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HULL VANE® VERSUS LENGTHENING

A comparison between four alternatives for a 61m OPV

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Abstract

The patented Hull Vane® is a fixed hydrofoil that can be mounted near the transom of the vessel, aimed at reducing the resistance of the vessel. Recent CFD studies, model tests, and full-scale trials have shown that the Hull Vane® is a very effective energy saving device for fast displacement vessels.

In a typical retrofit installation (attached to the transom), the Hull Vane® increases the total length of a vessel. In this paper, the question is answered whether lengthening the vessel by the same amount can achieve comparable results in case of a newbuilding. For this purpose, four versions of a state-of-the-art 61m patrol boat design by DAMEN are compared in terms of resistance and propulsion. These resistance and propulsion calculations are based on a series of CFD computations.

The results show that for a given operational profile, a Hull Vane® fitted behind the transom is the most beneficial solution in reducing the fuel consumption of the vessel. Incorporating the Hull Vane® within the existing length of the vessel also proves to be more beneficial for the yearly fuel consumption than extending the overall length of the vessel.

Introduction

The Hull Vane® is a primarily transverse wing mounted near the transom of a vessel. Its effectiveness in reducing resistance of fast displacement vessels has been proven in various CFD-studies, towing tank tests, and full-scale trials [1], [2], [3]. The Hull Vane® (HV) can be retrofitted to existing ships or integrated into new designs. The first option usually results in an increased overall length, since the wing is positioned behind the transom for maximum efficiency. This has posed the question if a mere extension of the ship to the same length yields the same resistance reduction and the associated benefits in fuel consumption and speed.

In this paper, four versions of a patrol boat design from DAMEN are compared in terms of resistance and propulsion. Furthermore, the power requirements and required fuel capacities to achieve certain ranges are checked against each other before calculating annual fuel consumption and emissions based on a given operational profile.

The paper is written assuming a project in the early design stages of a newbuild program. In this case, the naval architect has all options to choose from. Some of the options, such as the appended Hull Vane®, are also applicable to existing ships.

Candidates

Figure 1 shows a comparison of the side views of the four candidates analyzed in this study:

1. The benchmark vessel (L_{OA}= L) with optimized trim wedge
2. The retrofitted benchmark vessel, with the trim wedge removed and a typical appended Hull Vane® mounted on the transom (L_{OA} = L+L_{HV})
3. The modified benchmark vessel with a Hull Vane® integrated within the length of the vessel (L_{OA} = L)
4. The benchmark vessel extended to the same immersed length as version 2 (ΔL_{WL} = L_{HV}), with trim wedge

Candidates 1, 2 and 3 feature equal hull lengths. The lengthened benchmark has the same immersed length as the hull with the Hull Vane® appended to its transom.

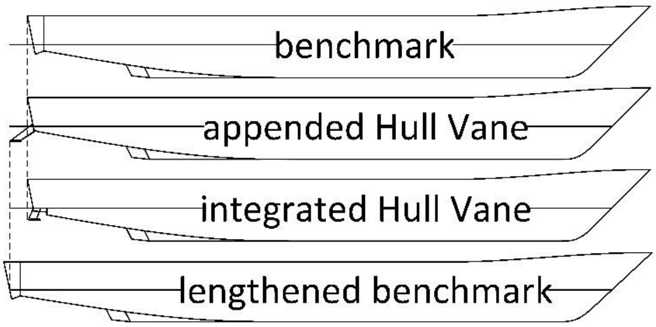

Figure 1 - Side views of design candidates

Figures 2 and 3 show a typical configuration of an appended and an integrated Hull Vane®, respectively. In case of the integrated Hull Vane®, a small recess is created in the aftship above the Hull Vane® to avoid an unfavorable interaction between the Hull Vane® and the hull.

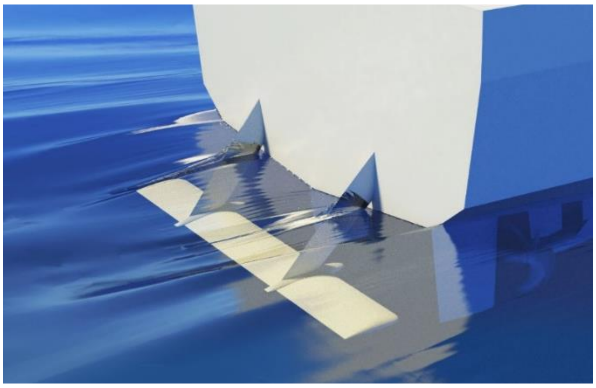

Figure 2 – Typical appended Hull Vane®

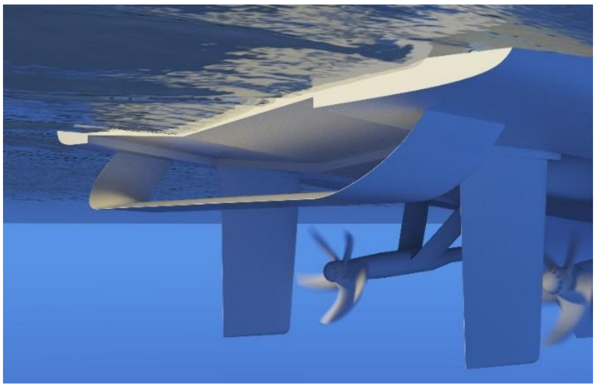

Figure 3 – Typical integrated Hull Vane®

The lengthened benchmark design was created by separating the trim wedge from the benchmark hull and stretching the remaining part of the hull longitudinally. The trim wedge was then added back on followed by a vertical scaling of the hull to reduce the displacement gain, because although some additional structure and hull surface is associated with the lengthening, it is important that - similar to the Enlarged Ship Concept [4] - the increased length and interior volume is not used for additional systems or interior, as this will increase the displacement, cost, and capabilities of the vessel, making the comparison invalid.

Table 1 lists characteristic lengths of the analyzed design candidates.

Table 1 – Characteristic lengths of design candidates

design [-]	L_{OA} [m]	L_{WL} [m]	$L_{immersed}$ [m]
benchmark	61.01	56.66	56.66
appended HV	62.75	57.05	58.97
integrated HV	61.01	56.66	57.13
lengthened benchmark	63.47	58.97	58.97

Method

Calm water resistance calculations have been performed using the FINE/Marine RANS-CFD software by NUMECA at speeds of 26 kts (top speed), 17 kts (intermediate), and 10 kts, employing a volume of fluid approach and the SST-Menter turbulence model. The symmetry about the centerline has been exploited by modeling only the portside half of the ship and applying suitable boundary conditions as per best practice settings recommended by NUMECA. A velocity along the longitudinal axis has been imposed and the pitch and heave motions were resolved, with all other degrees of freedom remaining fixed. At a later stage, CFD computations at 13, 20, and 23 kts were added to allow for more precise interpolation between the results.

One mesh per design candidate per speed has been created (24 meshes in total) to ensure meshes were suitable to Froude and Reynolds numbers at each resistance point. Vertical refinements were added to both sides of the free surface with additional longitudinal refinements inside the Kelvin angle to capture the ship's wave system. So-called viscous layer cells were employed to accurately resolve the boundary layer, with target y+ values ranging from approximately 100 at 10 knots to 200 at 26 knots. Mesh sizes ranged from about 1.8M cells for the hulls with trim wedge to 3.6M cells for the cases featuring a Hull Vane®. A typical mesh of the hull surface is shown in Figure 4.

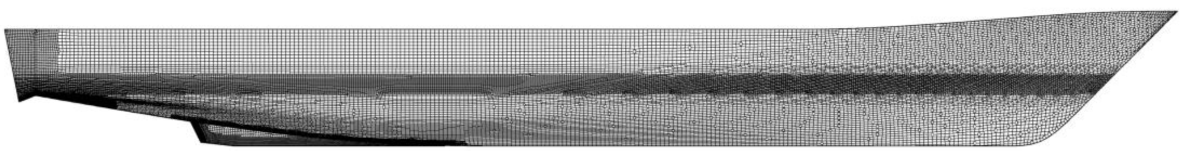

Figure 4 - Typical hull surface mesh

The results have been adjusted for additional friction (correction for surface roughness) as per ITTC guidelines [5]. The data has subsequently been used to tune empirical resistance estimates and generate a full resistance curve which in turn represents the input for a resistance and propulsion analysis to determine power requirements, fuel consumption, and achievable ranges. Range computations are based on a net fuel capacity of 90m^3 as typically found on the benchmark design.

A theoretical operational profile was provided by DAMEN and is presented in Table 2 along with two alternative profiles. Most of the sailing is expected to happen at speeds close to the cruising speed of 15 knots. The annual sailing hours are estimated at 4000 hours. Using this operational profile and the results from the resistance and propulsion analysis, the annual fuel consumption and emissions can be calculated.

Table 2 - Operational profiles

	operational profile	cruising speed profile	high speed profile
speed [kts]	sailing time [%]	sailing time [%]	sailing time [%]
26	6	6	10
20	0	10	15
17	47	15	44
15	0	44	15
13	0	15	10
10	47	10	6

Results

Resistance results

Figure 4 shows the total bare hull resistance coefficients of all candidates. To obtain non-dimensional coefficients, the resistances are normalized by benchmark wetted surface area and dynamic pressure. The coefficients include frictional and pressure resistance components of the hull, but no aerodynamic or appendage resistance. As can be seen, the benchmark hull has the highest resistance at all speeds. The Hull Vane® equipped hulls experience significantly lower resistance at low velocities. The range of values decreases with increasing velocity. At the same time, the performance of the extended benchmark hull improves until it becomes the most efficient hull at the top speed of 26 knots.

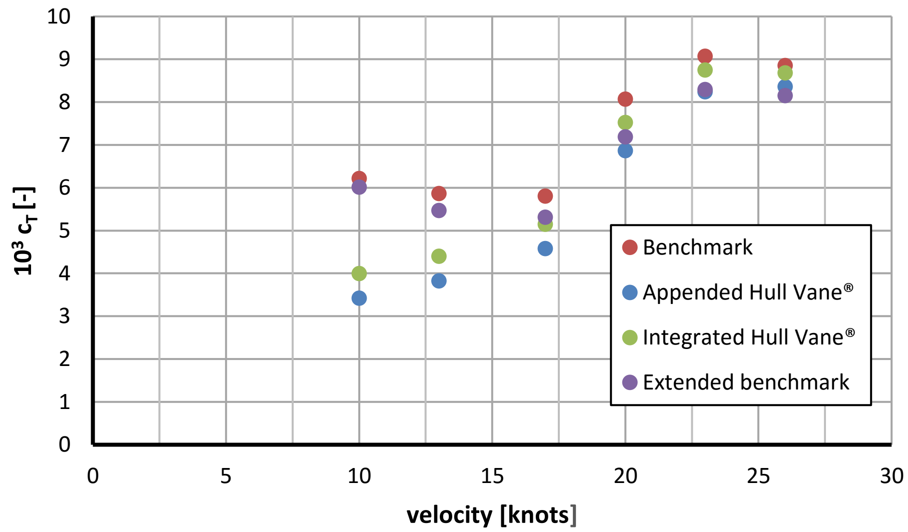

Figure 5 - Bare hull total resistance coefficients (calculated with benchmark wetted surface area)

Table 3 lists the resistance coefficients for all candidates.

Table 3 - Bare hull total resistance coefficients
(Fn based on benchmark L_{WL}, C_T based on benchmark wetted surface area)

		Benchmark	Appended Hull Vane®		Integrated Hull Vane®		Extended benchmark	
Velocity		10^3 C_T	10^3 C_T	R reduction	10^3 C_T	R reduction	10^3 C_T	R reduction
[knots]	**[Fn]**	[-]	[-]	[%]	[-]	[%]	[-]	[%]
10	**0.22**	6.21	3.42	45%	3.99	36%	6.02	3%
13	**0.28**	5.86	3.82	35%	4.40	25%	5.46	7%
17	**0.37**	5.81	4.58	21%	5.15	11%	5.31	9%
20	**0.44**	8.06	6.87	15%	7.53	7%	7.19	11%
23	**0.50**	9.07	8.24	9%	8.75	4%	8.29	9%
26	**0.57**	8.86	8.36	6%	8.68	2%	8.15	8%

Relative resistance results

Figure 5 shows the resistance differences of candidates two to four in relation to the benchmark hull.

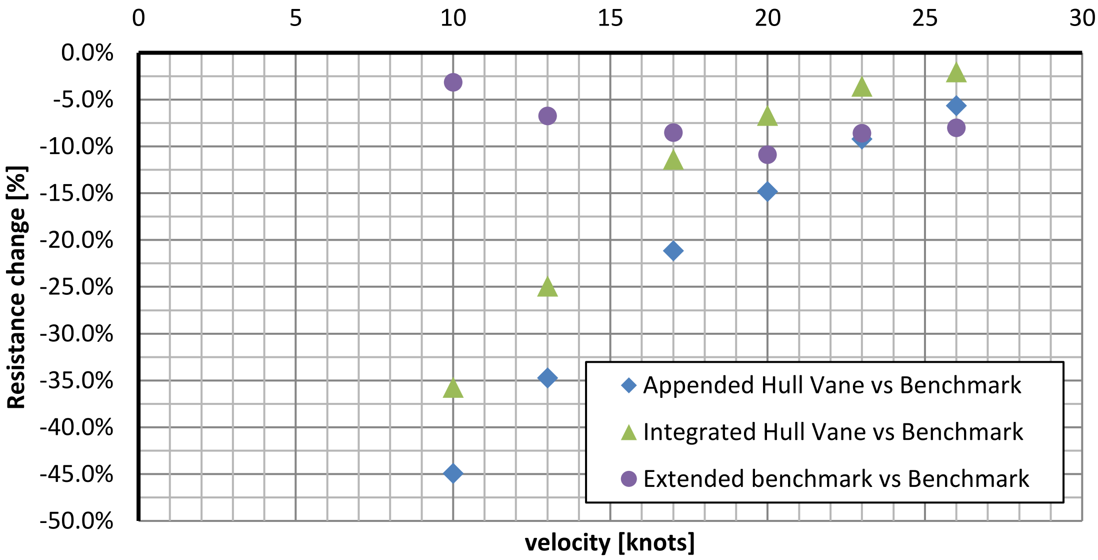

Figure 6 - Resistance reductions over benchmark

The Hull Vane® equipped hulls have significantly lower resistance, especially at the lower speeds. The integrated HV decreases the resistance by 36% at 10 knots, whereas the appended Hull Vane® cuts it down even further to achieve a reduction of 45%. In contrast, the lengthened hull only offers a small resistance reduction of 3% at 10 knots.

The resistance reductions achieved with the Hull Vane® seem very high at first. However, prior research shows that a trim wedge is very unfavorable at low Froude numbers [1], since it creates a large dead water zone behind the transom. Hence part of the gain can be attributed to the removal of the trim wedge for the applications of the Hull Vane®. This also explains why the extended hull, which uses a very similar wedge, performs only marginally better than the benchmark at low speeds. At the intermediate speed (17 knots), the result of the extended hull improves, leading to an advantage of 8.5% over the reference hull. The integrated Hull Vane®

exhibits a slightly superior performance with a resistance reduction of approximately 11%. The appended Hull Vane® gives the highest benefit with a 21% lower resistance compared to the benchmark.

At the top speed (26 knots) both the integrated and appended Hull Vane® offer an advantage over the reference hull of 2% and 6%, respectively. At this speed, the extended hull performs best with an 8% resistance benefit over the benchmark.

Comparison of forces

Figure 6 shows a comparison of pressure forces acting in the longitudinal direction along the hulls of all four candidates at a velocity of 17.0 knots. It can be seen that there is no significant difference between the first three candidates from forward until about the end of the skeg (negative peak at about 20% L). The pressure resistance curve of the extended benchmark differs from the others due to the stretching of the hull which among others gives a finer entry and a different skeg position. Hence it experiences slightly less resistance at the front and the favorable pressure starts further aft. However, the biggest differences can be found at the transoms. The trim wedges of the benchmark and extended benchmark cause significant resistance peaks whereas the Hull Vane®-equipped hulls show considerable negative resistance peaks (i.e. forward thrust) where the Hull Vanes® are positioned. Furthermore, it can be seen that the favorable pressure areas created in front of the trim wedges cannot offset their resistance.

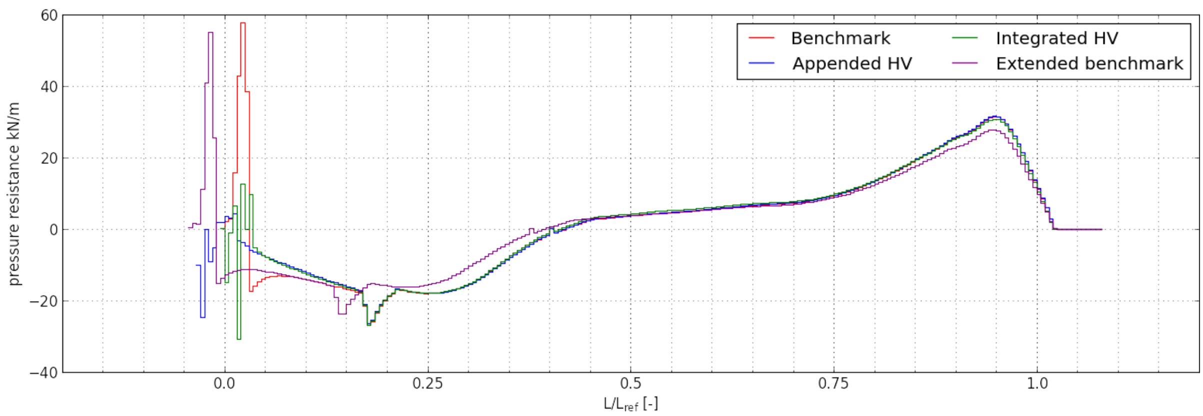

Figure 7 - Comparison of local pressure resistance along the hulls at 17 knots

Figure 7 shows a similar plot as above for frictional resistance. Again, there is very little difference along the forward two thirds of the hulls. The stretched skeg of the extended benchmark is evident in the plot. The most prominent differences can once again be spotted towards the back of the ships where the Hull Vane® creates a significant friction peak. Additionally, the effects of deceleration of the water in front of the trim wedges as well as the acceleration to pass them can be observed. The resulting local maxima are small in comparison to those caused by the Hull Vane®. There is a clear frictional resistance penalty associated with the Hull Vane®. However, since the frictional resistance scale is more than one order of magnitude smaller than the pressure resistance scale, this penalty is more than offset by the reduction in pressure resistance.

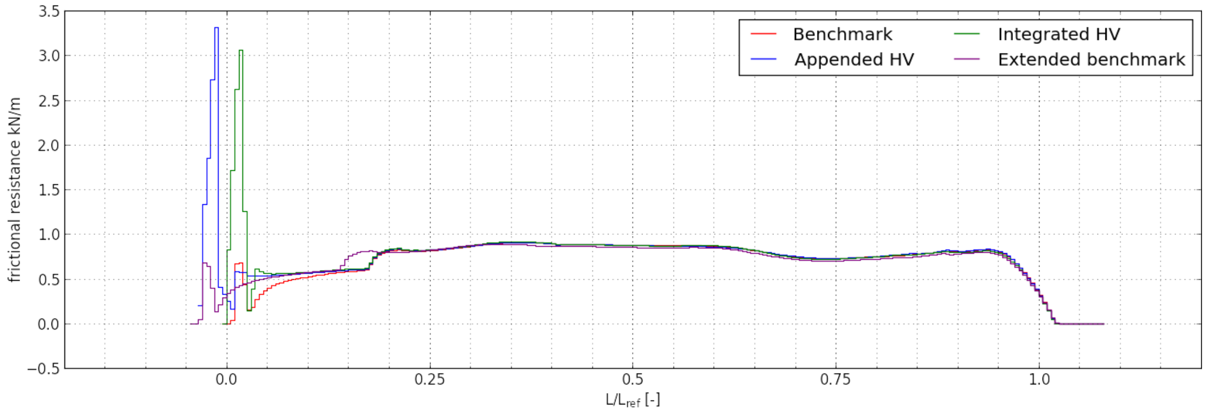

Figure 8 - Comparison of local frictional resistance along the hulls at 17 knots

Maximum speed and power demand

Combining the relevant input data from DAMEN with the resistance results yields enough information to perform a propulsion calculation on all four alternatives. The propellers were adapted to each candidate design for maximum propulsive efficiency under the constraint of a constant drivetrain configuration. Estimates have been applied for windage and appendage resistance to determine total resistance. The propeller diameter and tip clearance were assumed to be constant (no draft restriction). The resulting relative power/speed curves of these calculations is displayed in Figure 9.

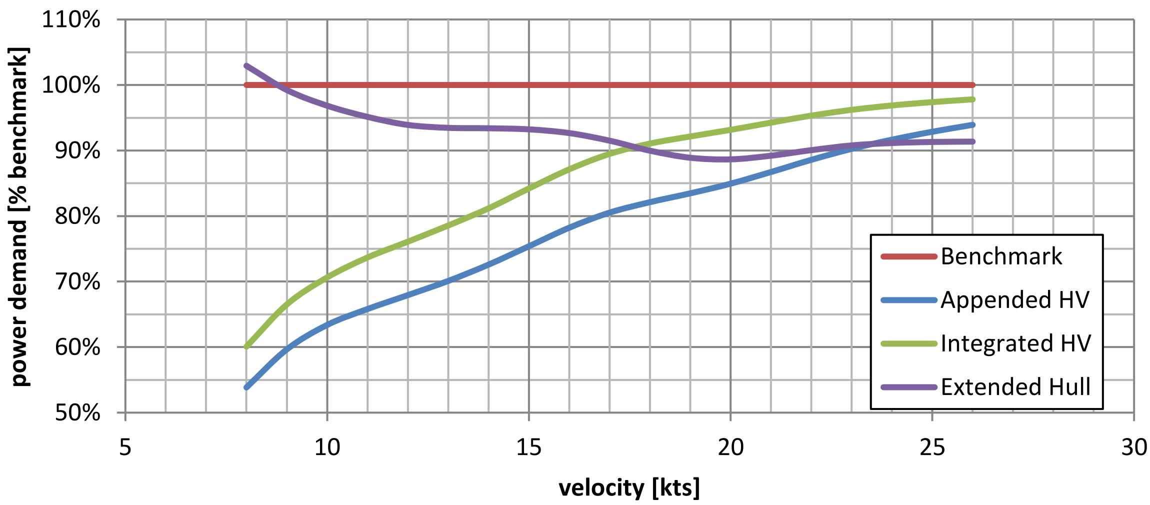

Figure 9 – relative power demand as function of velocity

For an installed power of 11.520kW the extended benchmark would reach a top speed of 27.0 knots. The geometries with the appended Hull Vane® and integrated Hull Vane® have top speeds of 26.6 and 26.3 knots respectively, while the benchmark vessel reaches the required design top speed of 26.0 knots.

Looking at the same results from a perspective of a reduction of installed power to achieve the required top speed of 26.0 knots, the extended benchmark has the lowest required installed brake power: 10,488 kW. The appended Hull Vane® follows at 10,782 kW, while the integrated Hull Vane® and the benchmark require 11,228 kW and 11,480 kW, respectively.

Fuel consumption and range comparison

With the propulsion data available, conclusions can also be drawn regarding the fuel consumption and the operational range that results from this fuel consumption. In Figure 10, the relative fuel consumption as a function of speed is displayed for all four alternatives.

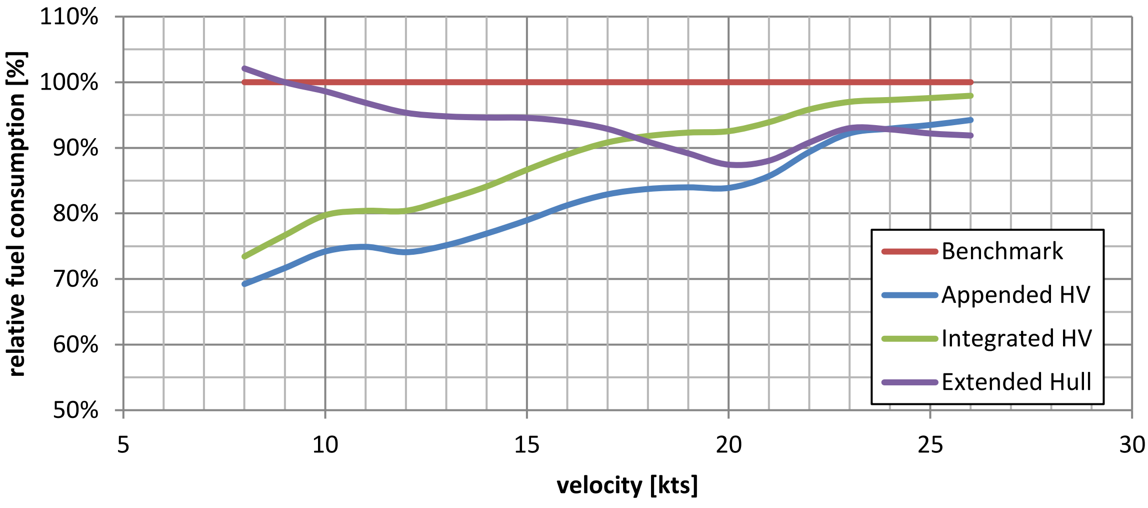

Figure 10 – relative fuel consumption as function of velocity

At higher speeds, the extended benchmark and the appended Hull Vane® clearly outperform the benchmark and the integrated Hull Vane®. At lower speeds however, the extended benchmark and the benchmark have a significantly higher fuel consumption than the alternatives with Hull Vane®. This difference becomes more obvious when looking into the resulting range of the vessels, as displayed in Figure 11.

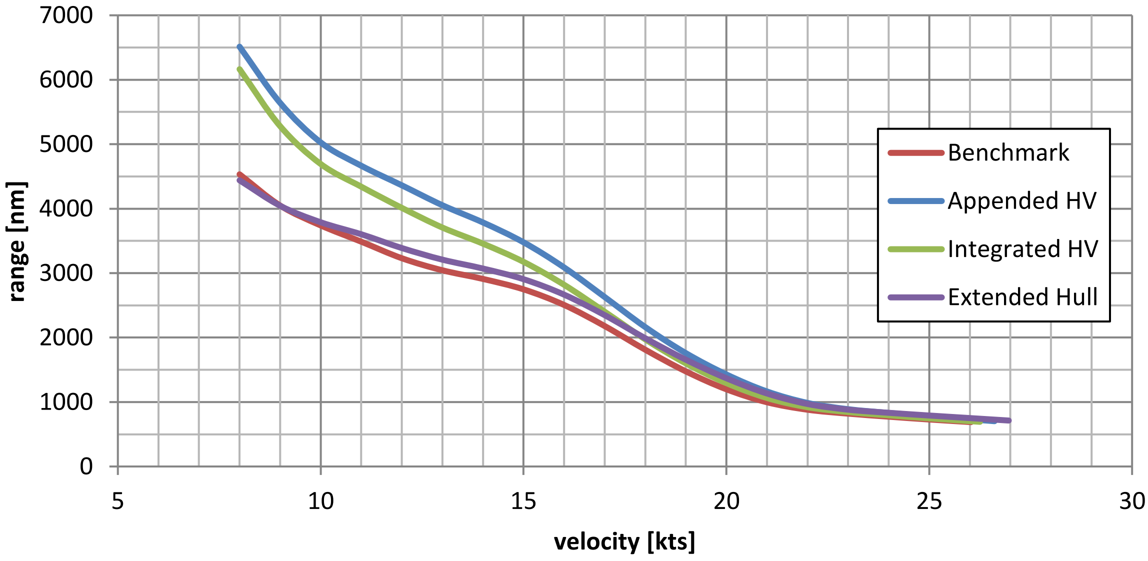

Figure 11 - Range as function of velocity

At a speed of 10 knots, the range increases from 3740 nm to 5028 (+34%) if a Hull Vane® is appended to the Benchmark. The extended benchmark has a marginally longer range than the original benchmark, with a range of 3789, while the alternative with integrated Hull Vane® reaches 4693 nm at 10 knots. The differences diminish with increasing speed. However, at the cruising speed of 15 knots the appended HV and the integrated HV still offer ranges that exceed the benchmark range by 27% and 16%, respectively. The range of the extended benchmark is 6% higher than the benchmark at cruising speed.

Ranges can also be viewed from the perspective of maximum attainable speeds and minimum transit times for certain passages like a typical transatlantic crossing from the Canary Islands to Antigua. With a small margin, this is a distance of about 2750nm, which the benchmark hull can cover at its cruising speed of 15knots in about 7days and 15.5 hours. The extended benchmark can sail slightly faster at 15.7knots, reducing the transit time to 7 days and 7.2 hours. For the hull with the integrated Hull Vane® the required time decreases further to 7days and 2 hours. With the appended Hull Vane® the passage can be made in less than 7 days (6 days and 20 hours) at a speed of 16.7 knots.

Annual fuel consumption and emissions

The results from the resistance and propulsion analysis have been used to determine the annual fuel consumption and emissions based on the operating profile specified in Table 2. Fuel consumption is calculated by weighing the fuel consumption at the discrete individual speeds with the associated sailing time and subsequent multiplication with the yearly sailing hours as per equation 1.

$$FOC = \left(\sum FOC_{v_i} * sailing\ time_{v_i}\right) * total\ sailing\ hours \tag{1}$$

The annual fuel consumption for each design is listed in Table 4. As can be seen in the table there is a significant fuel saving potential from the extension of the hull of about 6%. The integrated Hull Vane® offers fuel savings of 9% while the ship with appended Hull Vane® is the most economic, saving 15% compared to the benchmark.

For the computation of the CO_2 emissions a fuel density of 890kg/m3, a carbon content of 86.7% [6] and an oxidization rate of 99% have been assumed. This gives an emission factor of approximately 2.8t of CO_2 per cubic meter of fuel consumed. As can be seen in the table almost 1000 t of CO_2 can be saved per year by applying the appended Hull Vane®.

$$CO_2 - emmissions = FOC * 2.8t/m^3 \tag{2}$$

Table 4 - annual fuel consumption and emissions

design [-]	fuel consumption [m^3/y]	fuel saving [%]	CO_2 emissions [t/y]
benchmark	2329	0%	6524
appended HV	1979	15.1%	5542
integrated HV	2122	8.9%	5944
extended benchmark	2179	6.4%	6104

The annual fuel consumption has also been calculated for the two alternative operational profiles to check the sensitivity of the results to changes in the operational profile. The results are presented in Table 5. It can be seen that the fuel savings are not overly sensitive to changes in the operational profile as the fuel saving values stay within a band of two percentage points over all operational profiles. Furthermore, the appended Hull Vane® offers the highest fuel savings regardless of the operational profile. The extended benchmark becomes more competitive with an increasing emphasis on speed.

Table 5 - Comparison of annual fuel savings

	operational profile	cruising speed profile	high speed profile
design [-]	fuel saving [%]	fuel saving [%]	fuel saving [%]
benchmark	0.0%	0.0%	0.0%
appended HV	15.1%	15.8%	13.7%
integrated HV	8.9%	9.1%	7.2%
extended benchmark	6.4%	7.7%	8.4%

Discussion

Significant resistance reductions over the benchmark hull have been found for all three design alternatives which result in significant fuel savings. For a newbuild project, range and top speed are generally fixed from very early on. The savings will lead to a reduction in tank capacity (hence more useable interior volume) and a reduction in power (hence cost and space savings on main engines, exhaust systems and ventilation systems). In the case of a modification to an existing design, the savings will lead to a higher top speed and an increase in range. In all cases, the lifecycle costs are significantly reduced.

Furthermore, there are secondary effects associated with reducing installed power and fuel capacity in form of weight savings. Hence the displacement of the ship can be lowered resulting in even higher benefits. Although iterating through these design loops is outside the scope of this paper, the additional gains can be estimated. For example, if the range is selected to be fixed, carrying 15% less fuel enables a reduction in displacement of about 1.5%. The potential gains from a smaller main engine and machinery are expected to be of similar magnitude. Thus, these weight savings can translate into further relevant reductions in building and operating cost.

Conclusion

It has been shown that considerable savings in resistance can be achieved by applying one of the two variants of the Hull Vane®, or a hull extension to the OPV design over the whole speed envelope of the original design. For the Hull Vane® this is especially true for the low to medium speeds. The extended hull only offers a moderate advantage at these velocities. If the length-over-all is fixed, the choice is limited to the benchmark and the integrated Hull Vane®.

The reductions in resistance translate directly into fuel savings and range extensions. For the given operational profile, the appended Hull Vane® offers the highest annual fuel savings of about 15% over the benchmark, followed by the integrated Hull Vane® with 9% and the extended benchmark with about 6% lower annual fuel consumption. The advantage of the Hull Vane® can be traced back to the fact that it is typical for naval and coastguard ships to require a high top-speed, although they mostly sail at intermediate or low speeds. The longer range is considered very important, since the provisioning of warships can be very difficult during wartime.

The choice between any of the four alternatives will usually be limited by whether or not there is a length-over-all restriction. For existing ships, the appended Hull Vane® is the most logical choice, as it can be retrofitted in a very short time. For newbuilding projects, the resistance reductions offer the opportunity to decrease installed power and displacement which will lower the building cost and further enhance the economic benefits during operation.

Prior research has shown that both a hull extension and the Hull Vane® also generate considerable benefits for the seakeeping of a ship, most notably in the reduction of vertical accelerations. It is beyond the scope of this paper to quantify and compare these effects.

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