provides housing for the crankshaft. In large engines the crankcase and bedplate come in one piece.
- **Crankshaft and flywheel:** The crankshaft is one of the heaviest and costliest components of large marine engines. The flywheel ensures attenuation of the vibrations introduced by the discrete firings in each cylinder.
- **Engine body:** This part of the engine provides mechanical support for the engine cylinders and ensures the mechanical robustness and flexibility of the engine structure. Account is also taken that human access to the inner parts of the engine remains possible.
- **Cylinder blocks and liners:** In large marine diesels each cylinder is contained in it's own separate cylinder block.
- **Pistons and connecting rods:** Pistons consist of the piston crown, the piston rings and the piston rod. Their role is to deliver mechanical power

to the crankshaft to which they are connected through the connecting rods.

- Cylinder heads and exhaust valves: The cylinder heads secure the top of the combustion chamber and provide mechanical support for two other engine components, namely the exhaust valves and the fuel injectors.
- Camshaft(s): The camshaft is one of the most critical engine parts because it ensures timing of exhaust valves opening/closure, as well as fuel injection.
- Fuelling system: This is comprised of the high-pressure fuel pumps, the high-pressure pipelines and the fuel injectors (there can be more than one for each cylinder).

Diesel engines used for ship propulsion are tuned in order to operate near-steady-state on a well defined operating curve, the so called “fouled-hull” or propeller curve. The propeller loading curve, valid for full-bodied hulls, is of the form:

$$P = K_{pow} \cdot N^3 \Leftrightarrow \log P = \log K_{pow} + 3 \cdot \log N \quad (1.1)$$

where P is power in kW, N is shaft rpm and K_Q is the propeller law constant. In terms of torque Q the propeller curve (or law) takes the form:

$$Q = K_Q \cdot N^2 \Leftrightarrow \log Q = \log K_Q + 2 \cdot \log N \quad (1.2)$$

Note that the above cubic relationship for power represents propeller demands only in approximation, because it is dependent upon a variety of additional parameters for hull resistance and propulsion components; in effect, it assumes a more complex functional relationship. In practice, however, the cubic approximation is generally valid over limited power ranges.

Engine–propeller matching is done using a plot of the allowed engine operating envelope, where propeller power demands vs. shaft rpm are indicated, too, as in Figure 1.2. Major operational limit lines for large marine Diesel engines, shown below, are dictated by combustion efficiency (surge or smoke line) in the low rpm range and by shafting system bearings strength (torque limit line) in the higher rpm range.

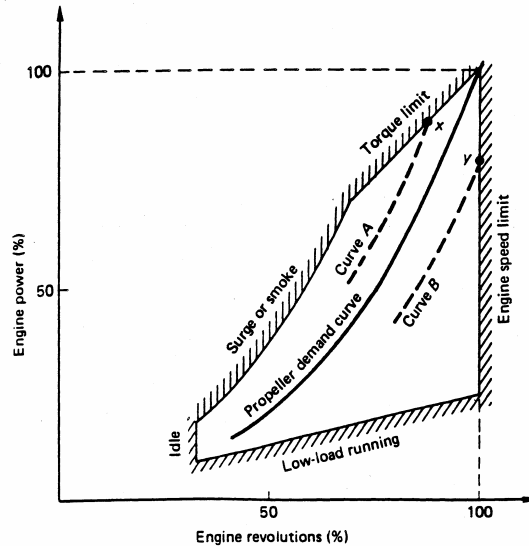


Figure 1.2 Chart used for engine-propeller matching

In the above plot, the propeller demand curve is shown to pass through the Maximum Continuous Rating (MCR) of the engine. The coordinates on the power-rpm chart of MCR represent the peak value of (continuous) engine power delivery and the engine speed limit.

Propeller power absorption characteristic can be modified during service due to a variety of factors such as sea conditions, wind strength, hull condition (roughness and fouling) and vessel displacement. Generally, increased severity of any of these factors requires a power increase in order to drive the ship at the same speed. In turn, this has an effect of moving the propeller power demand curve leftwards towards Curve A. Therefore, allowances need to be made for the propulsion plant to be able to develop full power under less favourable conditions due either to weather deterioration, deeper draught or hull fouling. In effect, a line like Curve B, located rightwards of the nominal (ideal) propeller demand curve, is selected for engine-propeller matching. This concept of difference in performance introduces the term “sea margin”, in order to ensure that the ship propulsion plant has sufficient power available in service and throughout the docking cycle.

Steady-state engine load is expressed in terms of power rating, *i.e.* as a fraction of MCR power. However, the operating point of the engine is controlled through the position (in mm) of the fuel index (rack) at the fuel pump(s), which in most cases is provided in dimensionless form reduced to the interval $[0,1]$ or $[0\%,100\%]$. The steady-state fuel index position is directly proportional to steady-state engine power, if the engine’s mechanical losses are neglected. This implies that the fuel pumps of marine Diesel engines are designed in order to provide linear operation in terms of generated power. In fact, the engine power delivery is not

linearly dependent on the fuel mass injected in the cylinders per cycle. The fuel mass required at each loading point on the propeller curve is calculated using a combination of theoretical thermodynamics, simulation results and testbed experimental data series. Then the mechanical design of the fuel pump is modified in order to achieve linear steady-state engine response in terms of power to fuel index changes.

Another important aspect of marine engine operation is that engine torque delivery is proportional to index position for constant engine speed (rpm) and provided that the engine is running with “excess air”, *i.e.* adequate air supply for perfect combustion. Note that this comes is not contradiction with the linear steady-state power dependence upon index, as the rpm value is different from one steady-state operating point to another.

The engine mechanical power delivery is determined by the following design features:

- Number of cylinders (z_c)
- rpm at MCR (N_{MCR})
- Cylinder bore and piston stroke; these parameters determine the volume V_h swept by the piston displacement during a stroke (piston displacement volume), *i.e.*:

$$V_h = \frac{\pi \times (\text{bore})^2}{4} \times (\text{stroke}) \quad (1.3)$$

- Brake Mean Effective Pressure (BMEP, p_e), defined as follows:

$$p_e = \frac{P}{z_c \cdot V_h} \cdot \frac{1}{N/60} \quad (1.4)$$

where P is power in watts and N is shaft rpm.

Maximum BMEP is observed at MCR where $P = P_{MCR}$ (MCR power) and $N = N_{MCR}$. As argued in the next chapter, engine torque is directly proportional to BMEP, and therefore maximum engine torque delivery is observed at MCR as well.

- Indicated Mean Effective Pressure (IMEP, p_i) and Friction Mean Effective Pressure (FMEP, p_f). IMEP features the per-cycle average in-cylinder pressure, and FMEP the mechanical losses of the engine. The relationship between IMEP, FMEP and BMEP is:

$$p_e = p_i - p_f \quad (1.5)$$

- The maximum in-cylinder pressure value (maximum pressure, p_{max}) achieved in-cylinder per cycle, which affects the power output of the engine. At peak load, state-of-the-art large marine engines achieve p_{max} values above 130 bar.

Taking into account the fact that direct coupling of the propulsion engine to the propeller imposes an upper limit in the range 60–250rpm, propulsion power is increased by increasing V_h , BMEP or the number of cylinders. This remark has

determined the trends in the evolution of the modern marine Diesel engine. Indeed, today's high-power-output engines are both long-stroke and very-large-bore in order to achieve maximisation of V_h . Piston stroke for these types of engine is typically larger than 2 m and cylinder bore above 900 mm (actually approaching 1 m).

BMEP of large marine engines has reached peak values of 18–18.5 bar. Limitations to further increase of BMEP originate from the fact that increasing BMEP leads inescapably to increasing in-cylinder p_{max} . Indeed, BMEP of 18 bar corresponds roughly to p_{max} values of about 130–140 bar. A great amount of research effort is nowadays invested in materials technology in order to manufacture combustion chambers with endurance to even higher values of p_{max} . This will enable a further increase of BMEP and engine power output.

Today's high-power two-stroke marine engines are most commonly built in arrangements of 6–12 cylinders according to requirements of propulsion power, engine room availability and mechanical vibration considerations. Indeed, more engine cylinders require larger engine rooms and impose modifications on the engine turbocharging system as the intake and exhaust manifolds grow longer. Also, a larger number of engine cylinders may be prohibitive due to limitations on crankshaft length and weight.

Finally, it is mentioned that today's marine propulsion engines are always turbocharged. For large two-stroke marine engines the constant pressure turbocharging system is used. With this type of turbocharging the exhaust ports of all cylinders are connected to a common receiver, whose volume is sufficiently large to damp the exhaust pressure pulses. Thus one or more turbochargers with a single turbine entry can be used. The main advantages of the constant pressure system are the simple exhaust receiver configuration and the almost steady conditions at the turbine inlet. The disadvantages are inadequate boost pressure at part engine loads and slow system response. Note that turbocharging introduces an intrinsic closed loop, as well as time delay and uncertainty to the system, and may, therefore, affect system relative stability and transient response. Specifically, the engine becomes more sensitive to torque demand fluctuations. Additionally, the linear relationship between engine torque and fuel index may cease to hold if the turbocharging system fails to deliver adequate air mass to the combustion chambers and, therefore, an incomplete combustion regime prevails.

1.1.3 The Screw Propeller

The single large-diameter propeller used for large cargo ship propulsion is a fixed-pitch mechanical construction with diameter that exceeds 5–6 m and composed of 3–6 blades. Screw propellers located aft-ship were introduced in ship propulsion not earlier than the 19th century. Their main advantage over other propulsion schemes is their superior hydrodynamic efficiency combined with simplicity of construction and operation. The efficiency of a propeller is defined as the ratio

between the thrust power transmitted to the water and the mechanical rotational power provided to the propeller by the shaft, *i.e.*:

$$\eta_B = \frac{P_T}{P_D} \quad (1.6)$$

The overall torque Q_L developed by the water and exerted to the propeller shaft is given as follows:

$$Q_L = K_{q,prop} \cdot \rho \cdot D_{prop}^5 \cdot N^2 \quad (1.7)$$

where D_{prop} is the propeller diameter, ρ is the sea water density and $K_{q,prop}$ is a constant. Comparing this relationship with the one given for direct-drive engine load torque given earlier (Eq. (1.2)) it is easily seen that:

$$K_Q = K_{q,prop} \cdot \rho \cdot D_{prop}^5 \quad (1.8)$$

Significant uncertainty is introduced to propeller torque, which forms the load torque applied to the propulsion engine, due to the fact that $K_{q,prop}$ is not a constant in the mathematical meaning of the word. Actually, coefficient $K_{q,prop}$ depends strongly, for fixed-pitch propellers, on the advance coefficient J_{screw} :

$$J_{screw} = \frac{v_{adv}}{N \cdot D_{prop}} \quad (1.9)$$

where v_{adv} is the advance speed (in m/s^2) of the propeller relative to the water, which is approximately equal to the advance speed of the ship. $K_{q,prop}$, and in effect K_Q , is a decreasing function of J_{screw} .

An rpm limitation is imposed on engine operation due to its direct coupling to the propeller as well. This limitation originates from the dependence of propeller efficiency η_B on coefficient J_{screw} . It can be seen that η_B has a peak value (maximum) when J_{screw} takes a specific value that lies somewhere between 0.3 and 0.7. As J_{screw} moves away from this specific value η_B degrades rapidly. This means that engine/propeller rpm must not exceed a comparatively small value (between 60 and 250 rpm), otherwise propeller efficiency becomes significantly poor (below 30%). Furthermore, $K_{q,prop}$ depends on the cavitation status of the propeller (quantified by the cavitation number σ_{cav} . In general, $K_{q,prop}$ increases with σ_{cav} . In turn, the cavitation number decreases as N (propeller rpm) and/or v_{adv} (propeller advance speed) increase.

Last, but not least, another important source of uncertainty is propeller inertia. When calculating the engine–propeller combined inertia, which determines the integration constant of the shaft, the propeller-entrained water inertia has to be taken into account, as well. However, the mass, and in consequence the moment of inertia, of the entrained water varies significantly. Expressed as a percentage of the propeller inertia, the entrained water can impose a surplus varying between 5 and 30%.

1.2 Contribution of this Work

1.2.1 Statement of the Problem

The subject of this text is robust control design methods for the marine Diesel propulsion system. Robust engine control should act in a manner ensuring safe powerplant operation, especially at near MCR and under conditions that may induce significant propeller load demand fluctuation. Such situations occur under a variety of vessel operating conditions, including mainly heavy weather and rough sea situations. Fluctuation of this magnitude has a number of undesirable effects on the powerplant operation, especially if it is run close to the upper bound of the allowed envelope. The most important consequence of such an event is the occurrence of critical main engine overspeed [4]. Indeed, a large propeller torque demand sink results in acceleration of the engine-propeller shaft. Then, if the engine rpm is set near MCR, the actual engine speed may exceed the maximum allowable limit, leading, thus, to critical overspeed and emergency shutdown, due to prohibitive main engine overloading. In order to avoid this undesirable situation, the main propulsion engine operating point is preventively reduced over the complete time interval for which heavy weather and rough sea conditions are experienced. As a result the ship speed decreases (voluntary speed loss).

As a consequence, it is required to increase the sea margin of the main propulsion engine in order not only to have reserves to overcome the inescapable hull fouling, but also to have the possibility to provide adequate propulsion power under rough sea conditions without the risk of an emergency engine shutdown due to critical overspeed. Therefore, the (resultantly larger) propulsion plant has increased installation costs and, possibly, increased running costs if the optimum operating point is positioned near MCR. Furthermore, even under fair sea states (*e.g.* sea state 3 or 4) it has been reported that significant propeller torque demand fluctuation can occur under certain conditions, *e.g.* when a large containership exhibits significant rolling due to beam sea encounter [4]. This can impose prohibitive limitations (enhanced fuel index limiter activity) to near MCR propulsion plant operation, and leading eventually to voluntary reduction of engine speed setpoint. This often results in failure in keeping the trading schedule, as it is another form of voluntary speed loss.

Limiters are incorporated in modern electronic control units for marine engines for protection against critical and off-design operation. Their effect is imposed directly on the control action, *i.e.* the fuel quantity injected per cycle in the engine cylinders, as expressed in dimensionless manner with the fuel index position percentage, and it is a non-linear and rather empirical feedback form of control. Specifically, a limiter is an upper or lower limit to fuel index position; the limit values are dependent upon the value of certain measured plant variables such as

engine speed (rpm) or boost (scavenging) pressure. If the value of the control action generated by the linear part of the controller (usually a PI control law with appropriate gain values) is outside the range dictated by the limits, then it is saturated to one of the two bounds.

Closer investigation of this operating deficiency, later in the text, demonstrates that if the worst-case disturbance has been taken into account during the controller's linear part synthesis then the limiter activity could have been greatly reduced or even completely avoided, ensuring at the same time reliable plant operation. Moreover, it was made clear, in the framework of a wider research effort on marine control [4,5], that a more systematic methodology is required for marine plants with enhanced capabilities of control. The objective in this effort is the development of marine engine control systems that are robust against operating conditions different than the "nominal" calm sea ones. The conventional PI speed governors, used today in practice, are most commonly tuned for calm sea conditions.

1.2.2 Overview of the Approach

The approach towards the solution of the operational problems stated above is based on: (a) engine and propulsion plant modelling for control, (b) model linearisation based on reasonable assumptions and, finally, (c) designs for two proposed robust controllers, one with the PID control law and one with full state feedback. The specific methodological steps are outlined below.

Prior to control system development, understanding of the open-loop (uncontrolled) plant dynamics is needed. In standard control engineering practice this is done by formulating transfer function or state space models that quantify the transient response of the open-loop plant. For a number of reasons explained later in this text (including ship trading schedule, feasibility limitations, *etc.*), it is preferable to tune either the transfer function or the state-space model of the marine plant using physical (thermodynamic) engine simulation models, rather than performing shipboard measurement and experimentation campaigns. Therefore, in the context of this work, engine operation from the energetic/thermodynamic viewpoint is analysed for control purposes. Then, based on the insight acquired, a transfer function and a state-space model are established for the marine propulsion installation.

Using a reduced-order transfer function for the marine plant, an alternative PI(D) speed regulator tuning method is presented. The method relies on loop-shaping for meeting the disturbance rejection specification of the closed-loop transfer function. Although the method is exemplified for marine propulsion plants, it can be extended to any process that is described by a transfer function with a single, stable dominant pole. As D-term control is needed, an alternative is given for the case of marine propulsion plants to overcome the difficulties encountered in the practical implementation of signal differentiators. The PI(D) controller design

method proposed is tested in the case of an actual propulsion plant of a large containership. A specific assessment of the PI(D) gains obtained using the proposed method is performed, confirming that they can provide adequate worst-case disturbance rejection, especially when compared to the PI governor used at the actual installation. Additionally, robustness against neglected dynamics is examined, using the full-order transfer function, which has been identified on the basis of thermodynamic engine simulation models.

The need for a systematic, formal approach to the design of the feedback propulsion controller is dealt with by firstly formulating a non-linear state-space model for the propulsion powerplant. This is achieved by combining the non-linear mapping abilities of neural nets with the extensive training (calibration) sets obtained using a cycle-mean, quasi-steady thermodynamic engine simulation model.

Finally, based on the plant state-space description, a full-state-feedback controller design methodology is proposed as an adaptation and application of the formal H_∞ -synthesis and real parametric uncertainty analysis frameworks of robust control theory. In this technique, the propeller fluctuation is treated as a disturbance signal that has to be rejected by feedback control. Then, due to the additional real parametric uncertainties introduced to the state-space model by the fluctuation of the thermodynamic properties, as well as by the varying propeller-entrained water inertia, robustness analysis theoretical tools can be employed for the closed-loop system with state-feedback controls. In effect, a complete propulsion control system is proposed combining supervisory control, for smoother engine running, with feedback control for bounding the actual powerplant operation in close vicinity to the nominal desired behaviour.

1.2.3 Text Outline

A brief reference to the topics covered in the upcoming chapters is now given.

In Chapter 2, thermodynamic analysis of the turbocharged marine Diesel engine is performed. The physical processes of power torque generation are examined, resulting in a cycle-mean, quasi-steady model of engine operation that provides adequate insight, as well as a validation platform for control developments. The simulation model is exemplified in the case of a typical marine engine and the numerical solution procedure is explained and evaluated.

Chapter 3 deals with the problem of propulsion powerplant modelling for control purposes. The modelling starts with shafting system dynamical analysis, aiming to depict the effect of the engine-propeller shaft dynamics on controller design. Then the transfer function of the propulsion powerplant is formulated, using the “black-box” approach, in combination with *ad hoc* assumptions for the dynamics of the marine plant. Finally, identification is performed by employing a detailed, filling-and-emptying thermodynamic model of the engine processes. The procedure is validated using the propulsion powerplant of a large containership.

In Chapter 4, PI and PID control of the marine propulsion powerplant are examined from the viewpoint of modern linear robust control theory, and, specifically, H_∞ disturbance rejection (attenuation). Based on the analysis of Chapter 3, the closed-loop (with a PI(D) controller) scalar transfer function is formulated. Then, the PI(D) gains are calculated so that the H_∞ -norm of the compensated system is equal or below specification. In effect, PI and PID compensated plants are assessed for robustness against neglected dynamics. From that perspective, superiority of PID over PI regulation is demonstrated. Finally, based upon the shafting system dynamical analysis of Chapter 3, an alternative method for implementing the D-term in practical installations is proposed. This method does not require differentiation of the rpm feedback signal, as the rpm derivative is calculated using the shaft torque feedback signal.

In Chapter 5, the state-space description of the marine plant is deduced from the thermodynamic engine model of Chapter 2. The analysis is based upon the neural net capabilities to depict non-linear mappings, if trained properly. State equations of the plant are then formulated, incorporating the neural torque approximators and the propeller law. The parametric uncertainty, present in the state equations, is in effect located and assessed. Next, the procedure for linearisation of the marine powerplant equations is applied. Finally, the open-loop transfer function matrix is determined and comparison to the scalar transfer function obtained in Chapter 3 is performed.

Chapter 6 deals with the marine propulsion powerplant control problem using state-feedback linear robust control theoretical results, in combination with open-loop optimised schedules for operating point changes. The disturbance rejection specifications are appropriately decomposed based on the analysis of Chapter 5. In effect, gains of the controller are calculated. Finally, applicability of criteria for robust stability and performance, as well as the effect of integral control on steady-state error, are briefly examined.

Chapter 7 concludes this work. Assessment of the modelling approaches, as well as of the PI(D) and state-feedback control options investigated, is done. Proposals are given for future research and investigations.



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