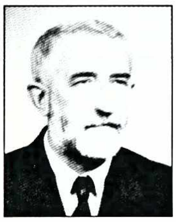

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A calculation method of ship frictional resistance on confined waterway

SUMMARY

In the paper a determination method of ship frictional resistance coefficient with accounting for limited waterway depth, is presented. The relationship between frictional resistance coefficient and the Reynolds number referred to waterway depth, was determined on assumption of fully developed turbulent flow.

Example calculations of total resistance of a motor cargo boat by using the proposed method is also attached.

INTRODUCTION

The confinement of a waterway results in phenomena which do not occur during ship operation on unconfined water. The phenomena include :

- a marked increase in resistance
- a change in the vessel's position relative to the undisturbed surface of water (sinkage and trim).

In addition, the so-called backflow occurs resulting in an increase of the flow velocity along the ship hull. Consequently, the velocity of the flow around the vessel is considerably higher than the sailing speed. An analysis of the ship motion on confined waterway shows that when the critical velocity is approached the ship's resistance and sinkage increases considerably. The critical velocity is linked with the rate of energy transfer in wave motion and for shallow water it is expressed as follows :

$$V_c = \sqrt{g \cdot h} \qquad (1)$$

where :
g – gravity acceleration
h – waterway depth.

In practice, freight vessels operating on confined waterways never reach the critical velocity. In this case the term limit speed [3] is used. The limit speed is estimated as [2] :

$$V_l = (0.5 \div 0.65) V_c \qquad (2)$$

The upper value applies to motor cargo boats, and the lower one to two - and three - row push trains. Tests show that ship resistance increases rapidly when the limit speed is exceeded. The increase is due to the increased wave-making resistance; also viscous resistance contributes to it. This problem has not been completely solved so far. Results of model tests show that - at speeds close to the limit one - the viscous resistance amounts to 30÷40% of the total resistance. This share increases at sailing speeds lower than the limit speed and it decreases at sailing speeds higher than the limit speed.

The ship resistance on a confined waterway is normally determined in the same way as for unconfined water depth. The frictional resistance is calculated from the ITTC-57 formula or the Schoenherr formula. In both formulas a free turbulent boundary layer is assumed. But in the case of shallow water the flow-around conditions are different. They are similar to those of a flow between two surfaces with the pressure gradient changing in the direction of the flow. Two methods of analyzing the resistance on shallow water were presented at the 20th ITTC conference [6]. One of the methods is based on the classical Froude method according to which the total ship resistance coefficient c_T is :

$$c_{TS} = c_{TM} - \left[c_{FM} - \left(c_{FS} + c_A\right)\right] \qquad (3)$$

where :
lower indices S and M refer to the vessel and the model, respectively;
c_T – total resistance coefficient
c_F – frictional resistance coefficient
c_A – roughness allowance.

In this method the Reynolds number Re is determined with taking into account the backflow velocity :

$$\mathrm{Re} = \frac{(V + \Delta V)L}{\nu} \tag{4}$$

where :

$$\left(\frac{\Delta V}{V}\right)_S = \left(\frac{\Delta V}{V}\right)_M \tag{5}$$

In the other method the shape factor *k* being a function of waterway depth, is used :

$$c_{TS} = c_{TM} - (1 + k)\cdot[c_{FM} - c_{FS}] + c_A \tag{6}$$

The obtained resistance forecasts differ depending on the test facility where the model tests were carried out. Generally, the classical Froude method yields slightly higher values than the shape-factor-based method does, provided that in both cases the ITTC-57 formula for the frictional resistance coefficient is applied.

INFLUENCE OF BACKFLOW VELOCITY ON FRICTIONAL RESISTANCE

The vessel sailing on confined waterway causes a reduction in the cross-sectional area of the flow. The reduction in the flow cross-sectional area results in an increase in the water flow velocity relative to the vessel's hull. The average rate of this increase (often referred to as the backflow velocity) depends on the degree of the area reduction and on the ship hull form, and it may be higher than the sailing speed of the vessel, i.e. $\frac{\Delta V}{V} > 1$. In the case of push trains its value is contained within the range :

$$\frac{\Delta V}{V} = 0.4 \div 1.2$$

and it depends on the type of train formation and the sailing channel. In the classical Froude formula (3) the total resistance coefficient applies to the ship sailing speed. An increase in the ship velocity relative to water is taken into account only by using (4) when calculating the Reynolds number. In such case the resistance forecast value will be lower than that obtained with neglecting the actual ship velocity relative to water. This follows directly from formula (3) and the relationship between the velocity increase and changes in the frictional resistance coefficient. For the ITTC-57 formula the relationship is as follows :

$$\frac{\Delta c_F}{c_F} = -\frac{0.4343}{\log\left(\frac{V\cdot L}{\nu}\right)}\cdot\frac{\Delta V}{V} \tag{7}$$

If the Reynolds number is assumed of a value from the interval of $10^5 \div 10^9$, the variation of the frictional resistance coefficient is contained within the range :

$$\frac{\Delta c_F}{c_F} = (17 \div 2.5)\%$$

If the frictional resistance share in the total resistance is taken into account, then the backflow velocity introduces no significant effect to the total resistance forecast. The effect is smaller than the differences (amounting to 8÷16%) between the propulsion forecasts from tests carried out in different model basins [6].

If the total resistance coefficient, frictional coefficient and reference velocity (being a sum of the sailing speed and backflow velocity) are used in the calculations, the obtained resistance values will be different than those yielded by the classical methods – mainly due to an increase in the frictional resistance. Due to the increase of the ship velocity relative to water, the frictional resistance coefficient decreases, but the frictional resistance itself increases. The following formula expressing the relationship can be derived :

$$\frac{\Delta R_F}{R_F} = 2\cdot\frac{\Delta V}{V}\left[1 - \frac{0.4343}{\log\left(\frac{V\cdot L}{\nu}\right) - 2}\right] \tag{8}$$

The increment of the frictional resistance – for possible values of the velocity increment ΔV and the actual range of the Reynolds number Re – can amount to several tens of per cent, causing a substantial increase in the total resistance.

If the backflow velocity is to be taken into account in a propulsion forecast it must be additionally measured as no such measurements are made in classical resistance investigations. As approximate estimation methods are not very reliable such method of determining the frictional resistance should be developed which will be able to take into account the effect of the limited water depth and to be based on the sailing speed only.

FRICTIONAL RESISTANCE ON SHALLOW WATER

The ship sailing on shallow water is strongly affected by the small distance between the bottom of the waterway and that of the ship, and in this case a free turbulent boundary layer hardly occurs. It can be assumed that the flow is similar to that between two parallel planes one of which is in rest while the other one moves with a velocity corresponding to the ship sailing speed. However this model does not take into account the effect of changes in the pressure gradient during the ship motion on shallow water. Hence it is assumed that the fully developed turbulent flow occurs in the entire flow space and the velocity distribution complies with the logarithmic distribution law.

$$V = V_*\left(A\cdot\ln\left(\frac{y\cdot V_*}{\nu}\right) + B\right) \tag{9}$$

where :

$V_* = \sqrt{\frac{\tau_o}{\rho}}$ – shear velocity

ν – kinematic viscosity coefficient

τ_o – shear stress in the entire flow space

y – distance from the plane in rest (Fig.1)

A,B – experimentally determined constants.

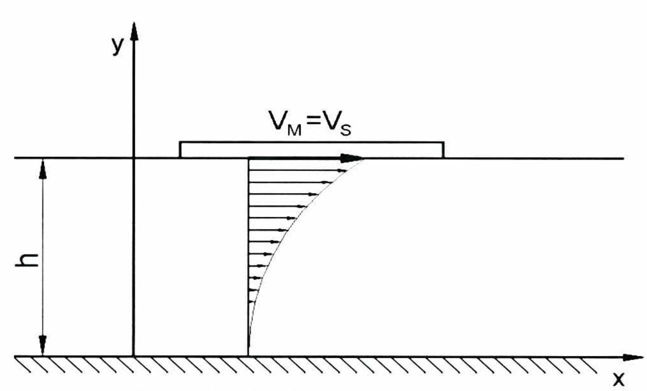

Fig.1. Model of flat plate moving on shallow water

For y = h, $V=V_M=V_S$

For the free turbulent boundary layer, A and B constant values are contained within the following ranges [1,5] :

$$A=2.4\div2.6 \qquad B=3.88\div5.0$$

For further considerations A=2.5 and B=5.5 was chosen (i.e. such as for flow through axially symmetric conduits).

By analogy to the flow through axially symmetric conduits, and assuming a unit plate width, the following relationship between the pressure loss along the length [x-axis] and the shear stress is obtained :

$$\tau_o = \Delta p \cdot \frac{h}{l} \tag{10}$$

If the relation for the frictional resistance coefficient c_F is introduced, namely :

$$c_F = \frac{\tau_o}{\frac{\rho \cdot V^2}{2}}$$

then the following formula is achieved :

$$\Delta p = c_F \cdot \frac{l}{h} \cdot \frac{\rho \cdot V^2}{2} \tag{11}$$

From (10) and (11), by applying the formula for shear velocity [see (9)], the following relation is derived :

$$\frac{V_*}{V} = \frac{\sqrt{c_F}}{1.414} \tag{12}$$

By introducing the notion of the mean velocity as follows :

$$V_m = \frac{\int_0^h V dy}{h \cdot l}$$

and taking into account that : at y = h, V = V_s the following relation is obtained from equation (9) :

$$\frac{V_m}{V_*} = 2.5 \cdot \ln\left(\frac{V_*}{V_m} \cdot \frac{V_m \cdot h}{\nu}\right) + 3 \tag{13}$$

next, by combining (12) and (13) the equation (14) for c_F is obtained :

$$\frac{1}{\sqrt{c_F}} = 1.768 \cdot \ln\left(\sqrt{c_F} \cdot Re\right) + 1.509 \tag{14}$$

where the Reynolds number is expressed as :

$$Re = \frac{V_S \cdot h}{\nu}$$

Formula (14) can be used to calculate the frictional resistance coefficient c_F in function of the Reynolds number related to the distance h between the waterway bottom and that of the hull. In Fig.2 the coefficient c_F versus Re as well as the frictional resistance coefficient computed by using the FLUENT software [4] is presented against the similar relation determined from the ITTC-57 formula. The calculation results from [4] can be regarded as accurate. They show a marked influence of waterway depth on frictional resistance. The values obtained from relation (14) can be regarded (to a certain extent) as average, numerically computed results which simplify the procedure of calculating the frictional resistance coefficient.

TRIAL COMPUTATIONS

The proposed method was applied to compute the frictional resistance of the Oder Motor Cargo Boat for which the model tests were carried out in the Ship Hydromechanics Centre of the Ship Design and Research Centre, Gdańsk [7].

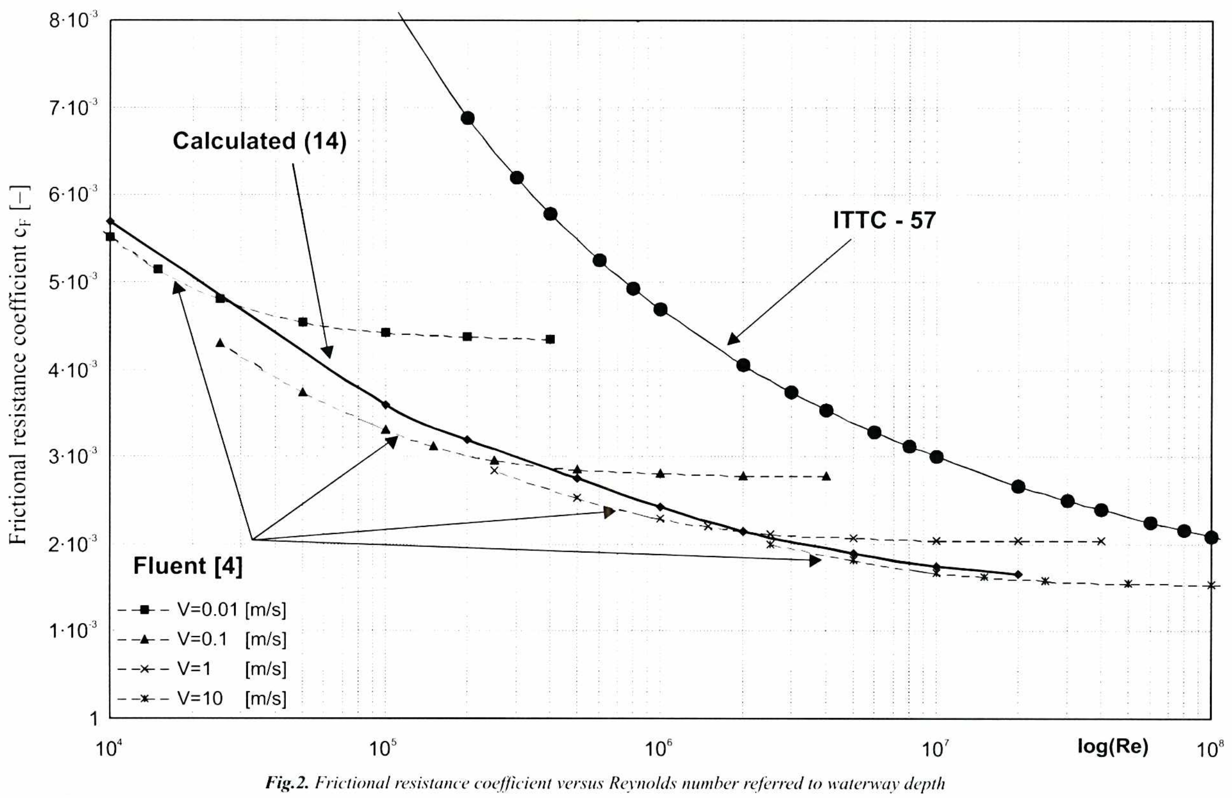

Fig.2. Frictional resistance coefficient versus Reynolds number referred to waterway depth

The draught of the barge was d = 1.6 m, the depth of the water-way: h = 2.0 m and the model scale: $\alpha = 16$.

The forecast based on the classical Froude method (neglecting the backflow velocity) and that based on relation (14) is given in Table, col. 4 and 5, respectively. The resistance values computed by using the proposed method are smaller than those yielded by the classical method. Similar results were obtained by means of the accurate method of calculating frictional resistance [4], which takes into account the limited waterway depth.

Comparison of total resistance calculated by using Froude classic method, (R_{TK}), and that obtained by means of formula (14), (R_T).

$V_S[m/s]$	$V_M[m/s]$	$c_{TM} \cdot 10^{-3}$	$R_{TK}[N]$	$R_T[N]$	$\frac{R_T}{R_{TK}}$
1	2	3	4	5	6
1.111	0.278	10.792	4061	3471	0.855
1.666	0.417	10.072	8629	7041	0.816
2.222	0.556	10.240	15976	14367	0.899
2.777	0.695	11.954	30516	28253	0.9258
3.333	0.833	18.920	74802	71841	0.960

$$d = 1.6\,m \qquad h = 2\,m \qquad \nu = 1.139 \cdot 10^{-6}\,m^2/s$$
$$L = 67.83\,m \qquad S = 773.1\,m^2$$

FINAL REMARKS

The proposed method of forecasting the resistance of a vessel on shallow water takes into account solely the influence of waterway depth on the frictional resistance, expressed in function of the Reynolds number calculated in a different way, namely in relation to the difference between waterway depth and vessel draught.

Acknowledgement

The reported work was supported by grant No. 9 T12C 007 19 of the State Committee for Scientific Research.

Appraised by Tadeusz Koronowicz, Prof.,D.Sc.

NOMENCLATURE

c_A – roughness allowance
c_F – frictional resistance coefficient
Δc_F – change of the frictional resistance coefficient
c_T – total resistance coefficient
g – gravity acceleration
h – waterway depth
k – shape factor
l – length, in general
L – ship length
Δp – pressures difference
Re – Reynolds number
R_F – frictional resistance
ΔR_F – change of the frictional resistance
R_T – total resistance
S – wetted surface
V – velocity, in general
V_c – critical velocity
V_l – limit speed
V_m – mean velocity
V_s – ship speed
V_* – shear velocity
ΔV – backflow velocity
ν – kinematic viscosity
ρ – density
τ_o – shear stress

Lower indices S and M refer to the ship and the model, respectiwely

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Conference

MMAR 2001

On 28 ÷ 31 August 2001 Międzyzdroje, a Baltic Sea side holiday resort, was the place of holding 7th IEEE International Conference on :

Methods and Models in Automation and Robotics

which was arranged by the Institute of Control Engineering, Technical University of Szczecin. The Conference was held under the auspices of IEEE Robotics & Automation Society, which added an importance to it. Also, taking part in its organization by Committee of Automation and Robotics as well as Committee of Metrology and Instrumentation, Polish Academy of Sciences contributed to the importance.

Over 210 papers were submitted to the International Programming Committee, 175 out of which were accepted for presentation. The papers were prepared by authors from 21 countries, inclusive of 12 from European countries, and of Iran, Japan, Singapore, China, New Zeeland, Algeria, Taiwan, Saudi Arabia and Brazil.

The following areas of basic and engineering sciences were covered by the conference papers :

- Control Theory
- Modelling and Simulation
- Computer Integrated Manufacturing (CIM) and Scheduling Problems
- Neural Networks and Fuzzy Logic
- Robotics
- Control Engineering, and
- Identification and Signal Processing.

Moreover one of the sessions, under Polish chairmanship, was devoted to Marine Automation, during which 9 invited papers were presented, namely :

- *A fuzzy fault tolerant control scheme for an autonomous underwater vehicle* – by R. Sutton, A.Pearson (UK) and A. Tiano (Italy)
- *Onboard experiments on weather vaning dynamic positioning/tracking and auotmatic berthing* – by H. Hagiwara, R. Shoji, H. Fukuda (Japan) and J. Pinkster (The Netherlands)
- *Predictive versus fuzzy control of autonomous underwater vehicle* – by M. Kwiesielewicz, W. Piotrowski (Poland) and R. Sutton (UK)
- *Problems of identification and validation of component elements of ship power systems* – by R. Arendt (Poland)
- *A neural network adaptive autopilot for a yacht* – by Ch. Yang (China), Ch. Xiao (New Zealand), A. Tiano and A.Zirilli (Italy)
- *A new fault-tolerant control scheme based on hybrid neural networks* – by T. Tang, Y. Chen and J. Li (P.R. China)
- *Intelligent marine traffic simulator for congested water-ways* – by K. Hasegawa, G. Tashiro, S. Kiritani and K. Tachikawa (Japan)
- *Tracking control of the ship using a nonlinear $H\infty$ control* – by E. Shimizu and M. Ito (Japan)
- *Recent developments in marine control systems* – by R. Śmierzchalski (Poland).