Performance Characteristics of a 260t Displacement SES

Leo Lazauskas
leo@cyberiad.net

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Introduction

The main objective of the present example is to estimate some performance characteristics of a 260t displacement SES with similar proportions to the Royal Norwegian Navy's P-960 KNM Skjold. Some of the principal dimensions of the Skjold are unavailable and had to be guessed or inferred from small photographic images at *http://www.knmskjold.org*.

We will first show the effect of squat on the components of resistance for off-cushion and on-cushion modes of operation. It will be demonstrated that for the on-cushion mode, squat makes very little difference to predictions at high speeds. We will also estimate the range of the vessel and its powering requirements in infinite depth and finite depth water. All calculations in the present report were made using the computer program `Flotilla`.

For details absent in the present report, see [3] which is available online at *http://www.cyberiad.net/library/pdf/llmsc.pdf*.

Mathematical Model

Environmental Variables

Table 1 shows the principal environmental variables used to produce the results to follow. Gravitational acceleration is denoted by g; ρ and ν are, respectively, the density and kinematic viscosity of water; ρ_{air} and ν_{air} are their counterparts in air at the same temperature.

	Value
g (ms^{-2})	9.80665
Water (15° C)	
ρ (kg/m^3)	1025.87
ν $(10^-6m^2s^{-1})$	1.18831
Air (15° C)	
ρ_{air} (kg/m^3)	1.226
ν_{air} $(10^-6m^2s^{-1})$	14.4

Table 1: Principal environmental variables.

Ship Dimensions

The present hypothetical vessel, "Giam260a", has Wigley parabolic sidehulls and a rectangular air cushion. *Skjold* is Norwegian for "shield": *Giam* is the equivalent word in a South-East Australian Aboriginal language. Front and side-views of the vessel are shown in Figure 1; a plan view is shown in Figure 2, and the principal dimensions are given in Table 2. The above-water components of the vessel are modelled as simple rectangular blocks in order to estimate the structural weight of the vessel and the air resistance. They do not affect the hydrodynamic resistance components.

The total displacement volume DOA is the sum of the volume displaced by the sidehulls and the volume displaced by the air cushion. LOA and WOA are the overall length and width of the vessel. HOA is the height of the vessel from the undisturbed free-surface to the top of the deckhousing.

The (static) displacement of the demihulls is D, their length is L_{WL}, beam B_{WL} and draft T.

The free-board height of the demihulls (i.e. from the undisturbed water surface to the bottom of the cross-structure) is H_{fb}. The superstructure (including the cross-structure connecting the demihulls) are defined by x_{ss}, the x-ordinate of the centre of the superstructure, and L_{ss}, B_{ss}, and H_{ss}, respectively, the length, width, and height. The location and dimensions of the deckhousing are specified in the same way.

The principal (nominal) dimensions of the air cushion are given in Table 3. In the present example, the cushion is represented by a simple constant-pressure rectangle. The x-ordinate of the centre of the air cushion is x_c. The length is L_c. The (nominal) beam of the cushion is B_{cnom}.

The last three parameters in Table 3 affect the momentum drag R_M, and the equivalent lift, R_L. The skirt clearance is h_{skirt}, the discharge coefficient is C_q, and the ratio of the thrust to the lift is η_T/η_L.

	Giam260a
VESSEL	
DOA (m^3)	260.000
LOA (m)	48.000
WOA (m)	13.500
HOA (m)	12.250
DEMIHULLS (Each)	
D (m^3)	130.000
L_{WL} (m)	40.000
B_{WL} (m)	3.250
T (m)	2.250
ABOVE WATERLINE	
C_{dair}	0.400
H_{fb} (m)	2.250
Superstructure	
x_{ss} (m)	0.000
L_{ss} (m)	48.000
B_{ss} (m)	13.500
H_{ss} (m)	6.000
Deckhouse	
x_{dh} (m)	9.000
L_{dh} (m)	16.000
B_{dh} (m)	6.750
H_{dh} (m)	4.000

Table 2: Principal dimensions and parameters of the vessel in the off-cushion mode.

	Giam260a
AIR CUSHION	
x_c (m)	0.000
L_c (m)	37.500
B_{cnom} (m)	9.500
h_{skirt} (m)	0.080
C_q	0.700
η_T/η_L	2.000

Table 3: Nominal cushion dimensions.

	Giam260a
DEMIHULLS (Each)	
D_{80} (m^3)	26.000
B_{WL80} (m)	2.047
T_{80} (m)	0.881
AIR CUSHION	
D_{c80} (m^3)	208.000
B_{c80} (m)	9.500

Table 4: On-cushion particulars.

The principal dimensions of the vessels in the on-cushion configuration are given in Table 4. In the present example, the cushion supports 80% of the weight of the vessel. The displacement volume of each demihull is D_{80}. The demihulls have a waterline beam of B_{WL80} and the draft is taken as T_{80}. The air-cushion displacement is D_{c80} and the nominal beam is B_{c80}.

In thin-ship theory it is consistent to use the separation distance between the demihull centrelines as the cushion beam, however slightly better agreement with experiments can be obtained by adjusting the cushion width (and hence the cushion pressure) to account for the beam of the demihulls.

Fuel and Efficiency

	Equation
η_p	$0.75 - 0.0007557(U - 25.7222)^2$
sfc $(g/kW.h)$	$155 + 884/P_s$

Table 5: Equations used to estimate the overall propulsive coefficient η_p of waterjets and the specific fuel consumption *sfc*. Speed U in ms^{-1} and shaft power P_s in megaWatts.

In the present example, we allow the overall propulsive coefficient η_p to vary with speed. The specific fuel consumption *sfc* varies with shaft power P_s. These quantities do not affect the hydrodynamics of the vessel. The equations used in the present example are given in Table 5.

Weight Group	Giam260a
100 Structure	0.4300
200 Propulsion	0.1400
300 Electrical	0.0300
400 Communications	0.0300
500 Auxiliary	0.0300
600 Outfit	0.0600
700 Armaments	0.1500
Light Ship	0.8700
Cargo	0.0000
Crew and effects	0.0100
Provisions	0.0025
Fresh water	0.0420
Oil	0.0100
Fuel	0.0655
Deadweight	0.1300
Full Load	1.0000

Table 6: Weights as fraction of total displacement.

Weights

The assumed disposition of weights in the present example is shown in Table 6. These quantities do not affect the hydrodynamics of the vessel.

The structural weight of the vessel can be estimated using, among many other methods, an equipment numeral approach. We assume that the equipment numeral E_2 for SES and catamarans is given by

$$E_2 = E_{2H} + E_{2S} + E_{2D} + E_{2X} \tag{1}$$

where

$$E_{2H} = 2L(B + T) \tag{2}$$

is the numeral of the hulls below the waterline,

$$E_{2S} = 0.85L(H_f + H_{ss}) \tag{3}$$

is the contribution of the sides of the demihulls and the superstructure,

$$E_{2D} = \left[0.70 + 0.15\frac{B_{dh}}{WOA}\right] L_{dh}H_{dh} \tag{4}$$

is the numeral of the deck-housing, and

$$E_{2X} = 1.6L(WOA - 2B) \tag{5}$$

is the numeral of the cross-structure.

The structural weight, assuming aluminium construction, is then estimated by

$$\begin{aligned} W_{structure} &= 0.00064E_2^{1.7} \ (E_2 \leq 3025) & (6)\\ &= 0.39E_2^{0.9} \ (E_2 > 3025). & (7) \end{aligned}$$

The estimates for the other weights are also little more than guesswork. The weight of the propulsion system (turbines, gearboxes and waterjets) is based on an assumed value for the maximum power available for propulsion (not including lift fans). We assume $P_{smax} = 12.0$MW for the off-cushion mode, and $P_{smax} = 14.0$MW for the on-cushion mode.

The fuel weight fraction is not specified in the model: it is the remainder after all other weights have been deducted from the total.

Resistance Components and Squat

The total resistance, R_T, is considered to be the sum of five non-interacting components

$$R_T = R_V + R_W + R_A + R_L + R_M \tag{8}$$

where R_V is the viscous resistance, R_W is the wave resistance, and R_A is the aerodynamic resistance of the above-water portion of the vessel. The last two components are applicable only to the on-cushion mode: R_L is the equivalent lift, and R_M is the air-cushion momentum resistance.

In the present report we use Michell's [4] thin-ship integral to estimate the deep water wave resistance of the hulls, and a modified version of Michell's integral to model the contribution to the resistance of the air cushion, as well as interference effects between the demihulls and the cushion, [8], [3]. Finite depth effects are modelled using Sretensky's [5] extension of Michell's integral.

The dynamic sinkage and trim is based on the method described by Tuck, Scullen and Lazauskas [6]. In the present report, the attitude of the vessel is iterated until equilibrium is achieved. Typically, between 3 and 10 iterations are required.

Viscous resistance is estimated using Grigson's [1] skin-friction line which is based on 2D boundary layer theory and is under consideration as an alternative to the usual ITTC 1957 line [2]. Form factors are not used in the present work.

A standard empirical formula is used to estimate the air resistance in the present example, namely

$$R_A = \frac{1}{2} C_{dair} \rho_{air} U^2 A_f \tag{9}$$

where U is the ship speed, C_{dair} is the (user-specified) air drag coefficient, and A_f is the frontal area of the vessel calculated from the above-water dimensions given in Table 2.

Air escaping from under the skirt transfers momentum from the vessel to the air which means that a force acts on the vessel. This "momentum air resistance" is given by

$$R_M = \rho_{air}\, Q_c\, U \tag{10}$$

where Q_c is the volume flow rate of air escaping from the cushion, assumed for purposes of calculation to be at uniform pressure p_c, the mean cushion pressure.

Q_c can be estimated as

$$Q_c = L_p\, h_{skirt}\, q\, C_q \tag{11}$$

where L_p is the perimeter of the skirt (equal to twice the tunnel width for a SES, or twice the sum of length and tunnel width for a "pure" ACV). h_{skirt} is the (nominal) distance between the water surface and the bottom of the skirt. The coefficient C_q accounts for the contraction of the escaping air jet.

The velocity of air escaping between the water surface and the skirt, q, can be calculated using Bernoulli's equation

$$q = \sqrt{2 p_c / \rho_a} \tag{12}$$

Inflating and sustaining an air cushion requires power. Although the power required is independent of speed, for the purposes of comparing resistance components we define an equivalent lift resistance

$$R_L = (\eta_T / \eta_L)(p_c\, Q_c / U) \tag{13}$$

where η_T is the thrust efficiency of the fans and η_L is the lift efficiency.

We ignore, among other drag components, splash, spray, and wave–breaking, which, in general, comprise a smaller proportion of the total resistance than the other components, particularly for the fine slender hulls we are using. We also ignore the drag of the seals containing the air cushion which can be quite significant for some (typically low) speeds.

Results and Discussion

Squat

The squat of the vessels is shown in Figure 3: dzb and dzs are the changes in the height of the bow and the stern from their static positions. Positive values mean that the bow or stern are raised out of the water.

It can be seen that squat is much larger for the off-cushion (i.e. $D_c/D = 0.0$) condition than for $D_c/D = 0.8$, the on-cushion mode.

For the off-cushion mode, the hull attitude begins to change rapidly for speeds between 12 knots and 18 knots, with the bow rising out of the water and the stern dropping. Above 18 knots, the bow stays at a relatively constant level and the stern begins to rise upwards from its minimum sinkage. At very high speed, above about 70 knots, both dzb and dzs are greater than zero. This suggests that the vessel is on the verge of planing, however whether this is actually achievable with Wigley demihulls is a hotly-disputed matter.

For the on-cushion condition, the changes in hull attitude are much smaller.

Resistance

Figures 4 and 5 show the total resistance and resistance components (non-dimensionalised by the total weight) for both the static and dynamic cases.

For the off-cushion case, wave resistance is the major component of the total for speeds between about 15 knots and 27 knots. Above 27 knots, viscous resistance dominates. Air resistance is small, but increases steadily and predictably as speed increases.

Interestingly, the wave resistance remains almost constant for the case where squat is included for speeds greater than about 30 knots.

As we will see in the next section, the vessel is unable to reach the highest speeds in these figures because of limitations placed on the turbine power output.

When the vessel is on cushion, wave resistance and the equivalent lift are the dominant components below about 38 knots. The differences between the static and dynamic cases are only significant at the highest speeds in the graphs.

Viscous drag dominates above 38 knots. Above about 62 knots, air resistance is greater than the wave resistance and the equivalent lift. For the assumed skirt clearance and other cushion parameters, momentum lift is negligible.

The wave resistance and equivalent lift are of very similar magnitude for speeds greater than about 30 knots, however this is pure coincidence.

Comparing the total resistance curves in Figures 4 and 5, it can be seen that the on-cushion mode is preferable for speeds greater than about 35 knots. Below that speed, the vessel should be operated either off-cushion, i.e. as a "pure" catamaran, or as an SES with the cushion supporting less than 80% of the total weight. (In the interest of brevity, we have omitted results for other D_c/D ratios.)

Power and Range

The top plot in Figure 6 is simply an illustration of the equation for the overall propulsive coefficient in Table 5.

The bottom plot shows the shaft power, here defined as $P_s = R_T U/\eta_p$, for both the off-cushion and on-cushion modes and for the static and dynamic conditions.

Under the many gross assumptions we have made, the maximum achievable speed off-cushion is about 50 knots. Differences due to whether squat is included or not are apparent for speeds above 25 knots

For the on-cushion mode, the maximum speed is about 65 knots, noting that this is under the ideal conditions of the mathematical model. The top sustained speed claimed for the Skjold is "in excess of 55 knots" [9].

The effect of squat is apparent, but small, for speeds greater than about 37 knots.

The top graph in Figure 7 shows *sfc* given by the equation in Table 5. Clearly, the lowest consumption occurs at the highest speeds.

The bottom graph in Figure 7 shows the Breguet range given by

$$s_{max} = -367098.8 \frac{\eta_p}{sfc} \frac{W}{R_T} \log_e(1 - \frac{W_f}{W}) \tag{14}$$

where s_{max} is the range in kilometres, W is the total weight of the ship, and W_f is the weight of fuel at the beginning of the voyage.

The maximum off-cushion range is achieved for a (constant) speed of about 37 knots; on-cushion, the maximum range is achieved at 50 knots.

At top speed, the range in off-cushion mode is about 600km, on-cushion it is about 730km.

Finite Depth Effects

The two plots in Figure 8 shows the effect of finite depth water on the wave resistance of the vessel in both the off-cushion and on-cushion modes of operation. The enormous increase of the wave resistance in shallow water is apparent for speeds around 12 knots to 17 knots. However, once the vessel is over the hump, the wave resistance can be much less than the infinite depth case.

Figure 9 shows the effect of finite depth water (for the extreme value $h = 5m$) on the required shaft power and the Breguet range. Here we can see that there are significant reductions in required shaft power and an increase in the achievable range It is also clear that speeds between about 12 knots and 15 knots should be avoided.

Conclusion

We have estimated the performance characteristics of a 260t displacement SES with similar proportions to the Royal Norwegian Navy's P-960 KNM Skjold. Having established a simple baseline vessel, we are now in a position to estimate the effect of many different variations: for example, different hull shapes, transom sterns, non-uniform air-cushion pressure distributions, disposition of weights etc.

Appendix

Imperial	Metric
1 ft	0.3048 m
1 lb	0.4535 kg
1 nm = 6080.2 ft	1853.245 m
1 short ton = 2000 lbs	907.0 kg
1 long ton = 2240 lbs	1015.84 kg
1 hp	745.7 Watts
1 lb/hp.h	606.2774 g/kW.h
1 knot	0.514444 m/sec

Table 7: Conversions.

References

[1] Grigson, C.W.B., "A planar friction algorithm and its use in analysing hull resistance", *Trans. RINA*, 2000, pp. 76–115.

[2] Proceedings of the 8th ITTC, Madrid, Spain 1957, published by Canal de Experiencias Hidrodinamicas, El Pardo, Madrid.

[3] Lazauskas, L., "Hydrodynamics of advanced high-speed sealift vessels", *MSc thesis*, Dept. Applied Mathematics, The University of Adelaide, April 2005. http://www.cyberiad.net/library/pdf/llmsc.pdf

[4] Michell, J.H., "The wave resistance of a ship." *Philosophical Magazine*, Series 5, Vol. 45, 1898, pp. 106–123.

[5] Sretensky, L.N., "On the wave-making resistance of a ship moving along in a canal." *Philosophical Magazine*, Series 7, Vol. 22, No. 150, 1936, pp. 1005–1013.

[6] Tuck, E.O., Scullen, D.C. and Lazauskas, L., "Sea Wave Pattern Evaluation, Part 5 report: Speed-up and squat", Applied Mathematics Department, The University of Adelaide, March 2001.

[7] Tuck, E.O., "Wave resistance of thin ships and catamarans", Report T8701, Applied Mathematics Department, The University of Adelaide, 1987.

[8] Tuck, E.O., Scullen, D.C. and Lazauskas, L., "Wave patterns and minimum wave resistance for high-speed vessels", 24th Symposium on Naval Hydrodynamics, Fukuoka, JAPAN, July 2002.

[9] Web page of the Skjold crew. www.knmskjold.org.

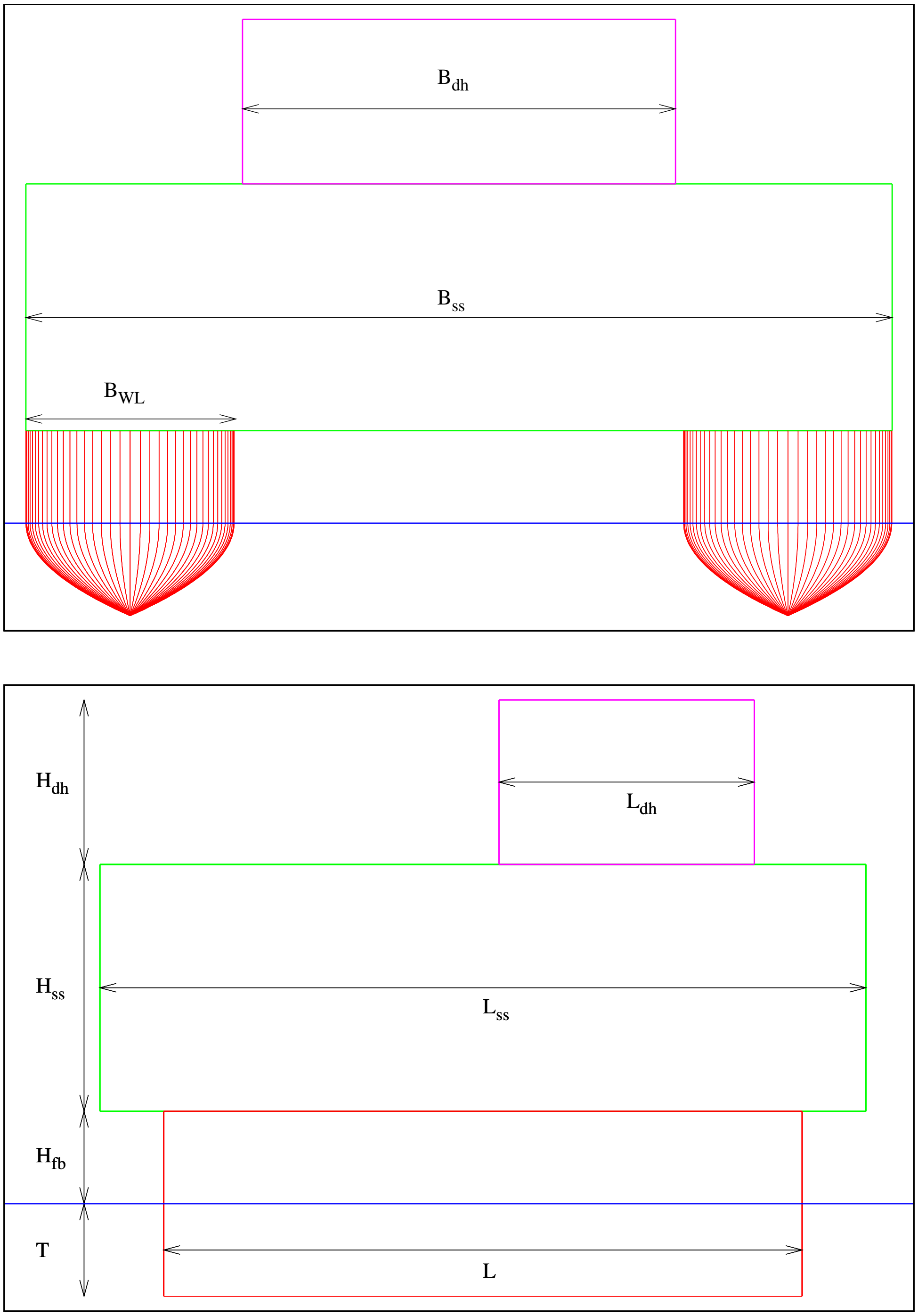

Figure 1: Front view (top) and side view (bottom).

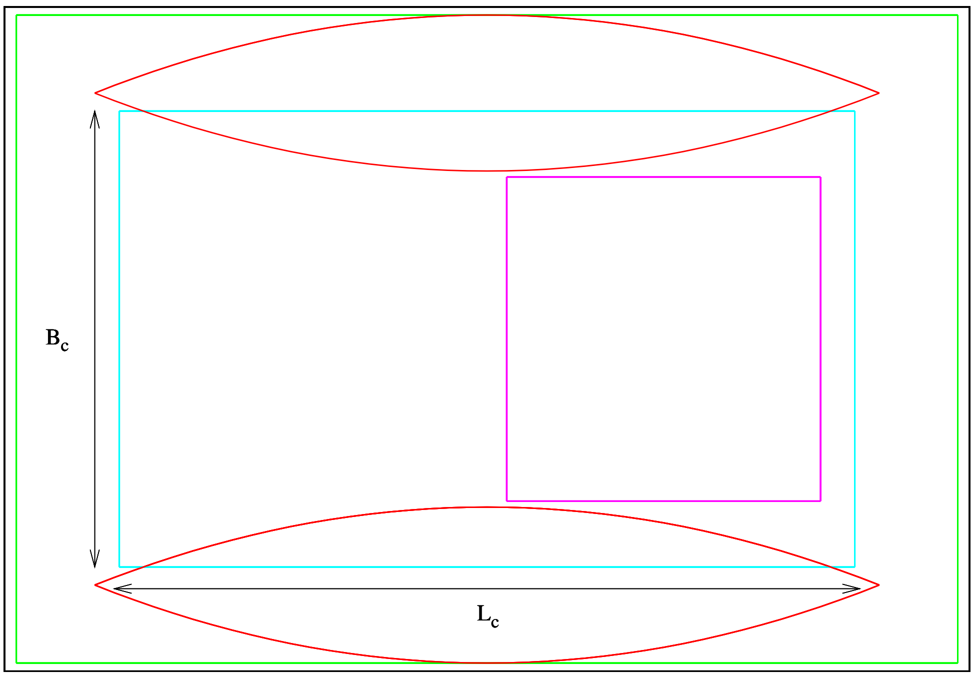

Figure 2: Plan view: Hull waterplanes are shown in red, the air cushion is cyan, the green rectangle is the super-structure, and the deck-housing is magenta.

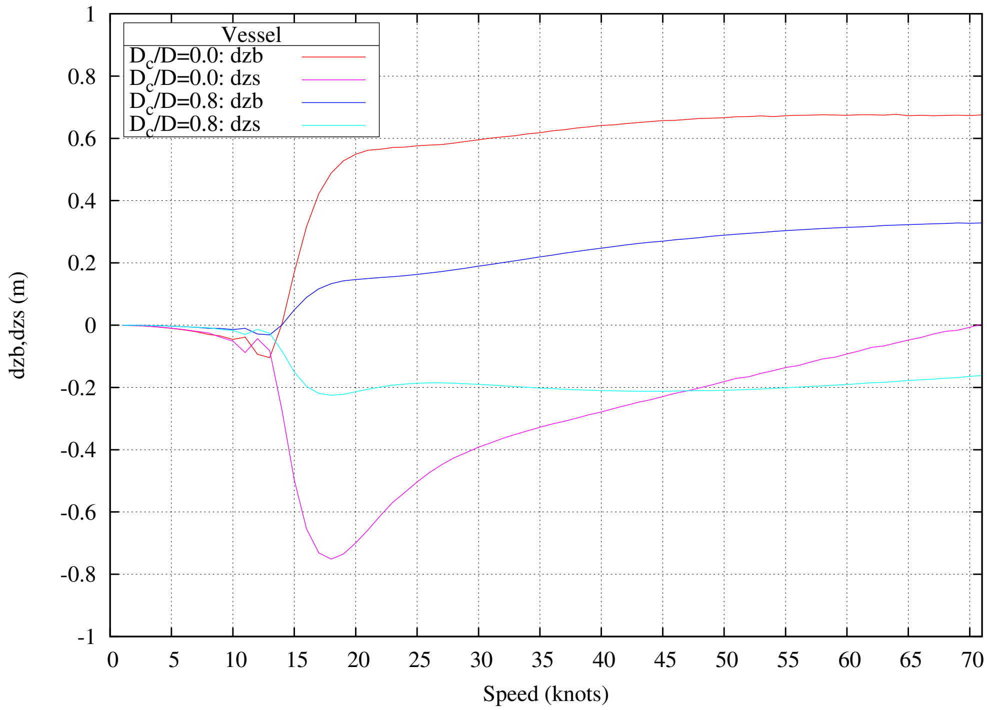

Figure 3: Squat for off-cushion and on-cushion modes.

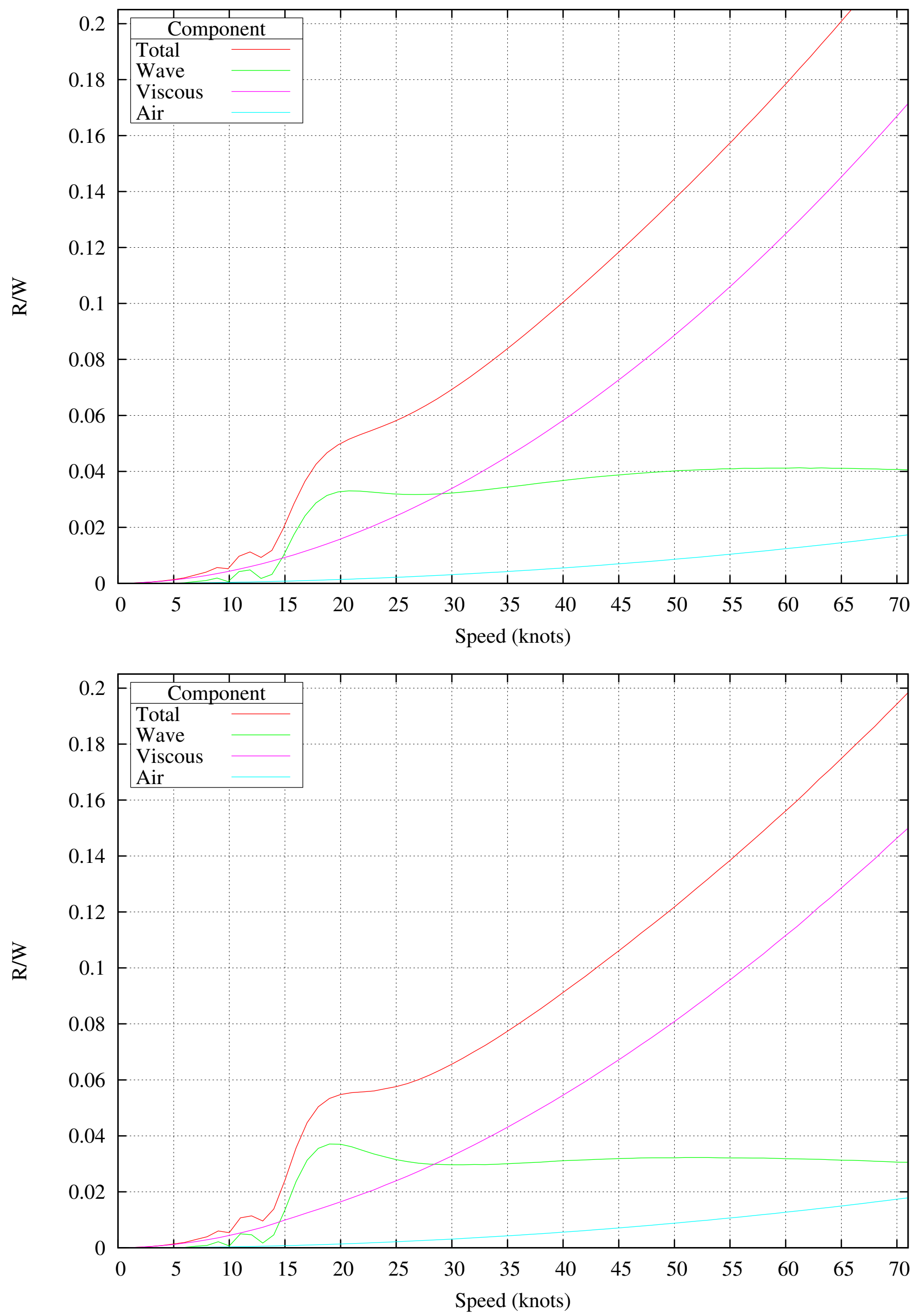

Figure 4: Resistance components of vessel in off-cushion mode. Static (top) and squatted (bottom).

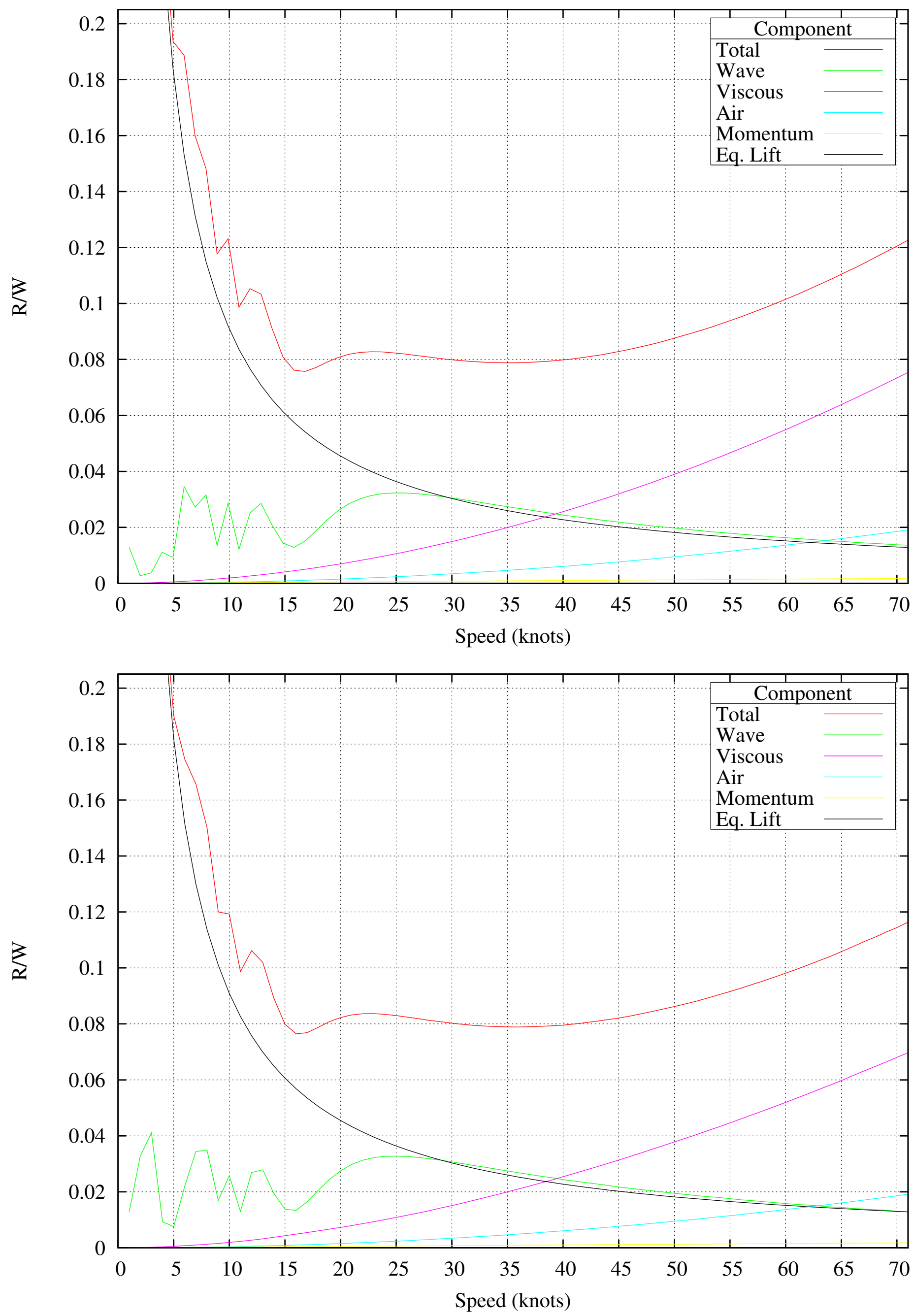

Figure 5: Resistance components of vessel in on-cushion mode. Static (top) and squatted (bottom).

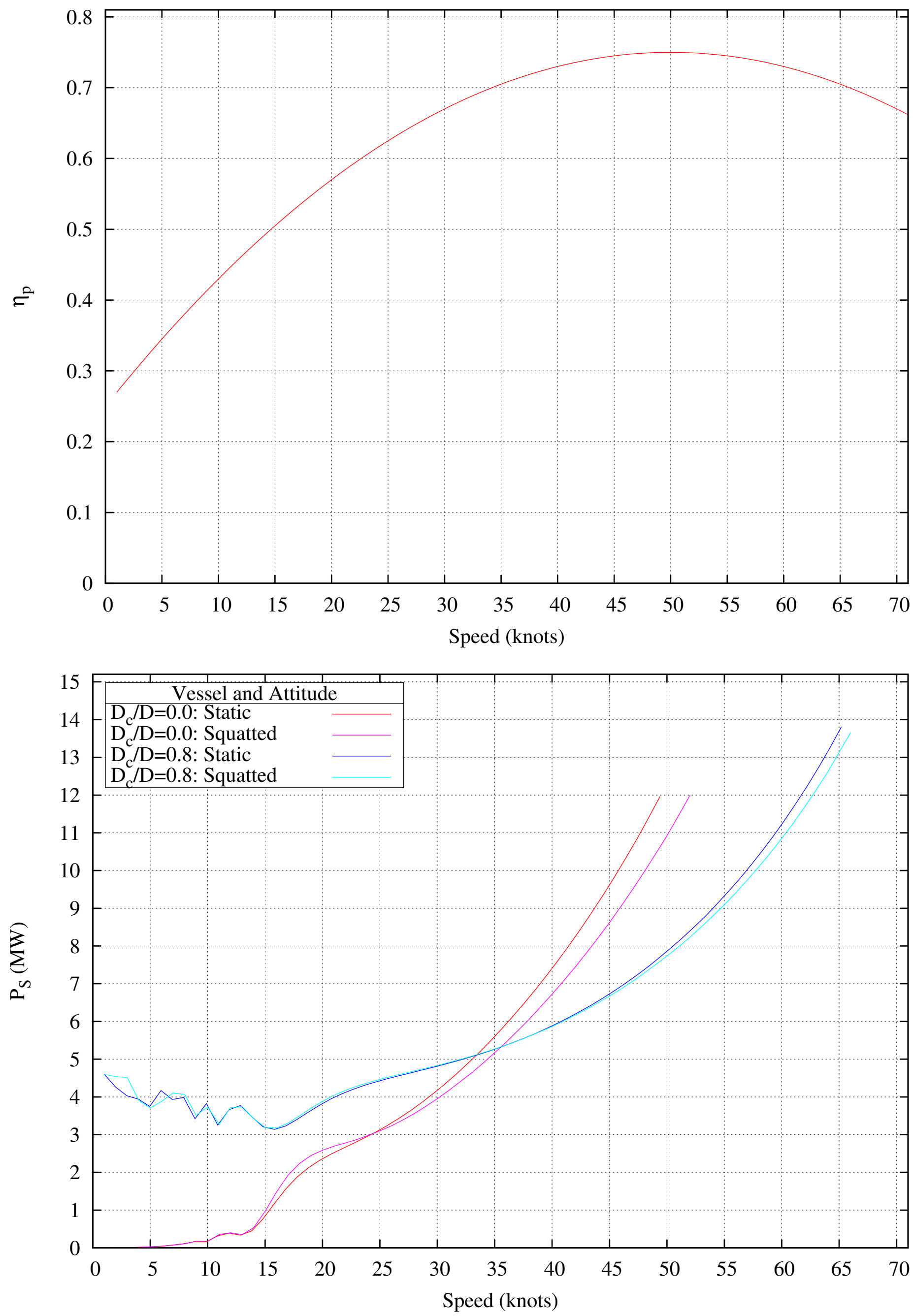

Figure 6: Overall propulsive coefficient (top) and shaft power (bottom).

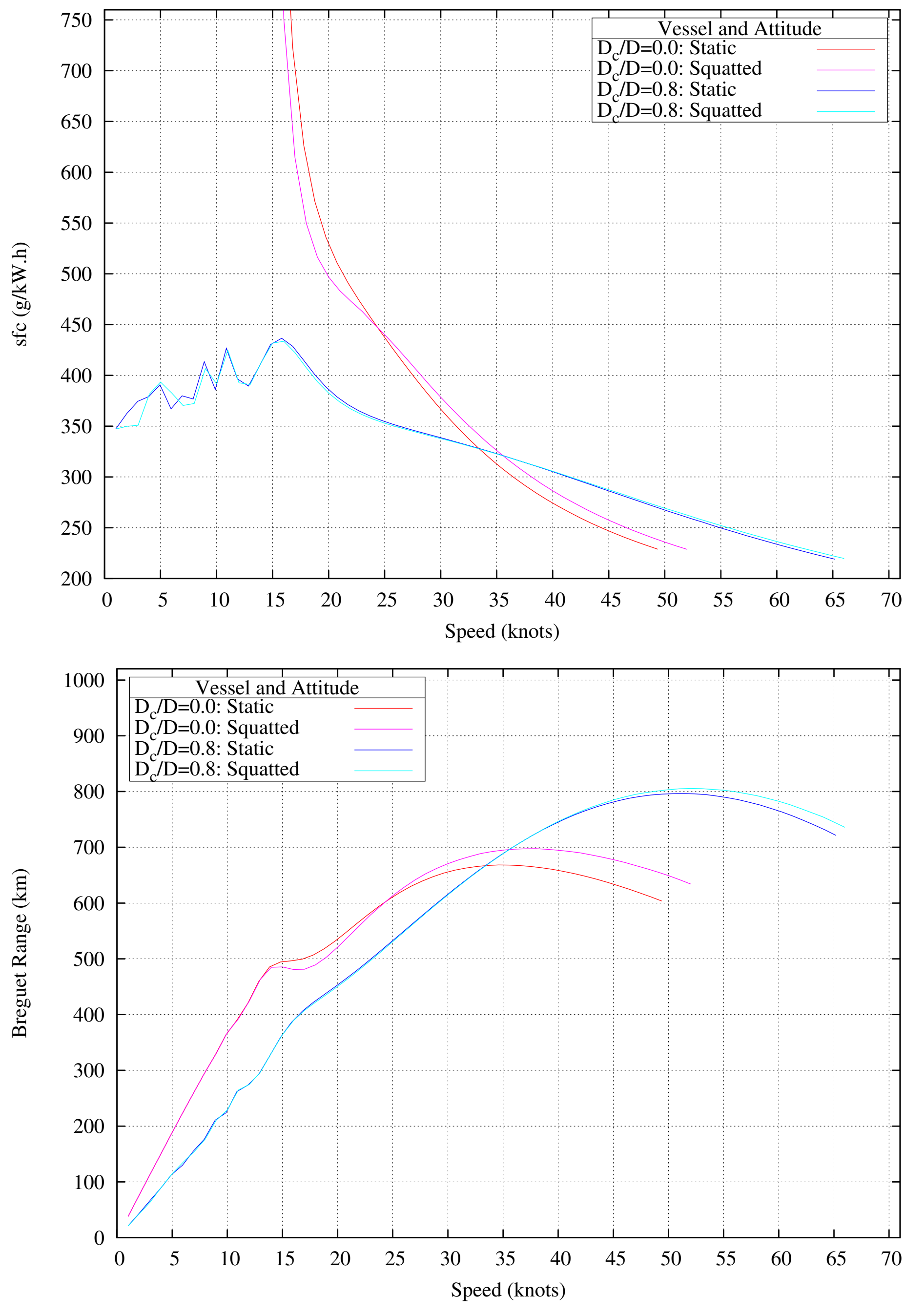

Figure 7: Specific fuel consumption (top) and Breguet range (bottom).

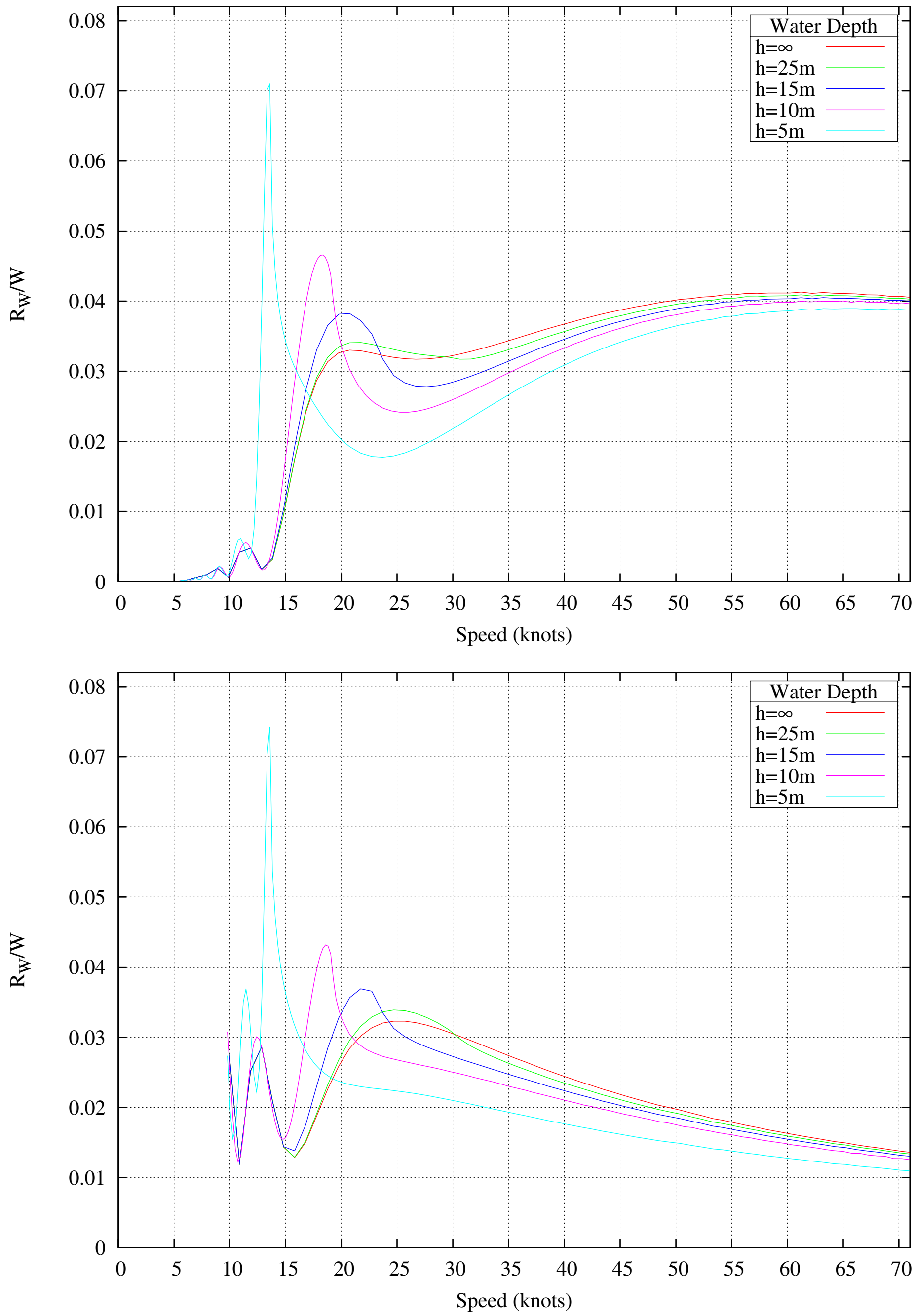

Figure 8: Finite depth effects on wave resistance: off-cushion (top) and on-cushion (bottom) in the static attitude.

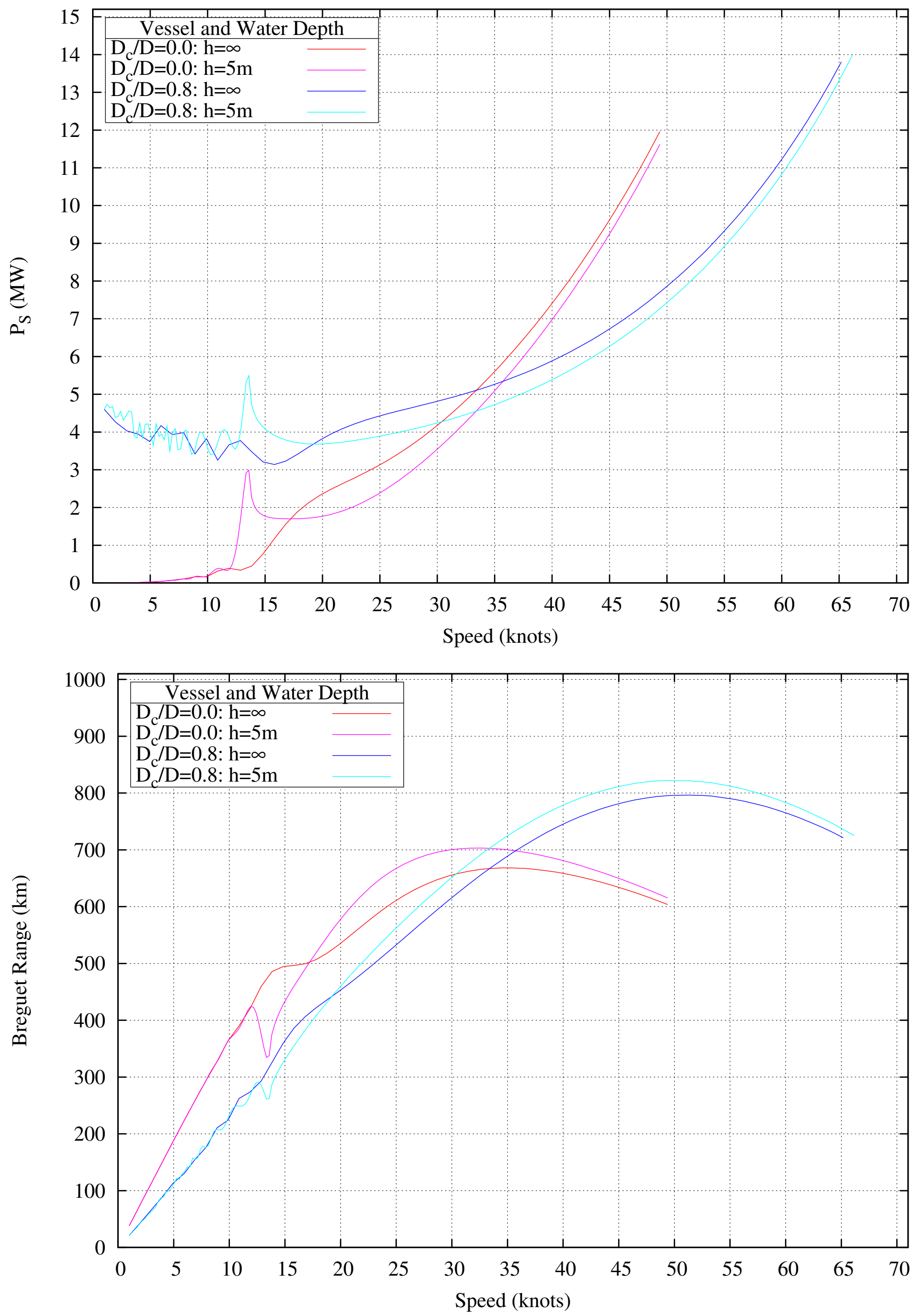

Figure 9: Finite depth effects on shaft power (top) and on range (bottom) in the static attitude.