STUDY OF FROUDE AND HUGHES METHODS BY NUMERICAL TOWING TANK

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SUMMARY

The resistance of a cargo ship is calculated by numerical towing tank. RANSE multi-phase parallel solver with K-ω SST turbulent model and VOF formulation is applied. Computational results from double model (without free surface) are used to obtain 1+k in Hughes' method and those with free surface are analyzed by both Froude and Hughes' approaches to investigate model and full scale correlation. ITTC recommended uncertainty study is carried out to evaluate numerical error due to grid density. The computed wave elevation, wake distribution and resistance components by fine, medium and coarse meshes are cross-compared and validated against experiment data where applicable. It is found that grid resolution has most effect on wave pattern. The predicted friction and viscous-pressure resistance coefficients are relatively grid independent from present numerical simulation.

NOMENCLATURE

K	Turbulence energy (kg m^2 s^{-2})
P	Pressure(N m^{-2})
S	External body force (N m^{-3})
$\vec{V}$	Velocity vector (m s^{-1})
$\vec{g}$	Gravity vector (m s^{-2})
τ	Stress tensor (N m^{-2})
ρ_w	Density of water (kg m^{-3})
ρ_a	Density of air (kg m^{-3})
ρ	Mixture density (kg m^{-3})
μ	Mixture viscosity (N s m^{-2})
r_w	Volume fraction of water
r_a	Volume fraction of air
μ_w	Dynamic viscosity of water (kg m^{-1}s^{-1})
μ_a	Dynamic viscosity of air (kg m^{-1}s^{-1})
ω	Specific dissipation rate (kg s^{-1})

1. INTRODUCTION

The classic Froude's law of comparison (1868) has being widely applied in the extrapolation of model test data. Later Hughes (1954) proposed the concept of form factor to consider three-dimensionality, which results in a variant of so called 1+k method. The form factor k is assumed to be independent of Reynolds number and is the same for all similar models and ships. How is the validity of this assumption? It is difficult to prove the definition from traditional towing tank tests due to limit of model size, influence of turbulent stimulator and wave-viscous interaction. The wind tunnel tests of double model may give answer to the question. But no such tests have been carried out until now. In the practice of towing tank, the form factor is derived from a series of towing tests having low Froude number rangeing 0.08-0.12 to minimise wave effects. The uncertainties of such measurements and therefore form factor are high due to size of facility, control of flow regime, small loads and approach of data processing. It was seen that wide scatter of form factor value exist for the same hull form from research organisations [1]. There is a necessity of international collaboration to resolve the inconsistency of form factor from ship research institutes.

With the rapid advance of Computational Fluid Dynamics, the numerical towing tank provides an economic and powerful tool to predict ship resistance and obtain form factor computationally. All limitations from model tests such as model size, turbulence stimulation, and wave-viscous interaction can be easily overcome by computer simulation.

In this paper, the results of RANSE simulations of flow around a cargo hull with and without free surface (double model) are presented. The commercial CFD package FLUENT 6.3.26 is used. The computed resistance is firstly verified by grid independent study and then validated by model test data. Afterwards, the form factor is derived by Hughes method. The validity of the method will be dscussed. Finally, the comparison will be made of data extrapolation from model to full scale by Froude and Hughes methods.

2. MATHEMATICAL FORMULATIONS

The Reynolds averaged Navier-Stokes equations with SST K-ω turbulence model were solved. The governing equations can be written as below.

2.1 CONTINUITY EQUATION

$$\nabla \cdot \vec{V} = 0. \tag{1}$$

2.2 MOMENTUM EQUATION

$$\frac{\partial}{\partial t}(\rho\vec{V}) + \nabla \cdot (\rho\vec{V}\vec{V}) = \rho\vec{g} - \nabla P + \nabla \cdot \tau + S \tag{2}$$

2.3 TURBULENCE MODEL

$$\frac{\partial}{\partial t}(\rho K) + \nabla \cdot (\rho\vec{V}K) = \nabla \cdot (\Gamma_K \nabla K) + G_K - Y_K \tag{3}$$

$$\frac{\partial}{\partial t}(\rho\omega)+\nabla\cdot(\rho\vec{V}\omega)=\nabla\cdot(\Gamma_{\omega}\nabla\omega)+G_{\omega}-Y_{\omega} \quad (4)$$

$$\mu_t=\alpha^{*}\rho K/\omega$$
$$\Gamma_K=\mu+\mu_t/\sigma_K$$
$$\Gamma_\omega=\mu+\mu_t/\sigma_\omega$$

2.4 VOF EQUATION

$$\frac{\partial}{\partial t}(r_w)+\nabla\cdot(r_w\vec{V})=0. \quad (5)$$

2.5 BOUNDARY CONDITION

The sketch of computational domain is shown in Figure 1 which constitutes inlet, side boundary, top, outlet and hull.

The inlet is located at one ship length in front of bow where flow variables are specified.

Side boundary is assigned as slip wall, which is located at one ship length away from axis X.

Outlet is two ship lengths behind where hydrostatic pressure is prescribed.

Wall function is used on hull boundary to save computer time.

2.6 NUMERICS

Second order upwinding interpolation for convection flux was used. SIMPLE method was applied to obtain pressure. Geometric reconstruction of volume fraction is used to capture free surface. The detail of theory and usage can be found in FLUENT manual.

3. TEST CASE

A Cargo ship (Series 60) was selected for numerical analysis. The main dimensions are listed in table 1.

Table 1: Main dimensions

	model scale	full scale
Lpp [m]	3.048	121.92
B [m]	0.406	16.256
T [m]	0.163	6.502
CB	0.60	0.60
Displacement [m3]	0.121	7744.0
Wetted surface [m2]	1.579	2526.4

The Froude numbers are 0.16 and 0.316, which give model speed of 0.87m/s and 1.73m/s respectively. The model tests data are available from IIHR at these speeds.

4. MESHES

For the purpose of uncertainty studies, three meshes with refining factor $\sqrt{2}$ were generated. Total cell numbers are 0.24M, 0.58M and 1.63M respectively. The Y plus value at hull boundary is around 50 for the use of wall function.

The boundary mesh is shown below

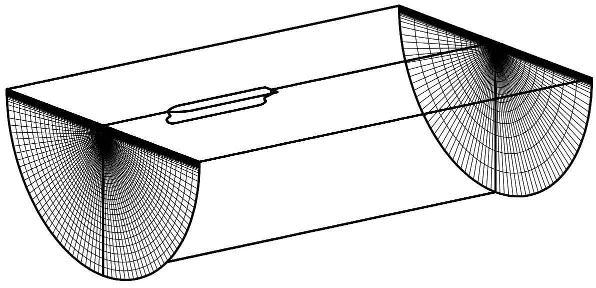

Figure 1 Boundary mesh and computational domain

5. VERIFICATION AND VALIDATION

The computations were run on 1080 processor-HPC (High Performance Cluster) of Faculty of Engineering. It normally takes a few hours for small meshes and 2-3 days for fine mesh for a calculation using 8 processors. The computational results of resistance at Froude number 0.316 by three refining meshes are given in table 2 and used for grid uncertainty study following the guideline of ITTC resistance committee. The verification and validation results of resistances are summarized in table 3-4.

Table 2 Grid convergence study

Grid	Coarse	Medium	Fine	Data
$1000C_T$	5.25	5.20	5.17	5.42
$1000C_P$	1.82	1.77	1.75	1.91(C_R)
$1000C_F$	3.43	3.43	3.42	3.51(ITTC)

Table 3 Verification of total resistance

R_G	P_G	C_G	δ_G	U_{GC}	$1000S_C$
0.55	1.7	0.81	-0.05	-0.01	5.12

Table 4 Validation of total resistance

	E%	U_V%	U_D%	U_{SN}%
E_C	8.1	2.51	2.50	0.2

The computed and measured pressure resistance and friction resistance are also shown in table 2. The comparison indicates the computed friction resistances by three meshes are close, but 3% lower than ITTC correlation line.

ITTC line: $C_F=\dfrac{0.075}{(LOG_{10}Rn-2)^2}$

This is attributed to three-dimensionality from bow and stern. ITTC friction line tends to overestimate 3D effects by 2-3% at model Reynolds number.

To compare, ATTC line and Hughes line are employed below.

Hughes line: $C_F = \dfrac{0.066}{(LOG_{10}Rn - 2.03)^2}$

ATTC line: $\dfrac{0.242}{\sqrt{C_F}} = LOG_{10}(Rn \times C_F)$

The friction resistance coefficients are 3.40E-3 and 3.13E-3 respectively by ATTC and Hughes line.

Therefore, the computed friction resistance coefficient agrees well with ATTC line but is higher than Hughes line. Hughes friction line is 10% lower than both ATTC and ITTC lines at model Reynolds number.

The effect of grid resolution on pressure resistance (wave resistance plus viscous pressure resistance) is clear in table 2. The change of pressure resistances due to mesh is roughly 3-5%. The good thing is pressure resistance converges consistently with increasing grid density.

We cannot separate wave and viscous-pressure resistance easily or conclude the effect of grid density on wave or viscous-pressure resistance by free surface calculation or model tests. The double model calculations can exclude wave effects from viscous resistance as shown in next paragraph.

In Froude's method, residual resistance coefficient is derived from measured total resistance coefficient as below:

$C_R=C_T- C_F$

Thus, C_R includes both wave resistance which is only Froude number dependent and 3d viscous effects subtracting 2D friction which is Reynolds number dependent. To use C_R from model scale to full scale extrapolation will result in overestimation of total resistance due to neglect of scale effect of viscous-pressure resistance. The introduction of form factor k enables to consider the scale effect of viscous-pressure resistance and makes the method of extrapolation more justified. In Hughes' method, residual resistance is derived as below:

$C_R=C_T- (1+k)C_F$

In this approach, we obtain corrected residual resistance which is only Froude number dependent (wave resistance). The viscous-pressure resistance is combined with friction resistance by form factor k, which will be discussed in next paragraph.

Based on the results in table 2, we can carry out verification study of total resistance. The converging criteria R_G is 0.5. The correction factor is 0.81. The order of accuracy, first order RE estimate, corrected uncertainty and solution are given in table 3.

Validation is performed using corrected computed results as summarised in table 4. The simulation uncertainty is smaller than data uncertainty. And validation uncertainty is smaller than validation error, which suggests simulation results are not validated. The similar results are obtained in ITTC quality manual [5]. Although the discrepancy was attributed to modelling error in ITTC report, it may come from stern tube and propeller hub which was not simulated in both calculations.

6. FORM FACTOR

Form factor 1+k is defined as the ratio of viscous resistance (total resistance subtracts wave resistance) to flat plate resistance. As wave resistance can not be easily deducted from measured total resistance, Prohaska method is widely used to estimate form factor. In Prohaska method,

$C_T = (1+k)C_F + CF_r^n$

Where C_F is ITTC friction line, Fr Froude number and n is integer of 4-6

As Min [1] pointed out in their study from a series of model tests spanning 15 years, 1+k is sensitive to model size and turbulence stimulator. It is actually Reynolds number dependent and increasing with the increase of Reynolds number. It is expensive to study the relationship of form factor and Reynolds number by model tests.

In our study, we use computational methods instead to derive form factor. Two computational methods are tested. One is the same as in towing tank practice. The resistance calculations are performed at low speeds (low Froude numbers) and results are regressed to obtain form factor. Another method is based on double model calculations (similar to wind tunnel model tests). The wave effect is excluded completely in this method.

Similar to the practice of towing tank, 5 RANS calculations with free surface were made. The computed total resistance are given in table 5.

Table 5 computed C_T vs Fr

Fr	0.08	0.09	0.10	0.11	0.12
$1000C_T$	5.58	5.34	5.10	4.99	4.77

Using Prohaska method, we obtain form factor 1+k=1.21 for power of 4. Obviously, the form factor based on free surface calculations is overestimated. The main reason is grid dependency. The wave pattern and wave resistance are not accurately evaluated. The theoretical wave length is calculated by $\lambda/L=2\pi Fr^2$ or 0.06 for Fr=0.1. At least 50 cells need to be distributed in a wave length for fine resolution. The total cells along the hull will be 800, which is too large for the present computer capability. About 200 cells were used in the present calculation. The theoretic wave height is $H/L=0.5Fr^2$, which is 0.5% for Fr=0.1. The vertical cell number in free surface region is less than 10 for fine mesh, which is not sufficient to capture wave trough and peak accurately.

The computed wave profile and wave pattern shows that more cells are needed in order to capture the short and small wave behaviour. Thus, the accuracy of RANS calculation at low Froude numbers with free surface is not satisfied for estimation of form factor.

Another option to obtain form factor is by double model calculations.

The similar grid uncertainty studies were carried out as with free surface. The results are given in table 6.

Table 6 Grid convergence study

Grid	Coarse	Medium	Fine	Data
$1000C_T$	4.35	4.35	4.35	
$1000C_P$	0.36	0.36	0.36	
$1000C_F$	3.98	3.98	3.98	4.01(ITTC)

As we can see from table 6, the difference of resistances due to mesh resolution is negligible. The symmetry condition on free surface saves computing time and reduces the error due to short and minor wave components. Actually, for the calculation of form factor, wave effect should be removed. Therefore, double body calculation is an ideal solution to obtain 1+k.

The computed results of double model using fine mesh are given in table 7. As we can see from the table, the computed form factors do depend on Reynolds number. The speeds of three calculations correspond to Froude numbers 0.16, 0.316 (model scale) and 0.316 (full scale). At model scale, the form factors from low and medium speed are identical. However, for full scale, Reynolds number is three orders higher and form factor is larger. The stronger bilge vortex at full scale tends to increase form factor.

Table 7 Grid convergence study

$Re*10^{-6}$	2.12	4.19	1059	
$1000C_T$	4.34	3.78	1.74	
$1000C_F$	4.01	3.51	1.52	ITTC
1+k	1.08	1.08	1.14	

In order to evaluate form factor effects, the scaled total ship resistances using Froude and Hughes methods are compared.

The scale ratio is 46.59 and Froude number is 0.316 corresponding to a speed 21.2 Knots at full scale. ITTC correlation line gives $1000C_{Fs}=1.52$ and $1000C_{Fm}=3.51$ Using corrected computed resistance coefficient, $1000C_{Tm}=5.12$

Froude method predicts:
$C_{Ts}=(C_{Tm}- C_{Fm})+ C_{Fs}=3.13*10^{-3}$

Hughes method gives:
$C_{Ts}=[C_{Tm}-(1+k)C_{Fm}]+ (1+k)C_{Fs}=2.97*10^{-3}$
Where: k=0.08 from model scale calculations.

If we take into account the difference of form factor at full scale and model scale and use form factor 0.14 from full scale:

$C_{Ts}=[C_{Tm}-(1+k_m)C_{Fm}]+ (1+k_s)C_{Fs}=3.06*10^{-3}$.

Therefore, the predicted total resistance by full-scale form factor is roughly 2% lower than that by Froude method but 3% higher than that by standard 1+k method.

The results show that total resistance by Froude method is overestimated due to neglect of shape effects while it is underestimated by standard 1+k due to neglect of the scale effect of form factor. The predicted total resistance using form factor computed by double model calculation at full scale provide more sensible results.

The results of flow field computed by three meshes are compared and validated by model test data in the following part to gain confidence of numerical results.

The wave profiles in Figure 2 shows the comparison of computation and model tests at Froude number 0.316. The computed peak and trough of wave profile generally agrees well with experiment data. The fine mesh gives slightly better prediction than coarse mesh.

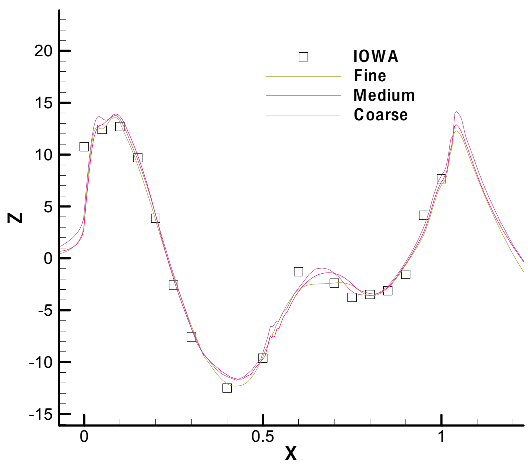

Figure 2 Comparison of wave profiles (Fr=0.316)

The comparison of wave elevation between calculation and measurement is given in Figure 3 and 4. The computed near field wave pattern coincides well with the measured as shown in Figure 3.

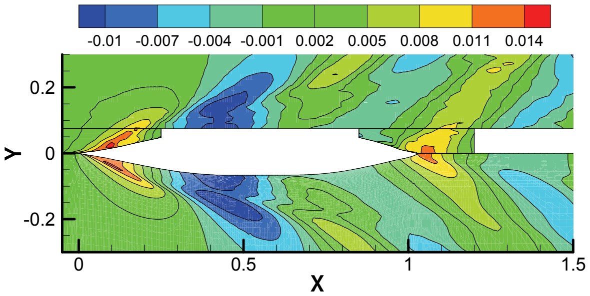

Figure 3 comparison of wave elevation
(Upper: measurement IOWA, Lower: present calculation)

The far field wave pattern is compared in Figure 4. As we can see, measured wave pattern is affected by severe tank wall deflection, while the computed one shows both divergent and transverse waves reasonably well.

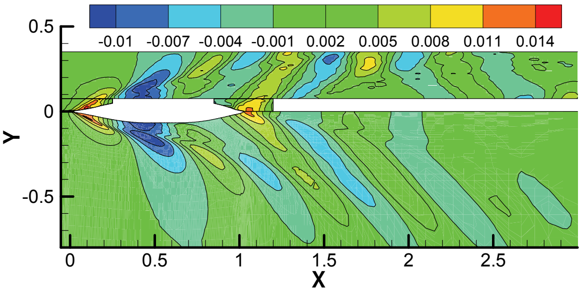

Figure 4: far field wave pattern
(Upper: measurement, IOWA, Lower: present calculation)

The computed wave profiles at Froude number 0.16 are shown in Figure 5. The differences among meshes are more noticeable than those at higher Froude number.

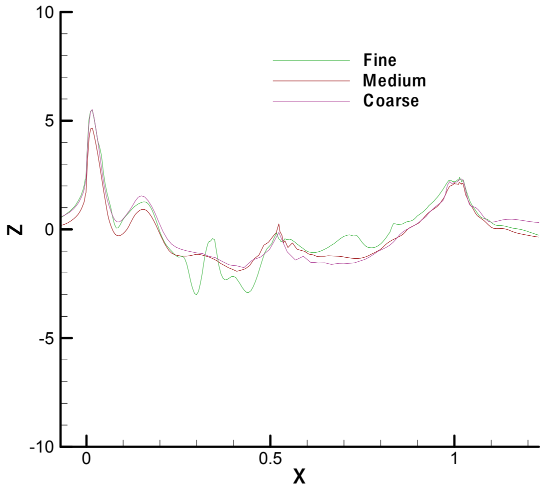

Figure 5 Comparison of wave profiles (Fr=0.16)

The computed wave profiles by coarse and medium meshes exhibit strong numerical diffusion due to insufficient mesh resolution. The result by fine mesh shows improved short wave detail. However, even for fine mesh, the mesh density is not sufficient for accurate prediction of low speed wave pattern as explained above.

The velocity distribution at propeller plane is shown in Figure 6. The computed contour of longitudinal velocity and cross vector generally agrees well with the measured.

In summary, we can conclude the predicted field quantities are of satisfying accuracy at service speed.

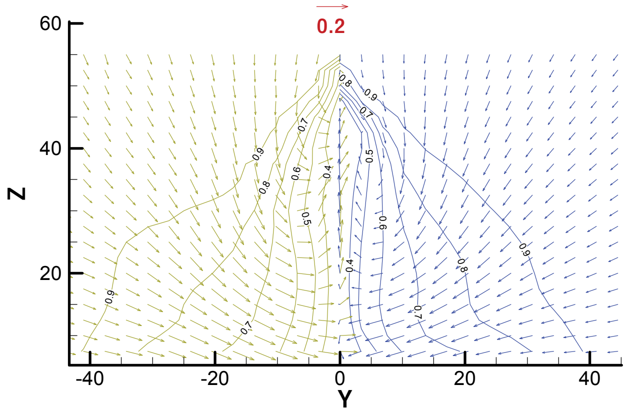

Figure 6: Flow field at propeller plane
(Left: measurement, IOWA, Right: present calculation)

5. CONCLUSION

The attempt to predict ship resistance by numerical towing tank is made in the study. The calculations include free surface flow around ship and double body simulation. Based on the verification and validation of numerical results, the following conclusions are drawn:

- Uncertainty analysis shows that the grid convergence is good for medium Froude number and poor for lower Froude number in the calculations with free surface. At medium Froude number, the grid convergence criteria is satisfied and order of accuracy is acceptable. However, at low Froude number, grid density is not fine enough to capture feature of short and minor wave. As the proportion of wave resistance is small at low Froude number, it is justified to neglect free surface effects and simplify to double model calculations.
- Grid density effect is not significant for double model simulations. This is possibly attributed to the use of wall function.
- Resistance coefficient is overestimated by Froude method due to overestimation of viscous pressure resistance and underestimated by standard 1+k method due to underestimation of form factor at full scale.
- The computed form factors show Reynolds number dependent. At full scale, 1+k is roughly 5% larger than that at model scale.
- As iteration is needed to consider trim and sinkage effects, the attitude change is not considered in the present work. Therefore, all calculations are for even keel at captive condition. In the future work, attitude effects will be taken into account for free model simulations.

6. ACKNOWLEDGMENTS

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7. REFERENCES

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