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Dimension prediction models of ship system components based on first principles

Douwe Stapersma[1] and Peter de Vos[2]

ABSTRACT

In this paper we describe a generic way of sizing main dimensions of primary equipment for marine applications. Contrary to a straight fit through a database the method tries to develop expressions in which, apart from the main specifications in terms of power and speed, the selection of the main machine parameters has an influence on the ultimate overall dimensions of the component. The result is a "rubber" design model that uses in an intelligent way the first principles underlying the design of machines and that can be used in preliminary design of complex maritime objects. The method will be illustrated for components as diverse as electric machines, gearboxes and diesel engines but is thought to be generally applicable.

KEY WORDS

Dimension prediction (sizing) models; first principles; electric machines; gearboxes; diesel engines;

INTRODUCTION

The basic idea behind first principle based dimension prediction models of ship system components (i.e. machinery or equipment like engines, motors, pumps, heat exchangers etc.) is that the dimensions of a type of machinery can be estimated by sizing the core of that machine to the required power output using first principles. The core of a machine consists of primary and secondary elements. The size of the primary elements can be determined from the required power output of the machine. The size of the secondary elements can be determined in a next step from the size of the primary elements. Together they determine the size of the core of the machine. In a final step the actual machine dimensions can be predicted from the size of the core using regression analysis.

To give an example for a diesel engine the primary element is the cylinder and indeed there is a first principle relationship between the size of the cylinder and the power output of a diesel engine. The cylinders need to be combined with a crankshaft, which is the secondary element, in order for the engine to work. The core of a diesel engine is therefore the combination of a number of primary elements (cylinders) and a secondary element (crankshaft). Together they determine the size of the core of the engine, which contributes significantly to the size of the overall machine.

In **Figure 1** the process of predicting machine dimensions using first principles based dimension prediction models is summarized in a flowchart.

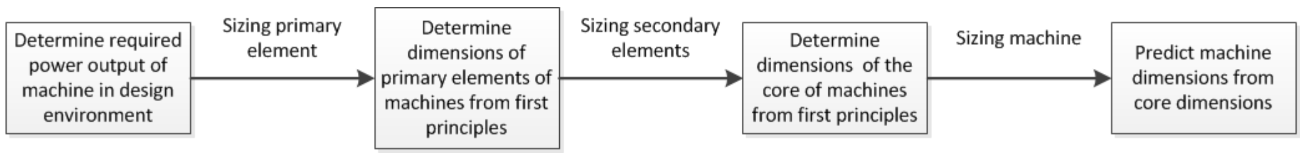

Figure 1: Process of predicting machine dimensions based on first principles.

The search for first principles based dimension prediction models of ship system components is part of the PhD research of the second author. The PhD project MOSES CD (Model-based Ship Energy System Conceptual Design) aims to improve the conceptual design of onboard service systems (also known as platform systems or main and auxiliary systems). It consists of three major parts: network modelling for variation of system topology (de Vos, 2014), performance modelling for prediction of system performance and dimensions modelling for prediction of component dimensions. This paper obviously is related to the latter.

This paper is divided into five sections. The first section elaborates on the general process for predicting machine dimensions from required power output using first principles as was already introduced above. The second section describes the first and

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second step in **Figure 1**, i.e. how the core of three different machines sizes with required power output. The third section discusses the final step of **Figure 1**, i.e. how the three different machines size with their core. The fourth section shows the correlation of the first principles based dimension prediction models with real machines, after which section five concludes the paper.

GENERAL PROCESS

In a ship design environment the first step to finding dimensions of machines is determining their required power output from a load balance or similar. This required power output is usually more or less fixed by the mission and size of the ship, which are of course related. From the required power output the size of the primary elements of machines can be found using first principles (step 1). In this paper we present the direct relationships that exist between the size of the primary elements and the power output of electric machines, gearboxes and diesel engines by deriving them from first principles. This paper focuses on dimension prediction models for these machines only. It is however expected that the methodology as presented here can be used for other ship system components as well. This is however left for future papers.

For diesel engines the primary element (and secondary and core) have already been introduced in the introduction. For an electric machine the primary element is the rotor and again there is a first principle relationship between the size of the rotor and the power output of an electric machine. The same can be stated for gearboxes for which the pinion proves to be the primary element. These first principle relationships can be applied in a design environment to size the primary elements of the machines, since in a design environment the power output of the machine is determined for maximum load conditions.

Once the size of the primary element is determined in a second step necessary additional elements can be sized as well by a geometric analysis of the construction of the machines. For the electric machine this second step means sizing the stator from the rotor dimensions. For a gearbox this means the wheel is sized from the pinion dimensions and for a diesel engine this means the crankshaft is sized from cylinder dimensions. Together the primary and secondary elements build up the core of the machines whose size is then determined. In a third and final step an estimate can be made for the dimensions of the entire machine on basis of the size of the core of the machine using regression analysis. We will hypothesize that a first order polynomial function exists between the size of the core and the size of the complete machine.

This approach to dimension prediction (refer to **Figure 1**) leads to expressions for the dimensions of the machines that have more physical meaning than standard regression analysis based dimension prediction models as normally used in comparative studies; e.g. in the All Electric Ship project in the Netherlands (van Dijk, 1998) and (Frouws, 2005), and also in (van Es et al., 2012). For the diesel engine the procedure was suggested earlier (Stapersma, 1998) but not implemented until now.

Since regression analysis is still necessary on a deeper level to fit the first principle based dimension prediction models to the dimensions of actual machines (by fitting the first order polynomial functions to dimensions data in a machinery database), it is not the objective of this paper to denounce regression analysis based models, but rather to add physical meaning to them. Furthermore this paper contributes to understanding the analogies that exist between the dimensions of apparently completely different machines.

SIZING THE CORE OF ELECTRIC MACHINES, GEARBOXES AND DIESEL ENGINES

As discussed the core of the machines consists of primary and secondary elements. In this section we will first focus on sizing primary elements from the required power output, i.e. the first step of the process. Then the second step follows: sizing secondary elements from primary element dimensions. Together they determine the size of the core of the machines.

Sizing primary elements

Start with the basic equation that relates power output to torque and angular speed:

$$P = M \cdot \omega = 2\pi \cdot M \cdot n \qquad [1]$$

Where P is power in W (or J/s), M is torque in Nm, ω is angular speed in rad/s and n is rotational speed in revolutions per second (rev/s or rps). From a more generalized perspective the torque is the “effort” variable and the rotational speed is the “flow” variable; power is the product of effort and flow irrespective of the energy form that is relevant. The three machines that are discussed here all deliver or transmit rotational mechanical energy, so equation [1] is true for all three machines (electric machine, gearbox and diesel engine).

Now it can be shown that for all three machines the power depends on:

- A characteristic mean shear stress τ or mean pressure p in N/m^2, which is determined by the manufacturer of the machine taking into account limitations due to material properties of the materials being used. The characteristic shear stress or pressure is related to the torque of the machines.
- A characteristic velocity v or c in m/s that is also determined by the manufacturer on basis of limiting inertial forces and/or wear and tear (i.e. life cycle) considerations. The characteristic velocity is related to the rotational speed of the machines.
- Characteristic dimensions of the primary elements of the machines.

The latter are of course of interest, since the objective of this section is to size the primary elements first. Applying the first two ensures that the expressions for primary element size are based on first principles. To derive these expressions note that the primary elements of the machines are all cylindrical volumes of which the characteristic dimensions are diameter D and length L (see **Figure 2**).

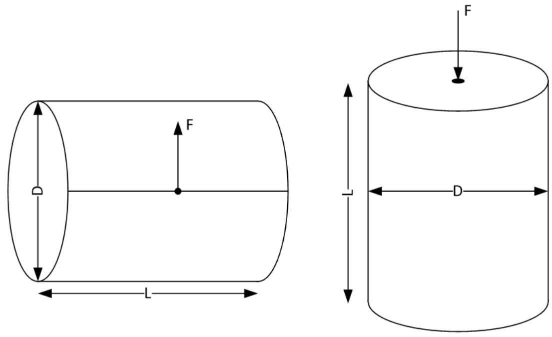

Figure 2: Cylindrical volumes - left depicting a rotor of an electric machine or pinion/wheel of a gearbox and right depicting a diesel engine cylinder.

For the electric machine and the gearbox the torque that is delivered or transmitted is determined by the force F that is acting at the circumference of the cylinder (as depicted at the left in **Figure 2**), i.e. at the radius r of the cylinder (r = D/2). For the electric machine the force F is the Lorentz force or EMF (ElectroMotive Force) that acts on the current carrying conductors along the side of the rotor as a consequence of the rotating magnetic field from the stator. For the pinion (and wheel) of the gearbox the force F results from mechanical interaction (action = reaction) between the teeth of pinion and wheel. For the diesel engine the torque that is delivered is determined by the pressure in the cylinder that results in a force F acting on top of the piston (as depicted at the right in **Figure 2**).

The forces F that are present in all three machines can be expressed as a mean shear stress or mean pressure in N/m^2 by dividing the force F by an area A. It should be emphasized that it is not the intention to calculate actual shear stresses or pressures present in the machines, but rather characteristic stresses or pressures that will help determine expressions that relate power and size of the machines. This means that the stresses and pressures that will be defined cannot be measured anywhere in the machine during operation even though they do have a relation with actual stresses and pressures. It also means that the area A can in principle be chosen arbitrarily, yet it is practical to choose the area rationally in order to increase the physical meaning of the model.

For the electric machine the area A that is chosen is the rotor circumferential area, which is calculated by $\pi \cdot D_R \cdot L_R$, where D_R is the diameter and L_R is the length of the rotor in m. Since the rotor contains current carrying conductors all around and the Lorentz force therefore acts all around the rotor it is logical to choose the complete circumferential area. The intensity of the EMF is thus characterized by a mean shear stress that is present everywhere at the rotor side. In fact the stress as defined is sometimes referred to as working force density or average air gap shear stress; amongst others (Hodge et al. 2001) and (Rucker et al., 2005). Again it is emphasized that this is not an actual shear stress, since the latter is not equally distributed over the area, but a means to relate power and dimensions from first principles. Thus for the electric machine:

$$M = EMF \cdot \frac{D_R}{2} = \tau_R \cdot A_R \cdot \frac{D_R}{2} = \frac{\pi}{2} \cdot \tau_R \cdot D_R^{\ 2} \cdot L_R = 2 \cdot \tau_R \cdot V_R \qquad [2]$$

This well-known expression, see a.o. (Kirtley, 2005), shows that there is indeed a first principle relation between torque of the machine and size of the primary element (the rotor) being characterized by its diameter and length. In fact it can be concluded that torque scales with volume. Also any torque can be delivered by a “short and fat” rotor (small L_R, large D_R) or a “long and slender” rotor (large L_R, small D_R).. The length/diameter ratio $\lambda_R = L_R / D_R$ of the cylinder is a shape factor that characterizes slenderness of the rotor and will be used later on to relate dimensions of the primary element to power output. Clearly the intensity of the force F characterized by the mean shear stress is a parameter that is determined by the manufacturer of the machine, who will most probably try to push the limits of material properties while maintaining reasonable safety margins in order to deliver an as-compact-as-possible machine.

For the gearbox the force F is a tooth force (TF) that also acts on the circumference of the cylinder, but only on a small part since the force is concentrated on one tooth pair. Therefore the circumferential area is reduced by two times the number of teeth on the pinion z_P, which means the intensity of the tooth force is characterized as a tooth shear stress that is present on a mean shear area of one tooth at the nominal contact diameter (disregarding the fact that the area at the base will be larger and that at that point there also is bending: again we are not calculating actual stresses but introducing reference stress).

$$M = TF \cdot \frac{D_P}{2} = \tau_T \cdot \frac{A_P}{2 \cdot z_P} \cdot \frac{D_P}{2} = \frac{\pi}{4} \cdot \frac{\tau_T}{z_P} \cdot D_P^{\ 2} \cdot L_T = \frac{\tau_T}{z_P} \cdot V_P \qquad [3]$$

As before torque is a specific shear stress times the primary element volume and the similarity between expressions [2] and [3] is clear, so the gearbox manufacturer can size his primary element in a similar manner as an electric machine

manufacturer. A new parameter is the number of teeth that on the pinion is limited to a certain minimum due to curvature and the risk of undercuttting the tooth profile.
For diesel engines similar expressions as equations [2] and [3] have been derived in (Klein Woud et al., 2003):

$$M = \frac{1}{8} \cdot \frac{i}{k} \cdot p_{me} \cdot D_B{}^2 \cdot L_S = \frac{i}{2\pi \cdot k} \cdot p_{me} \cdot V_S \qquad [4]$$

In this expression the mean shear stress as was used in equations [2] and [3] for electric machine and gearbox respectively has been replaced by the mean effective pressure p_{me} in N/m^2, which is a well-known performance parameter for marine diesel engines. The reason for this is that the force F acts on the top of the primary element (piston) instead of at the side of the primary element (rotor for the electric machine and pinion for the gearbox). The term i/k represents the number of cylinders i in the engine and the number of revolutions per power stroke (k=1 for 2-stroke and k=2 for 4-stroke engines). This is another difference between diesel engines and the other two machines; the fact that the torque of the machine is the result of a number of primary elements working together intermittently instead of one primary element working continuously. Still, the similarity between expressions [2], [3] and [4] is again apparent and for all three machines it can be concluded that torque scales with volume of the primary elements.
The three expressions [2], [3] and [4] for the "effort" variable torque in expression [1] have been listed in the second row of **Table 1**. In the third row the rotational speed of the machines (flow variable) has been related to a characteristic velocity; tangential velocity at circumference for electric machine and gearbox and mean piston speed for the diesel engine. These velocities are limited by inertial forces or by wear (i.e. life cycle) considerations. They are again "manufacturer" parameters and their values tend to lie in a limited range. All three expressions show that rotational speed is inversely proportional to one of the characteristic dimensions of the primary elements. This is echoed in the well-known fact that machines with higher rotational speeds are smaller.
In the final row the expressions in the second and third row are combined using equation [1] to arrive at expressions that relate power output to machine specific parameters determined by manufacturers and size of the primary elements. Note that power always is the product of a characteristic area of the element and a "Technology Parameter" (the latter being the product of a characteristic stress or pressure and circumferential or translational speed).

Table 1: **Overview of torque, speed and power relations as function of size and machine specific parameters.**

	Electric Machine	Gearbox	Diesel Engine
Effort	$M = \frac{\pi}{2} \cdot \tau_R \cdot D_R{}^2 \cdot L_R$	$M_P = \frac{\pi}{4} \cdot \frac{\tau_T}{z_P} \cdot D_P{}^2 \cdot L_T$	$M = \frac{1}{8} \cdot \frac{i}{k} \cdot p_{me} \cdot D_B{}^2 \cdot L_S$
Flow	$n = \frac{v_t}{\pi \cdot D_R}$	$n_P = \frac{v_t}{\pi \cdot D_P}$	$n = \frac{c_m}{2 \cdot L_S}$
Power	$P = \pi \cdot \tau_R \cdot v_t \cdot D_R \cdot L_R$	$P = \frac{\pi}{2} \cdot \frac{\tau_T \cdot v_t}{z_P} \cdot D_P \cdot L_T$	$P = \frac{\pi}{8} \cdot \frac{i}{k} \cdot p_{me} \cdot c_m \cdot D_B{}^2$

However in a ship design environment power as function of dimensions is usually not of interest. It is the inverse that is sought after: dimensions of machines as function of power, as was already discussed before. Also a designer may want to vary rotational speed (to vary size of the machine) and shape factor $\lambda = L / D$ (to fit the machine in a certain space). Therefore the power equations in **Table 1** are rewritten to include rotational speed and shape factors λ; see second row of **Table 2**. These relations are written such that the right hand side has the parameters that are more or less fixed by technology and the manufacturer while the left hand side gives the design choices of the user of which power and speed are prime of course. For the electric motor and pinion he/she can play with the L/D ratio and for the pinion perhaps somewhat with the number of teeth (but as said there is a minimum number, in fact around 20). For the diesel engine the choice between 2- and 4-stroke and number of cylinders is a further degree of freedom.

Table 2: Overview of dimensioning equations for primary elements based on known power.

Electric Machine	Gearbox	Diesel Engine
$\frac{P \cdot n^2}{\lambda_R} = \frac{1}{\pi} \cdot \tau_{EM} \cdot v_t{}^3$	$z_P \cdot \frac{P \cdot n_P{}^2}{\lambda_P} = z_W \cdot \frac{P \cdot n_W{}^2}{\lambda_W} = \frac{1}{2\pi} \cdot \tau_{TS} \cdot v_t{}^3$	$k \cdot \frac{P \cdot n^2}{i} = \frac{\pi}{32} \cdot \frac{p_{me} \cdot c_m{}^3}{\lambda_S{}^2}$
$D_R = \sqrt{\frac{1}{\pi} \cdot \frac{1}{\tau_{EM} \cdot v_t \cdot \lambda_R} \cdot P}$	$D_P = \sqrt{\frac{2}{\pi} \cdot \frac{z_P}{\tau_{TS} \cdot v_t \cdot \lambda_P} \cdot P}$	$D_B = \sqrt{\frac{8}{\pi} \cdot \frac{k}{p_{me} \cdot c_m} \cdot \frac{P}{i}}$
$L_R = \sqrt{\frac{1}{\pi} \cdot \frac{\lambda_R}{\tau_{EM} \cdot v_t} \cdot P}$	$L_T = \sqrt{\frac{2}{\pi} \cdot \frac{z_P \cdot \lambda_P}{\tau_{TS} \cdot v_t} \cdot P}$	$L_S = \sqrt{\frac{8}{\pi} \cdot \frac{k \cdot \lambda_S{}^2}{p_{me} \cdot c_m} \cdot \frac{P}{i}}$

Finally the equations in the second row of **Table 2** can be rewritten (using some algebra) to find expressions for the diameter and length of the primary elements as a function of power; these are, as promised in the introduction, first principle relationships between size of the primary elements and power, see row 3 and 4 of **Table 2**. Apart from power the size is determined by three main manufacturer choices: a typical stress (τ) or pressure (p), a typical circumferential (v_t) or translational (c_m) speed and a shape factor of the element (L/D), these are the three "players in the game" of which the first two seem to be limited by material properties for all three machines. For the gearbox additionally a number of teeth on the pinion (z_p) is required and for the diesel engine the number of cylinders (i) and the number of revolutions per cycle (k).

Sizing secondary elements and core

Now that the primary elements have been sized the first step of the process for dimension prediction of electric machines, gearboxes and diesel engines is finished. The second step is sizing additional elements whose size follow from the size of the primary elements.

For the electric machine this means adding an additional "manufacturer" parameter: the rotor/stator diameter ratio $s = D_R/D_S$. **Figure 3** shows a schematic of electric machine core construction (both in transverse and longitudinal direction) and defines important dimensional parameters.

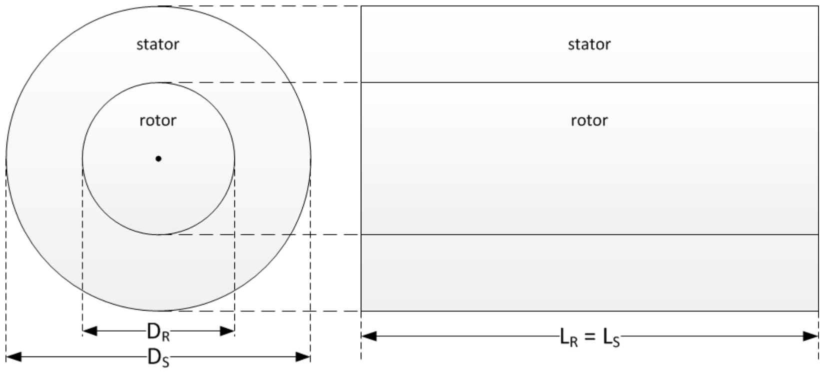

Figure 3: Schematic of electric machine core construction.

In contrast to the additional parameters that are needed for the gearbox and diesel engine for sizing secondary elements (which will be introduced shortly) the rotor/stator diameter ratio s regrettably is not a functional parameter that can be determined by the designer of ship systems. However typical values for s can be found in literature, e.g. (Miller, 1989); values are in the range of 0.45 – 0.55. This at least gives an idea for s which means the size of the secondary element (stator) is now found from the size of the primary element (rotor). Normally the stator length will be equal to rotor length as well. This also means the core dimensions of the electric machine are now found. In this case the primary element is completely surrounded by the secondary element, therefore:

$$\begin{aligned} L_{core,EM} &= L_S = L_R \\ W_{core,EM} &= D_S = D_R/s \\ H_{core,EM} &= D_S = D_R/s \end{aligned} \quad [5]$$

Note that for the gearbox, once the primary element (pinion) is sized and the gear ratio i_{GB} is given (or rather; chosen by the ship system designer), also the secondary element (wheel) is sized, since:

$$i_{GB} = \frac{z_W}{z_P} = \frac{D_W}{D_P} = \frac{n_P}{n_W} = \frac{\lambda_P}{\lambda_W} \quad [6]$$

The last ratio (of L/D ratios) only equals i_{GB} if the length of the teeth on pinion and wheel are the same ($L_P = L_W = L_T$), but this normally is the case of course. **Figure 4** shows a schematic of gearbox core construction (both in transverse and longitudinal direction) and defines important dimensional parameters. From this figure it can also be concluded that in this case the fact that the size of the secondary element can be found easily does not immediately result in the core dimensions of the machine, since the pinion can be horizontally or vertically offset with respect to the wheel (or something in between). The offset is characterized by the angle α. Many single marine gearboxes (SISO = Single Input Single Output) will have a vertical offset in order to be able to place the gearbox as far back (and low) in the ship as possible, but then again for double marine gearboxes (DISO – Double Input Single Output) a (more) horizontal offset may be chosen in order to obtain sufficient space between the two driving machines. In the latter case **Figure 4** should of course be expanded to include a second pinion as well.

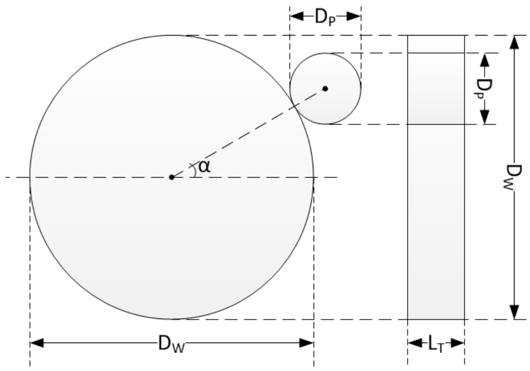

Figure 4: Schematic of gearbox core construction.

The core dimension of single marine gearboxes are determined by:

$$L_{core,GB} = L_T$$
$$W_{core,GB} = \max\left(\frac{D_P + D_W}{2} + \frac{D_P + D_W}{2} \cdot \cos(\alpha); D_W\right) \qquad [7]$$
$$H_{core,GB} = \max\left(\frac{D_P + D_W}{2} + \frac{D_P + D_W}{2} \cdot \sin(\alpha); D_W\right)$$

For the diesel engine the secondary element is the crankshaft. The dimensions of this secondary element are just as easily found as for the gearbox if one realizes that the diameter of the crankshaft must be equal to the stroke length. The length of the crankshaft is also relatively easy and is essentially determined by the number of cylinders that are connected to the crankshaft and their diameter. For a Line-engine the core length of the crankshaft is therefore estimated as $i \cdot D_B$. For a Vee-engine the story is different of course; here we estimate the core length of the crankshaft to be $i \cdot D_B / 2$.
Determining core dimensions for the diesel engine from primary and secondary elements is however rather difficult, at least more so than for the electric machine and gearbox, because of two reasons; the engine might be a L, V or even a boxer motor (horizontal cylinders opposite of each other to the left and right of the crankshaft – in fact a very specific V-engine) and the construction type may differ because of either trunk piston type construction or crosshead type construction.
In **Figure 5** again a schematic of core construction of the machine in question is given (in both transverse and longitudinal direction), but this time for a hypothetical V-engine of crosshead type construction. These types of engines do not exist (although designs for them have been made in the past), but this figure shows best how to derive general equations for the core of diesel engines in all possible cases: L- or V-engine is characterized by angle α and the difference between crosshead and trunk piston type (conceptually at least) is the extra stroke length that exists between the crankshaft outer diameter and the bottom of the cylinder. This extra stroke length does exist for crosshead engines, since the crosshead travels one L_S as well, but for trunk piston type engines this extra stroke length is zero, since for these engines the bottom of the cylinders conceptually "touches" the outer diameter of the crankshaft. To account for this the parameter "ct" (construction type) has been added; a basic assumption is ct = 0 for trunk piston type engines and ct = 1 for crosshead type engines. In a more refined model for ct an allowance can be made for the length of the connecting rod and the fact that the height of the piston at Bottom Dead Centre must be added to the stroke length. But information on connecting rod length and position height normally is not available so a pragmatic decision is to include both effects in the fit of the machine dimensions.

Careful analysis of **Figure 5** leads to the following expressions for the size of the core of diesel engines.

$$L_{core,DE} = i \cdot D_B \quad \textit{for L engines}$$
$$L_{core,DE} = \frac{i \cdot D_B}{2} \quad \textit{for V engines}$$
$$W_{core,DE} = 2 \cdot \max\left(\left(\frac{L_S}{2} + (1+ct) \cdot L_S\right) \cdot \sin\left(\frac{\alpha}{2}\right) + \frac{D_B}{2} \cdot \cos\left(\frac{\alpha}{2}\right); \frac{L_S}{2}\right) \qquad [8]$$
$$H_{core,DE} = \frac{L_S}{2} + \max\left(\left(\frac{L_S}{2} + (1+ct) \cdot L_S\right) \cdot \cos\left(\frac{\alpha}{2}\right) + \frac{D_B}{2} \cdot \sin\left(\frac{\alpha}{2}\right); \frac{L_S}{2}\right)$$

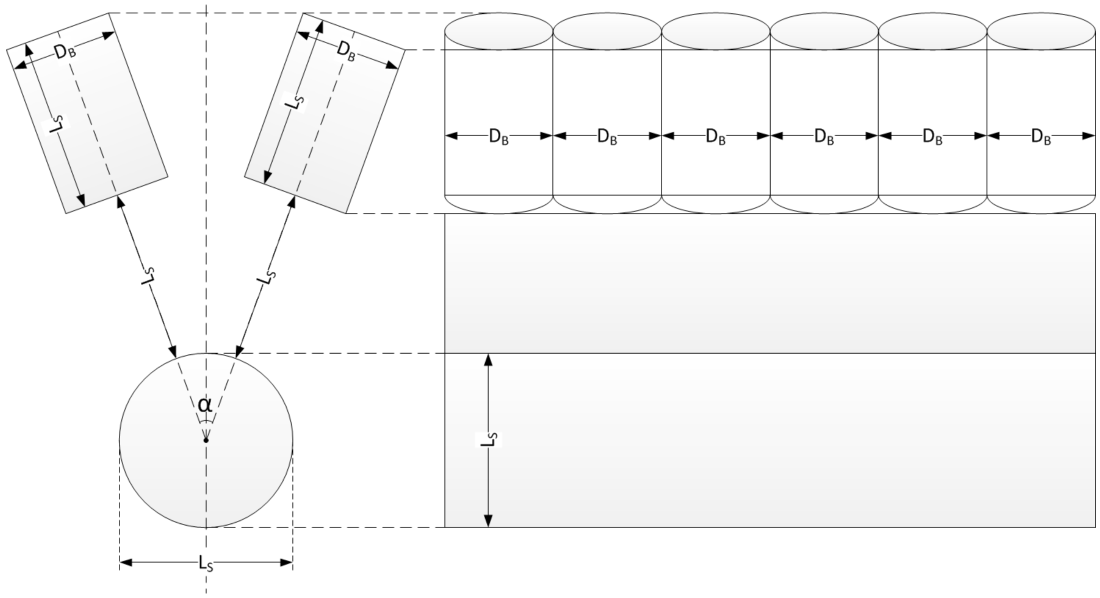

Figure 5: Schematic of diesel engine core construction.

SIZING MACHINE

Now that expressions have been given that relate dimensions of primary elements to power output of machines and core dimensions to dimensions of primary and secondary elements, the question becomes: how is size of the core related to machine size? This of course depends on the configuration of the machine that will be the topic of this section.

Start again with an electric machine. The shape of an electric machine actually resembles the shape of its core (stator + rotor) meaning the machine is also a cylindrical volume. This can also be seen from **Figure 6**, which is a drawing of an electric machine of a well-known electric machine manufacturer.

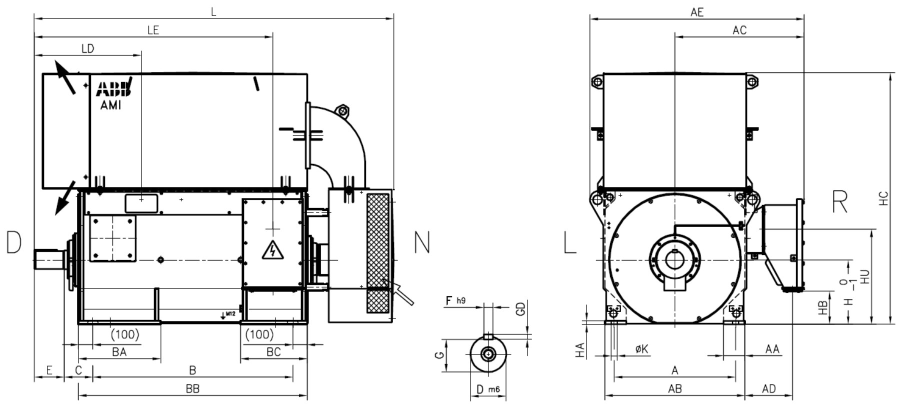

Figure 6: Typical electric machine construction. Source: online ABB catalog of HV induction motors (see references).

However, it also becomes clear from **Figure 6** that extra volumes are added to the machine. The main reason for this is required cooling of the machine (although the terminal box also requires quite some room). For smaller electric machines with open cooling a fan at the free end of the machine and cooling vanes around the housing will suffice for the required cooling. But for larger electric machines the heat cannot be dumped directly in the environment so they are closed and heat exchangers for cooling are required. The location of these heat exchangers represents another degree of freedom (on top or at the side or perhaps even some distance away from the electric machine). The sizing model may also be used for synchronous machines and in particular generators. Then the exciter and power electronics are further additions to the size of the electric machine.

Now a mathematical relation needs to be assumed between core dimensions and machine dimensions in order to be able to fit the model to actual machine data. Many options exist: polynomial functions, power laws or even Fourier series, but since the idea behind the model is that the core dimensions already represent a significant part of the actual machine dimensions a rather simple, but perhaps effective, relation is assumed. It is therefore hypothesized that the electric machine dimensions are related to the core dimensions with first order polynomials:

$$L_{EM} = A_0 + A_1 \cdot L_{core,EM}$$
$$W_{EM} = B_0 + B_1 \cdot W_{core,EM} \quad [9]$$
$$H_{EM} = C_0 + C_1 \cdot H_{core,EM}$$

In these equations the length, width and height of the entire electric machine are given as function of the earlier derived dimensions of the core (expression [5]). The polynomial factors A_0, A_1, B_0, B_1, C_0 and C_1 can be used to fit the dimensions of actual machines, so here regression analysis is needed to find appropriate values for the coefficients. In order to make the coefficients dimensionless, the actual dimensions and the core dimensions could also be normalized using a typical machine as a reference. This benchmark machine must preferably be somewhere in the middle of the design space. Note that it is possible to fit dimensions of actual machines using a database containing dimensions of real machines, but it is also possible to estimate machine dimensions with this model if a database is unavailable or outdated. The only information required to do so is a reasonable estimation of the coefficients A_1, B_1 and C_1 and neglecting A_0, B_0 and C_0 (as will be done shortly). One could say that this approach to dimension prediction models enables "rubber" machines that are dimensioned according to a designers insight instead of manufacturer data.

The success of fitting machine dimensions using the coefficients A, B and C is determined by the degree with which the dimensions of actual machine differ from the core dimensions, or rather the variance of this degree. In the case of electric machines for instance the already discussed different cooling methods cold pose quite a "disturbance" on the values of the coefficients A, B and C, if the cooling is to be included in these coefficients as well. If so, one can expect a step change in the value for C for instance when electric machines switch from open cooling to closed cooling by the "sudden" addition of a large heat exchanger on top of the machine. Such considerations could lead to more advanced mathematical relations to fit the dimensions, but it could also be accepted as a remaining weakness of the method and a reason for anyone that uses this method to think critically on what he/she is fitting. This is true anyhow, no matter in which manner regression analysis is applied to fit data.

Figure 7, which is another manufacturer drawing, suggests that a similar mathematical relation between machine dimensions and core dimensions can be assumed for a gearbox as for the electric machine:

$$L_{GB} = A_0 + A_1 \cdot L_{core,GB}$$
$$W_{GB} = B_0 + B_1 \cdot W_{core,GB} \quad [10]$$
$$H_{GB} = C_0 + C_1 \cdot H_{core,GB}$$

Especially from the front view (right hand side of **Figure 7**) it can be seen that the dimensions of the machine should size with the core dimensions: the bottom circle in the middle of the gearbox represents the flange of the output shaft which is attached to the wheel while the relatively large circle near the top represents the aft bearing of the input shaft that contains the pinion. The smaller circle above this "pinion" circle is the flange of another output shaft: for a PTI/PTO (Power Take In / Power Take Off). Whether or not a gearbox is equipped with PTI/PTO is another example of a disturbance that can cause variance in the coefficients A, B and C. Another example of such a disturbance for gearboxes is whether or not the thrust block is integrated in the gearbox. Note that angle α (refer to **Figure 4**) is in the example is 90°, which will often be the case for single gear units as especially width of a gearbox needs to be as small as possible to be able to place the gearbox low and far to the back in a ship; where the ship hull will be narrow.

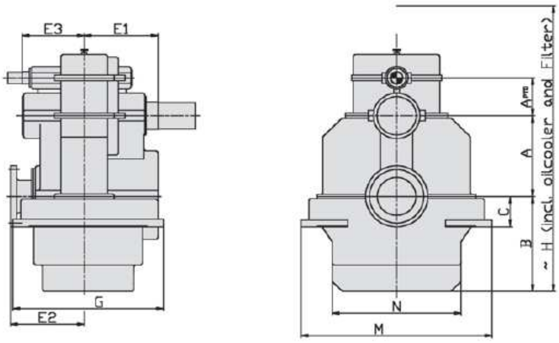

Figure 7: Typical gearbox construction. Source: online RENK catalog of single gear units (see references).

For the diesel engine again the same relationships are assumed between machine dimensions and core dimensions:

$$L_{DE} = A_0 + A_1 \cdot L_{core,DE}$$
$$W_{DE} = B_0 + B_1 \cdot W_{core,DE} \quad [11]$$
$$H_{DE} = C_0 + C_1 \cdot H_{core,DE}$$

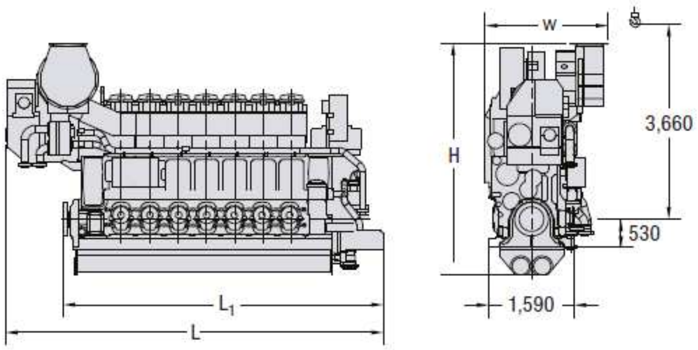

Figure 8: Typical diesel engine construction. Source: online MAN catalog of marine diesel engines (see references).

From **Figure 8** it can be seen that also for the diesel engine the core dimensions indeed represent a significant part of the machine dimensions. In this case a trunk piston type L-engine (ct = 0, α = 0; refer to **Figure 5**) is shown and length should indeed size with $i \cdot D_B$, width of the machine is approximately L_S (at least at the bottom) and height should indeed be a multiple of $2 \cdot L_S$, refer to expression [8]. On the other hand the turbocharger with its related air and exhaust gas channels (inlet receiver, charge air cooler, outlet receiver, etc.) represent quite a disturbance factor. The same applies for cooling water and lubrication oil provisions attached to the engine. Then again, such provisions are always there (even the turbocharger, which formally is optional, is a standard piece of equipment on marine diesel engines nowadays), so perhaps the variance of coefficients A, B and C is not so large.

MODEL CORRELATION

After having introduced the assumed relations between core dimensions and machine dimensions in the previous section the real question now is: what are typical values for coefficients A, B and C for the three machines and what is their variance. An equally important question is the range of the main parameters, i.e. the characteristic mean shear stress or mean effective pressure, mean circumferential or piston speed and finally the L/D ratio and whether these three “players in the game” can be easily selected.

For diesel engines the manufacturers provide a lot of data; therefore we will discuss the machines in reversed order and start with the diesel engine. First of all **Figure 9** shows the “players in the game” and in particular the two parameters that, when multiplied, make up the “technology parameter”. They are in a confined space with mean effective pressure between 20 and 30 bar and mean piston speed between 8 and 13 m/s. The “shape factor” shows two distinct groups, the low speed 2-stroke diesel engines having much larger L/D values then the medium and high-speed 4-stroke engines. The relative long stroke of the slow speed engines adapts their rotational speed to the propeller such that no gearbox is required. Also the long L/D together with a low mean piston speed is beneficial for the uniflow scavenging process that is critical for these 2-stroke engines.

Figure 10, **Figure 11** and **Figure 12** present at the left side the correlation between the actual length, width and height of the engines and the theoretical values as obtained from inspection of the core model.

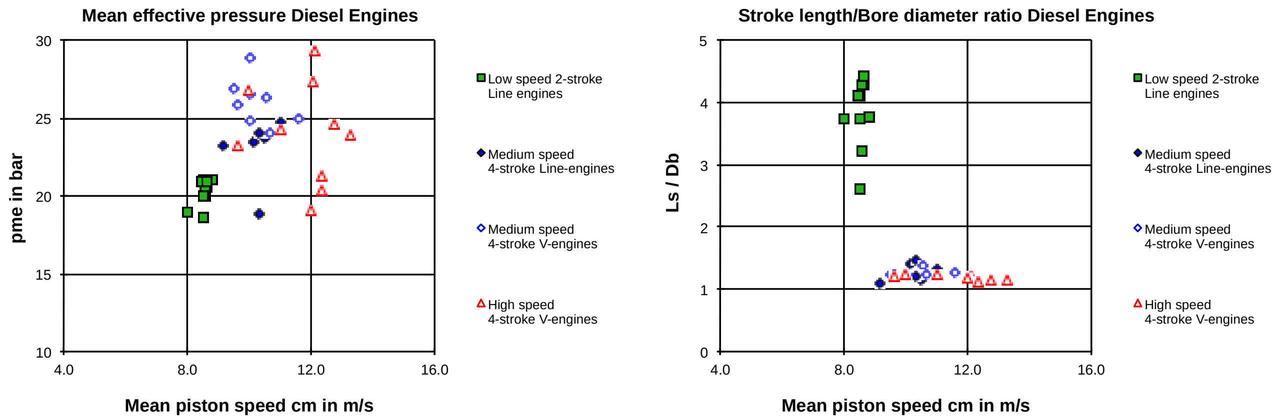

Figure 9: Range of mean effective pressure, mean piston speed and L/D ratio for diesel engines

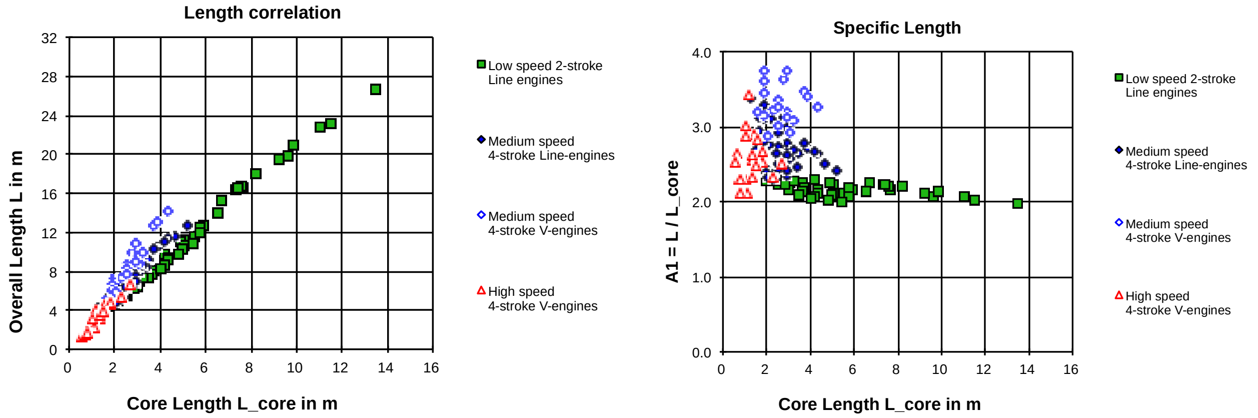

Figure 10: Correlation of actual length (left) and specific length (right) with theoretical core length for DE

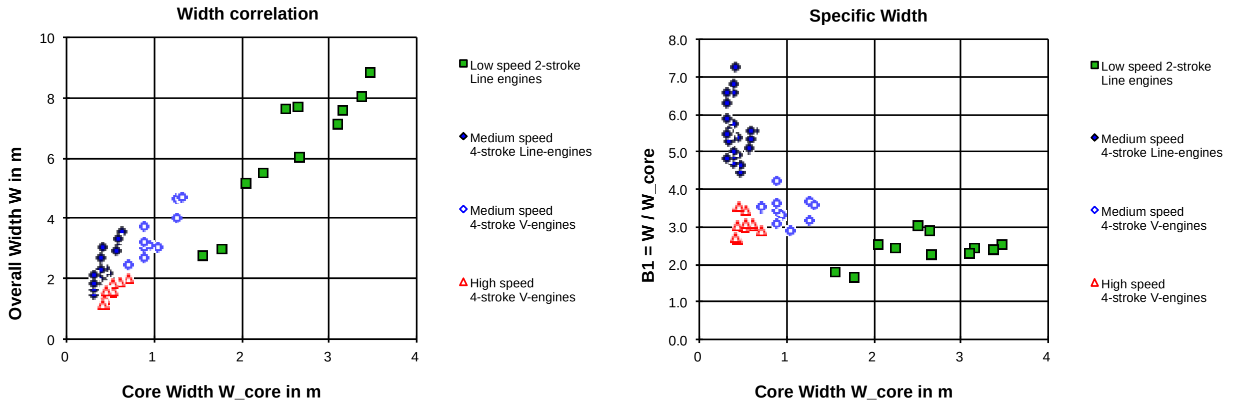

Figure 11: Correlation of actual width (left) and specific width (right) with theoretical core width for DE

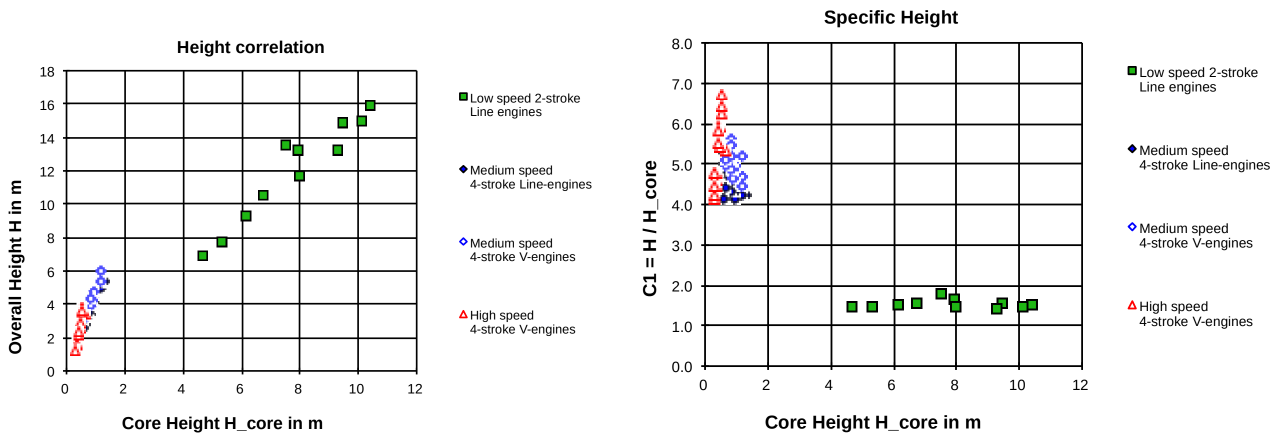

Figure 12: Correlation of actual height (left) and specific height (right) with theoretical core height for DE

All correlations show that the constants A_0, B_0 and C_0 can safely be discarded leaving the inclinations A_1, B_1 and C_1 to be fitted. These then effectively can be regarded as specific length, width and height that are shown at the right side of **Figure 10**, **Figure 11** and **Figure 12**.
Length correlates well, in particular for slow speed 2-stroke engines where specific length has a value of just over two times the core length. Scatter for medium and high speed engines is worse and also they seem systematically longer in terms of their specific length, in particular the medium speed V-engines. This of course is caused by the turbochargers being often mounted at the end of the engine.
Width correlates well within product groups of engines but not well across them. Low speed 2-stroke engines are between 2 and 3 times the core width. V-engines, both medium and high speed, are 3 to 4 times the core width and medium speed L-engines 5 to 7 times the core width. The latter seems surprising but the explanation must be that inlet and outlet receivers, camshaft and fuel pumps, that for these engines are attached to the sides, make them relatively wide.
Height correlates well for slow speed engines and reasonably for medium speed engines but less so for high speed engines. The reason could be that for these engines the turbocharger(s) are sometimes mounted on top, spoiling the picture. Somewhat surprising is the fact that slow speed crosshead engines are with a specific height of around 1.5, *relatively speaking* lower than medium speed engines with their specific height between 4 and 6 and certainly than high speed engines with a value between 4 and 7. Of course the extra height of the sliding bearing is in principle allowed for in the core height of the low speed crosshead engines, which is far larger than for the other engines as can also be concluded from **Figure 12**.

For gearboxes the data up to now are scarce: we have some data for eight Single Input/Single Output (SISO) gearboxes, connecting a medium speed diesel engine to a propeller and for one 2-stage locked train high speed gearbox for connection of a gas turbine to a propeller. In fact we only have number of teeth, input power and input speed, but no actual dimensions of the teeth. Therefore **Figure 13** not only gives the spread between the 9 gearboxes in the database but also a scatter in the way the mean tooth stress, circumferential speed and L/D ratio have been reconstructed.
Nevertheless by inspection of **Figure 13** spread of the range of value of the "players in the game" seems for gearboxes much wider then for diesel engines. The circumferential speed at least ranges between 10 and 80 m/s, but in super high-speed gearboxes (which are not included in the data set) they may nowadays be as high as 220 m/s. For the mean tooth shear stress as defined in this paper the knowledge really must be built up. A value of around 60 to 80 kN/mm^2 for medium speed gearboxes seems to be the case and a somewhat higher value for high-speed gearboxes for maritime gas turbines does not seem unreasonable.
For only 3 out of the 8 medium speed gearboxes dimensional drawings are available and for the high speed locked train gearbox there is an indication of the size. So the size correlations in **Figure 14**, **Figure 15** and **Figure 16** must be regarded with some reservation. Also these figures not only show the scatter between the (only 4) gearboxes but also the scatter as a result of the uncertainty of the reverse engineering process by which the points are reconstructed. For the high speed locked train gearbox the core model was expanded, but the details will not be explained in the paper.
Nevertheless the fact that in **Figure 14** the specific length of the medium speed single stage gearbox (with a value between 6 to 10 times the core length) is relatively longer than the 2-stage gearbox (with a value around 4) seems reasonable in view of the fact that the distance between the two gear trains in the high-speed gearbox "costs" relatively not much length.
For the width and height correlations in **Figure 15** and **Figure 16** the value for the specific width and height for the medium speed single stage and high speed 2-stage gearbox seems of equal magnitude between 1.5 and 2, so here, contrary to diesel engines the conceptual model work well across product groups. This is not surprising since width and height are dominated by the size of the big wheel.

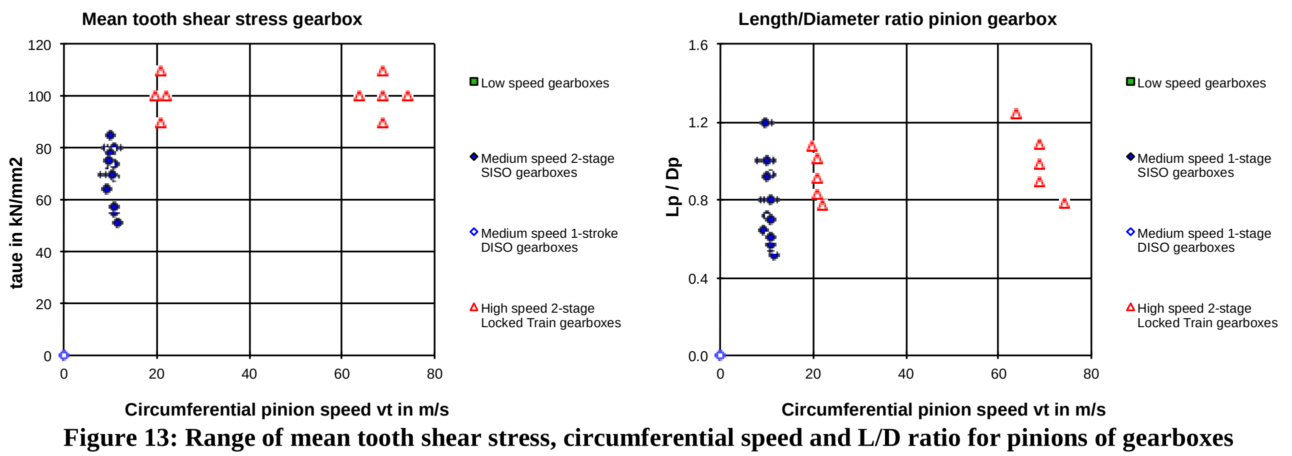

Figure 13: Range of mean tooth shear stress, circumferential speed and L/D ratio for pinions of gearboxes

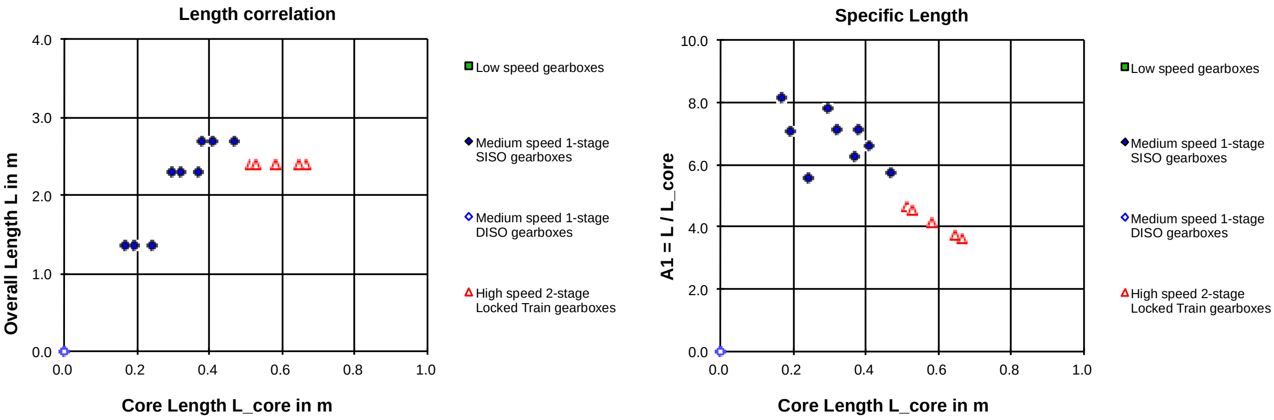

Figure 14: Correlation of actual length (left) and specific length (right) with theoretical core length for GB

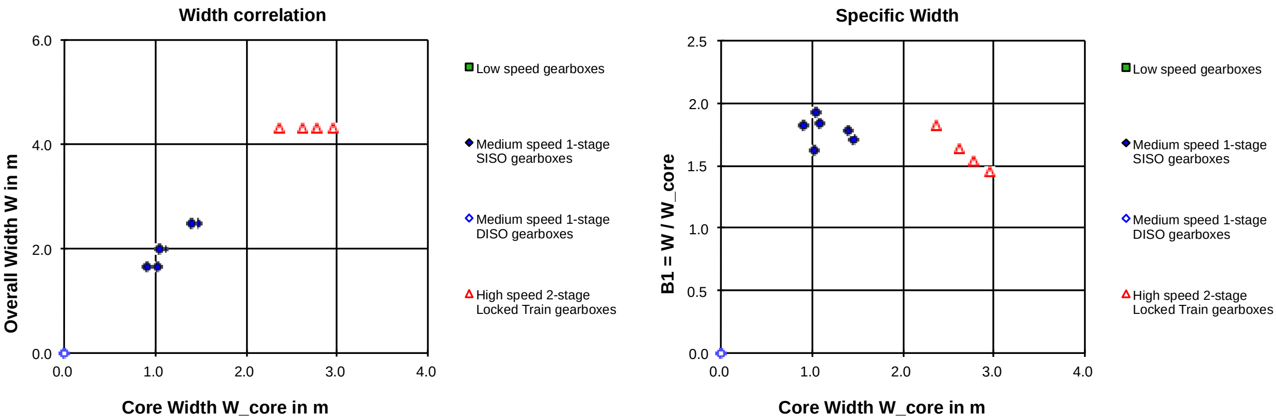

Figure 15: Correlation of actual width (left) and specific width (right) with theoretical core width for GB

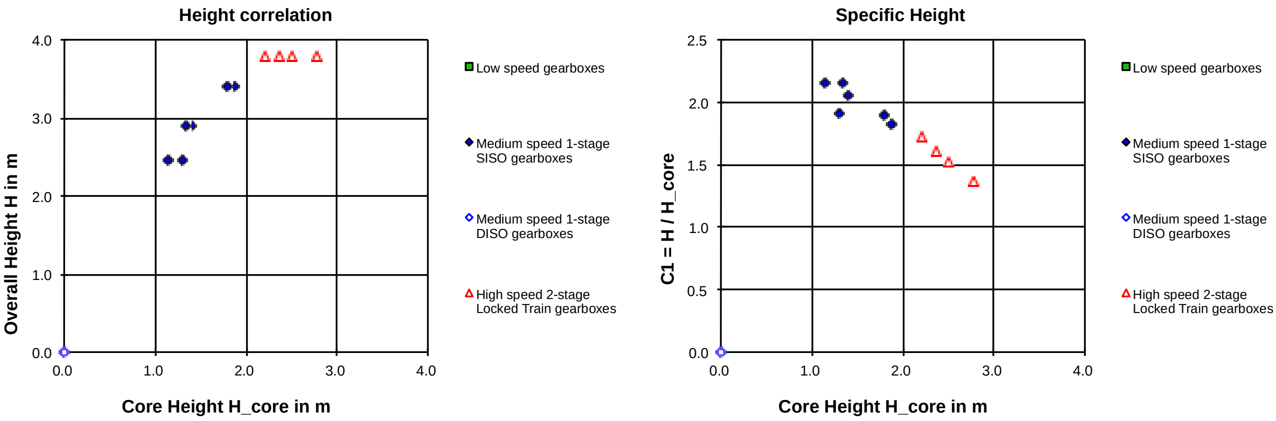

Figure 16: Correlation of actual height (left) and specific height (right) with theoretical core height for GB

Finally for the electric machines it must be emphasized that there is also uncertainty in the data gathered (though not so much as for the gearbox). Especially rotor dimensions have been hard to find and some uncertainty exists in the values for rotor/stator diameter ratio s. Still by careful inspection of machine drawings and application of typical values for s a reasonably well filled database could be constructed that provides some confidence in the figures below.
Figure 17 shows again the spread of the range of value of the “players in the game”, which for electric machines (like for gearboxes) seems to be much wider then for diesel engines. The circumferential speed ranges between 7 and 80 m/s according to the database, but (Rucker e.a. 2005) mentions a value as high as 200 m/s for a 13000 rpm PM generator for naval applications. They also mention that such a high circumferential speed limits the attainable mean shear stress, for which they assume a value of 15 kN/mm^2. From **Figure 17** it can be seen that higher values may be found, in fact the mean shear stress ranges from 15 to 80 kN/mm^2 in the database. The limits of mean shear stress (also referred to as force density) have been investigated by (Grauers e.a. 2004) and they conclude a maximum of 100 kN/mm^2 exists, but slightly higher values have already been found for special cases. Either way, both literature and the gathered values in the database suggest that the product of mean shear stress and circumferential speed, which was introduced as the “Technology Parameter”, is limited. This results in a decreasing line serving as a limit for mean shear stress and circumferential speed, which indeed can be observed in **Figure 17**. The L/D ratio for rotors was estimated on basis of machine drawings and show little spread, especially within product groups, which can be explained by the fact that product groups use similar housings. Machines with a low number of poles (2 or 4) might have somewhat higher L/D ratios for the rotor since they need more space for the salient poles of the stator. This does not change the dimensions of the housing, since the rotor/stator diameter ratio s then also changes (in the opposite direction).
Machine dimensions and core dimensions correlate well especially within each product group, like for the diesel engines, as can be seen from **Figure 18**, **Figure 19** and **Figure 20**. An exception (positively) to this rule is the length that also correlates well across all product groups. The reason for width and height only correlating well within product groups can be found in the fact that terminal boxes (TB) and heat exchangers (HE) have been taken into account in machine dimensions as well and they affect only width and/or height (depending on their location). The 3GBM product group does not include a terminal box or heat exchanger; the HXR group includes only a terminal box, while the AMI group includes both. A small experiment was done by not taking into account the TB and HE for the HXR and AMI group, from which could be concluded that all dimensions correlated well; also across product groups. A designer however is interested in the overall dimensions.
From the left hand side figures in **Figure 18**, **Figure 19** and **Figure 20** it can also be observed that, contrary to the diesel engines and the gearboxes, the coefficients A_0, B_0 and C_0 have a non-zero value (although a value of zero could be assumed of course; but this leads to a larger spread in coefficients A_1, B_1 and C_1). Coefficient A_0 has been estimated as 0.7, B_0 as 0.4 and C_0 as 0.5 by extending a linear line running through the data points. Values for A_1 (right hand side of **Figure 18**) can then be found to be in a narrow range of 1.15–1.45, if one discards the DC motor: this is a relatively fat machine (small L/D as can also be seen in **Figure 17**) and also has a very low speed (200 rpm), so it has a large number of poles. Ultimately the range of values for specific width B_1 and specific height C_1 is larger (0.6–2.1 resp. 0.4–2.4), which is again a result of inclusion of TB and HE in machine dimensions.

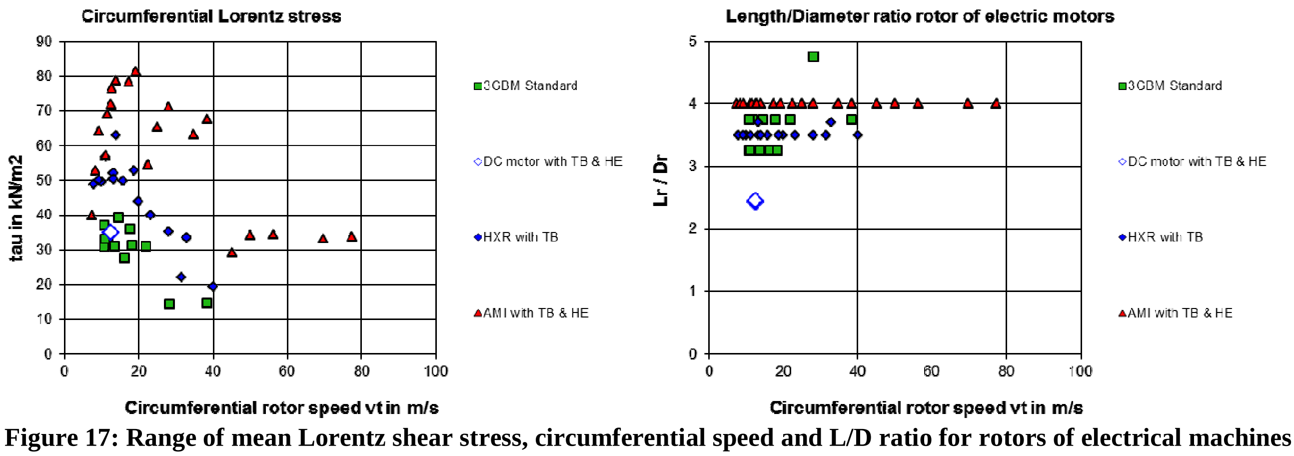

Figure 17: Range of mean Lorentz shear stress, circumferential speed and L/D ratio for rotors of electrical machines

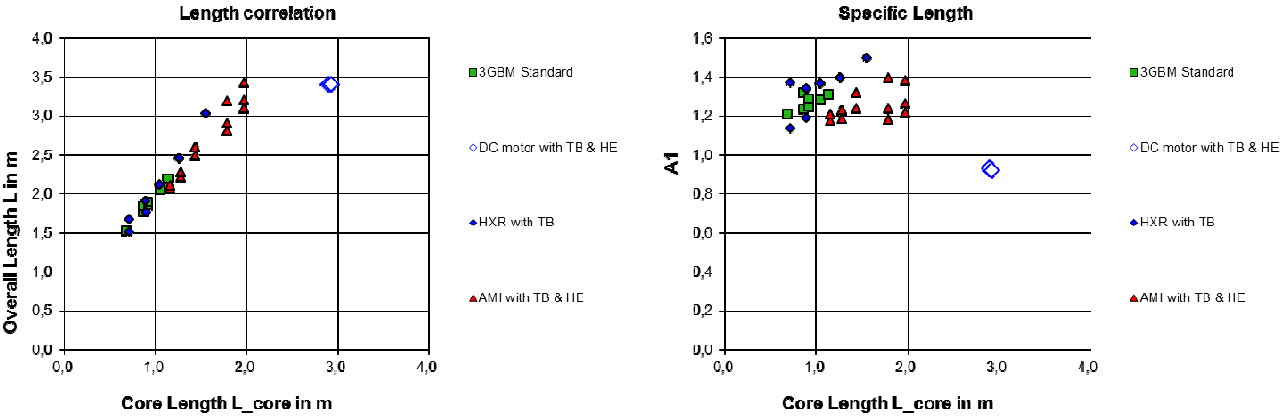

Figure 18: Correlation of actual length (left) and specific length (right) with theoretical core length for EM

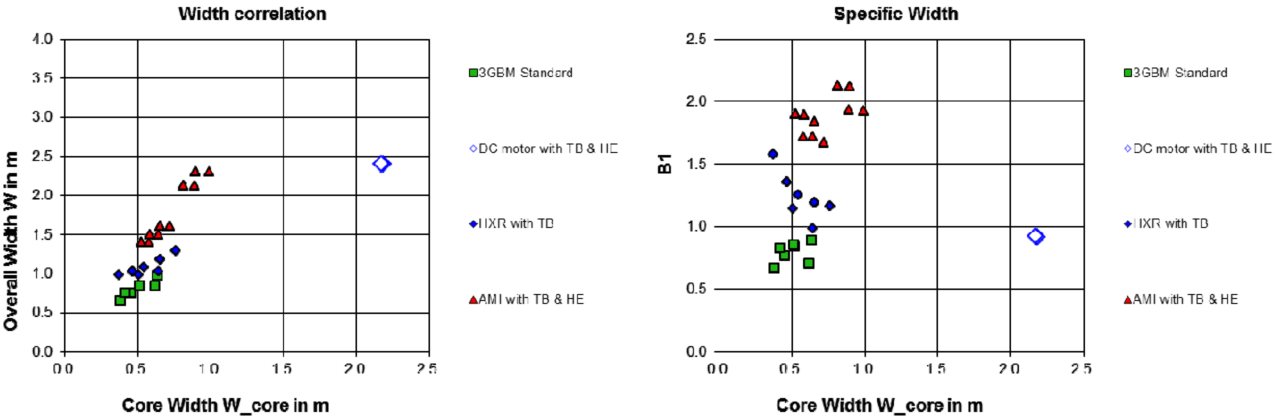

Figure 19: Correlation of actual width (left) and specific width (right) with theoretical core width for EM

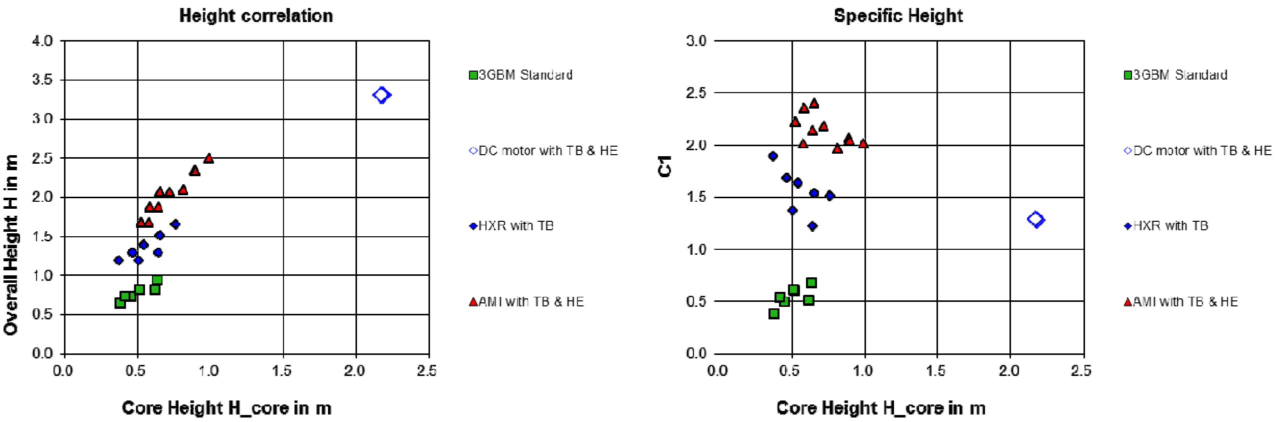

Figure 20: Correlation of actual height (left) and specific height (right) with theoretical core height for EM

CONCLUSIONS

In this paper, we have presented a generic method for predicting dimensions of primary ship system components. The method has successfully been applied to diesel engines, gearboxes and electric machines and is therefore thought to be generally applicable. Similar methodologies can be found in the domains of the respective machines, but as far as the authors know no one has ever tried to use them as generic as was done in this paper.

There are two main benefits of the proposed method over direct fitting of machine dimensions data as a function of power and/or speed as is often done. First it is possible to explore the influence of (future) technology on dimensions of equipment. Second the cause of the scatter in the regression analysis can be understood better and all models can, apart from mean values for the fit constants also be provided with a standard deviation for them. That way it is possible to introduce uncertainty analysis in the design process.

In order to apply the method to realistic design questions it is necessary not only to know the constants in the regression but also the limits in the characteristic stress and speed diagrams introduced in this paper in conjunction with any limits on the shape factor L/D. In particular for gearboxes and electrical machines the author's database at the moment is too scarce to provide that insight.

Future work therefore will focus on the one hand on gathering more information on diesel engines, gearboxes and electric machines in order to expand the database, thereby increasing the knowledge about the limits of the three main machine parameters and improving the fidelity of the first principle based dimension prediction models. On the other hand the methodology will be applied to more ship system components (pumps, gas turbines, heat exchangers, batteries, etc.) to assess whether it is indeed even more generally applicable as it seems to be now.

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