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COMPENDIUM Marine Engineering

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1.1 Marine diesel engines

Peter Boy, Roland Behrens

1.1.1 Mode of operation of diesel engines

Today, merchant ships (motor ships) are powered almost exclusively by diesel engines. Small sports boats running on petrol (gasoline) engines that use petrol as fuel as well as smaller ferries that are powered by gas-driven engines are the exceptions.

The diesel engine is characterised by internal mixture formation. A fine spray of fuel is injected into the highly compressed and thus heated air contained within the working cylinder. At the start of the injection cycle, the temperature of compressed air is above the self-ignition temperature of the fuel, so that it burns very rapidly. Heat released by the combustion process raises the temperature of gas, and thus the pressure in the cylinder, suddenly up to combustion peak pressure. Work is done at the working piston during the subsequent expansion of the gases.

Crankshaft mechanism

Diesel engines are reciprocating piston engines. Heat (pressure) generated in the power cylinder by internal combustion is converted into mechanical energy by means of a reciprocating motion of the power piston. The reciprocating motion of the piston is converted into a rotary motion of the crankshaft by means of a slider-crank mechanism, consisting of the connecting rod and crank, articulated to the power piston.

The driving mechanism of a piston engine consists essentially of the power (working) piston, the connecting (drive) rod and the crankshaft with its crank (Fig. 1.1.1). The kinematics and geometry of reciprocating engines are specified unambiguously by the following parameters:

- Stroke-bore ratio s/D
- Ratio of connecting rod

$$\lambda_{Pl} = r/l \tag{1.1.1}$$

- Compression ratio

$$\varepsilon = V_{max}/V_{min} = (V_h + V_C)/V_C \tag{1.1.2}$$

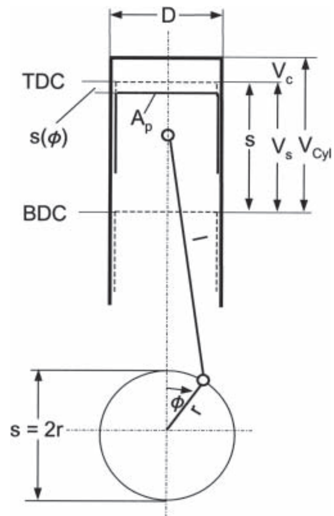

Figure 1.1.1: Crank gear

Here, volume V_{min} corresponds to compression volume V_C (piston in top dead centre position), and the maximum cylinder volume V_{max} is the sum of the swept volume V_h and the compression volume V_C. For the swept volume V_h of a cylinder:

$$V_h = \frac{\pi D^2}{4} \cdot s \tag{1.1.3}$$

The total swept volume of an engine with z cylinders is thus

$$V_H = V_h \cdot z \tag{1.1.4}$$

The cylinder volume V_{Zyl} of reciprocating piston engines changes periodically with the movement of the power piston between the limits V_{min} and V_{max}. The following relationship applies:

$$V_{Zyl}(\varphi) = Vc + \frac{\pi D^2}{4} \cdot s(\varphi) \tag{1.1.5}$$

There is no linear relationship between the path of the piston s(φ) and the rotational angle φ of the crankshaft; rather, starting from top dead centre, it approximates the following relationship:

$$s(\varphi) = r\left((1 - \cos\varphi) + \frac{\lambda_{pl}}{4}(1 - \cos 2\varphi)\right) \tag{1.1.6}$$

The following relationship between the rotational angle of the crankshaft (crank angle) φ and the angular velocity ω applies:

$$\omega = \frac{d\varphi}{dt} = 2 \cdot \pi \cdot n \tag{1.1.7}$$

Normally, in case of engines, the rotary speed n is not given in terms of the rotational frequency [s^{-1}], but as revolutions per minute [min^{-1}]. Thus the angular velocity ω is defined as

$$\omega = \frac{2 \cdot \pi \cdot n}{60} \tag{1.1.8}$$

The first derivative of the piston path with respect to the crank angle gives the momentary piston velocity, and the second derivative gives the momentary piston acceleration. This is necessary for calculation of the oscillating inertial forces of the drive mechanism.

The mean piston speed can be calculated from the piston stroke s and the rotational frequency n:

$$v_m = 2 \cdot s \cdot n \tag{1.1.9}$$

This is an important parameter for the kinematic and dynamic engine behaviour. Mean piston speed can be increased only to a limited extent, since with increasing v_m, inertial forces, friction, and wear also increase. As a result, a large engine with a long stroke can run only at a lower engine speed than a smaller engine.

Four-stroke engine

A four-stroke diesel engine's working process consists of four-strokes: “Suction”, “Compression”, “Expansion” and “Exhaust”. Each of these strokes requires movement of the piston from bottom dead centre to top dead centre or vice-versa, i. e. one stroke and half a crankshaft rotation for each. Therefore two complete revolutions of the crankshaft are required for completion of one work cycle of a four-stroke engine. Four-stroke engines possess a valve control, for gas exchange, with usually two intake and two exhaust valves per cylinder. Modern engines achieve firing pressures of up to p_z = 200 bar. Fig. 1.1.2 shows the p-V diagram of a work cycle and the valve timing diagram of a four-stroke diesel engine.

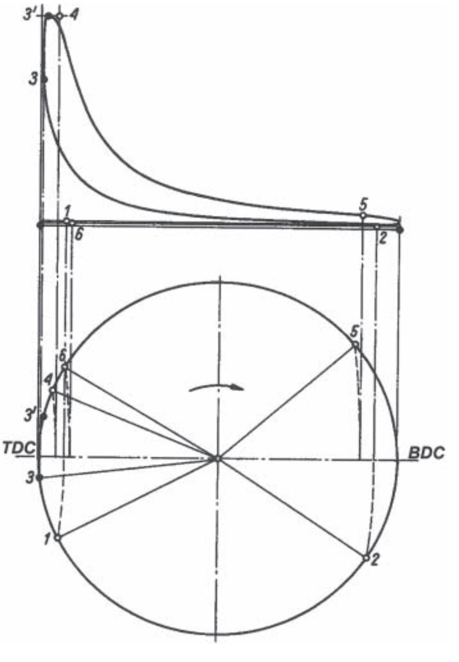

Figure 1.1.2: Working diagram and valve timing diagram of a four-stroke engine

Suction stroke

During the suction stroke, the piston moves from top dead centre (t.d.c.) to bottom dead centre (b.d.c.) and sucks in air through the opened intake valve (1). In order to exploit the inertia of the air column and to achieve a recharge effect, the intake valve shuts shortly after b.d.c., as the piston is already moving upwards (2). In case of naturally aspirating engines, there is a cylinder pressure somewhat under ambient pressure at “intake closes”, and in case of supercharged engines the pressure is higher than ambient according to the degree of supercharging.

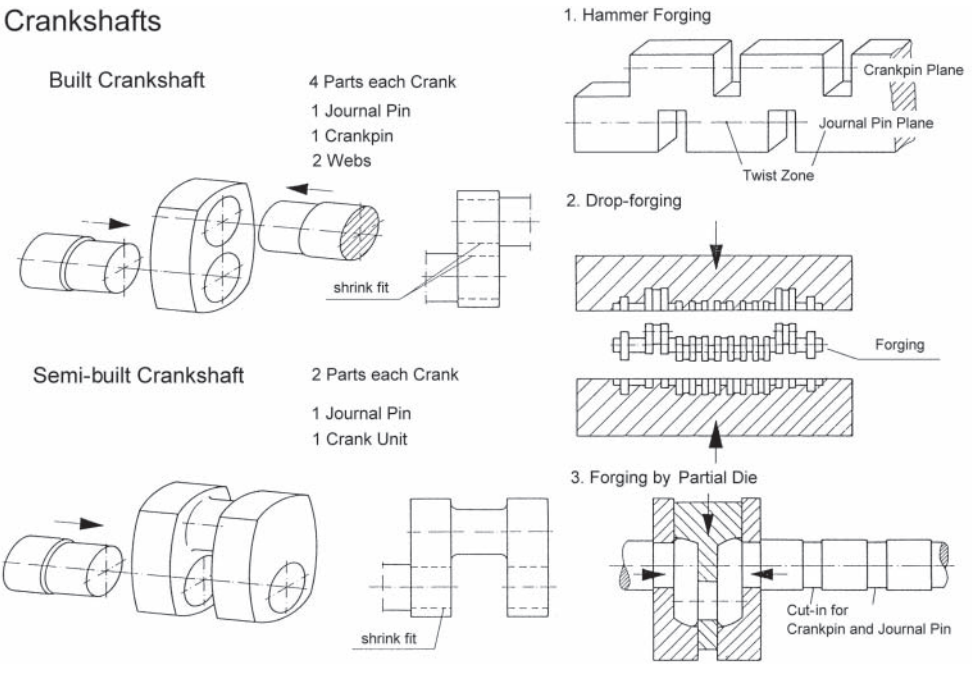

Figure 1.1.24: Crankshaft designs

Crankshafts used in ship engines are fully built and semi-built, forged, drop-forged and semi-drop-forged. Nearly all slow-speed two-stroke engines have semi-built crankshafts, whereas in case of medium-speed engines forged crankshafts are preferred. Crankshafts have a bearing mount after every crank. V-engines always have two connecting rods on one crank pin the pin must be of suitable length. When establishing the succession of cranks, one should keep an eye on a uniform firing order, mass balancing, and torsional vibrations. .

The crankshaft undergoes stresses resulting from gas forces, inertial forces and vibration. Above all, the design should ensure that sufficient bending and torsion strength as well as sufficient load-bearing capacity and lubrication of bearing seats are provided. Bending stress is caused by gas and inertial forces with stress peaks occurring in particular in the vicinity of dead centres. The combustion pressure and inertial forces at dead-centre positions of gas exchange causes tensile and compressive stress at the fillets of the cranks. Pulsating torque transmitted by the crankshaft, and torsional vibrations cause torsion loads in the crankshaft. Torsion stress caused due to variations in tangential force is mostly small in comparison to the torsional alternating stress due to the torsional vibrations.

For an approximate calculation, only one section between two bearings with articulated support assumed, is usually considered. In this way, a nominal bending stress for the middle of the crank web can be calculated. If the crankshaft is of known design, the maximum stress in the crank-pin fillet is calculated from the nominal stress using form factors based on empirical values, and combined with analogously determined maximum torsion stresses to form an equivalent stress. Simultaneous occurrences of the maximum torsion and bending stresses are assumed.

Ship engines must be equipped with crankshafts that satisfy standardised methods of computation of the classification societies (such as *Germanischer Lloyd* or *LRS*) which include corresponding safety margins.

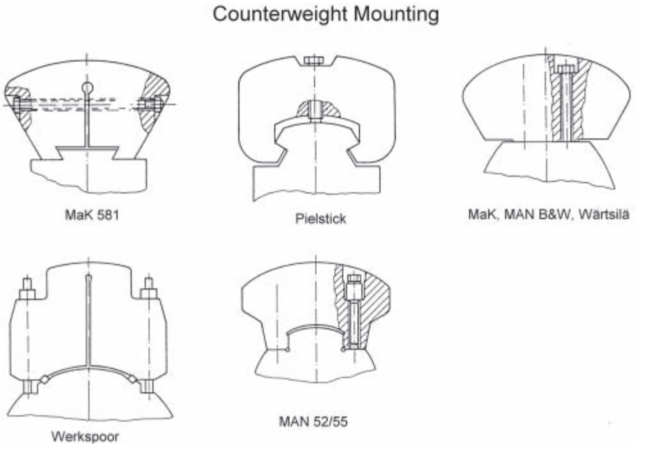

Figure 1.1.25: Four-stroke counterweight designs

Quenched and tempered steels such as CK 45 V for small, not highly-stressed shafts; 34 CrNiMo6V for more highly stressed engines, 42 CrMo4V for high-speed high-performance engines and CrMoV4 present-day highly-stressed ship engines, are used. Generally crankshafts for ship engines are not hardened and are polished to a surface roughness of up to 0.14 µm.

The counterweights required for counterbalancing inertial forces and for load-relief of bearings are either forged or welded on, or bolted on with tension bolts. Fig. 1.1.25 shows examples of the mounting of counterweights for four-stroke engines.

Connecting rod and crosshead

The basic functions of the connecting rod are transmission of the piston force and motion to the crankshaft, and where applicable, accommodating holes for supply of lubricating oil to the piston. Connecting rods are subject to gas forces and inertial forces. It is essential in designing them that sufficient mechanical strength, load capacity of the bearings, and an optimum length with respect to overall size and the friction through lateral force are given.

Connecting rod (four-stroke)

The connecting rod small top end, which is always of closed design, and through which the gudgeon pin passes, is connected to the connecting rod big end by way of the connecting rod shank. Generally, the connecting-rod big bottom end is split so that assembly on a one-piece crankshaft is possible.

The rising combustion pressures of recent years have led to a variety of connecting rods. The higher the combustion pressures implements, the larger the diameter of crank pin was required. Starting with the horizontally split and passing through obliquely split with different angles to the connecting rod axis, we have now reached the marine type and shank split type connecting rod, where the split can now be incorporated at various heights. Stepped connecting rod is also used in case of high combustion pressures in order to increase the bearing area of connecting rod and piston that can take up high gas forces.

Transverse forces in the case obliquely split and cap screws are taken up at the partition joint by the serrated profile in order to prevent movement in the partition joint. For this, the connecting rod bearing cap must be designed to be stiff and form-retentive. The connecting rod top end is supplied with lubricating oil from the big-end bearing via a bleeder hole in the connecting rod, which is then usually used as piston cooling oil through a spray nozzle.

4.2.3 Adjusting PID-controllers

Adjusting a PID-controller always requires a clear definition of the purpose of the control. In the example in figure 4.2.4, although temperature ϑ should be kept as constant as possible corresponding to the set-point value on the one hand, on the other hand disproportionately frequent actuation of the servomotor at the regulating valve should also be avoided. In most cases the servomotor is not designed for continuous operation.

Furthermore, adjustment of a controller essentially presupposes an analysis of the controlled system (e. g. with the result of the frequency response). Elementary estimation of the most important parameters suffices in the case of simple, non-critical control circuits that additionally permit trials. A detailed system analysis is imperative in other cases and simulation of the control response is common. The step response according to figure 4.2.5 can be estimated very well for the cooling circuit shown in figure 4.2.4. Even measurement is possible.

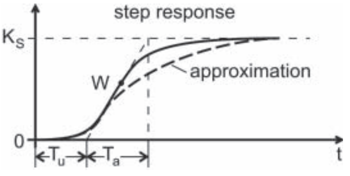

Figure 4.2.5: Typical step response of a controlled system and parameters to be used in the *Ziegler-Nichols* method

A progression similar to that of figure 4.2.5 can be found in many technical systems. The adjustment of a PID-controller according to the *Ziegler* and *Nichols* method shown below at least results in usable performance of the system and permits further experimental optimisation.

Once the step response (figure 4.2.5) can be measured, it is approximated through dead time T_u and subsequently P_{T1}-performance. Only the range up to turning point W of the step response is important. The very long-term changes for longer times correspond in the frequency response to very low frequencies and are therefore not critical. Setting values for the controller according to table 4.2.2 can then be found from the time periods T_u and T_a as well as the stationary ratio k_S ($k_S = \Delta\vartheta/\Delta\alpha$ in figure 4.2.4). Internal scaling must also be taken into account during conversion of values into parameters that can be used for controller setting (e. g. conversion of temperature by a Pt100 thermometer and an amplifier into an electrical voltage). Therefore, k_P is often provided with a dimension.

Controller type	"Step response" method			"Continuous oscillation" method		
	k_P	k_I	k_D	k_P	k_I	k_D
P	$\frac{1}{k_S} \cdot \frac{T_A}{T_U}$			$0.5 \cdot k_{P,\,Krit}$		
PI	$\frac{0.9}{k_S} \cdot \frac{T_a}{T_U}$	$\frac{0.3 \cdot k_P}{T_U}$		$0.45 \cdot k_{P,\,Krit}$	$0.85 \cdot k_P \cdot T_{Krit}$	
P/D	$\frac{1.2}{k_S} \cdot \frac{T_a}{T_U}$	$\frac{0.5 \cdot k_P}{T_U}$	$0.5 \cdot k_P \cdot T_U$	$0.6 \cdot k_{P,\,Krit}$	$0.5 \cdot k_P \cdot T_{Krit}$	$0.12 \cdot k_P \cdot T_{Krit}$

Table 4.2.2: Controller parameters according to the *Ziegler-Nichols* method

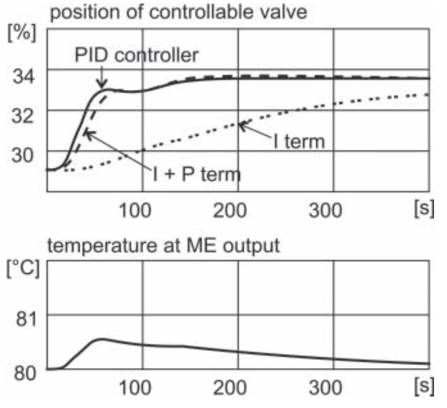

Figure 4.2.6: Reaction of the control loop on a sudden steep increase of the heat intake from the diesel engine. The controller output α is the sum of the P-, I- and D-terms of the PID controller.

Alternatively, the following procedure may be followed experimentally: to start with, the D- and I-component of the controller are disconnected. Gain k_P of the controller is then increased to the extent that it results in continuous oscillation, or that every impulse disturbance or step disturbance leads to a slowly decaying oscillation. Controller gain is termed $k_{P,crit}$. The period of the oscillation T_{Crit} is measured. Setting values according to table 4.2.2 are then determined from these two parameters.

Furthermore, both procedures for determining the parameters show an important interrelationship that can be formulated qualitatively in the following manner (figure 4.2.15):

1. The P-component must correct the majority of the disturbances in a relatively short time.
2. The I-component serves to correct longer-lasting control deviations. (The recommendation for the gain k_P in table 4.2.2 only depends to a relatively small extent on whether it is a P-, a PI- or a PID-controller.)
3. On the one hand, the D-component can accelerate the reaction of the controller at the start of the control process and, on the other hand, lessen overshooting, which is normal and unavoidable for a PI-controller.

4.2.4 Measures for improving the control response

Correcting variable limitation

It is often the case that the correcting variable is restricted beyond the actual controller due to technical reasons. The correcting range of the control valve as well as the displacement speed is restricted for the installation in figure 4.2.4. The controlled variable can therefore not change as quickly as was intended during controller adjustment. The effect is shown in figure 4.2.7. The integrator component would be much too large, and the integrator component becomes practically ineffective once the controlled variable has again reached the set-point value. To prevent such an effect, the integration of the deviation must be suppressed on reaching an external limit and the "filling level" of the integrator must trail the limitation. The D-component is not taken into account here. The following pseudo-code for implementing a PI-controller with set-point value w = 0 and limitation of the correcting variable y to $y \leq 1$ shows the principle

P_component: $= k_P \cdot x$
I_component: $= k_I \cdot x \cdot \Delta t$ + I_component

y: = P_component + I_component
If y > 1 then y: = 1
I_component: = 1 – P_component
End_If.

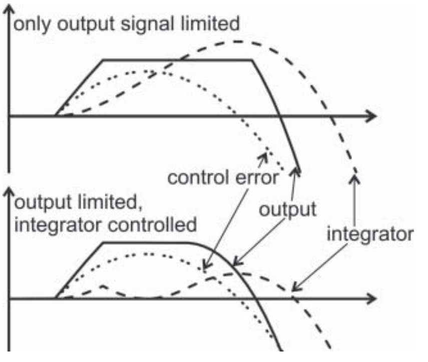

Figure 4.2.7: Effect of an external limitation of the controller output; upper curve: without limiting the integral term, lower curve: with limiting the integral term

Set-point filtering and disturbance variable feed forward

Large control deviations typically result if the set-point value changes suddenly by a large amount or if the disturbance variable changes suddenly by a large amount. Therefore, the set-point value entered by an operator is processed initially in such a way that the input variable for the actual control circuit (figure 4.2.1) changes only as fast as the system can follow this input (set-point filtering).

It is also possible in some systems to measure the disturbance. For example, the power output of the diesel engine can be measured in the example in figure 4.2.5. In the case of a sudden power increase, the control valve could be adjusted by a certain amount in the "cold" direction before the power increase is acknowledged as a temperature increase at the cooling water exit of the engine (disturbance variable feed forward).

It can make sense for systems with long delays between the correcting variable and controlled variable also to record intermediate variables, e. g. in the example of figure 4.2.4 the temperature at the entry to the engine. All these measures lead to the system becoming so complex that simulation becomes essential for the preliminary selection of setting parameters. Experimental setting is complicated due to a large number of controller parameters and barely feasible systematically.

Hierarchically structured controllers (cascade controllers)

The illustration in figure 4.2.4 is incomplete to the extent that conversion of the actuating signal α into switching commands for the motor at the valve is missing. This parameter is in fact preset for a subordinate control circuit as a set-point value (figure 4.2.8). In principle, two nested control loops exist: temperature is the controlled variable in the higher-level control loop and the valve position is the controlled variable in the lower-level loop. Often the qualifying time constants of the lower-level loop are much smaller than those of the higher-level loop, so that the subsystems can be examined separately from one another.

All the same, the gain and phase shift of the lower-level control loop must be suitably taken into account when designing the higher-level control circuit.

On-off and three-level control

Often there is a need to control the controlled variable only within a relatively wide tolerance range. If e.g. the pressure falls below the minimum in a compressed air bottle due to use, a compressor is started up, only to stop again on reaching the upper limit. Consequently, there are only two states for the correcting variable and the controller is just an on-off controller. A three-level controller is used if the correcting variable can assume three states, if e.g. air also has to be released from the compressed air vessel. Both controllers are quite simple and can be adjusted without any problems in their setting conditions.

In the case of three-level controllers, a small difference between the trigger points can lead to continuous oscillation. The trigger points must be spaced so far apart that the controlled variable has not already reached the opposite trigger point between the time of the command (e.g. to close a valve) and the time at which the command is executed (the valve is fully closed).

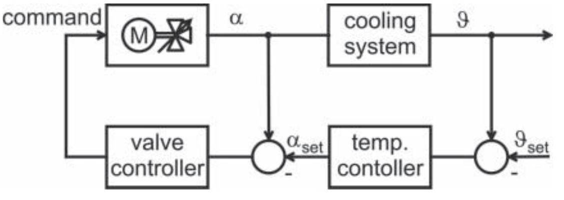

Figure 4.2.8: Cascade temperature control with control of the valve position in an inner control loop

4.3 Control engineering

4.3.1 Logic functions and sequences

Often the purpose of a specific control unit is to trigger certain switches depending on the switching state of other switches or depending on the time that has passed since the occurrence of a specific event. Upon receiving a starting command from the operator, for example, an automatic control for starting a motor according to the star-delta connection (figure 3.2.7) would have to react in the following sequence:

- Close star contactor
- Close main contactor
- Dwell awhile (until the motor has presumably reached its nominal rotation speed)
- Open star contactor

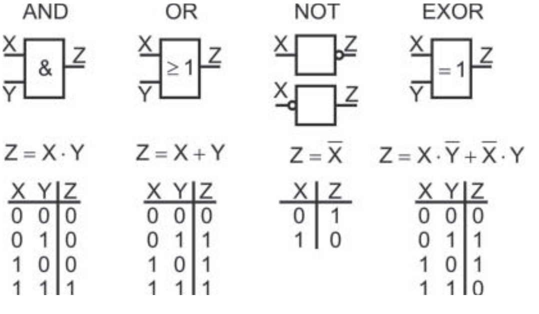

Figure 4.3.1: Logical operations with symbol, Boolean equation and truth tables

The foundations of this book were laid in 2006 with the German edition, which, as a follow-up to two previous well-known German books on marine engineering, was quite successful. However, as shipping is an international business and the working language on board thousands of ships nowadays is English, the authors, editors and publishers agreed to have the German book translated into English.

The book represents a compilation of marine engineering experience. It is based on the research of scientists and the reports of many field engineers all over the world.

Its principal aim is to gather the experience that has been gained by many engineers in the extremely broad field of marine engineering.

This book is mainly directed towards practising marine engineers, principally within the marine industry, towards ship operators, superintendents and surveyors, but also towards those in training and research institutes as well as designers and consultants.

Each author of this compendium is an expert in his field and has worked in the maritime industry, be it as seagoing marine engineer, surveyor or shipyard engineer, with an engine manufacturer or supplier, with a classification society, as a researcher in applied sciences or a lecturer in maritime training institutions or universities.

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