Scaling of RV Athena's Appendage Drag Using CFD

Iwan Mustaffa Kamal, Muhammad Safwan Rasahidan and Yaseen Adenan Ahmed

Abstract The drag or the resistance contribution of appendages such as brackets, skegs, shafts, rudders and fin stabilizers fitted to R/V Athena were investigated. The issue in estimating the appendage drag is that laminar flows occurred at lower Reynolds number typically in ship model used for ship hydrodynamic testing in a towing tank. In full-scale ship, the flow is usually turbulent. Uncertainties when scaling the model result to full-scale may arise as the flow regime in both cases are not similar, hence the full-scale estimate will be inaccurate. One method to resolve this issue is to scale the appendage drag using a fixed fraction which is known as the 'Beta' approach. The aim of this investigation was to determine the value of beta. R/V Athena was chosen as the case study in this investigation. The results obtained from SHIPFLOW 6.3 shows that the drag in model scale increased at 14% when fully appended. The drag in full-scale increased at 11% when fully appended. It was observed that the skeg contributed to the highest percentage of drag at about 7–8% of the total drag. Finally, the value of beta was estimated to be in between 0.38 and 0.39.

Keywords Reynolds scale effect · Appendage drag · CFD · Beta approach

1 Introduction

1.1 Ship Appendages

Ship appendages are external fittings fitted outside the main hull, such as shaft supporting brackets, struts, bossings, rudders, shaftings, skeg, bilge keels and control surfaces i.e. stabilizer fins etc. These appendages alter the flow around the hull and these result with an increase of drag to the total ship drag. The flow around a body or in this case a ship appendage can have two different flow regimes; it can be laminar

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or turbulent depending on the appendage length and the velocity of the flow around the appendages.

For full-scale appliances where in this case of a full-scale ship, the flow is usually fully turbulent. However, when considering model-scale ship used in hydrodynamic testing in a towing tank, the flow around the ship model can be in laminar flow or transitional. Uncertainties when scaling the model test results to full-scale may arise as the flow regime in both cases are not similar, hence the full-scale estimate will be inaccurate. Therefore this study investigated the appropriate appendages scaling method in addressing this issue.

Molland et al. [1] reported that there is only a small amount of actual data on scaling of appendages. One of the published work which deals with the appendage scale effect can be found in Allan [2]. Allan [2] presented the results of a study of both shafting with struts and bossings for three models, 18.70, 22.96, and 33.67 ft. in length, over a range of speed-length ratios from 0.50 to 0.80. Other published work can be found in Lackenby [3], where the British Ship Research Association (BSRA) in the 1950s, conducted a series of resistance experiment using a jet propelled 58 m ship called 'Lucy Ashton', which is fitted with various appendages. The resistance measurement was made possible by mounting the jet engine to the hull on a load transducer, which allowed the direct measurements of thrust, therefore the resistance of the ship. These sea trial results were compared with six geosim models tested at the National Physics Laboratory (NPL). The results of the appendage drag on 'Lucy Ashton' can be found in van Manen and Oossanen [4]. From the results it seems that at lower speed, the scale effect on the appendage drag is not apparent as at higher speed. A conclusion was made by Molland et al. [1] that the results from the 'Lucy Ashton' experiment and other such tests as in Allan [2] tends to be inconclusive.

1.2 Estimating the Appendage Drag

As recommended in ITTC [5] in the ITTC—Recommended Procedures and Guidelines for ITTC1978 Performance Prediction Method, Procedure no. 7.5-02-03-01.4, the full-scale appendage drag coefficient need to be added as C_{AppS} in the full-scale extrapolation of the total drag coefficient C_{TS} as in Eq. 1.

$$C_{TS} = C_F(1 + k) + C_R + \Delta C_F + C_A + C_{AAS} + C_{AppS} \tag{1}$$

where C_F is the frictional drag coefficient of the ship according to the ITTC1957 model-ship correlation line, k is the form factor determined from the resistance test, C_R is the residual drag coefficient calculated by subtracting the frictional drag coefficient from the total drag coefficient C_{TS}, ΔC_F is the roughness allowance, C_A is the correlation allowance, where C_A is determined from comparison of model and sea-trial results and C_{AAS} is the air resistance coefficient in full-scale.

According to Molland et al. [1] there are four primary methods in estimating the appendage drag coefficient C_{AppS}. The four primary methods as listed in Molland et al. [1] are as the followings:

1. Testing the hull model with and without appendages, where the difference of the model total drag coefficient C_{TM}, with and without the appendages represents the appendage drag as shown in Eq. 2.

$$C_{AppM} = C_{TM(appended)} - C_{TM(barehull)} \qquad (2)$$

where C_{AppM} is the appendage drag obtained from subtracting the bare-hull total drag coefficient $C_{TM\,(bare\text{-}hull)}$ from the appended total drag coefficient $C_{TM\,(appended)}$. This appendage drag is then scaled up to full-scale using Eq. 3.

$$C_{AppM} = C_{AppS} = \frac{R_{AppM}}{\frac{1}{2}\rho_{fw}S_{AppM}V_M^2} = \frac{R_{AppS}}{\frac{1}{2}\rho_{sw}S_{AppS}V_S^2} \qquad (3)$$

where R_{App} is the appendage drag in Newton, ρ_{fw} and ρ_{sw} is the specific density of fresh water and seawater respectively, S_{App} is the wetted surface area of the appendages and V is the model or the ship speed in m/s. The subscript M and S refer to model-scale and full-scale respectively. The scaling of the appendage drag can be re-written as in Eq. 4, where λ is the scale ratio of the linear dimensions of the model and ship.

$$\frac{R_{AppS}}{R_{AppM}} = \frac{\rho_{sw}}{\rho_{fw}} \cdot \lambda^3 \qquad (4)$$

2. The 'Beta' approach, where in this approach a beta factor is used. The beta factor is derived from a geosim set of appended models of varying scales. The scaling using the 'beta' approach is defined as in Eq. 5.

$$C_{AppS} = (1 - \beta)\, C_{AppM} \qquad (5)$$

3. Test a larger separate model of the appendage at higher speeds. With large models of the appendages and high flow speeds, for example in a towing tank, higher Reynolds numbers, closer to full-scale values can be achieved.
4. Use of empirical data and equations derived from earlier model tests which can be found in Lasky [6], Peck [7] and Kirkman and Kloetzli [8]. Some examples of the application of these empirical data and equations can be found in Mustaffa Kamal [9].

There are some issues in testing the hull model with and without appendages as mentioned in (1) above. The assumptions that the appendage drag coefficient for a ship C_{AppS} is similar to its corresponding model C_{AppM} is simply not true as there are scale effects present, where the appendage drag coefficient is smaller for a ship than for its corresponding model.

1.3 Appendage Scaling Using the Beta Approach

One solution in estimating the full-scale appendage drag is to scale the appendage drag coefficient using 'beta' approach as recommended by ITTC [5]. As mentioned above in Sect. 1.2 in (2), this beta factor is derived from a geosim set of appended models of varying scales tested in a towing tank. In the present case, SHIPFLOW 6.3 based on the Reynold's Average Navier Stokes Equation (RANSE) was used to derive the beta factor instead of using a geosim set of appended models tested in a towing tank.

Some recent study on the 'Beta' approach can be found in Oliva Remolà et al. [10]. In Oliva Remolà et al. [10], the scaling of appendages using the 'beta' method was explored. In his work, Oliva Remolà et al. [10] validated the 'beta' method using experimental through towing tank test and computational tools using a RANSE CFD code. However due to the existence of flow separation, reliable numerical results could not be obtained using CFD. The relative difference of the drag in full-scale between extrapolated value and the CFD simulated result is at 38.84%. Oliva Remolà et al. [10] commented that the one of the reason of the discrepancies could be the use of SST turbulence model which is not the most reliable model in computing the Reynold's stress. Therefore, it is the aim of this present work to explore the 'beta' factor using CFD.

2 Methods

2.1 Case Study

This case study used the hullform of R/V Athena, which is a high speed transom stern ship [11]. The hull lines and the particulars of R/V Athena or Model DTMB 5365 are shown in Fig. 1 and Table 1 respectively. Resistance experiments were carried out at the David Taylor Naval Ship Research and Development Center (DTNSRC)
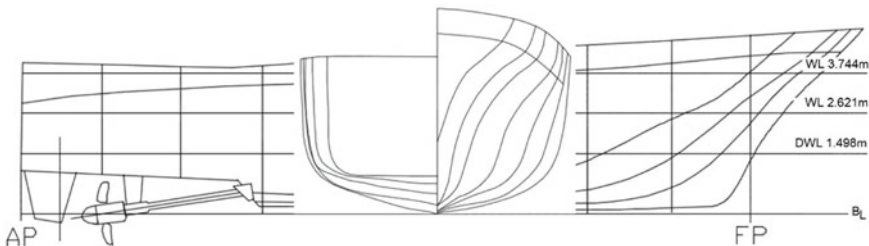
Fig. 1 The sheer and body plan of R/V Athena. note that the appendages of R/V Athena which consists of a pair of rudder, a pair of 'P' bracket, two propeller shafts and a pair of fin stabilizer (not shown in the figure)

Table 1 Ship and model particulars of R/V Athena

| Description		Ship	Model
Scale factor	λ	1	8.25
Length between perpendiculars	L_{PP} (m)	46.9	5.69
Length at waterline	L_{WL} (m)	46.9	5.69
Overall length	LOA (m)	47.03	5.7
Breadth	B (m)	6.9	0.84
Draught	T (m)	1.51	0.183
Trim angle	(deg)	0.0	0.0
Displacement	Δ(tonnes)	229.00	0.397
Wetted surface	S_W (m^2)	287.36	4.222
Block coefficient	C_B (−)	0.4775	0.4775

by Jenkins [12]. Jenkins conducted towing tank experiments on a 1/8.25 scale model of R/V Athena over a Froude number range of 0.28–1.00 with the model was made free to sink and trim. This hull form was chosen for this case study because of the extensive amount of experimental evaluations of total, residuary and wave resistance coefficient are available for this hull which can be found in Jenkins [12] and Gadd and Russell [13].

2.2 CFD Computations

The investigation was conducted using a commercial CFD code SHIPFLOW 6.3 which is available from FLOWTECH International AB. The solver used in the SHIPFLOW code is a viscous flow RANSE solver XCHAP. XCHAP is a finite volume computation using the Explicit Algebraic Stress Model (EASM) as the turbulence model. A global approach was chosen for the computation where the computational domain was built using the H-O grid topology. All the grids in this study were created using the SHIPFLOW in-house grid generation module XGRID.

The viscous flow computations were carried out with the computational domain having six boundaries as shown in Fig. 2. The distance between the inlet of the viscous flow and the fore-perpendicular of the ship is at half the length perpendiculars, L_{PP} of the ship. The outlet of the viscous flow is located at 1.5 of length perpendiculars, L_{PP} of the ship behind the aft-perpendicular of the ship. The radius of the cylindrical outer boundary is at 1.0 length perpendiculars of the ship. This is necessary to prevent from any influence of the blockage effect due to shallow depth of the domain boundary.

The main appendages of R/V Athena twin screw hull are two rudders, a pair of bilge keels, two shafts and two shaft brackets as shown in Fig. 3. The simulations in SHIPFLOW were conducted in model scale and in full-scale from bare hull until the hull was fully appended. Overlapping grids were used in the computation of the
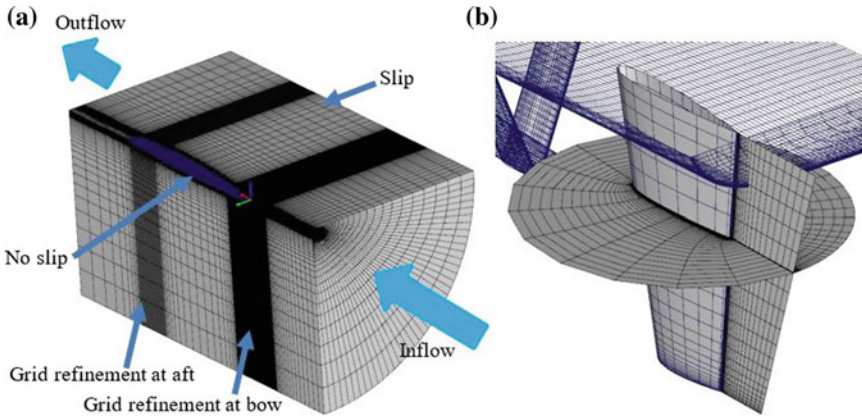
Fig. 2 **a** The computational domain used the H-O grid topology, with the inflow and outflow of the viscous flow computation shown above. No slip condition was applied to the boundary region close to the hull. Note that grid refinements were applied at the bow and the aft region. **b** The overlapping grid of the rudder. A grid clipping of the rudder grids is shown in the figure above
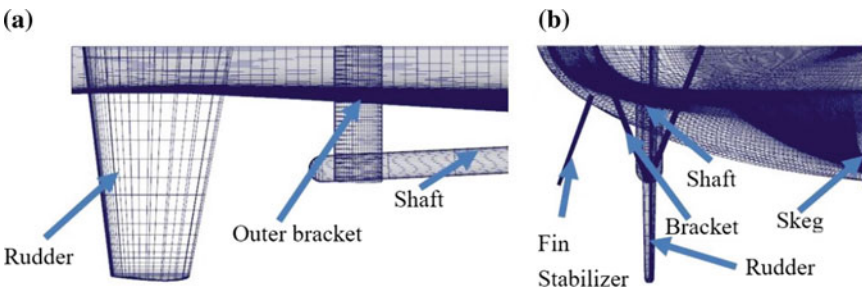
Fig. 3 **a** Profile view of the mesh of R/V Athena showing the mesh of the bare-hull with rudders, shafts and outer brackets. **b** The aft view of the mesh of R/V Athena showing the mesh of the bare-hull with rudders, shafts, fin stabilizer and skeg

appendage drag where the O-O grids of the appendages i.e. rudder overlapped with the cylindrical H-O grid which was meant for the hull geometry as shown in Fig. 2b.

2.3 Grid Dependence Study

A grid dependence study was performed in order to define the optimum number of grid cells to be used in the CFD simulations. The study was done using the results of total drag, frictional drag and the viscous drag coefficients of the Athena's bare hull as the main criterion in the study. It was found that the most optimum number

of grid cells are at 2.3 million grid cells. Fine grid settings were emphasized at the forward and at the midship region of the Athena's hull.

2.4 Validation Study

A validation study was made based on the experimental results which can be found in Jenkins [12]. In this study, the results of Athena's bare hull drag were compared with the experimental results from Jenkins [12]. The comparison of the simulated CFD total drag with the experimental total drag from Jenkins is shown in Fig. 4. The percentage differences between the CFD computed total drag and the experimental total drag were ranged from −14.8 to 16.3%. There are several reasons

for the disagreement beyond model speed of 5 m/s. The free surface computations was done using inviscid panel method instead of using viscous surface capturing method or Volume of Fluid method (VOF). Therefore the disagreement could be attributed by the inability of the inviscid panel method itself to capture the free surface accurately at higher speed where wave resistance dominates, over-estimating the wave resistance, hence the total resistance of the model.
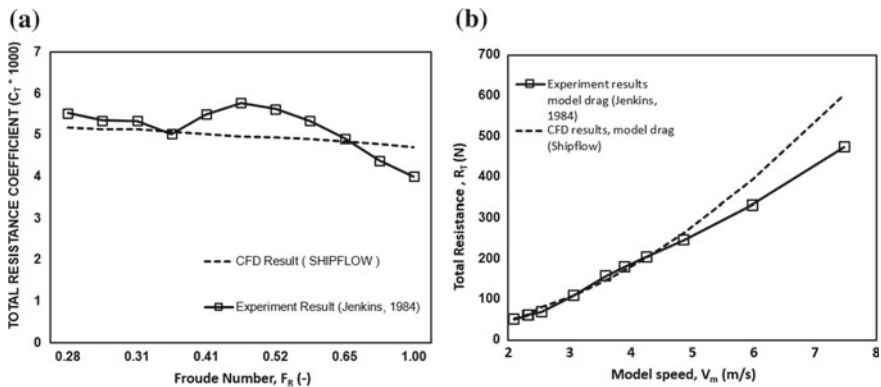

Fig. 4 Validation of simulated CFD results with experimental results from Jenkins (1984). **a** The total drag or resistance coefficient with respect to the Froude number. **b** The total drag or resistance with respect to the model speed in m/s

3 Results

3.1 Drag of Each of the Appendages

The simulations in SHIPFLOW on the appended hull of R/V Athena were done in different stages in order to determine the drag contribution of each of the appendages. The stages of computations were divided into five stages namely (A) Bare-hull with rudders (B) Bare-hull with rudders and shafts (C) Bare-hull with rudders, shafts and brackets (D) Bare-hull with rudders, shafts, brackets and fin stabilizer and (E) Barehull with rudders, shafts, brackets, fin stabilizer and skeg. The computations in stages are necessary in order to quantify the percentage increase in drag contributed by each of the appendages. The percentage increase in drag contributed by each of the appendages in model scale and in full scale is shown in Fig. 5a, b.

Overall, for both model scale and full scale, the drag increased by about 1%. The shafts, brackets and the fin stabilizer contributed to drag increase at all range of speed at about 1% and 2% respectively. The fully appended hull contributed to a bigger margin of appendage drag increase at about 8% in model scale and 7% in full-scale. The fully appended hull i.e. condition E, contributed to a bigger margin of appendage drag increase at about 14% in model scale and 11% in full-scale. It was observed that the skeg contributed to the highest percentage of drag at about 7–8% of the total drag.
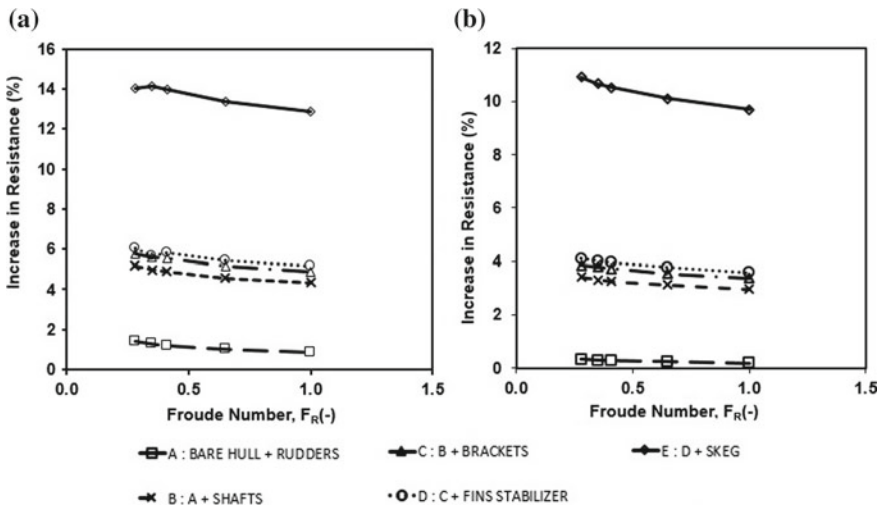
Fig. 5 The plot of accumulated increase in drag or resistance with respect to Froude number contributed by each of the appendages. **a** Model scale. **b** Full scale

Fig. 6 The comparison of appendage drag coefficients in model scale and in full-scale. It should be noted that the appendage drag coefficient in model scale was higher than the drag coefficient in full-scale
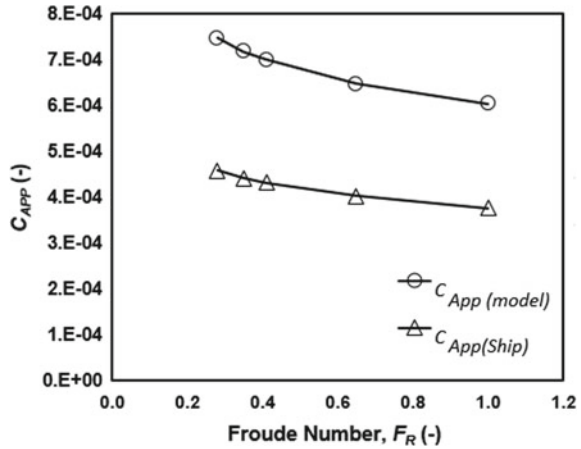
3.2 Estimation of the Beta Factor

Figure 6 shows that the appendage drag coefficient of the model is higher than the appendage drag coefficient in full-scale. The values of beta can be obtained using Eq. 2 in the form of $\beta = 1 - C_{AppS}/C_{AppM}$. Table 2 shows the calculated values of beta for Froude number 0.28 up to 1.0. It was found that the values of beta were almost consistent around 0.38–0.39. This agrees well with Oliva Remolà et al. [10] comments that the beta factor depends on the Reynold's number and it is usually in between 0.3 and 0.6. It should be noted that this beta factor found in this study applies strictly to R/V Athena and similar ship form.

4 Conclusion

In this paper, the scaling of appendages of R/V Athena using the beta method was explored. The drag or the resistance contribution of each of the appendages fitted to R/V Athena were investigated. The appendage drag on R/V Athena was highlighted through a comparison study between the model scale and the full-scale drag. The drag of the appendages was simulated using SHIPFLOW. The drag increased up to 14% in model-scale and up to 11% in full-scale. The skeg was the highest contributor of drag with 7–8% increase in drag. The drag coefficient in model-scale was found to be higher that the full-scale drag. This proved that the scale-effects were present in the model-scale which results in higher drag coefficient in model-scale. An equation for force scaling of the drag coefficient was developed. The equation for the scaling is $C_{AppS} = (1 - 0.38) \, C_{AppM}$. It should be noted that this equation only applies strictly to R/V Athena and similar ship form.

Table 2 Appendage scaling results of RV Athena

F_R	$C_{T(appended)\,Ship}$ * 1000	$C_{T(bare\text{-}hull)\,Ship}$ * 1000	$C_{T(appended)\,Model}$ * 1000	$C_{T(bare\text{-}hull)\,Model}$ * 1000	$C_{App\,Ship}$ * 1000	$C_{App\,Model}$ * 1000	β
0.28	4.661	4.20	5.926	5.18	0.458	0.748	0.39
0.35	4.586	4.14	5.808	5.09	0.442	0.719	0.39
0.41	4.536	4.10	5.727	5.03	0.432	0.700	0.38
0.65	4.393	3.99	5.502	4.85	0.403	0.648	0.38
1.00	4.267	3.89	5.307	4.70	0.377	0.604	0.38

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