Full-scale performance of the SES Motion Control System

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Figure 1 - Umoe Ventus, at the wind farm Borkum Riffgrund 1, Germany, and in the fjords of Mandal, Norway, by courtesy of Dong Energy and Umoe Mandal, respectively.

The pressurized air cushion on a Surface Effect Ship (SES) can lift up to 80% of total vessel mass.The SES Motion Control System (SES-MCS) controls the vent valves which again controls the air cushion pressure, assuming lift fan air flow is pressurizing the air cushion. By controlling the air cushion pressure one can significantly counteract vertical sea wave disturbances, ensure high passenger comfort and reduce sea-sickness. The case studied in this work is the Umoe Mandal Wave Craft prototype, 'Umoe Ventus', which is a high-speed offshore wind-farm service vessel specially designed for control in the vertical plane. The SES-MCS can adjust the draft from 1m to 3.2m in less time than the wave period. The SES-MCS can reduce motions significantly in order to perform Operation and Maintenance (O&M) in high seas. The craft is the fastest wind-farm service vessel of its size with high comfort in all relevant sea states. The performance of the SES-MCS is demonstrated through full-scale sea trials.

KEY WORDS
Surface Effect Ship (SES); Ride Control System (RCS); Boarding Control System (BCS); Marine Control Systems; Offshore wind; Accessibility; ship control.

INTRODUCTION
Several hundred SES-contributions have been published the past 55 years and over 550 SES have been developed and have been operative. Clark et al (2004), Butler (1985) and Lavis (1998) present three comprehensive articles on the SES history.

The modern development of the SES dates back to the early 60s. In 1959 the British initiated a commercial SES (Tattersall 1982) with the pioneer vessel D1, build by Denny & Bros. in cooperation with Hovercraft Development Limited. The vessel was a solid sidewall "hovercraft", launched in 1961. Simultaneously, in the U.S., the Naval Air Warfare Research Department of the Naval Air Development Center (NADC), led by Mr. Allen Ford, developed the XR-1. The XR-1 was a hovercraft with rigid side-hulls delivered in 1963 and went through eight major modifications before the XR-1E was launched 22 years later (Ford 1968). From 1965 to 1972, the US Navy focused on vessels with low cushion length-to-beam (l/b) ratio, including the XR-1B, XR-2, XR-3, SES-100A and SES-100B which all contributed to the ultimate goal: construct a 3K ton, 80 knot, SES. The XR-3 was developed and constructed at the David Taylor Naval Research Centre. In 1969, contracts for development and construction of the SES 100-A and SES 100-B were awarded to Aerojet General and Bell Aerospace, respectively. These craft weighed 100 -tons and regularly achieved speeds around 80 knots. After a hull modification in 1978, the SES 100-A served as a scaled vessel for the 3K ton SES. Unfortunately, the development of the 3K SES was terminated in 1979, only weeks before the hull construction were scheduled to start. The SES-100B, a 77 ft craft was launched in 1974 and sustained an impressive speed recorded of 94 knots; see Figure 2.

Figure 2 - SES-100B by courtesy of the U.S. Navy

From 1973 to 1982, the US Navy experimented with higher cushion l/b such as the XR-5 (l/b = 6.8) and the BH-110 which was later redesigned and became the SES 200 (l/b = 4.25). The main benefit of a low l/b is to cruise above the “hump” speed in cushionborne operation. However, these designs require high drag when exceeding the primary hump condition. A higher l/b ratio reduces the required drag force when passing the hump, lowers the operational cost below the hump speed and increases seakeeping. After a commercial breakthrough for SES ferries in the 80s the interest for SES, both military and commercial, has decreased. To the authors’ knowledge, the only successful yard developing SES the past 20 years, is Umoe Mandal (former Kvaerner Mandal) which has delivered 12 craft to the Royal Norwegian Navy, including the world fastest armed vessel: the Skjold-class which exceeds speeds of 60 knots. In 2015, the yard was introduced to the commercial marked when launching the Wave Craft series.

The majority of the mentioned vessels had, or has, some sort of automatic control of the air cushion pressure. An un-stabilized SES Vessel will 'bounce' at its natural spring mass damper frequency, and also can excite its various acoustic modes requiring a Ride Control System (RCS) during transit. While competing with other companies, Maritime Dynamics Inc., delivered the first functional RCS (Adams 1984). Additional work on the RCS, among others, is presented in (Kaplan and Davis 1978) and (Sørensen and Egeland 1995).

In this paper we present a novel method for low-speed, wave frequency vertical motion damping performed on a full-scale SES. We denote this system the Boarding Control System (BCS). The data is taken during operation in high-seas, next to an offshore wind-turbine.

This article deals with the two modes: the RCS and the BCS which together defines the SES-MCS. They can be manually toggled by the captain on the Wave Craft series, depending on the encountered sea-wave amplitude- and-period. The structure of the SES-MCS is summarized in Figure 3.

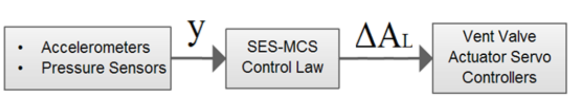

Figure 3 – The SES Motion Control System

What is a SES? The SES, or Surface Effect Ship, is a rigid side hulled hovercraft. The shape of the twin hulls resembles a typical catamaran-hull. The forward and aft end contains flexible structures denoted ‘seals’. The typical, generic seal setup consists of a multiple-loop, flexible rubber bag in the stern and bow fingers in the forward end. The hull, seals and the water surface below the craft forms a large air pocket, also called the air cushion. The air cushion is pressurized using air flow effectors, typically centrifugal lift fans, which can reduce draft significantly. Properly done, a SES vessel can achieve very high speeds while maintaining very high transport efficiency. Lift power is required to maintain the pressurized air cushion but the reduction in resistance at high speed results in less required propulsion power: the overall required power on a SES is less compared to the equivalent monohull traveling at the same speed. A generic SES is shown in Figure 4.

Figure 4 – A generic SES

The air cushion pressure on a SES can typically lift zero-to-eighty percent of the total vessel mass. These two extremes are often referred to as the hullborne- and cushionborne-mode, see Figure 5. The air leakage area out of the cushion is controlled using a set of ventilation valves

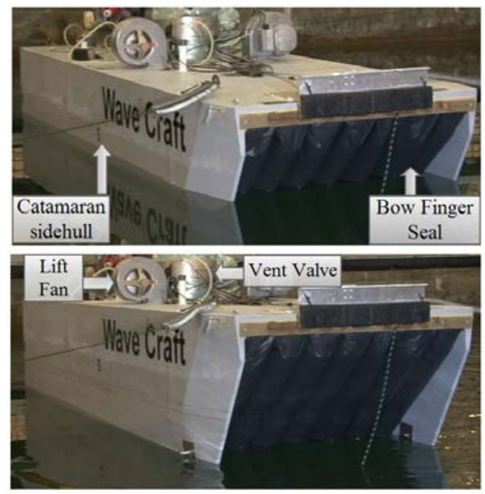

Figure 5 – Top: hullborne mode, bottom: cushionborne-mode

The SES-MCS automatically controls the air cushion pressure to reduce vertical motions:

- In low speed and high seas, the sea wave frequency is dominating the vessel motion frequency. In this case, we present the BCS which operates around a semi-cushionborne mode. This mode has shown to be beneficial in terms of fuel consumption, minimization of water spray, and performance. The enhanced performance of a semi-cushionborne vessel, compared to the two extremes, includes a beneficial compromise between a “jumpy cushionborne SES in low speeds" and “sufficient available cushion pressure for conducting vertical motion damping”.
- During high-speed, it is desired to minimize the draft to significantly reduce the resistance to forward motion; hence the cushionborne mode is preferred. A RCS is required to avoid uncomfortable accelerations, mainly in the following modes:
    - The fundamental heave-to-cushion pressure resonance, which is spatially constant and excited by an encountered wave train of the

same frequency. Mathematically calculated to occur at 1.2 Hz (Kaplan and Davis 1978) on our test craft.

- The 1st acoustic mode, which is a half-wave response with cushion pressure maxima at the bow and stern. Mathematically calculated to occur at 8 Hz (Sørensen and Egeland 1995) on our test-craft.

The paper is organized as follows:
The first section characterizes the cushion response to vent valve motion and therefore deals with system identification. The second section presents the BCS. The full-scale result sections, presents the performance of the SES-MCS system while the work is concluded in the final section.

SYSTEM IDENTIFICATION

Figure 6 illustrate the feedback system for the SES-MCS.

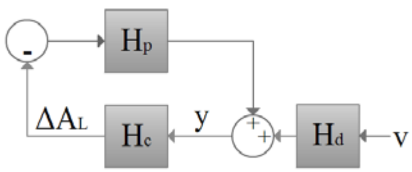

Figure 6 – Feedback system

Table 1 presents the notation and description of the symbolic terms used in this paper.

Table 1

Symbol	Description
H_P, H_c, H_d	Process-, control- and disturbance-operator, respectively. The operators, or transfer functions, maps the input- to the output-signal.
y	The measurement signal provided by accelerometers or (cushion) pressure sensors.
$\Delta A_L, A_L$	ΔA_L is the controller signal and commanded dynamic air cushion leakage area. The total cushion leakage area: $A_L = \Delta A_L + A_0$, where A_0 is some constant. In practice, we control the position of vent valves to obtain ΔA_L.
v	The vertical disturbance vector. Consist of air cushion sea wave volume pumping and hydrodynamic excitation force in heave and pitch.
c	A scalar parameter to normalize y and ΔA_L in Figure 6 and Figure 7.
μ_u	Dynamic air cushion pressure. $\mu_u = \frac{P_u - P_0}{P_0}$, where P_u and P_0 denotes air cushion excess- and ambient-pressure, respectively.
acc_CG	Vertical acceleration at center of gravity (CG).
acc_BOW	Vertical acceleration at the vessel bow tip.
$z_{C/O}$	Vertical displacement at the vessel bow tip

The closed-loop control objective is to design a high-performance, stable controller H_c that damps vertical motion. In order to do so, information regarding H_p is preferred as this reveals, or highlights the complex magnitude and phase relationships attributable to fan dynamics, air inertia and flow lags and points to the need for carefully determining the response of the air cushion over the frequency range of interest. Auestad et al (2014; 2015) presents a time domain, control plant model using mathematical formulation of the SES in- and not in-contact with an offshore structure. H_p can be extracted from these equations. However, such a model suffers from parametric- and linearization-uncertainty, and is not necessary to develop the required process response model (H_p) for the heave and cushion pressure in the frequency range of interest; we can simply obtain this information using forced response system identification tests performed directly on the full-scale craft. Such an open loop test provides the required information for designing and populating the parameters in the control law which could include filters, scalars, tunable integrators etc…

The open loop test is characterized by the following properties and the result is shown in Figure 7:

- The test was intentionally performed in calm seas, hence $v = 0$.
- The vent valves (ΔA_L) are following a sinusoid with varying frequency. We are measuring the corresponding response in cushion pressure ($y = \mu_u$). The time series for ΔA_L and y were stored. In this case $y = \mu_u$, but if one wanted to capture the response between vent valve to heave acceleration, we simply set $y = acc\_CG$ and use data from an accelerometer mounted at CG.
- The ticks on the y-label are unified in this paper, since our goal is to illuminate the identification procedure and show the effect of the control system (on/off). Hence, Umoe Ventus performance is not shown directly, except for in Figure 11.
- Based on experimental-measured results for H_p we use Matlab System Identification toolbox to estimate a 5th-order polynomial, mathematical transfer function for H_p which is given in Appendix A. See Figure 7.

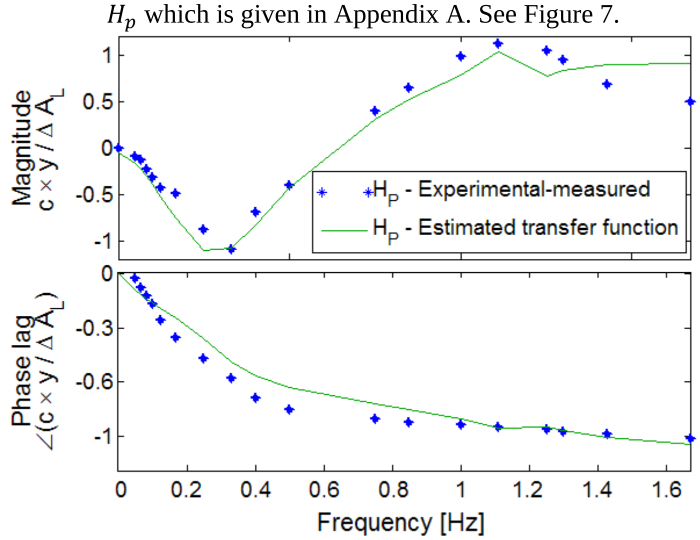

Figure 7 – The response between the air cushion pressure and vent valve dynamics.

As mentioned, the point in this paper is not to dive into quantitative data, but Figure 7 provide useful information:
The response of the vessel to vent valve activity can be seen to always lag the change in vent position and that phase lag increases quite quickly over the low-frequency range. Typically, the results are used to design a robust filter algorithm that compensates for the phase delay and reduced amplitude response in the range we want to control.

THE BOARDING CONTROL SYSTEM

The BCS controls the air cushion pressure to counteract vertical motion induced by v. v is mathematically formulated in (Auestad 2015) and is a sum of the following sea wave disturbances:

- excitation force in heave,
- excitation force in pitch,
- sea wave volume pumping.

As a consequence, the BCS damps the dominating disturbance. For instance, if the sea waves are of such a character that volume pumping is dominating (typically a short wave period), then large cushion pressure (P_u) variations occur in the air cushion, leading to vertical motion. This is the case in Figure 8 where a 1/8-scaled model-craft is thrusting towards a wind-turbine. When the BCS is OFF, the volume pumping effect denies the bow tip of the vessel ($z_{C/O}$) to remain fixed towards the turbine structure. When the BCS is ON, the wave volume pumping effect is damped which results in a bow that is in fact fixed to the turbine. This allows for a safe transport environment for the service-personnel to board the turbine.

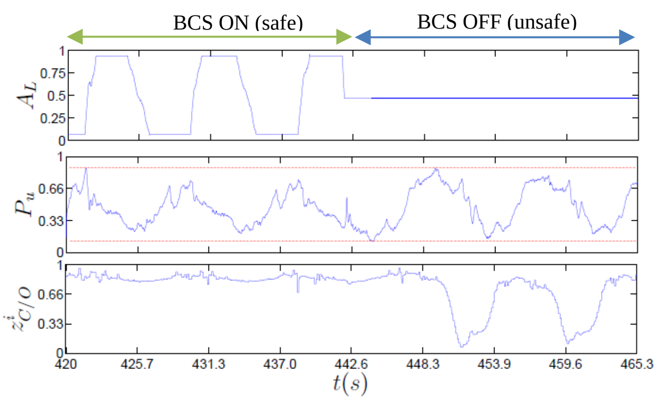

Figure 8 – Wave volume pumping is dominating motion.

Figure 8 shows that motion is reduced, and therefore turbine-access is possible, by decreasing cushion pressure amplitude. Figure 9 illustrate a somewhat oposite case where the BCS is increasing the cushion pressure amplitude.

The reason behind this phenomen (Figure 9) is that the sea wave disturbance is dominated by excitation force in heave and therefore, the BCS commands the cushion pressure to act as a heave-compensator which lifts the vessel mass in a wave-through (max pressure) and releases the mass in a wave crest (min pressure).

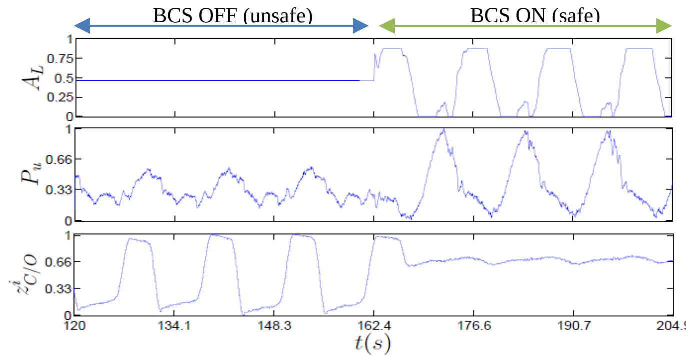

Figure 9 – Excitation force in heave is dominating motion.

The input, measurement signal of the BCS consist of weighted accelerometers located near CG and the bow. The signal is filtered, sent through a tunable integrator and then sent to the vent valve servo-controllers, see Figure 3. The control law coefficients were found in a somewhat similar approach as presented in the system identification section.

FULL-SCALE RESULTS

Boarding Control System

- The following test occurred in head sea with significant wave height around 1.8-1.9m and wave period 4-5s.
- BCS is initially ON, turned OFF, and then back ON.
- Free vessel motion, turbine contact doesn't apply.

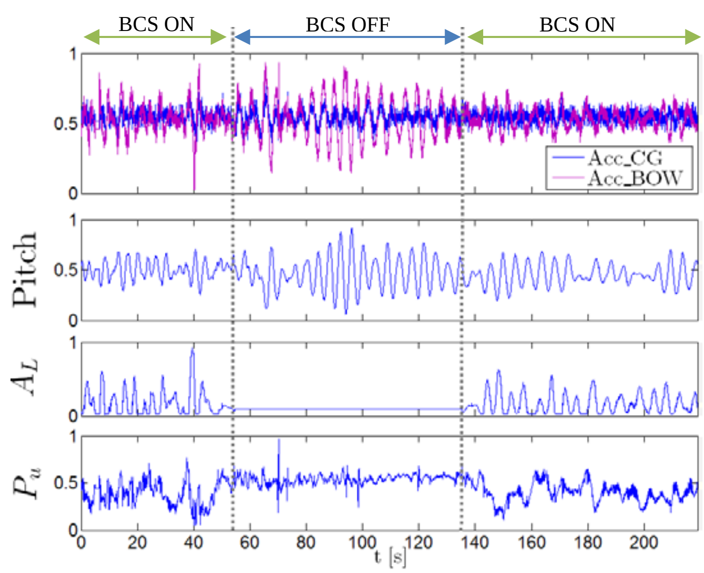

Figure 10 – Timeseries, BCS OFF/ON

Figure 10 shows that vertical motion (acceleration and pitch) is significantly reduced when the BCS is ON. The measurement for $Acc_{CG}(t)$ can be separated into segments: the BCS OFF where $t \in \{54, 134\}$ and the BCS ON segment, where $t \in \{135, 215\}$. These two segments can be plotted as Root Mean Square (RMS) values per 1/3 octave band and directly compared to the ISO 2631-1:1997 (ISO, 1997) standard which deals with the MSI (Motion Sickness Index). The MSI presents criteria for acceleration limits as a function of frequency and exposure time.

Figure 11 illustrate that with the BCS ON, in this sea-condition, then sea-sickness is most likely not going to occur, even after exceeding an exposure time of 8 hours. Note that the presented frequency plots are linear spectrums and therefore the maximum-peak difference between the control system on-and-off modes, are “squared-times-less” compared to those in a Power Spectrum Density (PSD) plot.

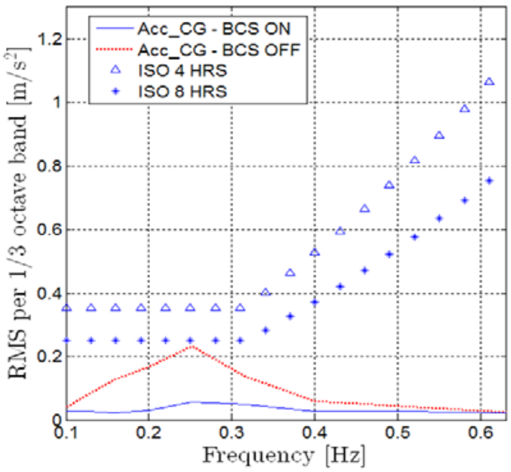

Figure 11 – Heave acceleration

Figure 12 shows the RMS per $1/3$ octave band for the pitch angle using the same test run. The BCS removes more than half of the pitch motion. Note that the BCS ON curve has a smaller peak frequency than that of the BCS OFF curve. This illustrates the variance in sea wave period during a $2x80$ second test run.

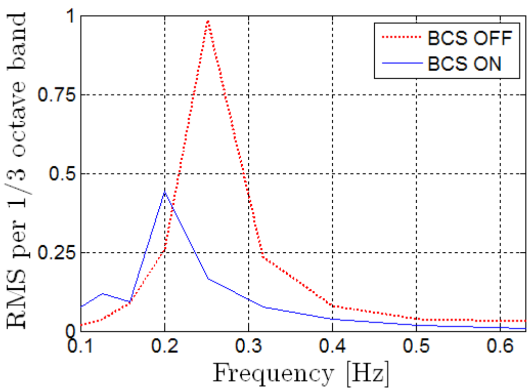

Figure 12 – Pitch angle

The damping of pitch motion, as seen in Figure 12, has an important effect for the captain as this simplifies the process of docking to an offshore structure. The sea waves in Borkum Riffgrund 1, often contains time periods around $4 - 6$ seconds which typically falls into to the natural frequency in pitch for a 25m long vessel. Also, when docking to a turbine, the captain usually encounters strong currents and wind, and the pitch damping has proven crucial for increasing turbine-accessibility.

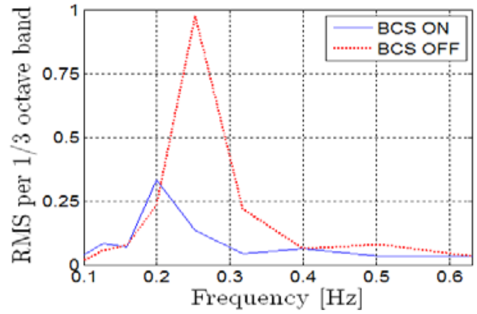

Figure 13 – Vertical acceleration bow

Figure 13 proves that the BCS significantly reduces acceleration on the bow tip. The bow tip is the location on the vessel where the offshore service workers start climbing the turbine latter- and when they return back to the vessel upon completion of the O&M mission.

Ride Control System

- The following test occurred in head sea with 0.5m significant wave height and 38 knot speed.
- RCS is toggled ON/OFF, each with a time segment of 45.7 s.

Recall from the intro section that the non-spatial resonance frequency in cushion pressure-and-heave is estimated to occur at 1.2 Hz and that the spatially varying first acoustic mode occurs at 8 Hz. Figure 14 shows that the amplitude of the non-spatial resonance is damped by 50% and the spatial resonance is also significantly damped. As the figure illustrate, the acoustic modes aren’t a huge issue in the first place on the Umoe Ventus hull.

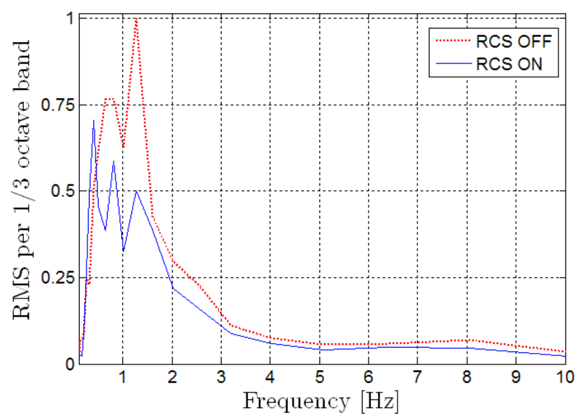

Figure 14 – Acceleration CG

Figure 15 shows that the air cushion pressure variations are damped when the RCS is toggled on.

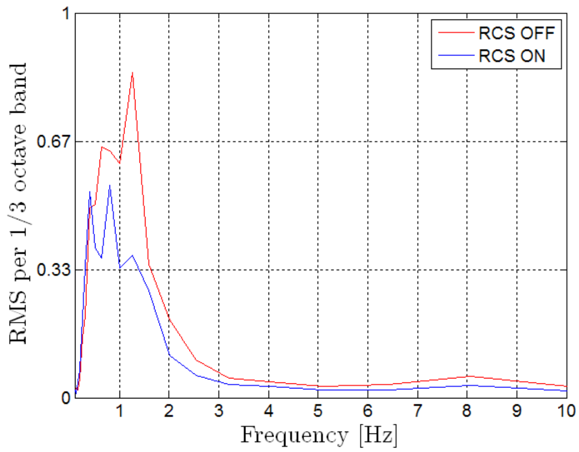

Figure 15 – Cushion pressure

CONCLUSIONS

The BCS has been tested in up to 2m significant wave heights, with no signs of losing its grip to the turbine with calm, damped motions. The BCS significantly simplifies the process of docking to an offshore structure with perceptible reduced motion in heave and pitch.

The RCS performance is as good-, or better, -as expected and results in a comfortable transit for the crew and personnel onboard.

Future work includes further full-scale-testing of the BCS in higher seas, as the authors of this paper are confident that its potential has not yet been reached. Model-scaled results indicates that the vessel is capable of accessing turbines in up to 2.5m Hs and even higher in swell seas (Auestad 2015).

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APPENDIX A

The estimated transfer function H_p:

$$H_p = \frac{2.38\,s^5 + 4.714\,s^4 + 125.3\,s^3 + 301.4\,s^2 - 178.9\,s + 1616}{s^5 + 7.483\,s^4 + 107.6\,s^3 + 429.8\,s^2 + 2891\,s + 1731}$$

APPENDIX B

Umoe Ventus has an overall- length and-width of 27.2 m and 10.4m, respectively. The draught is 1-to-3.2 meters. Max speed is in excess of 40 knots. The cushion l/b ratio is 2.2 with narrow side-hulls. The propulsion machinery consists of 2 x MTU 16V 2000 M72 diesel engines each with an output of 1440 kW which is connected to the 2 x 360 kW lift fan engine (Scania DI13 78M) and 2 x MJP 650 CSU water jet construction.