Development of the Wärtsilä EnergoFlow: An innovative energy saving device

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ABSTRACT

The quest for a solution of an energy saving device which gives sufficient power reduction, fulfils structural requirements and facilitate an easy installation paved way to the development of an innovative pre-swirl stator (PSS) commercially known as The Wärtsilä EnergoFlow. This device finds its suitability both for new builds and the retrofits. The innovative design features the curved stator fins and the connecting ring at the fin tips amongst others. The results of the CFD analyses shows that the generated pre-swirl considerably increases the propeller blade efficiency in the quadrant of the upcoming blade. The device can generate power reduction for a given sailing speed, without experiencing much resistance of the curved fins. Results show that highest power reductions can be achieved when the EnergoFlow and propeller are designed as a combined pair. The ring connecting the fin tips reduces the tip vortex intensity while contributing to the robust design of the structure. Calculations suggest that the stress levels during conditions such as slamming are reduced by 40 % when the ring is applied. Investigations were carried out in the manufacturing method of the curved fins and was concluded that they can be manufactured either as a welded structure or as a casting or even the combination of both is a possibility. The connection of the fins to the stern structure is established by welding them to the cast bossing or the stern tube. The determination of the loads acting on the fins is carried out by the Ship Motion Methodology. The method for accessing the fatigue is described in the paper. The maximum occurring loads are used as input for the FE assessment of the entire stern structure including fins for integration within the ship’s hull.

Keywords
Pre-swirl stator, Innovative design, fuel saving, retrofit, hull integration, fatigue, EnergoFlow

INTRODUCTION

Pre-swirl stators (PSS) have been on the market for many years now, but not always have they lived up to expectations. One reason is that the hydrodynamic effects of an upstream stator are quite complex, comprising not only the interaction between stator and propeller but also the effect on the hull and the change in the rudder inflow. The maturity of today’s numerical calculation tools such as CFD are crucial to take all these factors into account. Another critical aspect can be found in the structural design and hull integration of the stator fins given the hydrodynamic loads resulting from ship motions. The development and technical findings of a next-generation pre-swirl stator, are shared here. Both hydrodynamic and structural aspects are highlighted.

1 ENERGOFLOW

The EnergoFlow is a structure consisting of multiple fins attached to the ship's hull for preventing the losses occurring in the slipstream of the propeller or in other words it provides a favourable inflow. The EnergoFlow comes with curved fins and a ring at the tip of the fins, this distinguishes it with the other available pre-swirl stators. The use of curved fins enhances the propeller efficiency keeping the resistance at the acceptable levels. The ring reduces the tip vortex and also levels out the peak stresses occurring in severe loading conditions such as slamming. The development of EnergoFlow is done keeping in mind the retrofits market, but it very well finds its suitability on the newly built vessels. Although the fuel savings are always case dependent, up to 10 % are possible in the cases where the combination of EnergoFlow and the new propeller are optimised as a combination. Figure 1 below shows a graphical impression of the EnergoFlow.

Figure 1: Graphical impression of EnergoFlow
WHERE DOES THE POWER SAVING COME FROM?

Sea trials conducted as part of the European project GRIP (Green Retrofitting through Improved Propulsion), demonstrated a clear benefit of using a pre-swirl stator, showing a reduction in the range of 7% for the selected bulk carrier (Prins, et al., 2016).

The propeller is located at the stern of a ship and during its operation, it experiences non-uniform inflows. In Figure 2 on the left, a typical velocity profile at the propeller plane is given with the contours of a clockwise rotating propeller. The colours indicate the axial velocity ratio with respect to the vessel speed, while the vectors show the combined radial and tangential component of the inflow. The downward moving blade experiences a counteracting tangential velocity, which has a positive effect on the blade loading. At the upcoming side, the tangential flow has a negative effect on the blade loading. Higher the blade loading brings higher efficiencies and vice versa. So ideally, the circumferential velocities in the propeller plane should all have the same counteracting direction, as shown on the middle chart in Figure 2.

To influence the tangential components towards an ideal situation as shown in Figure 2, a pre-swirl stator can be applied. A PSS guides one side of the stern flow in the opposite direction of the propeller rotation, generating pre-swirl for the propulsor. An example of the effect of the PSS on the wakefield is shown in Figure 2 on the right side. It is important to note that the PSS is more effective on the lower radii. At the higher radii, the circumferential propeller velocity becomes so dominant that the effect of the tangential velocity in the wakefield becomes negligible (Schuiling, 2013).

The propeller’s blade efficiency varies during one revolution dependent on the local inflow (wakefield). A typical example is given in Figure 3. In the figure the angular clockwise position is represented by X-axis and the efficiency is shown on the Y-axis. The shown efficiency is represented by non-dimensional thrust and torque coefficients. (KT/10 KQ). The blue curve shows the variation in blade efficiency over one rotation for a 4 bladed propeller design in the wake of a bulk carrier. Once applying a PSS the efficiency clearly increases between 0° and 120°, which equals the zone of the upward moving blade. The efficiencies shown are based on a CFD numerical self-propulsion test. (Bulten, 2015)

An overall reduction in power can only be achieved if the positive effect on the propeller’s performance from the imposed pre-swirl outweighs the added resistance caused by the stator fins.

3 NUMERICAL HYDRODYNAMIC ASSESSMENT

The hydrodynamic performance and the interaction between the Ship’s hull, PSS, the propeller and the rudder need to be quantified by calculations based on CFD numerical self-propulsion. This will indicate the power reduction, resistance added by the PSS, propeller inflow and the changes in the propulsive efficiency. Model test, sea trials, and the voyage data monitoring can be used for validation purpose.

The CFD assessment was done with the commercial software STAR-CCM+ on a high-performance computing cluster. The simulation was set up with an unstructured mesh consisting of trimmed cells using a High y+ Wall Treatment. The free surface is solved via a Volume of Flow (VOF) approach. For the turbulence modeling, the standard k-ε model was used. To obtain reliable results for the numerical self-propulsion calculations a moving
mesh was applied. The transient results were averaged over a number of propeller revolutions.

Validation of the methodology was done both in a qualitative and quantitative sense. Bare hull resistance calculations with fins were compared with available data from e.g. the GRIP project, which showed a good correlation. Typically added resistance values of the PSS were 2-4 % of the ship’s resistance. The effect of PSS on the wakefield can also be derived from the bare hull resistance calculation. Again those wakefield data were reviewed with GRIP data and showed proper agreement concerning the expected trends.

In Figure 5 an example is shown of the tangential velocities when a PSS is applied. The left plot shows the circumferential inflow without the PSS while the right plot shows the propeller’s inflow with the PSS. From these standard used plots, it is not easy to judge the improvement.

Circumferentially averaging the tangential velocity components and the corresponding standard deviations provide a better insight, see Figure 6. The blue line depicts the case without pre-swirl stators, the red line the case with pre-swirl stators. The difference in average tangential velocity shows the change in the rotational velocity in the propeller plane.

For lower radii the standard deviation is lower, this indicates a more homogenous inflow and therefore a higher efficiency. Note this improvement might not be fully utilized for the existing propeller since the new inflow angle might not be optimal. It is understood that the change of the standard deviation of the tangential inflow is a good indicator of the effect of the PSS on the propulsive efficiency. This indicator will be used during the development process.

From the self-propulsion assessment the propeller’s efficiency, RPM, the effect on hull and rudder resistance can be derived. Comparison with earlier mentioned reference cases shows that the calculated RPM reductions correlate reasonably well (as shown in Figure 7). In the first two cases, a PSS was fitted, whereas in the third one a new propeller design was additionally retrofitted.

The effect of the PSS on the tangential inflow, is inversely proportional to the propeller speed, in other words, more effect of the PSS will lead to the reduction of a propeller speed. These aspects are to be taken during the design of the propeller.
Apart from the added resistance of the stator fins (see next section) it was observed that the change in rudder resistance also cannot be neglected. A closer look at the rudder performance in self-propulsion calculations shows that the rudder drag can increase in the presence of the PSS. The combination of the PSS and propeller leads to a less intense rotational flow downstream of the propeller. The rudder also acts as a stator fin as it partly recovers the rotational energy in the flow. Due to the PSS, the rudder can only recover less energy because there is less, to begin with.

![Figure 8: Rudder pressure. Left without PSS, right with PSS](image)

The pressure distribution over the rudder for the case with and without a PSS can be seen in Figure 8. A typical case for a bulk carrier is shown. When applying a PSS, the rudder’s resistance increases from 6 to 7% as part of the total vessel’s resistance. The differences are subtle, however, in order of 1% or more power demand due to the interaction with the rudder must be taken into consideration

The variation in the propulsive efficiency with a PSS can be derived from the numerical propulsion test. A graphical overview is shown with the kt/kq ratio in Figure 3. The example is of a bulk carrier with a 3 fins PSS. The effect of the fins is clearly visible. Propulsion efficiency gains exceeding 5% are calculated. The adverse effect of added resistance needs to be taken into account to an overall power saving. In order to judge the ratio of propulsion gain in relation to resistance loss on the power reduction, a quality index for a PSS is defined according to the equation [1]. This equation is based on and derived from the classical definition of propulsion efficiency; related to ship resistance, sailing speed and needed power.

$$QI = 1 - \frac{\Delta R/R}{\Delta \eta_D/\eta_D}$$  \[1\]

This PSS quality index is helpful to compare various PSS designs. It also shows the insight of the added resistance versus propeller efficiency gain since both can be combined via the power definition. In general, a good value for the quality index is in between 1-2, depending on ship type and propeller design. We chose to use an index with values > 1, showing the positive effect of an efficiency increase and the adverse effect of the added resistance, although this requires a negative sign for the relative change in resistance.

Figure 9 shows the power change when applying a PSS in combination with a new propeller design for a container ship. The results are derived from a numerical propulsion test. The figure shows the relative propulsion gain in comparison to the added resistance. The quality index is 1.26, which indicates that a large part of the increase in propulsion efficiency is transferred into a power reduction.

![Figure 9: Relative calculated power change for a container ship with a PSS and new propeller design](image)

Based on the available validation material, numerical simulation results show there is a tendency for underestimating the power reduction. These differences might be attributed to the simplification of turbulence as a hydrodynamic phenomenon can have a significant effect on local flow phenomena such as wake of the vessel. To improve the performance prediction it is worthwhile to investigate the effect of other turbulence models such as the Reynolds Stress model or SST k-ω with Algebraic Reynolds Stress Model combined with curvature correction. (Guirard, 2013)

More validation of the simulations by means of model test measurements is planned. However, it must be noted that model tests suffer from scaling effects, especially when it comes to Energy Saving Devices (J. Dang, 2012), (G.J. Zondervan, 2011).

4 PARAMETRIC VARIATION AND DESIGN OPTIMIZATION

The EnergoFlow’s fin shape is the result of an extensive stator fin design study in which a large number of resistance calculations have been performed. The analysed models have been defined based on a process of generic optimization instead of systematic variation. At the start of the project, several parameters were identified for following this setup; the complexity of the fin design,
diversity in parameters and the interaction effects to name a few.

An important part of the design strategy is the placement of the fin on the hull. A fin can deliver thrust and redirect the flow but it can also contribute to the hull resistance and thereby affecting negatively on the propulsive gain. This means that the number of fins, placement on the hull and rake distribution are decisive for the EnergoFlow’s success. In Figure 10 the effect of EnergoFlow design on the resistance and the field averaged per radius standard deviation of the tangential velocity ($\sigma_{\text{vtan}}$) can be seen. EnergoFlow fins which resulted in a small resistance increase while lowering $\sigma_{\text{vtan}}$ showed the most gain.

Figure 10: the effect of EnergoFlow designs on the resistance and the field averaged per radius standard deviation of the tangential velocity ($\sigma_{\text{vtan}}$)

Another interesting finding is the cavitating fin tip vortex and its possible adverse effects on the propeller performance and the cavitation behaviour. In Figure 11 an example of a cavitating fin tip vortex can be seen. By adding a ring at the tip of the fins the vortex strength is distributed, resulting in a more uniform inflow (see Figure 12). Furthermore, the span of the fins is chosen such that the tip vortex is less likely to enter the propeller plane.

Figure 11: Example of a fin tip vortex as witnessed during the GRIP PSS full-scale trials.

Figure 12: The effect of the fin tip vortex on the tangential velocity at the propeller disc, no ring (TOP) and with a ring (Bottom).

As mentioned earlier, directing the tangential flow is more effective in increasing the propeller performance at the lower radii. Therefore, the pitch and camber of the fin sections vary from being positive in the root towards being negative at the tip section. Effectively redirecting the flow at the lower radii, minimizes the losses at the higher fin span.

5 LOADS ACTING ON THE PRE-SWIRL STATOR

The hydrodynamic forces induced due to the ship motion in waves is the determining load case of a PSS. These forces fluctuate over time and are dependent on wave height and the ships dynamic response. Fatigue and incidental peak stresses are the main criteria for the PSS strength assessment. The loads required for this assessment are determined using a statistical approach based on the Ship Motion Methodology (SMM) as developed by S. Paboeuf (Prins, et al., 2016).

The SMM is based on the Response Amplitude Operators (RAOs) that can be calculated for a certain loading condition using a 2D Strip Theory or a more advanced method. These RAOs are used together with a JONSWAP spectrum ($S_T(\omega)$) that represents the sea state and the result is the response spectrum ($S_\mathbf{R}(\omega)$) as shown in equation [2].

\[
S_\mathbf{R}(\omega) = |\text{RAO}(\omega)|^2 S_T(\omega) \tag{[2]}
\]
In the wave scatter diagram (see Figure 13) the possibility of certain sea states is shown. For the peak stress, a sea state is chosen that corresponds with a probability of exceedance of $10^{-8}$ which is equal to a 25-year wave. For the fatigue calculation, all the sea states with their probability of occurrences are taken into account to determine the loads. The loads are determined by specifying the operational speed profile of the ship in question and the ship motion is converted to an angle of attack on the fins of the PSS.

**Figure 13: Wave scatter diagram, typical example per 100000 observations**

Compared to the initially used SMM, the method of calculation of the lift and drag coefficients has been improved. Instead of evaluating the lift and drag coefficients in 2D and then using an empirical correlation, the lift and drag coefficients were now calculated using lifting-line theory (A. Houpt, 2016) where 3D effects were also taken into consideration. Another improvement is that the fin is divided into 12 sections instead of 3 and at each section, the geometry, inflow angle with respect to the horizontal, zero lift angle and the velocity were determined.

With these parameters known, the angle of attack is determined and applying the lifting line theory results in the lift and drag coefficients which can be used for the lift and drag force calculations, see equations [3] and [4]. The lift and drag can then be transformed to forces in the earth-fixed coordinate system taking the positioning of the fins into account.

\[
F_L = \frac{1}{2} \rho u^2 C_L A \tag{3}
\]

\[
F_D = \frac{1}{2} \rho u^2 C_D A \tag{4}
\]

$C_D$ & $C_L$ = Lift and drag coefficients $\rho$ = density $u$ = speed $A$ = Area $F_D$ & $F_L$ = Lift and drag

The same example case as in the SMM is used to validate the Lifting Line Methodology (LLM) results. The corresponding PSS is shown in Figure 14.

**Figure 14: PSS as used for the load calculation**

The results are compared with the CFD results as obtained from HSVA during the GRIP project. A statistical approach was simplified for the CFD assessment by means of defining a regular design wave for the CFD calculation. A 5.5 m wave height with a period of 4.93s and an 80° heading was used for this calculation. The same condition and regular design wave were used in the LLM and the results are shown in Table 1. For the lower fin, the results are in nice agreement, while for the other two fins the deviation is more than 50%. Model tests are planned to measure forces coming on the fins in the waves to further improve the prediction of the loads.

<table>
<thead>
<tr>
<th>Resultant force [kN]</th>
<th>CFD</th>
<th>LLM</th>
</tr>
</thead>
<tbody>
<tr>
<td>Upper (315°) fin</td>
<td>74</td>
<td>56</td>
</tr>
<tr>
<td>Middle (270°) fin</td>
<td>48</td>
<td>72</td>
</tr>
<tr>
<td>Lower (225°) fin</td>
<td>50</td>
<td>49</td>
</tr>
</tbody>
</table>

**6 STRUCTURAL DESIGN**

Apart from the design aspects of fatigue, lifetime and hull integration, the focus areas for structural design of the EnergoFlow are stiffness, robustness, determination of stresses and avoidance of the resonance. An accurate method of calculation for determining the natural frequencies is necessary.

The starting point is the outer contour resulting from the hydrodynamic optimization. Based on this generic geometry, two structural concepts were developed: a welded structure and a casted variant. A gradual transition in thickness and selection of the stiffeners balance the stress levels required for desired stiffness and robustness.
Investigation shows that both the welded and casted variants are feasible, and even a combined solution is possible. The loadings from the LLM are used as input for the FE assessment. It was observed, that it was very well possible to keep any occurring stresses within 100 MPa Maximum loading of the fins might occur during conditions such as slamming, where typically one fin is exposed to the peak load. Different solutions were reviewed to mitigate the risk for overloading of a single fin. A ring connecting the fin tips was found to be effective to avoid such conditions, since it distributes the peak load from one fin to the others, and lowers the maximum occurring peak stress. Figure 15 shows an example of the stress distribution during a simulation of slamming for a PSS with and without a ring. The calculated max stress within the device is reduced with approx. 40 %.

Resonance can be avoided by keeping a sufficient margin between the excitation frequency (often induced by the rotating propeller) and the natural frequency. The determination of these natural frequencies of the PSS is not that straightforward because of the so-called added mass of the water. This numerical fluid-structure interaction problem was solved by using an acoustic-structural coupling that models the interaction mechanisms of a fluid and a structure (FSI) (Moosrainer, 2009).

Initially, the natural frequencies for the cast and the welded structure were investigated, it was observed that the added mass of water is considerably larger than the mass of the stator and the concepts showed similar natural frequencies. In one particular case resonance frequencies of a single fin was observed in a range of 12-17 Hz.

Similar calculations were conducted by including the ring connecting the tip of the fins and it was observed the ring considerably changes the response on the excitations. The methodology is developed to assess the risk of resonance in different cases.

Figure 16 Natural frequency of a PSS with and without ring in air and in water.

7 FATIGUE

The occurring load and the corresponding number of cycles are the two operating parameters decisive for the determining the fatigue lifetime. The associated stress levels and the frequency of occurrence are the input for the estimation of the fatigue lifetime for the corresponding detailed construction. The chosen method for this will be discussed in this section.

One can use either global or localized stress for fatigue assessment. Using global forces has the downside that localized stress concentrations might not be considered correctly. Therefore the structural hot spot stress approach, as defined by DNV-GL (DNV-GL, 2015) is used.

Fatigue is considered for the steel plates and also the welds. Since the steel welds for the hull integration are in general decisive for the lifetime assessment, we focus on these welds from here on. The structural hot spot approach is based on the joint geometry and the vicinity of the weld and it provides the stresses at the weld toe (Figure 17) two reference points at a certain distance from the weld toe are used to extrapolate the stresses at the weld toe. Due to the complex geometrical structure, a Finite Element assessment is used to calculate the stresses at the reference points.

The International Institute of Welding developed a fatigue resistance S-N curve which defines the maximum fatigue strength at a specific amount of cycles and is shown in Figure 18. The figure shows an additional decreasing
curve in which each stress level corresponds to a specific number of cycles. The decreasing line indicates that the lower the stress level, the longer the fatigue lifetime is. The diagram incorporates various so-called FATigue (FAT) classes. This FAT class represents a specific fatigue strength and the selection of the FAT class depends on the used approach. The structural hot spot approach uses a FAT class of 90 (Hobbacher, 2008).

![Fatigue diagram of steel with various FAT class values.](image)

Figure 18: Fatigue diagram of steel with various FAT class values.

The next step is to cumulatively summate the individual load cases during the lifetime into a total fatigue lifetime. For this, the use of so-called damage ratio at a specific stress level was considered. This ratio represents the safety factor against a predefined lifetime of e.g. 25 years. Since the maximum fatigue strength (red line) and the calculated stresses (orange line) are continuous, the line is divided in multiple load cases. The damage ratio is calculated for each load case by dividing the calculated amount of cycles from the orange line by the maximum amount of cycles indicated by the red line.

\[ D = \sum_{i} \frac{n_i}{N_i} \]  

[6]

Here, D is the damage ratio, i the index for the block number in the load spectrum, n, the number of cycles of the calculated load stress level in load spectrum block i and N, the number of cycles to failure at design stress level obtained from the fatigue resistance curve.

The total damage ratio is the sum of all damage ratios of each load case.

The final step is the calculation of the design fatigue using equation

\[ \text{actual fatigue lifetime} = \frac{\text{design fatigue lifetime}}{D} \]  

[7]

The actual fatigue lifetime will be shorter if the sum of damage ratios is higher than 1 but higher if the total damage ratio is lower than 1.

This derivation shows that the stress concentration should not be larger than 130 MPa in order to maintain a sufficient high fatigue lifetime.

8 HULL INTEGRATION

The connection of the PSS to the vessel’s hull is a critical aspect, especially for retrofits, and therefore needs to be investigated in detail. Merchant ships such as bulk carriers and container ships are generally fitted with one of the two types of stern structures; a heavy stern boss casting or a stern structure with the stern tube integrated into the hull plating and frames. As shown in figure 20. Both have been part of the development.

![Typical stern designs for merchant vessels; stern tube with hull plating (left) and heavy stern boss casting (right).](image)

Figure 19: Typical stern designs for merchant vessels; stern tube with hull plating (left) and heavy stern boss casting (right).

In the due course, the investigations were narrowed down to 2 important aspects.

- Can the loads acting on the fins be transmitted and absorbed by the stern structure?
- The quest for the best method to physically connect the fins to the cast bossing or the stern tube.

It is identified that for a full FE assessment, modelling of the fins and the stern are needed to assess the transmission of the loads and its resulting stresses. As an example case, a typical bulk carrier with a stern tube was taken under consideration. The resulting stress distributions are shown in Figure 20.

![Stress distribution of fins and stern structure for typical bulk carrier with reference load on the fins](image)

Figure 20, Stress distribution of fins and stern structure for typical bulk carrier with reference load on the fins.
Several possible ways for hull integration were investigated and it was concluded that the welded connection between the fins and the stern structure proves to be most beneficial and preferred by the class. The connection provides a large cross section area with a high load carrying capability and apparently it is widely used in the maritime industry.

Strength analysis showed that the highest structural strength is generated when the complete structure including internal stiffeners are welded directly to the stern boss casting or the stern tube.

CONCLUSION
- It is possible to design a pre-swirl stator showing a considerable power reduction as well as meeting fail safe design requirements.
- Curved blades combine a large increase in propulsive efficiency due to local fine-tuning of the pre-swirl, in combination with low added resistance. This leads to the largest possible power reduction and fuel savings.
- A ring connecting the tip fin reduces the vortices leaving the fins and increases the structural robustness of the device in terms of max occurring peak stresses.
- The loads acting on the fins can be quantified via the ship motions occurring during the lifetime of the vessel.
- The EnergoFlow can be installed both on newly built vessels and also be used as a retrofit. For retrofits, a welded connection to the cast bossing or stern tube is strongly recommended. A full FE assessment of the stern is needed in order to judge fail-safe operation during the lifetime.

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REFERENCES


