Assessment of Aframax Tanker Hull-Girder Fatigue Strength According to New Common Structural Rules

The paper describes the fatigue strength assessment of ship hull girder according to Common Structural Rules for Oil Tankers (CSR). Additional criteria for hull girder fatigue calculation have recently been introduced into CSR because of frequent crack appearances on the main deck structure of large tankers. Hull girder fatigue check in CSR is performed in two steps: preliminary "fatigue section modulus" verification and detail fatigue calculation of deck longitudinals. The analysis is performed for an Aframax oil tanker fully complying with "old" rules of classification societies. Since the results of fatigue calculation for initial structure have not been found acceptable, a significantly increased hull section modulus is necessary as the only practical way for the deck longitudinal fatigue life improvement. In practice, the vertical wave bending moment at midship, as the primary cause of hull girder fatigue damage, is calculated according to a simplified CSR formula. In order to improve the knowledge of its influence on the calculated fatigue life, the wave bending moment is also determined directly by a hydrodynamic and statistical analysis. In that analysis, the North Atlantic navigation is assumed as design wave environment for three predominant loading conditions. It is obvious that such an approach enables a more detailed and rational fatigue analysis than the one carried out according to the CSR rules.

Keywords: Aframax tankers, bending moment, fatigue strength, hull girders, vertical waves

1 Introduction

The Common Structural Rules (CSR) for Double-Hull Oil Tankers have been developed by a group of IACS classification societies in response to a consistent and persistent call from industry for an increased standard of structural safety of oil tankers. The recently published statistics indicate a significant number of defects, especially fractures, occurring in tankers less than 10 years old. It is the intent of CSR rules to reduce the possibility of so many defects [1],[2]. New CSR rules implement advanced structural and hydrodynamic computational methods to establish new criteria applied in a consistent manner, which will result not only in a more robust, safer ship, but will also eliminate the possibility of using scantlings and steel weight as...
a competitive element when selecting a class society to approve a new design.

Possibly, the most important new CSR rule requirement is the one for ultimate vertical bending moment capacity of hull-girder, which was not prescribed in previous versions of ship classification rules (with the exception of the Rules of Bureau Veritas that adopted the ultimate strength criterion in the year 2000 [3]). A “net” thickness approach is also an important new feature of CSR, where the structural capacity for different failure modes is to be calculated by assuming that the thickness of structural elements is reduced because of corrosion effects. CSR proposes a corrosion deduction thickness for different structural elements and different levels of calculation. Design scantlings of structural elements are then obtained by adding this corrosion deduction thickness to the minimum calculated “net” thickness.

Fatigue and corrosion are recognized as predominant factors which contribute to the structural failure observed on a ship in service. Fatigue may be defined as a process of cycle by cycle accumulating of damage in a structure subjected to fluctuating stresses. Until recently, the fatigue was considered as a service-ability problem rather than a hull girder strength problem [4], [5]. However, the latest researches conducted for the development of the new CSR showed that the majority of cracks are caused not only by local dynamic loads but also by global dynamic loads such as the wave bending moment. In other words, fatigue of the hull girder may be a governing strength criterion for oil tankers, in particular if higher tensile steel is implemented [6].

The aim of the present paper is to present the hull-girder fatigue analysis of an existing Aframax oil tanker according to new CSR.

A brief description of the Aframax tanker used in the present study is given in the first section of the paper. The following section describes the methodology proposed by CSR for fatigue life calculation of deck longitudinals of a double hull oil tanker. The next section presents results of the application of previously presented methodology to the Aframax tanker, showing that the fatigue life of the deck structure is significantly below 25 years. Although the fatigue life in general depends on many factors, such as design shape of structural details, material grade, scantlings of details, etc., a decrease in fatigue stresses is found to be the only convenient way to improve the global fatigue behaviour. Therefore, the section modulus of midship section is to be increased in order to reduce fluctuating stresses and to improve the fatigue behaviour of a ship as a hull girder. Finally, in the last section of the paper, the wave bending moment, as a primary cause of fatigue in the deck structure of oil tankers, is calculated by a direct hydrodynamic and statistical analysis using the linear strip theory and the IACS Recommendation No. 34 for extreme wave loads [7]. The obtained results are compared to those obtained by a pure “rule” approach and corresponding conclusions are drawn.

The main conclusion of the study is that a satisfactory fatigue life may be achieved only by a significant increase in the midship section modulus. Therefore, the study supports the opinion that fatigue becomes a governing criterion in ship design, requiring a lot of additional steel-weight to be added to the hull structure.

2 Ship description

The ship analyzed in the present study is an existing Aframax oil tanker with the Centre line plane bulkhead fully complying with “old” rules for the design and construction of steel ships, including IACS UR S11. The main particulars of the Aframax tanker are presented in Table 1. Deck and bottom areas of the ship are made of higher tensile steel AH32, while the region around the neutral axis is made of mild steel ST235. Since this tanker has the ICE-1C class notation, the side shell in the ice belt region is made of higher tensile steel AH36. In addition, the whole center line bulkhead is made of higher tensile steel AH32 due to shear stress requirements.

The general arrangement of the vessel is shown in Figure 1, while the midship section of the vessel is presented in Figure 2.

<table>
<thead>
<tr>
<th>Table 1 Main characteristics of the Aframax tanker</th>
<th>Tablica 1 Osnovne značajke aframax tankera</th>
</tr>
</thead>
<tbody>
<tr>
<td>Length between perpendiculars, (L_{pp})</td>
<td>236 m</td>
</tr>
<tr>
<td>Moulded breadth, (B)</td>
<td>42.0 m</td>
</tr>
<tr>
<td>Moulded depth, (D)</td>
<td>21.0 m</td>
</tr>
<tr>
<td>Scantling draught, (T)</td>
<td>15.6 m</td>
</tr>
<tr>
<td>Deadweight, (DWT)</td>
<td>114000 dwt</td>
</tr>
</tbody>
</table>

Figure 1 Aframax tanker

Figure 2 Midship section of the Aframax tanker
### 3 Fatigue in CSR

Hull girder fatigue calculations in CSR are performed in two steps: a simplified check of hull girder fatigue section modulus and a detailed fatigue life assessment of main deck longitudinals. These two calculation methods are briefly described in the following sections.

#### 3.1 Hull girder fatigue requirement

Hull girder fatigue strength is checked by a simplified fatigue control measure against dynamic hull girder stresses in the longitudinal deck structure. The required hull girder fatigue section modulus $Z_{v-fat}$ is given in CSR, Section 8.1.5:

$$Z_{v-fat} = \frac{M_{v-hog} - M_{v-sag}}{1000 \cdot R_{dl}} (m^3)$$

Where:
- $M_{v-hog}$ = hogging vertical wave bending moment for fatigue (kNm)
- $M_{v-sag}$ = sagging vertical wave bending moment for fatigue (kNm)
- $R_{dl}$ = allowable stress range (N/mm²)

The actual section modulus to be compared to the minimum required value $Z_{v-fat}$ is calculated by deducting half of the rule corrosion wastage ($0.5 \cdot t_{corr}$) from the gross thickness of all structural elements contributing to the hull girder longitudinal strength. It should be pointed out that this requirement is not mandatory, but recommended to be applied in the early design stage in order to avoid significant reinforcements in the later design stage when detailed fatigue calculations are carried out.

Hogging and sagging vertical wave bending moments for fatigue are obtained by multiplying rule wave bending moments for strength assessment by a factor of 0.5. In that way, the representative probability level of wave bending moments is reduced from $10^{-8}$ to $10^{-4}$. This aspect is described in the following sections.

#### 3.2 Detailed fatigue assessment of deck longitudinals

The calculation of hull girder stress for the detailed fatigue strength assessment of deck longitudinals is based on the fatigue hull girder sectional proprieties calculated by deducting a quarter of the corrosion addition ($-0.25 \cdot t_{corr}$) from the gross thickness of all structural elements comprising the hull girder cross section.

The capacity of welded steel joints with respect to fatigue strength is characterized by the Wöhler curves (S-N curves) of all structural elements contributing to the hull girder longitudinal strength. It should be pointed out that this requirement is not mandatory, but recommended to be applied in the early design stage in order to avoid significant reinforcements in the later design stage when detailed fatigue calculations are carried out.

Hogging and sagging vertical wave bending moments for fatigue are obtained by multiplying rule wave bending moments for strength assessment by a factor of 0.5. In that way, the representative probability level of wave bending moments is reduced from $10^{-8}$ to $10^{-4}$. This aspect is described in CSR Section 7.3.4.1.3.

When the cumulative fatigue damage ratio, $DM$, is greater than 1, the fatigue capability of the structure is not acceptable. $DM$ is determined according to CSR Appendix C 1.4.1.1:

$$DM = \sum_{i=1}^{2} DM_i$$

Where:
- $DM_i$ = cumulative fatigue damage ratio for the applicable loading condition
  - $i = 1$ for full load condition
  - $i = 2$ for normal ballast condition.

The fatigue capability of the ship’s life:
- $t_f$ = number of cycles for the expected design life. The value is generally between $0.6x10^9$ and $0.8x10^9$ cycles for a design life of 25 years.

$$N_L = \frac{f_o U}{4 \log L}$$

Where:
- $f_o = 0.85$, factor taking into account non-sailing time for operations such as loading and unloading, repairs, etc.
- $U$ = design life (s) = $0.788x10^9$ for a design life of 25 years
- $L = rule length [2]$
- $m = 3$-
- $S-N$ curves exponent as given in CSR Table C.1.6
- $K_\gamma = 0.63 \cdot 10^{12}$ - $S-N$ curves coefficient as given in CSR Table C.1.6
- $\alpha_i$ = proportion of the ship’s life:
  - $\alpha_i = 0.5$ for full load condition
  - $\alpha_i = 0.5$ for ballast condition
- $S_{Ri}$ = stress range at the representative probability level of $10^{-4}$ (N/mm²)
- $N_k$ = 10000, number of cycles corresponding to the probability level of $10^{-4}$
- $\xi$ = Weibull shape parameter
- $\Gamma$ = Gamma function
- $\mu_i$ = coefficient taking into account the change in the slope of the S-N curve

$$\mu_i = 1 - \gamma \left( 1 + \frac{m}{\xi}, v_i \right) - \frac{S_{Ri}^m}{S_{Ri}^m + m} \cdot \gamma \left( 1 + \frac{m + \Delta m}{\xi}, v_i \right)$$

$$v_i = \left( \frac{S_{Ri}}{S_{Ri}^m} \right)^{\xi} \cdot \ln N_k$$
\( S_{\alpha} \) - stress range at the intersection of two segments (“knee”) of the S-N curves. CSR Table C.1.6.

\( \Delta_{\alpha} = \) 2 - slope change of the upper-lower segment of the S-N curve

\( \gamma(a, x) \) - incomplete Gamma function, Legendre form

The Weibull shape parameter \( \xi \) is calculated as:

\[
\xi = f_{\text{Weibull}} \left( 1.1 - 0.35 \cdot \frac{L - 100}{300} \right) \quad (12)
\]

The cumulative fatigue damage ratio, \( DM \), is finally converted into a calculated fatigue life:

\[
\text{Fatigue life} = \frac{\text{Design life}}{DM} \quad (13)
\]

According to CSR requirements, the calculated fatigue life should be more than 25 years.

4 Result of the analysis

Stress range \( S_{\alpha} \), required for the calculation of accumulated damage in Eq. (8), is calculated by the simple beam theory assumptions, i.e.:

\[
S_{\alpha} = \frac{M_{\alpha}}{Z_{\text{net75}}} \quad (14)
\]

where \( Z_{\text{net75}} \) is the “net” section modulus (-0.25 \( \gamma_{\text{net}} \)) of the midship section cross section, while \( M_{\alpha} \) is the range of wave bending moment at a representative probability level of 10\(^{-4}\). \( M_{\alpha} \) is calculated as:

\[
M_{\alpha} = M_{\text{wv-hog}} - M_{\text{wv-sag}} \quad (15)
\]

where \( M_{\text{wv-hog}} \) and \( M_{\text{wv-sag}} \) are hogging and sagging vertical wave bending moments for fatigue, respectively, as given in CSR Section 7.3.4.1.3. For the Aframax tanker analysed in the present paper, the range of the vertical wave bending moment reads 3864 MNm. It should be noted that the stress range and all calculation parameters are the same for ballast and full load conditions. Consequently, the same results have been obtained for both conditions.

4.1 Initial structure

The existing “net” section modulus of the midship section calculated with the appropriate corrosion deduction, CSR Table 6.3.1-Corrosion addition, should be over the CSR minimal required fatigue section modulus. However, as may be seen from the results presented in Table 2, the actual section modulus should be increased by more than 15% to comply with the CSR minimal required value.

<table>
<thead>
<tr>
<th>Initial structure</th>
<th>&quot;Reinforced&quot; structure</th>
</tr>
</thead>
<tbody>
<tr>
<td>Actual sectional area, ( A ) (m(^2))</td>
<td>5.30</td>
</tr>
<tr>
<td>Actual section modulus, ( Z ) (m(^3))</td>
<td>31.74</td>
</tr>
<tr>
<td>Allowable fatigue stress, ( R_{\alpha} ) (N/mm(^2))</td>
<td>125.7</td>
</tr>
<tr>
<td>Required fatigue section modulus, ( Z_{\text{req75}} ) (m(^3))</td>
<td>30.75</td>
</tr>
</tbody>
</table>

Also, the detailed fatigue calculation of reinforced structure leads to a fatigue life of deck longitudinals of 25 years, which is satisfactory in accordance with the CSR (Table 5).

5 Fatigue analysis with loads from the hydrodynamic analysis

The vertical wave bending moment is the dominant dynamic loading component for the hull girder fatigue analysis. New CSR continue using simple IACS UR S11 formulae for the design...
wave bending moments in sagging and hogging. The rule vertical wave bending moments are defined as the bending moments with the exceeding probability of $10^{-8}$. In other words, the rule values are the most probable extreme values for the return period of 20 years, which is the ordinary ship lifetime. The rule design wave bending moments are based on the main dimensions of the ship: length, breadth and block coefficient. Operational profile, mass distribution and hull form are not taken into account by the rule formulae.

As an alternative to the application of IACS UR S11, a direct hydrodynamic analysis of ship motion and load may be performed to determine the long term distribution of wave bending moments for fatigue assessment. The direct analysis requires more detailed and elaborated input data and it is of interest to see its implication on the hull girder fatigue life.

Evaluation of the wave-induced load effects that occur during long-term operation of the ship in a seaway was carried out for areas in the North Atlantic in accordance with the IACS Recommendation Note No. 34. Although this recommendation is basically conceived as guidance for the computation of extreme wave loads, it seems to be appropriate for fatigue analysis as well [7]. The basic assumptions proposed by IACS for the calculation of long-term extreme values of wave bending moments are:

- The IACS North Atlantic scatter diagram should be used. This scatter diagram covers areas 8, 9, 15 and 16, as defined in Global Wave Statistics (GWS). The data from the GWS are further modified by IACS in order to take into account the limited wave steepness more properly.
- Only ship speed equal to zero is to be taken into account.
- The two-parameter Pierson-Moskowitz spectrum (ITTC spectrum) is recommended.
- Short-crested waves with the wave energy spreading function proportional to $\cos^2(\theta)$ are to be used.
- All heading angles should have equal probability of occurrence and maximally 30° spacing between headings should be applied.

The calculation of transfer functions of wave-induced load effects is performed by the program WAVE-SHIP, based on the linear strip theory [8]. The strip model of Aframax tanker is shown in Figure 3, while the transfer functions of vertical wave bending moments for the full load condition and different headings are presented in Figure 4.

The long-term analysis according to IACS procedure is performed for three loading conditions: full load (FL), ship in ballast (BL) and partial loading condition (PL). The long-term analysis is performed by the computer program POSTRESP, which is a part of the SESAM package [9]. After that, the range of wave bending moments corresponding to the probability level of $10^{-4}$ required for fatigue analysis is easily determined. Parameters of Weibull distribution, used to approximate the long-term probability distribution of vertical wave bending moment, are also computed easily.

Fatigue analysis according to CSR considers that the tanker spends 85% of the time on sea, equally in ballast and full load condition. In the direct analysis, the partial loading condition is also considered. The percentage of time that a ship spends in either of these loading conditions may be estimated based upon the statistical analysis of load duration data for tankers performed by Guedes Soares [10], as presented in Table 6.

Finally, it should be mentioned that the results from hydrodynamic analysis are reduced by 10% for the application in fatigue calculation. This reduction is a consequence of the fact that the wave bending moments determined by linear strip theory overestimate the measured wave bending moments in average by 10% [11].

Input parameters and results of the detailed fatigue analysis of deck longitudinals are presented in Table 7. Calculated fatigue life is estimated to be 16.6 years, which is lower comparing to the CSR approach. It can be seen from Table 7 that the full load condition gives the largest contribution to the total fatigue damage.
6 Conclusion

The purpose of the paper is to point out that the fatigue failure is recognised as one of the governing failure modes in newly developed CSR for Double Hull Oil Tankers. Thus, fatigue is not only important for design of ship structural details, but also may be a governing criterion for the required section modulus at midship, i.e. for ship longitudinal strength, affecting thus the overall dimensions of structure subjected to fatigue.

Fatigue analysis of the connection of the main deck longitudinals with web frames and transverse bulkheads, which are among the most important ship structural details from the fatigue point of view [12]. Since the governing fatigue loading of side shell longitudinals is the local dynamic pressure, a significantly different approach would be necessary, which is outside the scope of the present study.

References

ABSTRACT
Violent fluid motions, high speed marine vehicles and Computational Fluid Dynamics (CFD) are selected as main topics. Violent fluid motions deal with green water on deck, sloshing and slamming. Slamming involves many physical effects. When analyzing slamming, one must always have the structural reaction in mind. This necessitates that hydroelastic effects are considered. Many hydrodynamic phenomena matter for the three main categories of high-speed vessels, i.e., vessels supported by the hull, foils and air cushions. Dynamic instabilities, cavitation and ventilation are limiting factors for their performance. The coupling with automatic control is discussed. A brief overview of the many different CFD methods is given and advantages and disadvantages are discussed.

Keywords:
Green water on deck, sloshing, slamming, hydroelasticity, high-speed marine vehicles, CFD.

1. INTRODUCTION
There is a broad area of marine structures needed for ocean transport, exploitation of subsea hydrocarbons and wave energy, sea food production as well as for marine infrastructure. Challenging hydrodynamic aspects are focused on. However, hydrodynamics must be linked to other disciplines such as structural mechanics and automatic control.

There is a general tendency that specialists in marine hydrodynamics work on separate disciplines such as resistance, propulsion, manoeuvring and seakeeping without always combining the knowledge. For instance, the presence of sea waves can clearly influence the manoeuvring ability of a ship. This is of practical concern in, for example, replenishment operations between ships at sea. An important aspect is to properly account for the mean wave forces and moments in the mathematical manoeuvring model. Figure 1 shows
results for the 161m long *Mariner* in a starboard turning manoeuvres. The rudder angle is 20.9 degrees and the approach forward speed is 15.4 knots. The period, height and direction of the incident waves are 7s, 3m and 150 degrees, respectively. The figure demonstrates clearly that the presence of waves can have a large effect on the turning ability of a ship, with a noticeable wave-dependent loss in speed.

![Figure 1](image)

**Figure 1:** Turning circle manoeuvre of *Mariner* in calm water and waves, respectively (Skejic, unpublished).

A ship is often hydrodynamically optimized in calm water conditions. Because good seakeeping behaviour is an important feature of a high-speed vessel, optimization in calm water conditions may lead to unwanted behaviour in a seaway. Both wave resistance and wave radiation damping are due to the ship’s ability to generate waves. Because low wave resistance may imply low wave radiation damping in heave and pitch, the result can, for instance, be unwanted large resonant vertical motions of a catamaran with slender side-hulls and no automatic control.

Another example is the recent designs of passenger cruise vessels with very shallow local draught and nearly horizontal surfaces in the aft part of the ship. This was the result of hydrodynamic optimization studies in calm water. One does not need to be a hydrodynamicist to understand that this caused slamming (water impact) problems. Aft bodies with shallow draught should also be of concern for directional stability and for ventilation of water jet inlets in waves. Hydrodynamic optimization studies must therefore consider resistance, propulsion, manoeuvring and seakeeping. There are obviously also constraints of non-
hydrodynamic character. For instance, minimizing ship motions may lead to higher global structural loads.

Several of our examples in the main text deal with ship applications and we will in particular focus on violent fluid motions and hydrodynamics of high-speed vessels. The future increased role of computer simulations relative to experiments is also addressed.

Examples on marine structures that are left out of the detailed discussions are fish farms and fishing nets. Increased knowledge about the flow through fishing nets in current and waves with due consideration of the net deformation is important, for instance, for the design of mooring systems of fish farms. Vortex induced vibrations is a challenging problem, for instance, for the design of pipelines. The towing of flexible long seismic cables is difficult to theoretically predict under dynamic conditions due to limited knowledge about flow separation at small angles of attack.

2. VIOLENT FLUID MOTION
Examples on fluid motions that will be discussed are green water on deck, sloshing and slamming. Capsizing of damaged ship with water ingress /egress in waves has similarities both with sloshing in tanks and green water on deck. Sloshing in moon pools is another example that could be mentioned. An important issue is to design efficient damping devices, such as perforated walls in the pool. A similar resonance phenomenon occurs between two adjacent ships in waves, or for a ship alongside a terminal. Conventional engineering tools based on linear panel methods cannot accurately describe the flow.

2.1 Green water on deck
There is extensive work worldwide in applying Computational Fluid Dynamics (CFD) to green water on deck. However, most work is related to two-dimensional flow. An attempt has been made by Greco et al. (2007) to classify how the different green water phenomena occur as a function of wave parameters in head sea conditions for stationary ships with blunt bows, based on experimental and theoretical studies of a restrained two-dimensional body. A typical application is to a Floating Production Storage and Offloading (FPSO) vessel. Interaction between the incident waves and the hull plays an important role. One type of green water is the so called dam-breaking phenomenon, where a vertical wall of water is generated at the edge of the deck due to the relative vertical motion between the ship and the waves. The
subsequent motion of the water resembles the breaking of a dam, water flooding at high speed (15-20 m/s) along the deck. Its impact against deck structures and equipment can cause serious damage.

A second scenario is water hitting the deck as a plunging breaker. Actually, the plunging breaker may hit a deck house in the forward part of the ship. A third case is the "hammer fist" effect of water on deck, using an analogy to karate. A large mass of water rises above the deck and collapses heavily over a substantial area of the ship during a "hammer fist" type of water on deck. Figure 2 shows how these different phenomena occur as a function of the incident wave steepness and the $W_w/W_b$-ratio. Here $W_w$ is the maximum vertical velocity of the incident waves and $W_b$ is the maximum vertical fluid velocity at the ship's bow.

![Green water occurrence](image)

**Fig. 2:** The effect of green water on deck as a function of the incident wave steepness and $W_w/W_b$-ratio. $W_w$ is the maximum vertical incoming wave velocity and $W_b$ is the maximum relative fluid velocity at the bow.

Wave-body interaction plays a similar decisive role in bow stem slamming as in the green-water situation.

### 2.2 Sloshing

Sloshing in a partially filled container is a violent *resonant* free surface flow with strongly nonlinear behaviour. Sloshing must be considered for almost any moving vehicle or structure containing a fluid with a free surface.

A partially filled ship tank will experience violent fluid motion when the ship motions contain energy in the vicinity of the highest natural period for the liquid motion inside the
tank. Impact between the liquid and the tank roof is then likely to occur for larger filling ratios. The consequence is wave breaking, spray and mixing of gas and liquid. Actually, extreme cases with gas bubbles everywhere in the liquid have been experimentally observed.

Because sloshing is a typical resonance phenomenon, it is not necessarily the most extreme ship motions or external wave loads that cause the most severe sloshing. This implies that external wave induced loads can in many practical cases be described by linear theory. However, nonlinearities must be accounted for in the tank liquid motions. Because it is the highest sloshing period (natural period) that is of prime interest, vertical tank excitation is of secondary importance. Lateral and angular tank motions cause the largest liquid response in the frequency range of interest. An increased tank size increases the highest natural period of the liquid flow. As a consequence, higher sea states and larger ship motions excite sloshing around resonance. The less internal structures obstructing the flow in the tank are present, the more severe sloshing is.

There is a variety of ship tank shapes. This includes rectangular, prismatic, tapered and spherical tanks as well as horizontal cylindrical tanks. The fluid may be oil, liquefied gas, water or heavy density cargoes like molasses and caustic soda. Ideally one should be able to predict two phase flow due to strong mixing of gas with the liquid. However, it is hard enough to predict one phase flow.

Sloshing has always been an important design criterion for oil tankers, even though partial filling is rare in actual operation. Environmental concerns have led to requirements about double hull tankers. Ship owners try to avoid internal structures in cargo tanks for cleaning reasons. The resulting wide and smooth oil tanks increase the probability of severe sloshing. Sloshing is also of concern for FPSO units and shuttle tankers. The severity of sloshing is connected to possible filling height restrictions for oil tankers, gas carriers, shuttle tankers and FPSO units. Often, operators require no restrictions on filling heights to achieve loading flexibility. Because ballast exchange is required outside the port for a bulk carrier, there are possibilities for sloshing damages. Particularly, the hatch cover is vulnerable. Partial fillings in LNG carriers are a consequence of gas boil-off during operation. Sloshing in tanks has received increased attention due to the design of new types of prismatic LNG tanks. The interior tank surface is relatively smooth.

The hydrodynamic loading inside a tank can be classified either as impact loads or ‘dynamic’, non-impulsive, loads. In this context dynamic loads mean loads that have dominant time variations on the time scale of the sloshing period, while impact loads may last only from \(10^{-3}\) to \(10^{-2}\) seconds. It is the slamming pressures and resulting stresses in the
membrane structure that are of the main concern in the design of prismatic LNG tanks. Sloshing loads are of significance for both fatigue and ultimate strength. Local structural response due to liquid impact is an important response variable. Loads on possible internal structures must be considered. Some internal structures, as a horizontal stringer on the wall or web-frame at the tank roof, may be in and out of the liquid so that impact loads as well as dynamic loads may matter. Hydroelastic effects are sometimes of importance for impact loads. Total dynamic loads on the tank are of interest in order to estimate tank support reactions as for instance for a spherical LNG tank and possible global interaction with ship dynamics.

The ship motions excite sloshing, which in turn affects the ship motions. Ships equipped with anti-rolling tanks utilize this effect. The sloshing induced roll moment on the vessel will cause roll damping by properly choosing the highest natural sloshing period close to the roll natural period. FPSO units sometimes have several partially filled tanks during operation. The wave induced motions and loads on these ships will then be influenced by the dynamic motion of the liquid in the tanks. Because ship motions can strongly affect the mean and slowly-varying wave drift forces and moments, sloshing may also matter in a station-keeping analysis.

The tank shape, the level of filling and the characteristics of the tank motion, for example amplitude and frequency content, make up the principal parameters that determine the nature of the free surface flow. The relative importance of the different parameters depends on the characteristics of the flow, i.e. the response. There is a clear difference between sloshing in a shallow liquid condition and higher filling level conditions. For small ratios between liquid depth and tank length and an excitation frequency around resonance, a hydraulic jump or bore, which travels back and forth in the tank, is formed. When the steep front of the bore hits the tank wall, an impact occurs and a thin vertical jet shoots upwards. When the liquid depth is non-shallow and the liquid motion is two-dimensional, the free surface motion resembles a standing wave. Swirling or rotational flow is a special feature of three-dimensional flow, for instance, in a spherical tank, a vertical circular tank or a square-based tank.

When the interior tank surface is smooth and there are no internal structures as stringers obstructing the flow, the viscous damping of the resonant fluid motions is small as long as wave breaking does not occur. The damping for a smooth tank can be very small at finite depth, i.e. it takes a very long time for transience to die. The damping increases in general with decreasing liquid depth. Large amplification of the liquid motion occurs in a
resonant condition. This leads to important nonlinear liquid behaviour inside the tank associated with nonlinear transfer of energy between different modes of liquid motion. Secondary resonances may matter. This means that higher harmonic oscillations caused by nonlinearities excite resonance periods lower than the primary resonance period(s). Secondary resonance increases in importance with decreasing liquid depth and increasing excitation amplitude.

It is popular to use Computational Fluid Dynamics (CFD) to model sloshing. We will address this approach in a more general way later in a separate section. However, the analytically based multimodal modal time domain method presented by for instance Faltinsen & Timokha (2001) and Faltinsen et al. (2005) is more suitable to understand the many different flow configurations that can occur during sloshing. The assumptions are potential flow, no overturning waves and infinite tank roof height. The tank wall surface must be vertical in the free surface zone. The method is extremely fast from a computational point of view. The method has been derived for two-dimensional flow in rectangular tanks and for three-dimensional flow in prismatic tanks with a rectangular base. The effect of small chamfer at the tank bottom has been studied in a two-dimensional case. The method has been extensively validated by experimental results for wave elevation, lateral force and roll moment. The 2D studies have examined how the flow changes physical character by going from finite through intermediate to shallow liquid depth. The excitation amplitude is also an important parameter. The physical dissipation increases with decreasing depth.

The steady-state solutions may have several branches. Some of the branches may be unstable. We cannot have steady-state solutions along an unstable branch. Jumps can happen between different branches of solutions.

The multimodal method has been applied to three-dimensional flow with a square-based tank. There are three types of possible dominant steady-state wave response for longitudinal harmonic excitation, namely, ‘planar’ (two-dimensional), ‘swirling’ (rotary motions) and so-called ‘squares’-like three-dimensional steady-state waves formed by a combination of diagonal wave patterns. Even with longitudinal excitation along a tank wall, 3D waves will occur due to non-linear interaction in a nearly square-based tank.

By adopting a stability analysis scheme, one can calculate effective frequency domains for the different wave behaviour and find critical depths where either the frequency domains of stable regimes or their wave response may change dramatically. Frequency domains with no steady-state solutions, i.e. “chaos” exist. This has been experimentally confirmed and gives important guidance for CFD calculations.
Both prismatic and spherical tanks are commonly used on LNG carriers. The most important load on a spherical tank is the sloshing-induced hydrodynamic force, the predominant component of which is the lateral force. Swirling occurs readily so there is in general a force component perpendicular to the forced oscillation direction.

2.3 Slamming

Slamming is of concern in many marine applications. Slamming on ships is often categorized as bottom, bow-flare, bow-stem and wetdeck slamming. Wetdeck slamming has similarities with other nearly horizontal parts of a ship such as large overhanging sterns. Green-water impact on deck structures and bow-stem slamming are of concern for FPSO units. Slamming is important in the design of a ship tank. There are similarities between slamming on ships and offshore platforms. Breaking waves can impact on a ship hull or the columns of a platform. Run-up along the columns can cause local damage of the platform deck. A platform is normally designed with an air gap to avoid global water impact. However, slamming may happen due to unanticipated large waves (Figure 3) or due to the subsidence of the sea floor for bottom-mounted platforms. Bottom slamming should be considered on shallow-draft barge-type Very Large Floating Structures (VLFS) proposed as floating airports in the coastal zone. Examples on more special types of water impacts are slamming on air bags of Surface Effect Ships (SES), drop of mines, accidental drops of pipes from platforms and analysis of free-fall life boats. The deceleration, slamming loads and hydrodynamic loads on the top cover of the life boat may matter. This issue has lately got increased attention for life-boats on platforms in the North Sea. A too low maximum operating sea state can have serious economic consequences due to stop in oil and gas production.

![Figure 3: Wetdeck slamming on a semi-submersible platform and artist’s impression of bow slamming causing global elastic vibrations (whipping) of the ship hull.](image)

Slamming in a ship tank is associated with violent liquid motion and many possible impact situations have to be considered. For instance, large filling ratios can cause important
slamming loads. Examples are high curvature free surface impact and impact with an oscillating gas cavity. Another possible scenario is a sudden flip-through of the free surface at the tank wall. A liquid wedge with a high velocity will as a consequence impact on the tank roof. A chamfered tank roof is likely to reduce the severity of slamming.

Steep waves impacting on a vertical tank wall represents an important consideration for shallow and intermediate liquid depths of a tank. An example is hydraulic jumps that can be formed at resonant conditions for shallow liquid conditions.

It is common to analyze slamming in a tank by assuming a two-dimensional flow. However, swirling may cause important impact against the tank roof corners for non-small liquid depths in a nearly square-based tank.

Both local and global slamming effects must be considered. This is illustrated in the case of bow slamming on a ship in Figure 3 where the resulting global elastic vibrations (whipping) of the ship are exaggerated. The local slamming analysis is typically done by first calculating the ship motions without impact and then consider the impact with given conditions for the impacting hull. However, this procedure does not account for the mutual interaction between slamming and the global ship behaviour and needs to be improved in the future.

Many physical effects may have to be considered such as gas cushions, liquid compressibility and hydroelasticity. When analyzing slamming, one must always have the structural reaction in mind. An important consideration is the time scale of a particular hydrodynamic effect relative to wet natural periods for structural modes contributing significantly to large structural stresses. If the time scale of a hydrodynamic effect is very small relative to important structural natural periods, the hydrodynamic effect can be neglected. When the hydrodynamic loads occur on a time scale of important structural periods, hydroelasticity must be considered. This implies that the fluid (liquid, gas) flow must be solved simultaneously with the dynamic elastic structural reaction.

Local external slamming effects on ship structures of steel and aluminum are first discussed. Drop tests with horizontal plates with correctly scaled elastic properties of steel and aluminum on calm water as well as varying wave conditions and impact conditions have shown recorded maximum pressures with very large variations for a given drop speed. A maximum pressure of about 80 bar was recorded with a drop speed of 6 m/s. However, the recorded strains were not sensitive at all and showed a time variations dominated by the lowest beam mode. The maximum strain occurred approximately one quarter of the highest natural period after the impact. The effect of the impact was to cause a force impulse to the
plate. The very high slamming pressures are unimportant, i.e. they cannot be used as a static loading and there is no correlation between pressures and strains. Numerical simulations of water entry of wedge-formed ship cross-sections with stiffened plating have shown the importance of the ratio between the loading (wetting) time and the highest natural period for the structure. If these results are translated into realistic ship structural dimensions and relative vertical motions between the ship and the waves, a rough guidance is that local hydroelasticity should be considered when the angle between the impacting free surface and the hull surface is less than 5 degrees.

We then consider global slamming effects and choose wet deck slamming on a catamaran in head sea as an example. Because of the very different time scales of local and global structural vibrations, we can consider the local structure of the wetdeck as rigid in the global analysis. Heave, pitch and two-node longitudinal bending are important modes of the ship. Because the duration of the impact and the water exit (decrease of wetted surface) is smaller than one quarter of the natural period of heave and pitch, the force impulse becomes important. This enables us to simplify the analysis. Both water entry and exit must be considered. Nonlinear hydrostatic and Froude-Kriloff loads matter. Studies have shown that the Wagner method is unnecessary. Further, only the added mass force is needed during water exit.

Slamming in ship tanks will now be focused on. The discussion has also relevance for slamming in other applications. It is common in tank design to do model tests for sloshing induced slamming effects by means of forced oscillation tests. However, the scaling of the model test results represents a challenge due to the many physical effects that may matter.

Because sloshing is associated with gravity waves, we must require the Froude number is the same in model and full scale. Further, the wave induced ship motions that excite sloshing, is also Froude scaled. If harmonically forced oscillation of the tank with frequency $\sigma$ is considered, Froude scaling implies that $\sigma\sqrt{L/g}$ must be the same in model and full scale. Here $L$ is a characteristic tank dimension such as the tank breadth. A conventional model test approach does only consider the effect of Froude scaling. However, other scaling parameters of possible importance are summarized below.

If hydroelasticity matters during impact, we must ensure that the relevant natural frequencies for the elastic structural vibrations are Froude scaled. For instance, let us consider a steel tank. The bending stiffness matters for the natural elastic frequencies of importance. Further, the length of the elastic plate must be geometrically similar in model and full scale.
The scaling of the natural frequency may be achieved by having different bending stiffness $EI$ in model and full scale. The bending stiffness may be properly scaled by considering a different material and/or changing properly the thickness of the material. Because the main interest is to find the slamming induced structural stresses, care must also be shown in scaling structural stresses. More structural modes may be needed for membrane structures than for steel structures. Some of the important structural modes for membrane structures may have relatively lower natural periods than for steel structures.

If slamming is associated with the formation of gas pockets, the Euler number must be the same in model and full scale. The *Euler number* is defined as $Eu = p_a / \rho U^2$. Here $p_a$ is the ullage pressure, i.e., pressure in the ambient gas of the tank. The consequence of both Froude and Euler number scaling is that the ratio between $p_a$ in model and full scale is equal to the ratio between the length scale $L$ in model and full scale.

A gas pocket has a natural frequency associated with the compressibility of the gas and a generalized added mass due to the liquid oscillations caused by the gas cavity oscillations. If only Froude scaling is used, it has been shown that Froude scaling is conservative. Further, even though the gas cavity oscillations may be linear in model scale they may be strongly nonlinear in full scale with a time history that can cause larger hydroelastic effects.

Because LNG is boiling, the cavitation number is a factor. The *cavitation number* is defined as $\sigma = (p_a - p_v) / (0.5\rho U^2)$ where $p_v$ is the vapor pressure. There is strongly limited knowledge about the effect of boiling on the slamming loads in an LNG tank.

The *Cauchy number* $C_a = \rho U^2 / E_v$ characterizes the effect of compressibility on the flow. Here $E_v$ is the bulk modulus for elasticity. The effect of liquid compressibility is less important in the case of a liquid with no gas bubbles and may be disregarded for a steel tank. This is a matter of what are the important wet structural natural periods. However, the speed of sound in a liquid can be substantially lower in the case of a mixture of gas and liquid. The consequence is an increased characteristic time scale for the effect of liquid compressibility. Because a mixture of gas and liquid can happen during violent sloshing and because LNG is boiling, we cannot out rule the effect of compressibility on slamming loads. The mixture of LNG and gas is not homogeneous. It is only the upper layer of LNG that is boiling. Because the sloshing and the impact affect the pressure distribution in the fluid and the pressure determines if cavitation occurs, the amount of bubbles is time dependent. The void fraction
is therefore a function of time and space. This is an area requiring future research. Surface tension and viscosity are believed to be less important effects during slamming.

3. HIGH-SPEED MARINE VEHICLES

Figure 4: Examples on high-speed marine vehicles supported by foils or an air cushion.

The focus is on the three main categories of high-speed marine vehicles supported by the submerged hull, foils or air cushion. It is important for high-speed marine vehicles to consider all hydrodynamic aspects in design. Considered items have to be limited in this presentation. A more comprehensive discussion of physical effects is given in the book by Faltinsen (2005).

3.1 Surface Effect Ship (SES)

An air cushion is enclosed between the two side hulls and by flexible rubber seals in the bow and aft end of a Surface Effect Ship (SES) (Figure 4). The air cushion raises the vessel, depresses the water surface between the two hulls and lowers the zero-speed metacentric height. However, static stability does not represent a problem at zero speed. Further, the air cushion causes a wave resistance. Generally speaking, the resistance and needed power of a SES is clearly lower than a similarly sized catamaran at corresponding speeds. A danger is leakage from the air cushion in higher sea states with the consequence that the submergence of the vessel increases and the vessel speed drops. Other problem areas for an SES are wear of skirts and as well as power peaks and wear and tear of propulsion/machinery systems caused by ventilation and cavitation due to the low
submergence of the water jet inlet. The latter effect is most probable to happen in a seaway or during a turning manoeuvre. Wetdeck slamming and bow diving may occur. Further, berthing of an SES at high wind speeds may be difficult.

Cobblestone oscillations cause unpleasant vertical accelerations in small sea states and are the result of resonant compressible flow effects in the air cushion. It is called cobblestone oscillations to make the resemblance to driving a car on roughly laid cobblestones. Compressibility effects of the air in the cushion are essential. The oscillations are excited because the waves change the enclosed air cushion volume. It is the vertical vessel accelerations that are of concern. When cobblestone oscillations are not excited, i.e. for higher sea states, an SES will in general have good seakeeping behaviour relative to a similarly sized catamaran.

Figure 5 gives an overview of the physical effects that matter in describing the cobblestone oscillations (Ulstein, 1995). The 1-D wave equation referred to in the figure means that spatially varying one-dimensional standing acoustic waves and spatially uniform dynamic air cushion pressures are studied. Representative values for a 30 m long SES of the spatially uniform pressure resonance frequency and the one node longitudinal acoustic resonance frequency are 2Hz and 5Hz, respectively. There are effects of the dynamic pressure in the air bag and the fact that the water waves impact on the bag and elastic vibrations of the bag. The vibrations of the bag are like a wave maker for the acoustic wave motions in the air cushion. The figure also mentions spatially varying air pressure in the vicinity of the air bag. Because this occurs on a length scale that is short relative to the important acoustic wavelength, it can be analyzed by assuming incompressible fluid. Due to continuity of fluid
mass, the escaping air flow under the air bag must have a mean velocity that is dependent on the local height between the air bag and the water surface. Because high velocity implies small pressure, the escaping air flow causes a suction force on the air bag. This influences the mean escape area of the air from the air cushion. The leakage and inflow to the air cushion influences the damping level of the cobblestone oscillations.

In reality one would use an automatic control system to damp out some of the "cobblestone" effect. This is done by controlling the air flow out from the cushion in such a way that it effectively acts as a damping on the system. In order to do that properly one needs a simplified but rational mathematical method that accounts for the dynamic pressure variations in the air cushion in combination with the global heave and pitch accelerations of the vessel. Sørensen (1993) used a louver system consisting of two vent valves at the deck in the front of the air cushion. The opening and closing of the vent valves control the air flow from the air cushion so that one gets a damping effect on the system. The placement of the louver system is essential. For instance, if the louver system is placed at midships, it will have a negligible effect on the acoustic resonance mentioned above. The reason is simply that the acoustic pressure component has a node, i.e. no amplitude, at midships, while it has its maximum amplitude at the ends of the cushion.

Both the Euler and Froude numbers matter for cobblestone oscillations. A depressurized tank with a wave maker generating high quality waves with small wave lengths is needed. Simplified numerical methods accounting for automatic control are commonly used instead of model tests.

3.2 Hydrofoil vessels

A hydrofoil vessel with a fully-submerged foil system is illustrated in Figure 4. It has in foilborne conditions in general good seakeeping characteristics, create small wash and have small speed loss due to incident waves. Foils are normally designed for subcavitating conditions. However, the possibility of cavitation is then an important issue. Our discussion assumes subcavitating foils.

The designer tries to maximize the foil’s lift-to-drag ratio and the speed for cavitation inception. Further, the weight of the strut-foil system must be minimized with due consideration of structural strength.

Relevant structural loads are slamming, hull bending moments in foilborne condition and bending of the forward foil and strut during recovery from a forward foil broach.
Slamming on the side hulls of a hydrofoil catamaran is not considered as a problem. The reason is large deadrise angles. Because a monohull hydrofoil vessel uses typically a planing hull with relatively small deadrise angles, slamming loads matter. If a hydrofoil catamaran is hullborne in bad weather, wetdeck slamming must be considered. The possibility of grounding and hitting of objects like logs against the strut-foil system must also be evaluated.

Flutter of foils and struts could cause catastrophic failure, but this has never occurred yet. The classical flutter scenario is dynamic instability of combined bending-torsion vibrations of a strut-foil system. The mass ratio, i.e. the ratio of a typical density of structural material to the density of the fluid is an important parameter. This difference creates a clear advantage for a hydrofoil vessel relative to an aircraft when it comes to flutter.

A surface-piercing hydrofoil system in foilborne condition stabilizes the vessel in heave, roll and pitch. This can be understood by means of a quasi-steady analysis. Consider for instance that the heave motion increases. Here heave is positive upwards. This causes the reduction of the wetted foil area. Because the lift is proportional to the wetted area, the lift due to the foils decreases. The weight of the vessel balances the lift in the equilibrium position. The increased heave implies that the vessel weight will force the vessel downwards. Another way of saying this is that there is a restoring force in heave bringing the vessel back to the equilibrium position. Similar static analysis can be made for heel and trim.

A hydrofoil vessel with a fully-submerged foil system needs automatic control to stabilize the vessel in heave, pitch and roll. Platforming and contouring modes are used in connection with an active control system. The contouring mode is used in longer waves to minimize relative vertical motion between the vessel and the waves and to avoid ventilation and broaching of the foils. The platform mode is used to minimize vertical accelerations of the vessel in relatively short waves.

The Reynolds number scaling of foil tests and how to ensure turbulent boundary layers during model tests is an important issue. One effective way is to use Hama strips. The basis of a Hama strip is a tape. A saw-tooth shape is made on the upstream edge by means of a scissor.

Cavitation on foils designed for subcavitating conditions limits the vessel speed to the order of 50 knots. Cavitation appears when the pressure in the water is equal to the vapour pressure, i.e. close to zero. The pressure distribution along the foil should be relatively flat in order to minimize the possibility of cavitation, i.e. there must not be pronounced local pressure minima (suction peaks). The consequence of cavitation is that the material of the foil can be quickly destroyed and the lift capabilities of the foils can be significantly reduced. Another consequence is that the drag on the foils increases. Because cavitation is
accompanied by noise generation, one can hear on board the vessel when there is the possibility of damage due to cavitation. If the vessel speed should be increased substantially beyond 50 knots, supercavitating foils must be used to avoid cavitation damage. They are characterized by much lower lift-to-drag ratios and lift coefficients than subcavitating foils.

The importance of cavitation and ventilation for hydrofoil vessels will be illustrated by model test results for the side force coefficient of a strut used as a rudder on a hydrofoil catamaran. Ventilation means that air enters from the atmosphere to low pressure areas on the strut. It occurs typically at angles of attack $\psi$ higher than $4^\circ$ to $6^\circ$ for a strut on a hydrofoil vessel in foilborne condition. However, this depends on the Froude number. Ventilation will cause a significant drop in the force. Hysteresis is associated with ventilation. This means that $\psi$ may have to be lowered significantly to restore non-ventilated conditions. The detailed physical understanding of ventilation is limited.

**Figure 6:** Model test results of the side force coefficient $C_L$ of a front strut-foil system used as rudder on a hydrofoil catamaran (Minsaas, unpublished).

Figure 6 shows model test results for a front strut-foil system used as a rudder on a hydrofoil catamaran. The model tests were done in the depressurized circulating free surface tank at the Technical University of Berlin. The side force coefficient $C_L$ is presented as a function of the yaw angle $\psi$ both at atmospheric air condition and at the cavitation number
$\sigma = 0.349$ corresponding to the full-scale condition. The pitch angle is $2.7^\circ$ and the submergence Froude number $Fn_h$ is 5.96 corresponding to a full-scale speed of 50 knots. The drop in the absolute value of $C_L$ with either increasing positive $\psi$-values or decreasing negative $\psi$-values is an indication of the start of ventilation. This occurs, for instance, at $\psi \sim 4^\circ$ when $\sigma = 0.349$ and at $\psi = 6^\circ$ at atmospheric air condition.

A consequence of the ventilation at the strut is that air penetrates to the foil and causes a significant drop in the lift force on the foil. This drop occurred at $\psi$ approximately equal to $\pm 6^\circ$. The lift force on the ventilated foil was the order of 50% of the non-ventilated lift.

### 3.3 Semi-displacement and planing vessels.

Semi-displacement vessels without damping systems may have unsatisfactory seakeeping behaviour. One reason is that the excitation loads in heave and pitch at resonance condition in head sea increases with forward speed. Increased heave and pitch damping of semi-displacement vessels is commonly achieved by using T-foils in the bow (Figure 7). Because ventilation and cavitation may limit the operability in higher sea states, wrong conclusions about seaworthiness can result from conventional towing tank tests. While the wave-induced response of displacement vessels can normally be adequately described by linear theory, this is not true for planing vessels. A linear 2.5D (2D+t) theory based on potential flow theory has become popular in describing wave resistance and wave induced motions and loads on semi-displacement vessels.

![Figure 7: T-foil placed below the keel and used to damp vertical wave-induced ship motions at high speed (SEASTATE).](image)

The steady wave pattern (wash) associated with fast semi-displacement vessels are of concern in coastal and inland waters. The wash may cause nearby small boats to capsize or ground or large moored ships to move and mooring lines to break. The waves may cause erosion or even collapse of a bank. When the waves approach a beach, the amplitudes
increase and the waves break. This may happen when the ship is out of sight, surprise swimmers, and represent a risk factor.

An example of the measured wave elevation in the near aft of a trimaran in deep water is shown in Figure 8. Free surface nonlinearities due to flow separation at the transom stern and breaking waves matter. Numerical methods have clear limitations in describing the wave field.

Figure 8: Model tests on high-speed trimaran in calm water. Wave elevation measurements at Froude number 0.4, and colour code for the ratio $\eta / L$ between the wave elevation and the ship length (INSEAN).

The water depth can have an important influence on the wave pattern when the water depth-to-ship length ratio $h/L$ is small. An important parameter is then the depth Froude number $F_{\text{hn}} = U / \sqrt{h g}$ where U is the ship speed. $F_{\text{hn}} = 1$ corresponds to the critical speed. Large changes occur at the critical depth Froude number. The ship waves are very different for subcritical, critical and supercritical speeds in shallow water. The wave pattern at subcritical speed consists of transverse and divergent waves while only divergent waves exist for supercritical speed. The supercritical waves have a very small decay with distance which makes them of particular concern from a wash point of view. The angle (Kelvin angle) between the boundary of the wave system and the ship course is $19^\circ 28'$ in deep water and when $F_{\text{hn}} < 0.5 - 0.6$. A rapid increase in the Kelvin angle occurs for $F_{\text{hn}} > 0.9$, and the angle is $90^\circ$ for $F_{\text{hn}} = 1$. False wall effects can therefore occur during shallow water model tests around the critical Froude number. If the vessel is in the vicinity of critical speed and in a channel, large waves can build up, be unsteady relative to the ship and travel faster than the ship. Russel described in 1865 that one day, the happiest of his life, something unexpected
happened during his towing tests. A large solitary wave occurred ahead of the ship. He could not follow it by foot and got on the horseback and followed it for more than a mile.

Dynamic instability has increased importance with increasing Froude number. One reason is the increased importance of hydrodynamic pressure relative to hydrostatic pressure with increasing forward speed. There is a broad variety of dynamic stability problems. The loss of steady restoring moment in heel with forward speed can cause a sudden list of a round bilge monohull to one side. This can at high speed be followed by a violent yaw to one side. The consequence can be capsizing. This “calm water broaching” is the main reason round-bilge hulls should not operate beyond a Froude number of 1.2 (Lavis, 1980). The loss of steady restoring heel moment with speed should be accounted for in the design by having sufficiently high metacentric height at zero speed.

When a high-speed catamaran in following waves has a speed close to the phase speed of the incident waves, the catamaran can come in a position relative to the waves so that the fore part of the vessel dives into a wave crest. If there is not sufficient buoyancy in the fore part of the vessel, a critical situation may occur.

Examples of dynamic instabilities for planing vessels are “corkscrew” pitch-yaw-roll oscillations, chine walking and porpoising. Porpoising is dynamic instabilities in the vertical plane and is commonly seen. Cavitation and ventilation in the aft body can cause dynamic instabilities.

The Froude number dependence and roll, sinkage and trim matter for manoeuvring of semi-displacement vessels. Strong nonlinearities may occur for a trimaran. For instance, steady heel may cause sign change in steady yaw moment for relatively small heel angles. Numerical methods have difficulties in describing those effects.

Global wave loads matter in the design of vessels longer than 50m. Important global loads for catamarans are transverse vertical bending moment (often called split moment), vertical shear force and pitch connecting moment, as illustrated in Figure 9. Torsional moments, vertical shear forces and vertical bending moments at transverse cross-sections are also of concern, as for monohulls. Both continuous wave loading and slamming will cause global loads. When wetdeck slamming-induced global loads are analyzed, the vessel must be considered elastic. The response is called whipping. When we consider the effect of continuous wave action, the catamaran is often considered as rigid in the determination of the global loads. However, it should be ensured in a linear analysis that the natural frequencies of the global elastic modes are higher than the encounter frequencies of practical interest. When
the global elastic modes are accounted for during continuous wave loading, the phenomenon is called springing. Nonlinear effects may also cause springing.

Figure 9: Examples on global wave loads on a catamaran.

The largest global wave loads in a longitudinal cut of a catamaran occur in oblique sea conditions. The time window for testing in existing model tank facilities in oblique sea can be limited due to the high speed and the fact that transients must die out. Experimental error sources associated with non-constant wave conditions in a basin should also be considered.

4. COMPUTATIONAL FLUID DYNAMICS

It has become popular to use Computational Fluid Dynamics (CFD) to solve fluid flow. There is a broad variety of numerical methods. An overview is given in Figure 10. By CFD we mean that the fundamental governing equations, with initial conditions and nonlinear boundary conditions based on either potential flow, inviscid or viscid fluids are solved. For instance, we do not mean linear strip theories for ship motions.
Figure 10: Overview of numerical methods in fluid dynamics with emphasis on possible solution strategies within Navier-Stokes solvers (Greco, unpublished).

We may ask: What are then the advantages and disadvantages of using CFD? Advantages are that complex structures and general excitation may in principle be considered. A CFD method may provide good flow visualization including details such as the vorticity distribution. Flow separation around structures can be simulated. It seems generally accepted that CFD codes have difficulties in predicting impact loads when the angle between the impacting free surface and the body surface is small. Actually hydroelasticity may then play an important role. Most computations do not consider this fact and assume a rigid structure.

Another disadvantage is that the CFD methods are time consuming which makes statistical estimates of response variables in a stochastic sea difficult. Some methods may not be robust enough. For instance, a Boundary Element Method breaks down when an overturning wave hits the underlying free surface. Numerical problems may also arise with a BEM at the intersection between the free surface and the body boundary. When a BEM
works, it is in general a fast and accurate method relative to other CFD methods. Nonlinear effects associated with water entry of two-dimensional bodies can in many cases be adequately described by potential flow theory and the Boundary Element Method.

CFD methods based on the Navier-Stokes equations are often very robust. Movies based on CFD may look convincing without always representing the reality. Care must be shown that the solutions are true physical solutions. If sufficient care is not shown, some of the methods may numerically lose or generate liquid mass on a long time scale. This is of particular concern in sloshing simulations. Because the highest natural period of the liquid motion is strongly dependent on the liquid mass, this can result in an unphysical numerical simulation. Verification and validation are a must. By verification is meant that the solutions are consistent with the governing equations, initial and boundary conditions that have been used. By validation is meant comparisons with model tests and full scale trials. Experiments are, of course, not free of errors. So it is important that the experiments are accompanied by an error analysis. Verification can be done by temporal and spatial convergence tests and by controlling that conservation of global mass, momentum and energy are satisfied. Comparisons should be made with analytical methods such as the linear potential flow theory.

Because commercial CFD codes are generic of nature and special physical features are specific for the different application fields, one cannot necessarily trust documentation of verification and validation in other applications than those one is interested in.

If a step should be made in improving seakeeping calculations, we have to rely on Computational Fluid Dynamics (CFD) and solve the Navier-Stokes equations. A graph was made by the author in 1988. Major previous achievements in computations of wave induced motions and loads were listed as a function of available computer power and year. A prognosis of the computer power in year 2000 was made and it was asked if it would be common to solve the 3D Navier Stokes equations in waves for ships and ocean structures. Even though the available computer power was underestimated, we are still not at the stage where solving the Navier Stokes equations in waves is standard practice.

A domain decomposition method is suggested for strongly nonlinear flow at the ship. The violent flow near the ship including possible green water on the deck and water impact is handled by a CFD solver based on Navier-Stokes equation with two-phase flow while the more moderate free surface motion at the rest of the sea is solved by a Boundary Element Method. The following example assumes the near-field solver to be a Finite Difference Method combined with a Level Set-method to describe the interface between water and air. Computational time is roughly estimated by Greco (unpublished) to be 16 hours for one wave
period with a PC Pentium IV 2.8 GHz. If 1080 oscillation periods is assumed in a 3 hour storm, this means a computational time of approximately two years. Further, many realizations of each sea state as well as many sea conditions are needed in the design analysis. Computational speed may not be an obstacle with the combined use of supercomputers, future improvements in numerical methods and alternative design procedures by selecting time windows with extreme conditions. However, we have still not solved all physical problems needed for design such as ventilation, air cushions and proper turbulence modelling.

5. CONCLUSIONS

There is an increased focus on Computational Fluid Dynamics (CFD). A broad variety of numerical methods exist and it is difficult to favour one particular method to solve all marine hydrodynamic problems. A straightforward application of CFD in solving the behaviour of ship and ocean structures in severe weather is not conceivable in the near future if proper response statistics should be derived. CFD for high-speed vessels is, in general, not ready for design applications.

Simplified numerical methods based on rational physical approximations should be further developed. An example is on describing cobblestone oscillations of an SES.

A balance between analytical methods, CFD, model tests and sea tests is recommended. Further, links between the different disciplines of hydrodynamics as well as other topics such as structural mechanics and automatic control have been stressed.

Research on violent fluid motions such as slamming, green water on deck, sloshing in tanks, damaged ship with ingress/egress of water are important future activities. Knowledge about the effect of boiling on slamming loads in a LNG tank is needed. A physical understanding of what flow parameters matter for response variables of importance in design and operation is essential. The time scale of a particular physical effect in order for the structure to react, for instance in terms of maximum stresses, is a basis for deciding what physical effects matter.

Dynamic instability, ventilation and cavitation are limiting factors for operation of high-speed vessels. There is limited knowledge about ventilation. A depressurized towing tank with wave maker and 6DOF forced oscillator is desirable. Large horizontal tank dimensions may be needed for seakeeping and manoeuvring tests of high-speed marine vessels.
6. REFERENCES


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 Zaštita brodskoga trupa od korozije i obraštanja

Pregledni rad

Tehnološki korektno izvedena zaštita od korozije znatno utječe na produljenje životnog vijeka broda. Najraširenija tehnologija zaštite brodskoga trupa od korozije jest zaštita premazima, ali u obzir dolaze i koriste se i druge metode, npr. katodna zaštita. Fouling (engl. obraštanje) su kolonije biljnih i životinjskih morskih organizama na uronjenim površinama brodova, pluća, offshore i drugih potopljenih objekata koje izazivaju ozbiljne probleme: povećanje mase uronjene konstrukcije, povećanje otpora i smičnih naprezanja, anaerobnu koroziju kovnih površina. Kroz povijest su se pojavile nebrojene metode zaštite od obraštanja, a danas su u uporabi isključivo premazi koji se razvijaju u sve kvalitetnija i okolišu ugodna rješenja. U tehnologiji nanošenja AF premaza jednako su važni priprema površine, okolišni uvjeti i pravilno nanošenje kvalitetnog premaza.

Ključne riječi: obraštanje, zaštita od korozije, zaštitni premazi

Protecting the Ship's Hull Against Corrosion and Fouling

Review paper

Corrosion protection if carried out technologically correctly, significantly affects the extension of ship's lifetime. The most widely used method for protecting steel is application of coatings, but other methods such as cathodic protection are also applied. Fouling is the settlement of animal and vegetable marine organisms on underwater surfaces of ships, buoys, offshore structures and other immersed objects which cause severe problems: increased weight of the structure, increased drag and shear stress, anaerobic corrosion of metal surfaces. The history records numerous methods of fouling protection, but today coatings are applied exclusively (development of advanced, environment friendly coatings). Equally important in antifouling technology are surface preparation and climatic conditions as well as correct application of a high-quality coating.

Keywords: corrosion protection, fouling, protective coatings

1. Uvod

Brod kao složen i skup proizvod svoju funkciju mora obavljati tijekom životnog vijeka od nekoliko desetljeća, u uvjetima koji su s korozijskoga stajališta iznimno nepovoljni. Korozijskga oštećenja mogu uzrokovati velike probleme na brodskoj konstrukciji, a s vremenom i kolaps konstrukcije što osim materijalne štete može uzrokovati ljudske žrtve, ekološke probleme i sl. Tehnički ispravno i pravodobno izvedena antikorozivna zaštita znatno utječe na produljenje životnog vijeka broda.

2. Zaštita brodskoga trupa od korozije

Najraširenija tehnologija zaštite brodskoga trupa od korozije jest zaštita premazima, ali u obzir dolaze i koriste se i druge metode. U brodogradnji se najčešće koriste zaštitni premazi iz grupe organskih prevlaka, dok je užišta primjene svih ostalih tipova prevlaka bitno manji. Ovisno o dijelu konstrukcije koji se štiti, premazi imaju i druge namjene osim zaštitne (protuobraštajni, protuklizni, protupožarni itd.). Kod izvođenja zaštite od iznimne je važnosti odgovarajuće pripremiti podlog, korektno nanositi premaz, te osigurati dobre radne uvjete (osvjetljenje, dostupnost površine, ventilacija) uz odgovarajuću mikroklimu (temperatura okolice, relativna vlažnost i sl.). Pravilan izbor metode bojenja bitno utječe na cjelokupnu zaštitu površine (brzinu zaštitivanja i kakvoću izvedene operacije). Najčešće se koristi: bojenje četkom, valjima, prskanje sa zrakom, bezračno i elektrostatsko prskanje [1].

Uz zaštitu premazima, važna je metoda zaštite od korozije u brodogradnji katodna zaštita. Postupak se temelji na privođenju elektrona kovini, bilo iz negativnoga pola istosmjerne struje (na-rinuta struja) bilo iz neplenumiti kovine (čvrtovana anoda), sve dok potencijal objekta ne padne ispod zaštitne vrijednosti, jednako ravnopravnu potencijalnu anodu korozijskoga članka, čime nastaje afinity za koroziju, tj. kovina postaje imuna [1].

Inhibitore korozije definiramo kao supstance koje usporavaju korozijске procese kada se u malim koncentracijama dodaju okoli-com svojim efektom usporavajuće ili lakši preparirajući (VCI – engl. Volatile Corrosion Inhibitors). To su organske tvari u čvrstom stanju koje imaju dostatno visok tlak para da bi sublimirale (izravno isparavanje čvrtne fazne) učinile nekorozivim okolni zrak ili neki drugi plin bez potrebe izravnog nanošenja na kovinu. Zbog svojega specifičnog djelovanja inhibitori nalaze primjenu u zaštiti nepristupačnih mjesta brodskih konstrukcija kao što su npr. koblica, unutrašnjost lista kormila, rog kormila, bokostitnik, cjevovodi, brodska oprema, električni kontakti itd.
Veća primjena inhibitora u brodagradnji bez sumnje bi značila tehnološki, ali i ekonomski napredak u području primjene tehnologija zaštitne od korozije [2].


Koroziju je moguće usporiti i različitim konstruktivnim i tehnološkim mjerama tj. kreativnom primjenom teorijskih temelja pri oblikovanju konstrukcije.

3. Zaštita brodskoga trupa od obraštanja

Fouling (engl. obraštanje), sasvim uobičajen izraz i u hrvatskom jeziku, označava kolonije biljnih i životinjskih morskih organizama na uronjenim površinama brodova (slika 1), bova, offshore i drugih potopljenih objekata (slika 2). Najvećim problemom je ubrojavanje i rast većih brodova koje se sa nekih oblika takvog obraštanja su balanidi (priljevci, engl. barnacles), mekušci i morske trave. Nadzirati obraštanje u naravi znači riješiti problem adhezije morskih organizama koji se odvija u četiri glavna stadija. Obraštanje počinje na površini tijela broda. Ako se ovakav proces ne riješi, organizmi koji se uzdržavaju mogu izazvati velike probleme i u potrebi za obnovom zbrane. U svrhu obezbjeđivanja stabilnosti trupa koristimo antifouling razrjeđivački materijali.

3.1. Zašto trebamo antifouling (AF, antivegetativne) premaža?

Glavni su ciljevi brodovlasnika u životnom vijeku broda maximizirati učinkovitost u eksploataciji i minimizirati potrošnju goriva. Obraštanje i dotrajalost površine glavni su uzroci povećanja hrapavosti, a hrapavost izaziva potrošnju goriva u obliku katastrofalnih neprijetnih stanja. U cilju rješavanja ovih problema brodovlasnici koriste antivegetativne premaže čime se izbjegava usporavanje brodskog trupa dobro je definirana [6]:
- na površini od vodne linije pa 1-2 metra u dubinu dominiraju algije i možda nekoliko školjaka (balanidi). Obraštanje u ovom pojasu nastupa prvo i najjače je izraženo.
- niže od toga pojasu pojavljuju se razrasne školjke, mahovnjaci, crvi crjevači
- ravnim dnom broda dominiraju hidroidi (engl. hydroids), balanidi (priljevci, engl. barnacles), školjke (engl. mussels), plaštenjaci (engl. tunicates) i mahovnjaci (engl. bryozoa).

Osim povećanja mase uredjenih konstrukcija zbog obraštanja, povećanja otpora i smičnih naprezanja problem je i anaerobna korozija kovnih površina koja nastaje kad organizmi s ljuskama stvore barjeru između morske vode i površine. Takva barjera stvara mikrokrokolicu s pH vrijednosti u kiselom području, visokim sadržajem C i iона i bez prisutnosti kisika. U takvim uvjetima pojava se intenzitet korozije pogotovo kad je potaknut razvoj sulfatno-reducirajućih bakterija. Te bakterije generiraju sulfidne i prerezvode enzime koji ubrzavaju koroziju, pa je mogućnost pojave hidrodiokvinih i strukturalnih problema velika [7].
ukupnom otporu može iznositi i do 90 posto. Općenito se smatra da prisutnost sluzi na podvodnom dijelu oplate broda uzrokuje povećanje otpora od 1-2 posto, morske trave otpor će povećati za 10 posto, a školjke na dnu za čak 40 posto. S pojavom obraštanj rastu troškovi održavanja broda (brod mora češće u dok, priprema površine i nanosnje premaza iziskuju više vremena i sredstava), a smanjuje se upravljaljivost broda. Utjecaj na okoliš evidentno je i odbijal, jer povećana potrošnja goriva rezultira povećanjem emisijom štetnih plinova (CO₂, NOₓ, SO₂) i uzrokuje širenje morskih organizama iz prirodnoga staništa u područja gdje mogu predstavljati prijetnju ekološkoj ravnoteži.

3.2. Povijesni prikaz

Prvi pisani dokazi o tretiranju dna broda datiraju iz 5. st. pr. Krista. Smatra se da su već drevni Fenici i Karantanški koristili katran, spominje se uporaba arsena i sumpora pomićanih s uljem, a Grči su koristili katran ili vosak, te različite gume, ebonitu (tvrda guma), plutu i papiru, te različite miješavine koje su se u obliku premaza nanosile na podvodni dio trupa i štitile ga od obraštanja uporabljani su bili: smole, pčelinji vosak, sirovterpentin, pčelinji alkohol, usitnjeno staklo, katran, vapno, lijevani kositar, cink, željezni sulfaat, asfalt, riblje ulje itd. Nakon 1835. godine ozbiljno su shvaćena problem galvanske korozije čeličnoga trupa i rastuća potreba za sredstvom protiv obraštanja koje neće proizvoditi galvanske efekte na trupu, pa se počinju razvijati premazi koji iz neke vrste matrice otpuštaju otrovne tvari. Najčešće korištene otrovi bili su bakar, arsen i živa zajedno s njihovim spojima, kao i otapala služili su terpentin, nafta i benzen, a matrice su činili laneno ulje, šelak (‘prirodna plastika’), katran i razne smole. Sredstva protiv obraštanja bila su preskupoa, često kratkog vijeka trajanja, a ponekad i nepouzdana. Formula Norfolk sadržavala je crveni živioksid raspršen u šelaku, pčelinji alkoholu, terpentinu i ulju borovoga katranu uz dodatak cinkovog oksida i cinkove prašine. Kvalitetan šelak nabavljao se iz Indije što je sa širem uporabom postalo preskupoa i prekomplicirano za nabavu, pa je dobro zamjena šelaku uvedena prirodna smola. Početkom 20. st. vladalo je mišljenje da je Talijanska Moravia (premaz na osnovi vrele plastike, smjesa prirodne smole i bakrenih spojeva) jedan od najboljih antivegetativnih premaza. Prije nanošenja bilo ga je potrebno grijati što je predstavljalo problem zbog kojega se uvode premazi koji su se suše isparavanjem otapala [9].

Bakrene obloge

Iako su stare civilizacije poznavale bakar i broncu i znale su njihovih brodova i veze ih vrlo dobro tehnološki obraditi, pa šira uporaba ovih materijala u brodogradnji nije isključena. Bakrene obloge prestale su se koristiti [9]. Bakrene obloge se koriste u modernim brodogradilištem, ali su toplinski pougljenili izvanjsku površinu brodova u dubinu i smanjili ih vrlo dobro tehnološki obraditi, pa šira uporaba ovih materijala u brodogradnji nije isključena. Bakrene obloge prestale su se koristiti [9]. Bakrene obloge prestale su se koristiti [9]. Bakrene obloge prestale su se koristiti [9]. Bakrene obloge prestale su se koristiti [9]. Bakrene obloge prestale su se koristiti [9].

Zaštita čeličnoga trupa

Smatra se da je važnost obraštavanja pridana tek s uvođenjem čeličnih brodova i većim brzinama plovidbe. Bakrena se obloga pokazala neuporabljivom, pa je pokrenuta potraga za manje štetnom kovnom oblogom i načinom izolacije bakrene obloge i čeličnoga trupa. Prilikom obraštavanja, munt metal (vrsta mjeđi), pocićeni čelik, nikal, slitine olova i antimona, cinka i kostru, a ispitivanje se suvremene tehnologije zaštite brodskoga trupa od obraštanja

TBT spojevi

Premazi na osnovi organokositrenih spojeva (TBT) komercijalizirani su 60-ih godina 20. st. i pozdravljeni kao „čarobno oružje“, jer su pružali potpunu zaštitu od obraštanja u razdoblju od 5 godina, a bilo ih je jednostavno nanositi. Danas se u zaštiti brodskoga trupa od obraštanja gotovo isključivo koriste premazi koji iz neke vrste matrice otpuštaju otrovne tvari. Najčešće korištene otrovi bili su bakar, arsen i živa zajedno s njihovim spojima, kao i otapala služili su terpentin, nafta i benzen, a matrice su činili laneno ulje, šelak (‘prirodna plastika’), katran i razne smole. Sredstva protiv obraštanja bila su preskupoa, često kratkog vijeka trajanja, a ponekad i nepouzdana. Formula Norfolk sadržavala je crveni živioksid raspršen u šelaku, pčelinji alkoholu, terpentinu i ulju borovoga katranu uz dodatak cinkovog oksida i cinkove prašine. Kvalitetan šelak nabavljao se iz Indije što je sa širem uporabom postalo preskupoa i prekomplicirano za nabavu, pa je dobro zamjena šelaku uvedena prirodna smola. Početkom 20. st. vladalo je mišljenje da je Talijanska Moravia (premaz na osnovi vrele plastike, smjesa prirodne smole i bakrenih spojeva) jedan od najboljih antivegetativnih premaza. Prije nanošenja bilo ga je potrebno grijati što je predstavljalo problem zbog kojega se uvode premazi koji su se suše isparavanjem otapala [9].
Prema mehanizmu otpuštanja biocida suvremena tehnologija AF premaza dijeli se na [11]:

1) tehnologija premaza temeljenih na prirodnim smolama koji mogu biti:
   a) premazi s topivom matricom (engl. controlled depletion polymer, CDP)
   b) premazi s netopivom matricom (engl. contact leaching antifoulings)

2) tehnologija samopolirajućih kopolimera (engl. selfpolishing copolymer, SPC)

3) tzv. hibridna SPC/CDP tehnologija.

Antivegetativni premazi s topivom matricom - Controlled Depletion Polymer, CDP

Sadrže više od 50 posto prirodne smole ili njezinih derivata u vezivu, a biocid je bakreni oksid zajedno s pojačivačima. Iako se u teoriji ovi premazi mogu otapati i imaju polirajući efekt, u praksi se to ne događa zbog gomilanja bakrenih soli i ostalih netopivih spojeva što stvara debeli iscrpljeni sloj. Značajke su im zadovoljavajuće za primjenu u područjima s niskom stopom obraštanja i na brodovima s kratkim intervalima između dokiranja [11].

Premazi protiv obraštanja s netopivom matricom - Contact Leaching Antifoulings

Kod ove vrste premaza zbog malog udjela smole nema otapanja tijekom vremena, pa se na površini stvara debeli iscrpljeni sloj (difuzija biocida iz dubljih slojeva je usporena, a prazna matrica povećava hrapavost). Premaz je moguće reaktivirati struganjem prazne matrice, ali to može izazvati ponovni rast morskih trava koje su se naselile u šupljinama [11].

Samopolirajući premazi protiv obraštanja - Self Polishing Copolymer, SPC

Biocidi se otpuštaju u procesu hidrolize ili ionskom zamjenom između akrilnoga polimera i morske vode isključivo u blizini površine (sloj tanji od 30 μm) što omogućava nadzor otpuštanja biocida i proizvodi efekti samozaglađivanja (potvrdjeno su s hydrodamičkih stajališta). Idealni su za primjenu na novogradnjama (čvrst i trajan film premaza), Glavni biocid je bakreni oksid zajedno s cinkovim oksidom (ZnO, ZnO2), pojačivačem koji se brzo razgrađuje a ne akumulira se u morskom okolišu. Mnogo su učinkovitiji od CDP premaza (stopa otpuštanja biocida je konstantna dokle god postoji sloj premaza) [11].

Hibridni SPC/CDP premazi protiv obraštanja


4.2. Neobraštajući premazi – Foul Release Coatings, FRC

Sa stajališta zaštite okoliša najpoželjniji pristup zaštiti broda od obraštanja svakako je onaj koji se ne oslanja na otpuštanje biocida u morski okoliš. Od mnogih zamisli samo je foul release tehnologija uspješno komercijalizirana. Foul release silikonski premazi su vrlo glatki što otežava adheziju morskih organizama (morski organizmi radije obraštao ravnim sami prema površini - tigmotaktička priroda obraštanja). Dok samopolirajući (SPC) premazi imaju otvorenu teksturu s učestalim šiljcima i udolinama nalik na površinu planinskog lanca (slika 3), foul release sustavi imaju površinu otovorene teksture nalik na malo valovitu morsku površinu (slika 4).

Fluorirani silikonski elastomerni polimeri imaju dobra svojstva stvaranja tankoga filma i kemijsku i biološku inertnost, ali su im mehanička svojstva loša. Na brzim brodovima premaz se čisti od obrašta samo prolaskom trupa kroz vodu dok je na sporim brodovima potrebno provoditi čišćenje (ispiranjem pod niskim tlakom) [11]. Na sveučilištima u Newcastleu, Hamburgu i Hirošimi razvijaju se daljinski upravljani roboti za podvodno čišćenje. U tablici 1 dan je prikaz glavnih značajki pojedinih vrsta AF premaza.
Unatoč atraktivnim značajkama neobraštajućih premaza do njihove šire primjene nije došlo uglavnom zbog toga što većinu svjetske flote čine tankeri i brodovi za rasuti teret koji ne plove pri dostatno velikim brzinama i nemaju dostatnu aktivnost da bi danas dostupni foul release premazi pokazali svoju učinkovitost i opravdali svoju cijenu koja je 5 do 10 puta viša od cijene ostalih danas dostupnih AF premaza bez kositra

<table>
<thead>
<tr>
<th>Vrsta premaza</th>
<th>Svojstva</th>
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| Premazi s netopivom matricom        | - visoka mehanička čvrstoća  
- pražna matrica pridonosi povećanju hrapavosti brodskog trupa  
- premaz je moguće reaktivirati struganjem pražne matrice, ali to može izazvati ponovni rast morskih trava koje su se naselile u šupljinama  
- kratki životni vijek, do 18 mjeseci                                                                                                                                                                                                                                           |
| Premazi s topivom matricom, CDP     | - visoki udio prirodne smole, bakar kao glavni biocid  
- niska mehanička čvrstoća  
- debeli iscrpljeni sloj zbog gomilanja netopivih spojeva (soli, nečistoće)  
- pruža zaštitu u trajanju do 36 mjeseci  
- cijena najniža među AF premazima bez kositra                                                                                                                                                                                                                                   |
| Samopolirajući kopolimeri, SPC     | - otpuštanje biocida i otapanje polimera u tankom površinskom sloju – efekt samozaglađivanja  
- čvrst i trajan film premaza – idealno za primjenu na novogradnjama  
- glavni biocid je bakarni oksid sa cinkovim oksidom kao pojačivačem  
- stopa otpuštanja biocida je konstantna dokle god postoji sloj AF premaza  
- zaštita do 60 mjeseci, ovisno o uvjetima u službi                                                                                                                                                                                                                           |
| Hibridni CDP/SPC                    | - mali udio otapala, kontrolirana stopa otpuštanja biocida  
- trajniji film u odnosu na CDP premaze  
- učinkovitost i cijena između performansi CDP i SPC tehnologija  
- životni vijek do 36 mjeseci                                                                                                                                                                                                                                                     |
| Foul-release premazi                | - bez biocida  
- silikonska baza stvara vrlo glatku površinu koja otežava obraštanje  
- vrlo mekani, podložni mehaničkim oštećenjima  
- za samočišćenje potrebna je velika brzina plovđive ili visoka aktivnost broda  
- 5-10 puta skuplji od ostalih AF premaza                                                                                                                                                                                                                                         |

5. Alternativni pravci u zaštiti od obraštanja

Kristokijorno rješenje nakon zabrane korištenja TBT spojeva je premazi na bazi bakra koji su danas još uvijek dopušteni, ali strahuje se od trovanja neciljanih skupina organizama. Da bi se razvili netoksični (ili vrlo malo toksični) sustavi potrebno je bolje shvatiti biokemiju morskih mikroorganizama i mehaniku njihove inicijalne adhezije.

Biološke boje

Znanstvenici su identificirali i sintetizirali tvari koje nekim morskim organizmima omogućavaju izbjeći obraštanje (npr. koralji), a razvijaju se i tehnologije koje bi omogućile zamjenu biocida u AF premazima s enzimima koji ometaju mehanizme biološke adhezije. Načini industrijske proizvodnje i aplikacije bioloških repelenta još nisu razrađeni [12, 10].

Elektrovodljivi premazi

Površina brodskoga trupa koja je u doticaju s morskom vodom prevučena je elektrovodljivim slojem premaza. Hidrolizom morske vode i reakcijom s Cl ili ionima nastaju ClO ioni s antivegetativnim djelovanjem. Mala struja prolazi kroz fazi izgradnje broda i privlači ClO ione koji tvore tank sloj, a nestaju kad se odvoje od trupa (reagiraju s drugim sastavnicama iz morske vode i ne uzrokuju kontaminaciju mora). U procesu ne dolazi do otpuštanja čestica, pa premaz ostaje gladak. Sustav se testira na nekoliko velikih i srednje velikih brodova [10].

6. Tehnologija nanošenja premaza protiv obraštanja

Nanošenje premaza protiv obraštanja na novogradnjama izvodi se u dvije faze: 1) na navozu i 2) u doku. Razlozi tome su sljedeći:

- brod će nakon porinuća provesti od 2 do 3 mjeseca u moru, privezen na opremljenoj obali, pa ga je potrebno zaštititi od obraštanja već prije porinuća  
- radi ležanja na potkladima nemoguće je zaštititi sva mjesta na podvodnom dijelu trupa  
- u fazi izgradnje na navozu na oplućenom sloju broda zavaren je određen broj uški i proširi  
- prije porinuća brod se zaštićen od obraštanja u fazi izgradnje broda  
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- a provesti od 2 do 3 mjeseca u moru, privezen na opremljenu obali, pa ga je potrebno zaštititi od obraštanja već prije porinuća
Nakon pripreme površine pristupa se nanošenju slojeva antikoro-
vizne zaštite i prvoga sloja AF (flekanje). Završna faza tehnologije zaštite od korozije u doku jest nanošenje završnoga sloja AF premaza (full coat) na podvodni dio brodskoga trupa (slika 6). AF premazi kojima se zaštićuju bokovi podvodnog dijela (vertikale) i dno broda (flat bottom) moraju odgovarati ponešto različitim zahtjevima, pa se iz tog razloga na bokove i na dno nanose različiti premazi u različitim debljinama (premaz na bokovima u eksploata
tciji se više troši od premaza dna, organizmi koji naseljavaju dno i bokove se razlikuju). Nadzor nad cijelim procesom obavljaju tri
inspektora: inspektor brodogradilišta, inspektor proizvođača boje
i inspektor brodovlasnika. Nakon svake pojedine faze inspektor
svojim potpisima na Prizemljenom listu označavaju završetak
iste. Tek kad se utvrdi da nema primjedbi ili da su one uspješno
ukonjenje prelazi se na sljedeću fazu.

7. Zaključak

Primjenom tehnologije zaštite od korozije osigurava se po-
stojanost konstrukcije tj. sposobnost broda da dugi niz godina
obavlja svoju funkciju. Zaštita od obraštanja osigurava se po-
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I. JURAGA, I. STOJANOVIĆ, T. NORŠIĆ
ZASTEBA BRODSKOG TRUPA OD KOROZIJE I OBRAŠTANJA
The KRALJEVICA Shipyard, shipbuilding and shiprepairing company, is the oldest shipyard on the eastern coast of the Adriatic Sea. The continuity of shipbuilding in KRALJEVICA has been lasting uninterrupted since the year 1729.

The KRALJEVICA Shipyard ranks, in view of its capacities, among medium-sized shipyards (420 employees, area is 110,000 m²)

Main activities:

- newbuilding of ships and other marine constructions of up to 120 m in length, up to 10,000 dwt (passenger/car ferries, tugs, supply vessels, tankers, dry cargo vessels, Ro-Ro vessels, multipurpose/container and paper carriers, etc).

- newbuildings, retrofitting and repairing naval (gun boats, patrol vessels, missile corvettes), coast guard boats, special-purpose ships, fast crafts, light commercial crafts and yachts, built of ordinary or high strength shipbuilding steel and aluminum.

- engineering, consulting and trading, projects, drawings and technical documentation.

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“TROGIR” SHIPYARD was founded in 1944 as a small repair yard and now is equipped to accept construction of the most sophisticated vessels. Since 1960 when construction of steel vessels begun the Shipyard has delivered 100 ships of various types and sizes plus 17 floating docks mainly for foreign buyers.

Shipbuilding works dispose of two slipways, the smaller one 20m wide and bigger one 47m wide. The slipways can accommodate ships up to 50,000 dwt ranging from oil tankers, cargo ships, ferry boats, supply vessels, tugs rescue vessels as well as floating docks of 60,000 tons lifting capacity built in sections and thereafter connected afloat.

“TROGIR” SHIPYARD is today a shipyard that offers the buyers all over the world its cooperation in designing and building vessels of different purpose making its best to comply with the request of potential buyers.
BRODOSPLIT - Croatian shipyard with a long tradition and experience in designing and building various types of ships, always significant in their class.

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e-mail: uprava@brodosplit.hr, web: www.brodosplit.hr
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**ULJANIK Shipyard**
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- Educated personnel
- CAD/CAM system
- ISO 9001, ISO 14000
- Diesel engine MAN-B&W licence
- High quality of various types of ships

**ULJANIK Brodogradilište d.d.**
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HRVATSKA GOSPODARSKA KOMORA

Sektor za industriju
Zajednica proizvođača brodske opreme

Članica Europskog vijeća proizvođača brodske opreme - EMEC

Zajednica proizvođača brodske opreme okuplja proizvođače uređaja i opreme te pružatelje usluga povezanih s brodogradnjom radi osvrivanja njihovih interesa.

ZAJEDNICA SVOJIM ČLANOVIMA OMOGUĆUJE:
zajednički nastup pred državnim institucijama radi osiguranja što povoljnijih uvjeta poslovanja;
jedinstvenu promociju kod domaćih i inozemnih brodograditelja, tiskanje promotivnih materijala i organiziranje izlaganja na sajmova u inozemstvu u suradnji s Hrvatskom brodogradnjom - Jadranbrodom d.d.;
mogućnost povezivanja s inozemnim partnerima radi izvoza ili kooperaicije;
usuglašavanje razvoja proizvodnih programa u suradnji s Hrvatskom brodogradnjom - Jadranbrodom d.d. odnosno hrvatskim brodogradilištima.

CROATIAN CHAMBER OF ECONOMY

Industry and technology department
Affiliation of marine equipment manufacturers

Member of the European Marine Equipment Council – EMEC

The Affiliation of Marine Equipment Manufacturers gathers the manufacturers of marine instruments and equipment, and service providers to the shipbuilding industry.

THE AFFILIATION ADDRESSES ITS MEMBERS NEEDS IN THE FOLLOWING WAYS:
joint approach to government institutions in order to ensure the most favourable business conditions;
joint promotion among both the domestic and foreign shipbuilding companies - publication of promotional materials and organizing the display of products at international fairs in cooperation with the Croatian Shipbuilding Corporation (Hrvatska brodogradnja - Jadranbrod d.d.);
establishment of links with foreign partners, with a focus on exports and cooperation;
coordination of the development of manufacturing programmes in cooperation with the Croatian Shipbuilding Corporation, i.e. with Croatian shipyards.