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COMPENDIUM Marine Engineering

Operation – Monitoring – Maintenance

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Preface

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.

We are grateful to our brave publishers for being willing to accept the risks that accompany a project to translate a book on such a specific technical subject as marine engineering into another language. Thanks is also due to the many translators who have so painstakingly translated the German text and without whom it would have been impossible to produce this book.

As the esteemed reader may gather from the curricula vitae, every one of the 21 authors 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 sub-supplier, with a classification society, as a researcher in applied sciences or a lecturer in maritime training institutions or universities.

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.

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

The 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.

Chapter 1 deals with the principal elements to be found on board ships. The following chapters are then devoted to the various systems such as the propulsion and drive systems (chapter 2), electrical and electronic systems (chapter 3), measuring, monitoring and control systems (chapter 4), supply and disposal systems (chapter 5), ventilation, air conditioning and refrigeration systems (chapter 6), cargo handling systems (chapter 7), manoeuvring systems (chapter 8), ship types and their governing technical details such as hydrodynamics, stability, strength, sea-keeping properties and other aspects, presented in an extremely compact form (chapter 9), followed by fire fighting, safety and rescue systems (chapter 10), the impact of maintenance and viable concepts for keeping all systems fit for service (chapter 11), damage and how to deal with it (chapter 12), the regulations and institutions governing ship operation (chapter 13), and, finally, details of how to convert traditional measuring units into modern SI units (chapter 14).

In each of the chapters of this book, an attempt has been made to achieve a fair balance between theoretical considerations and practical experience, so that the information presented will be of value to those practising marine engineering.

For more advanced studies, particularly of a theoretical nature, the information presented here may act as a starting point for exploring an individual problem in greater depth.

At the end of each chapter, a series of references is given so that, if necessary, the reader may refer to the original source containing full details of the topic under consideration. The majority of these references, however, are written in the German language. This is due to the fact that this book is a translation of the original German edition. But, as the engineer's language consists of the sketch, the formula and the diagram, we are confident that this will not really present a problem.

Glücksburg, Germany, June 2009
Hansheinrich Meier-Peter

Warnemünde, Germany, June 2009
Frank Bernhardt

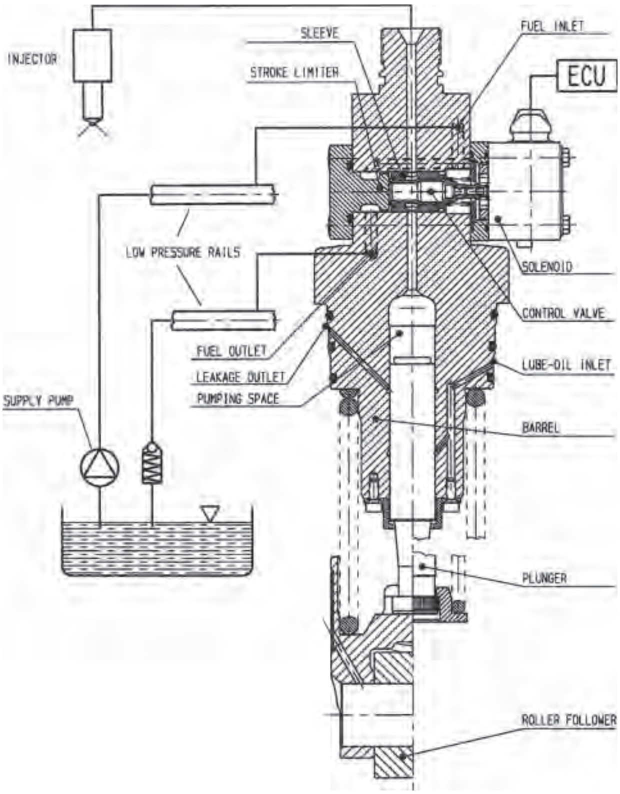


Figure 1.1.78: Injection system controlled by solenoid valve (unit pump system) (Courtesy: *Bosch*)

of the pump chamber and gradual shutoff must be effected by the solenoid valve, so fuel injection does not continue with the maximum quantity if the valve blocks when closed. Fuel injection with the maximum quantity can occur only once, which the engine must be able to withstand, since the pump chamber is not filled when the valve is closed.

One design advantage is the low installation height of solenoid-valve pumps, since the piston need have only the minimum length necessary for sealing, because of the absence of the control edge, and the solenoid valve can be installed on the side of the pump casing.

Stepped or pilot injection is possible with a fuel injection pump controlled by a solenoid valve. Furthermore, by (cyclical) switching-off of individual pumps under part load, a more favorable performance of the other cylinders of the engine can be achieved [1.1.5.2].

Fuel injection pump with hydraulic drive

The goal of developments in combustion is to disassociate the injection times and injection pressure from their association with cam speed and cam shape, and to set these parameters only according to the thermodynamic requirements of combustion.

This effect can be achieved using a hydraulic-powered fuel injection pump. Here, instead of by a cam, the pump piston is driven by a hydraulic working piston drawing its energy from a servo oil system at a pressure of about 200 bar. Under this arrangement, the engine camshaft can be dispensed with, provided the gas exchange valves and starting valve are also controlled hydraulically/electronically. Fig. 1.1.79 shows the hydraulic system of a slow-speed two-

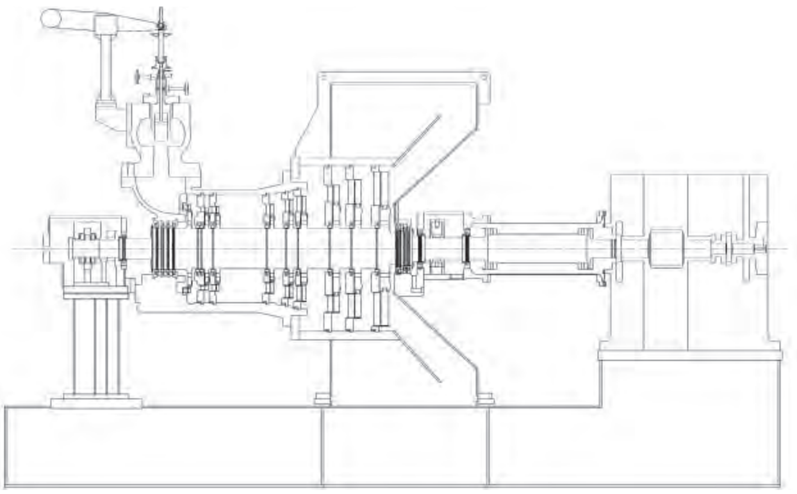


Figure 1.3.12: Impulse-type condensing turbine for a generator set [1.3.15]

is controlled depending on a temperature difference characteristic of these stresses. Steam condenses while preheating the turbine and this condensate can accumulate in the casing. During start-up, this condensing water must be drained from the casing through drain lines closed by automatic steam traps, so that it does not damage the rotor blades.

Depending upon temperature and pressure, the casing is almost always made of heat-resistant cast steel. Only large low pressure sections are of welded construction. The casing rests directly on the foundation or indirectly via the bearing pedestals, and is fixed to the foundation at one point, the fixed point. At the other support points, the casing can slide and expand horizontally and vertically when it heats up.

Blading: Blades must absorb forces of flow and pressure, and the pulsating flow excites oscillations. The rotor blades are also subject to centrifugal forces. Blade profiles, design, and structure have a significant influence on turbine efficiency.

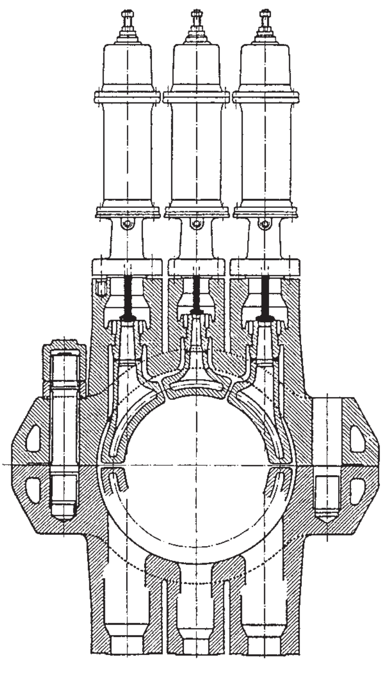


Figure 1.3.13: Section through the live steam entry of an HP-turbine (acc. to [1.3.3])

Steam turbines, especially condensing turbines, have a high expansion ratio (v_{out}/v_{in} > 1000). Therefore, the blade length L increases greatly from the regulating stage to the last-stage blades, despite the increasing mean diameter D_m. The clearance σ between blade and housing or rotor changes only slightly. In the high pressure section (short blades, low L/D_m and high σ/L), blades have a cylindrical profile and terminate in shroud plates or riveted-on shroud bands, in order to limit clearance losses and fasten the blades to one another (Figure 1.3.14a). For long blades, these leakage losses are not relevant (σ/L is small) and shrouds can be dispensed with. On the other hand, tangential

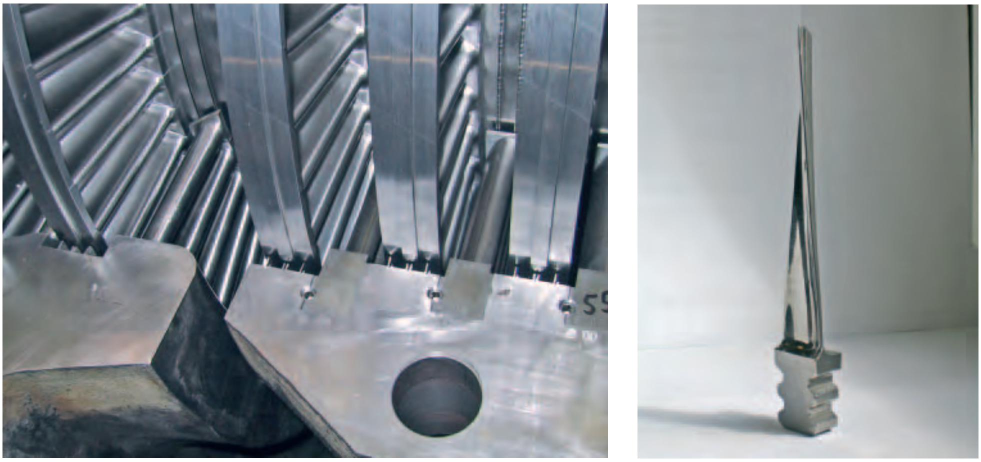


Figure 1.3.14: Turbine blade designs: a) cylindrical blades with shroud b) twisted blade [1.3.14]

velocity changes substantially from blade root to blade tip (L/D_m is large). The blade profile is made to suit changing velocity triangles, resulting in twisted blades (Figure 1.3.14b). Blade profile design has a considerable effect on stage efficiency. The blade lengths and relative clearance, as well as the effort devoted to sealing the blades and adjusting blade profile to the real velocity triangles are particularly important in this respect.

Figure 1.3.15 shows stage efficiency as a function of the product of volumetric flow coefficient $\varphi = c_m/u$ and relative blade length L/D_m for blades of full admission stages. Partially admissioned blades have a lower efficiency because of the additional windage losses.

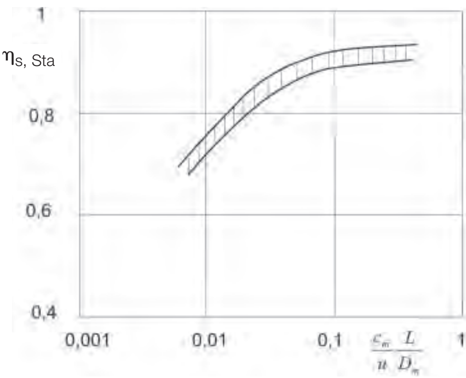


Figure 1.3.15: Dependency of stage efficiency from the blade length

The blade anchoring transmits the forces and moments of the blades to the rotor or housing. The specially profiled blade root is inserted into the ring grooves of similar counter profile at the rotor. Fir-tree roots and multiple fork roots are used for highly stressed blades, such as last-stage blades.

The blades are fastened in the casing or rotor by way of the root. The blade root of the rotor blades must absorb centrifugal forces and transmit the forces resulting from the flow to the rotor. Grooves for rotor blades are located in the wheel disks (chamber design) or in the drum-type rotor. Guide vanes have the ring grooves formed directly in the casing or in horizontally split guide vane carriers or diaphragms that are fixed to the upper and lower housing sections. Often a guide vane carrier will carry a blade group between two extractions (Figure 1.3.11).

Vibrational fatigue failure of the blades is amongst the most frequent causes of severe damage. Measures to prevent blade vibrations are:

– Natural resonance frequencies of blades should be sufficiently remote from frequencies of excitations;

$$VCF = \frac{980\,\frac{kg}{m^3} - 0{,}63\,\frac{kg}{m^3 \cdot K} \cdot 25K}{980\,\frac{kg}{m^3}} = 0.984 \qquad (1.16.3)$$

results in

$$V_t = \frac{V_{15}}{VCF} = \frac{1000\ m^3}{0{,}984} = 1016{,}3\ m^3 \qquad (1.16.4)$$

Product		Density correction factor k [kg/m^3 K]
IFO	Heavy fuel oil	0.63
MDO	Marine diesel oil	0.68
MGO	Marine gas oil	0.70
Lube oil	Cylinder oil	Dependent upon base oil and additives

Table 1.16.1: Density correction factors for different types of oil

The pressure-dependence of volume and density is relevant only at high pressure (Common Rail Injection, hydraulic power packs). Density in air is about 0.1 % greater than in a vacuum. In many countries, trading in mineral oil is based on mass-in-air and not on mass-in-volume because of customs duties and tax structures, regulations or out of habit. If a tank truck is weighed e. g. after being loaded with lubricants, the weight is based on mass-in-air. If the quantity of delivered lubricant is determined, however, through tank sounding or with a flow metering device, i. e. on the basis of volume, then conversion needs to be carried out using correction factor f (f ≈ 0.999) from density-in-vacuum to density-in-air (factor for converting weight-in-vacuum to weight-in-air). For mineral oils, the result for mass-in-air Is about 0.1 % lower than for mass-in-vacuum. The following example should clearly show their interrelationship:

– Delivered volume, corrected to 15° C: 10 m^3 or 10,000 litres
– Density at 15° C: 939 kg/m^3
  Mass-in-air = 10 m^3 * 939 kg/m^3 * 0.999 = 9380.6 kg
  Mass-in-vacuum = 10 m^3 * 939 kg/m^3 = 9390.0 kg

An interrelationship exists between the molecular mass, ratio of carbon to hydrogen (C/H-ratio) and density of a material. Material with a higher molecular mass normally has higher density. In the case of various materials with the same molecular mass, the material with the higher C/H-ratio normally also has the higher density. Thus the density of a fuel increases significantly with increasing sulphur content.

Viscosity *(DIN 51 562, IP 71, ASTM D445)*

Viscosity is temperature-dependent and one of the most important attributes of oil. It is a measure of inner friction in a liquid. Thick-flowing, semifluid or highly viscous oils have high viscosity. Thin-flowing or low viscous oils have low viscosity. A description of the flow behaviour of materials is discussed under Rheology.

Dynamic viscosity is also termed absolute viscosity. Let us consider a homogeneous fluid in a system at a constant temperature and imagine two parallel liquid layers therein with area A at spacing y. If the upper liquid layer is now displaced by a tangential force F with velocity c – or the areas are displaced with respect to one another by displacement path u or differential velocity dc – the action of shear force F is proportional to velocity c and inversely proportional to distance y for a Newtonian liquid. The velocity difference dc per unit of film thickness dy is defined as velocity gradient D or also as shear rate with

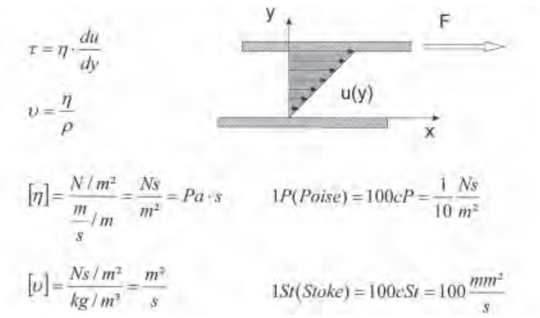


Figure 1.16.1: Viscosity of a Newtonian liquid

$$D = \frac{dc}{dy} \qquad \left[s^{-1}\right] \tag{1.16.5}$$

The shear stress τ is the shear force F or tangential force per unit of area A to be applied in the flow direction for deformation of the plastic system.

$$\tau = \frac{F}{A} \qquad \left[Pa\right] \tag{1.16.6}$$

The proportionality factor of shear stress τ and shear rate D is termed dynamic viscosity η. It is specified in Pascal second [Pa s] or previously in Centipoise [cP].

$$\frac{F}{A} = \eta\,\frac{dc}{dy} \qquad \left[Pa\right] \tag{1.16.7}$$

Depending on the shear rate, the dynamic viscosity and shear stress are represented as follows:

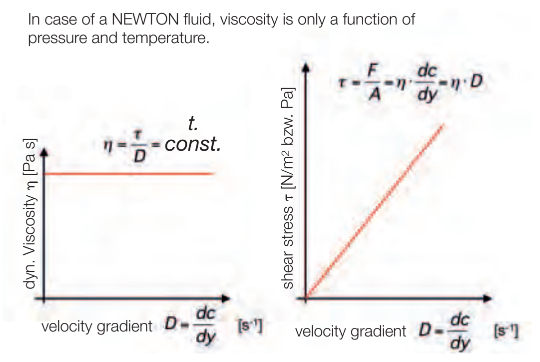


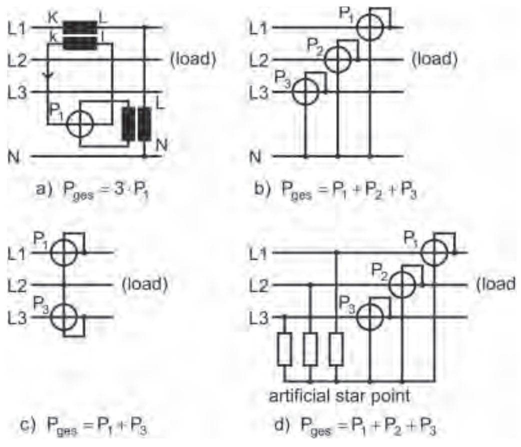
Figure 1.16.2: Shear stresses and dynamic viscosity as a function of the shear rate

Limits have been established for the so-called HTHS-viscosity (High Temperature High Shear or high oil temperatures (150° C), high shear rate ($D = 10\ s^{-1}$)). These ensure that engine oils have the necessary lubrication safety in the bearing area on the one hand and on the other make fuel savings possible through a reduction in dynamic viscosity without increased wear.

Viscosity also increases with rapidly rising pressure. The pressure-dependence of viscosity depends on the base oil or oil composition, on the viscosity itself and on the oil temperature. The change in viscosity due to pressure is stronger in the case of paraffin-based oils[1] than naphthene-based oils[2].

1 Paraffins = alkanes = aliphatic hydrocarbons with single bonds (saturated)

2 Naphthenes= cycloalkanes= cycloparaffins = cyclic hydrocarbons with single bonds



a) symmetric system with neutral N
b) 3-wattmeter method
c) 2-wattmeter method
(Aron's technique)
d) 3-wattmeter with artificial star point
(current and voltage transformers for
b to d corresponding to a)

Figure 4.1.9: Power measurement in a three-phase system

which all lead to the same result following the above example. Connection of all converters and measuring instruments with the correct signs is especially important with this two-wattmeter circuit (Aron measuring circuit).

4. Symmetrical or unsymmetrical mains without N-conductor: An artificial neutral point can be created (figure 4.1.9d) using three equal resistances or other impedances. This method offers no advantages compared to two-wattmeter circuit and is therefore seldom used.

4.2 Closed-loop control engineering

4.2.1 Design of a control circuit, transfer function, Bode diagram

A closed-loop controller (figure 4.2.1) adjusts a controlled variable x (e. g. temperature in a cooling water circuit) to a specific value determined by the command variable w (set-point value). The controller manipulates the process through the correcting variable y. Consequently, any disturbance z does not affect the controlled variable x, rather in the ideal case they are evened out by the controller through the correcting variable y. The closed loop is thus characteristic of a closed-loop control system. Compared to this, the term "open-loop control" designates systems with an open loop, where the actual adjustment is made by the system operator.

The performance of the technical system to be controlled as well as the controller is described in an abstract manner in closed-loop control engineering. The equipment used for the the controller, sensors and actuators can be determined only after the system has been analysed and a controller has been designed.

The controlled system and the controller are characterized by their transient response. It describes how the output variable reacts to an input variable. Often this reaction depends on the installation's operating point. Therefore, as a first step, only the reaction to very small changes in the input variable is described, starting from a given operating point. In all following calculations, the variables x, y and z in figure 4.2.1 are the deviations from the selected operating point. A linear interrelationship results for small values, i. e. k-times the input variable $k \cdot (z - y)$

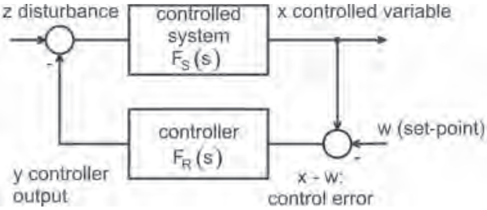


Figure 4.2.1: Basic form of a control loop

results in k-times the output variable $k \cdot x$ at the controlled system. Furthermore, it is presumed that the transient response is independent of time. For simplification, pure transient-time effects are initially excluded. It is presumed that a change in the input variable of a system immediately causes a change in the output signal which, of course, can initially be small. Immediately, in this sense, means that there is at the most an infinitesimal delay. Every controlled system and every controller can be described by a linear differential equation under these conditions, e. g. in the case of the controller for w = 0

$$y + a_1 \cdot \frac{dy}{dt} + a_2 \cdot \frac{d^2y}{dt^2} \ldots = b_0 x + b_1 \cdot \frac{dx}{dt} + b_2 \cdot \frac{d^2x}{dt^2} \ldots \qquad (4.1)$$

Further consideration can be considerably simplified and also made considerably clearer by applying the Laplace transformation. A sinusoidal, exponentially increasing or decreasing progression is presumed for describing the transient response of x,

$$x(t) = e^{\delta t} \cdot \cos \omega t \qquad (4.2)$$

which is determined by the two constant factors δ and ω. Terms with sin ωt are created through derivation (or even through integration). Both sides of the equation (4.1) are equal only if all cos terms are equal and all sin terms are equal. Consequently, y(t) consists only of terms with cos ωt and sin ωt. No frequency other than ω occurs.

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