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A study on viscous drag reduction of three dimensional double model

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3次元 二重 模型의 粘性 抗力 減少化 硏究

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3次元 二重 摸型이 粘性 流體中을 운동할 때 발생하는 점성 摩擦力을 減少시키기 위하여 船體 표면에 流線 방향으로 V-홈(Riblet)의 띠를 그 표면에 부착하였다고 가정하여 점성 마찰력 감소에 관하여 亂流 境界層의 特性을 계산하는 수정된 방법을 구하고 그 계산을 위한 프로그램을 작성하였다. 계산 방법으로서는 Hess & Smith의 방법에 의하여 포텐시얼 유동을 계산하고 그것으로부터 구한 유속 값들을 Momentum 적분 방정식에 이용하였다. 補助 방정식으로서는 Head의 식과 점성 마찰력에 관한 Clauser의 식을 사용하였다. 그리고 Riblet의 효과로서는 Gaudet의 실험식을 이용하였다. 그 계산 결과 선체 전표면에 유선의 방향으로 Riblets를 付着하였다고 가정 하였을 경우 상당한 점성 마찰력 減少效果를 나타냈으며 Riblet를 선체 전체길이 4등분하여 각각의 표면에 부착시켰을 때도 현저한 점성 마찰력 감소 효과를 나타냈으며 특히 선수 부분의 25% 표면에 부착되었을 때가 다른 영역에 부착하였다고 가정하였을 경우보다도 가장 優秀한 效果를 나타내었다.

The practical application of riblet to three dimensional double model, for viscous drag reduction, was studied analytically by intergal solution to three dimensional turbulent boundary layers. The case of a V-groove riblet technique on the shear stress and boundary layer velocities were incooperated in the computation of the flow over a smooth slender ship hull. As the results the possible mechanism of turbulent drag reduction by riblets are then suggested based on detailed studies of near-wall turbulence characteristics. And a turbulent boundary layer calculation scheme based on a momentum integral method was modified for the computer program. An example of the calculation results is presented.

Nomenclature

C_F : mean skin friction coefficient
C_f : local skin friction coefficient
G : Clauser's pressure gradient parameter
H : shape factor, $\delta * / \theta$
$\overline{k_B}$: mean apparent negative riblet height
L : ship length
R_n : length Reynolds number
$R_{\delta *}$: displacement Reynolds number
U_δ : local potential flow velocity
$\overline{U}$: temporal mean velocity in x-direction at y
$\Delta \overline{U_R}$: shift of logarithmic profile due to riblets, ΔB
$\Delta \overline{U_P}$: shift of logarithmic profile due to pressure
u_τ : friction velocity, $\sqrt{\tau_W / \rho}$
β : angle between surface shear stress and surface potential flow streamline
δ : nominal boundary layer thickness
δ_* : displacement thickness
θ : momentum thickness
θ' : nondimensional momentum thickness, 2θ/L
ν : kinematic viscosity
ρ : density
τ_W : shear stress at the wall

Introduction

The possibity of getting a net drag reduction by riblets has initially been pursued by Walsh(1982) and his colleagues at NASA. The fact that the riblet can be reduce turbulent viscous shear stress has been ascertained by many studies(1982, 1986) and some models have been proposed to explain the mechanisms of the drag reduction by riblets(small streamwise groove). One of the next interesting step is to apply the riblet to industrial fields. In particular, an application to ship model expected to provide major energy savings. However, most of the riblet studies were carried out in flat plate boundary layers with few works in the past.

The use of riblet for the 1987 America's Cup winner "Star and Stripes" was undoubtedly one of the most sucessful example of marine application of the riblet device. The work described with one third model is a general study of experiments supported largely by British Maritime Technology using their Towing Tank at Feltham(1987).

The purpose of the paper was to find a practical calculation technique of assessing the role of riblet to turbulent boundary layer for the drag reduction over the bow of model hull. The study has been carried out experimentally and will be further pursued. And the present work also is to present the performance of riblet for drag reduction in ship of double model through comparisons of ribleted ship surface with those from smooth ship.

Calculation Methods

There are several methods available for the computation of the potential flow and boundary layers of ships. The present author's have given preferance to personal computer time and convenient usage, and have chosen a Hess and Smith method (1967) for calculation of the potential flow, described by Kim and Lewkowicz(1992). In this method as a basis for the boundary layer characteristics and skin friction resistance calculations, the streamlines and pressure distribution along the streamlines must be determined. For this purpose a potential flow technique modified by Kim and Lewkowicz(1992) has been used. For boundary layer calculation the Kim and Lewkowicz's Version Ⅱ(1992) method was chosen, assuming a small cross flow. Momentum integral equations were used as basic equations, viz.

$$\frac{d\theta_{11}}{d\alpha}+\Theta_{11}(2+H)P-K_1(\Theta_{11}-\Theta_{22})$$
$$=1/2C_{\tau 1} \qquad (1)$$

$$\frac{d\theta_{21}}{d\alpha}+2\Theta_{21}(P-K_1)+\Theta_{11}K_2(1+H)$$
$$+K_2\Theta_{22}=1/2C_{\tau 2} \qquad (2)$$

Head's entrainment equation was chosen as auxiliary equation,

$$\frac{d(\theta_{11}\cdot H_E)}{d\alpha}+\Theta_{11}H_E(P-K_1)=F(H_E) \qquad (3)$$

where

$$F(H_E)=0.0306(H_E-3.0)^{-0.653} \qquad (4)$$

$$H_E=\frac{\delta}{\theta_{11}}-H \qquad (5)$$

$$H_E \approx 1.535(H-0.7)^{-2.715}+3.3 \qquad (6)$$

Kim's profile(1992) was used for the crose-flow velocity representation. To close the system of equations, a skin friction law was added in the following form :

$$\sqrt{\frac{2}{C_{\tau 1}}}=2.5\,ln\,R_{\delta*}+4.1-\frac{\Delta\overline{U_R}}{u_\tau}+\frac{\Delta\overline{U_P}}{u_\tau} \qquad (7)$$

where $\frac{\Delta\overline{U_R}}{U_\tau}$ represents the downward shift in the velocity profile due to riblet as found from experiments by Gaudet(1987). Again, for the turbulent boundary layer calculation the Kim and Lewkowicz method(1992) was chosen. It assumes a small cross flow and is connected with the Kim and Lewkowicz method for the slender body potential flow calculation. As on auxiliary equation Head's added to the momentum integral equation. The following form of the skin-friction law for the boundary layer calculation on the smooth and ribleted surface in moderated pressure gradient was used :

$$\sqrt{\frac{2}{c_\tau}}=2.5\,ln\,R_{\delta*}+4.1-\frac{\Delta\overline{U_R}}{u_\tau}+\frac{\Delta\overline{U_P}}{u_\tau} \qquad (8)$$

The function $\frac{\Delta\overline{U_R}}{u_\tau}$ was found from Gaudet(1987) described below and the pressure gradient contribution to velocity defect law $\frac{\Delta\overline{U_P}}{u_\tau}$ was calculated as function of parameter :

$$\frac{\Delta U_P}{u_\tau}=1.253(G-6.7) \quad \text{for } G\geq 6.7 \qquad (9)$$

or

$$\frac{\Delta U_P}{u_\tau}=0.404(G-6.7) \quad \text{for } G<6.7 \qquad (10)$$

For the estimation of the G value the following numerical approximation was used :

$$G(H)=3.3+\exp[0.4667-2.722ln(H-0.6798)] \text{ for } H\geq 1.6 \qquad (11)$$

or

$$G(H)=3.3+\exp[0.4383-3.064ln(H-0.6798)] \text{ for } H>1.6 \qquad (12)$$

The procedure for the boundary layer calculation described above was programmed for a 80486DX personal computer. The program itself was kept basically intact in its lay-out and structure but physical improvements were incorporated Version V. Ⅱ by making the shift by smooth and riblet.

Thus altered program run well for all tried flow cases free from any numerical peculiarities nor significant increases in the CPU time.

The time is need to calculate velocities, pressures and boundary layer characteristics along 20 streamlines. A practical feature of this program is that it can be used to calculate the boundary layer on a smooth ship hull, on a ribleted hull and on a hull with riblet only on some parts along its length(the location of those riblet areas can be changed). To assess seperately the influenc of the riblet effects above, it was possible to 'switch' them on and off within the program.

Drag Characterization in the Case of a V-groove Riblet

In this study the modified program was applied to a slender hull(Liverpool Model) model shown in Fig. 1 which was tested experimentally in a wind-tunnel at Liverpool University.

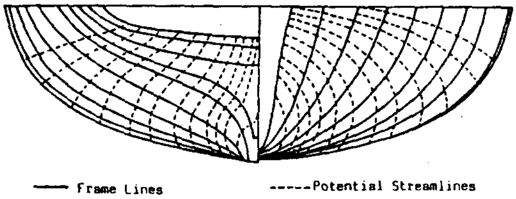

Fig. 1. Body Plan and Potential Streamlines of Model.

The test results provided a data bank of the wall friction coefficient values on the hull in a smooth and ribleted ship modle. As the example, the similarity laws are applied to the case of a V-groove riblet where the lenth factor is h, the depth of the groove and s, the lateral spacing of the groove, respectively. For a particular case of riblets of grooves with $s/h=1.28$, Gaudet(1987) presents drag characterizaition data, ΔB, against $\log_{10}h^+$. Here $h^+=u_\tau h/v$. The beginning point for $\Delta B=0$, $\log_{10}h^+=0.62$, is inferred from a $\Delta B-s^+$ correlation shown by Gaudet.

Three regimes are delineated as follows :

1. $\Delta B=0$, $\log_{10}h^+ < 0.62$. This regime may be termed hydraulic smoothness.
2. $\Delta B > 0$, $0.62\le\log_{10}h^+\le 1.37$ This is a regime of drag redution.
3. $\Delta B < 0$, $\log_{10}h^+ > 1.37$. This is a regime of drag argumention or hydraulic roughness.

The equations may be analytically fitted in piecewise segment as follows :

1. $\log_{10}h^+ < 0.62$, $\Delta B=0$
2. $0.62\le\log_{10}h^+ < 1.06$,
   $\Delta B=1.13-5.84(1.06-\log_{10}h^+)^2$
   $1.06\le\log_{10}h^+ < 1.37$,
   $\Delta B=1.13-11.76(\log_{10}h^+-1.06)^2$
3. $\log_{10}h^+ > 1.37$,
   $\Delta B=-A\log_{10}[1+0.07(h^+-23.44)]$

For maximum drag redution, $\Delta B=1.13$ at $\log_{10}h^+=1.06$, $h^+=11.48$. The effect of riblet on the boundary layer was studied by means of a specially designes balance fitted into the cavitation tunnel at Liverpool

University. The balance records the total drag and the difference in drag between a smooth and a riblet floating element in the two surface sides of a slender hull. The velocities in the boundary layer were measured by means of a pitot tube.

The velocity decrement in the logaritmic part of the boundary layer may be found and based on the friction velocity u_t. The variance of skin friction over the floating element is very near linear, so that mean values for C_t and u_t can be taken for the element.

For a smooth, flat plate we may write according to Clauser and Hama :

$$\sqrt{\frac{2}{c_\tau}} = A\ln\frac{U_\delta\,\delta*}{\nu} + B, \qquad (13)$$

and for a riblet surface :

$$\sqrt{\frac{2}{c_\tau}} = A\ln\frac{U_\delta\,\delta*}{\nu} + B - \frac{\Delta U_R}{u_\tau}$$

Hence at the same displacement Reynolds number :

$$\frac{\Delta\overline{U_R}}{u_\tau} = \sqrt{\frac{2}{c_{\tau Smooth}}} - \sqrt{\frac{2}{c_{\tau Rough}}} \qquad (14)$$

Calculated Example

The boundary layers were calculated for at length Reynolds number $R_n=3.58 \cdot 10^6$ corresponding to a ship velocity of 13.8knots. As starting section station no 19 was chosen and the calculations followed to station no.1. The initial values of R_θ and C_f for each streamline at the starting point were required.

We assumde that these values were the same for all streamlines and equal to 0, 0.7 · 10^{-4} and 0.6 · 10^{-3} respectively for all the smooth hull surface and equal to 0, 0.55 · 10^{-3} and 0.5 · 10^{-3} for riblet hull surface. The cases of six(All Smooth, All Riblet, 1st Riblet, 2nd Riblet, 3rd Riblet, 4th Riblet) correspond to the situation when the whole hull surface is smooth or riblet respectively. In the four remaining cases(3, 4, 5 and 6) 75% of hull surface is riblet and the rest is smooth. This ribleted stripe with length equal to L/4 has a different location along the ship length for each particular case. In the cases when the riblet was taken into consideration the function $\frac{\Delta\overline{U_R}}{U_\tau}$ was calculated by equation :

$$\frac{\Delta\overline{U_R}}{U_\tau} = 0.385\sqrt{\frac{U_\tau\,k_B}{\nu}} \qquad (15)$$

corresponding to experimental results in Gaudet(1987) for V-groove plate with a mean apparent riblet amplitude. The results of the boundary layer calculations are shown in Fig. 2 through 10. In Fig.1 the streamline plots are presented. In Fig. 2, 6, 8, 9 and 2 the changes of calculated local skin friction coefficient and nondimensional momentum thickness along the ship

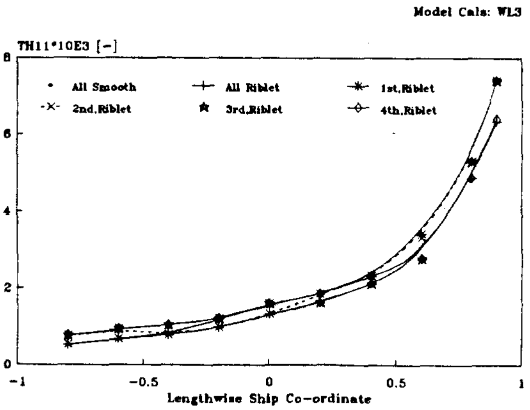

Fig. 2. Streamline momentum thickness.

length and in he cross-sectional direction are shown.

In this figures both results for the completively smooth and completely ribleted surface are presented(case 1 and 2). In Figs. 5, 6, 7, 8, 9, and 10 are shown the local skin friction along the streamline "3" (passing the bilge) for all six cases. As can be seen in these figures the transition from smooth to riblet surface produces a marked jump in C_f values above those for a completely riblet hull surface(case 2). Similarly, the transition from a smooth to a riblet surface results in C_f values below those for a completely smooth surface(case 1). For other streamlines the C_f changes have a similular form.

The nondimensional momentum thicknesses along the streamline "3" for cases 1 to 6 are shown in Fig. 2. For the cases with partially riblet surface the θ' values never exceed the values for a completely smooth surface and never go above the values for a completely riblet surface. The total friction coefficients C_F for the ship model were calculated by integration of C_f over the hull and they aren't given in here.

First, the outcome of the present computations is shown in Fig. 2 where the values of the streamwise momentum thickness, for the model, at probably the most representative WL3, are displayed for six different computational cases : (i) All smooth hull computed(All Smooth), (ii) All riblet hull computed(All Riblet), (iii) First quarter area ribleted ribleted hull computed(2nd, Riblet), (v) Third quarter area ribleted hull computed(3rd, Riblet) and Forth quarter area ribleted hull computed(4th, Riblet). Interestingly, only very little effect on can be seen on either 4th ribleted hull.

The basic streamwise shape parameter, $H=\theta_{11}/\delta_{*1}$, along WL3 is plotted in Fig. 3. It contains six curves for the same cases as shown in Fig. 2. It can be seen that the case from 1 to 6 modification in order imposes a much weaker effect on the parameter, although forth case does reduce slightly H values on the smooth hull. As expected, a strong effect of the surface riblet on ΔH is expectedly conveyed

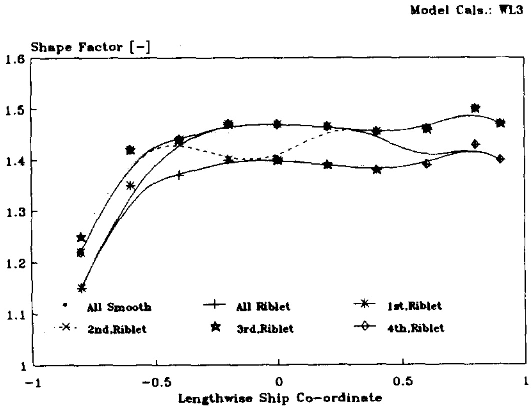

Fig. 3. Streamwise shape factor.

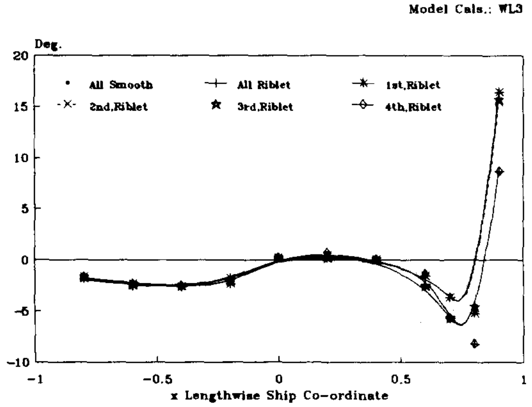

Fig. 4. Surface flow angle.

by the second case modification. What must be realised is that the TBL along WL3 is largely of a flat plate nature(typical to midships). The all riblet covered case and first case come to fore in pressure gradient TBL flows where the upstream history is of importance.

Figure 4 demonstrates the development of the limiting wall cross-flow angle β_0, as invented by Kim(1992), along WL3 :

$$\frac{U_1}{U_2} = (\tan \beta_0 - C\eta)(1-\eta)^2 \qquad (16)$$

where $\eta = \frac{x_2}{\delta}$ transverse co-ordinate; δ= B.L. absolute thickness; C=proportionality parameter in the cross-flow model; subscripts 1 and 2 denote streamwise and crosswise cmponents, respectively. For almost the entire length of hull(extremely except stern), the value of $0° < \beta_0 < -7°$ indicating, as expected, a weak three-dimensional TBL. All computational variants more or less mutually agree throughout the hull and there is minimal surface riblet influence on β_0. Nearer to aft some differences appear: the riblets seems to change things somewhat but it is not easy to discern any systematic trend. At stern, on approach to separation, rapidly grows to values greater than +16° in case of 1, 2 and 3, but except in case of 4th case.

Of main interest in this exercise was to calculate the wall friction along WL3 of the slender hull and compare the predictions with measurements published by Okuno and Lewkowicz(1987). These results were assembled together in Fig.5. From which it is apparent that the riblet modification is beneficial to the predictions on reduction of resistance of hulls alike, although more strongly so for the latter, especially, when combined with local riblet correction.

Datas utilised in Fig. 5 were further processed to obtain the girth averaged values and as such are given in Fig. 6.

Here also it is clear that riblet surface yields quite encouraging drag reduction with most of the smooth of surface.

The wall friction distribution around the hull allows to calculate the friction resistance of the hull(at Fr=Froude number=0) by integrating the C_f over its wetted surface(between WL0 and LWL). This cal-

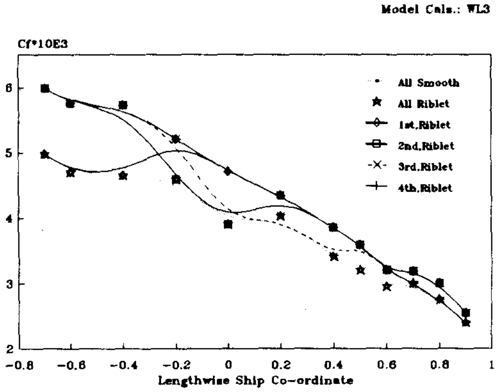

Fig. 5. Local wall friction coefficient.

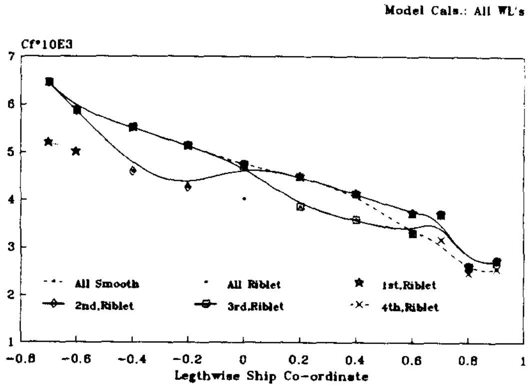

Fig. 6. Girth averaged wall friction coefficient.

culation is only approximate as it is assumed that past separation the wall friction was zero and of course that the presesnt method was predicting separation correcrtly. It transpires from the model test experimental data that the surface ribleted accounts for $\Delta C_f = 1.31 \cdot 10^6$ decrease in the total hull friction coefficient over that for the smooth hull.

Figure 7 shows that modification program underestimates the decrease of total skin friction of wetted surface with rilbet surface. The largest decrease in C_F compared to the smooth hull values was obtained for case for completely rebleted of surface when the ribleted strip was covered between station 1 and station 20. In this case the decreasement of friction by all covered with riblet surface amounts to about 8.2% of the total wetted surface. Case 2, i.e. a riblet stripe between station 1(from F.P.) and station 20(stern) - gives the least decrease in C_F in relation to case 6, but simultaneously the ribleted surface is also the smallest one, about 1.5% of the total wetted surface only.

Zornally Smooth Hull

Policy decisions regarding the economy and effectiveness of hull cleaning, antifouling protection and riblet surface may be made on the basis of the idea of 'zonal' hull treatment. It had been considered in the past. Notably, Baba and Tokunaga (1980), Kauczynski and Walderhaug(1987) as well as Okuno and Lewkowicz(1987) addressed this problem.

Zonal analysis of the hull riblet is usually executed by assigning a state of surface riblet to selected quarters of the hull where each quarter is LWL/4 long.

It was convenient to apply the present modified version in this manner in order to assess how zonal smoothing affects the girth averaged C_f along the slender hull.

The final output of the exercise, shown in Fig. 8, seems to suggest that the effect diminishes quite markedly if the 'riblet' quarter moves progressively aft. Of course, this exercise must be regarded with caution since no proper account could be made of the consequences of zonal smooth-

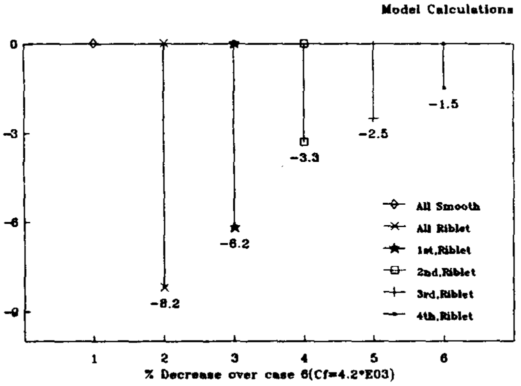

Fig. 7. Riblets caused decrease in hull friction resistance(% below smooth).

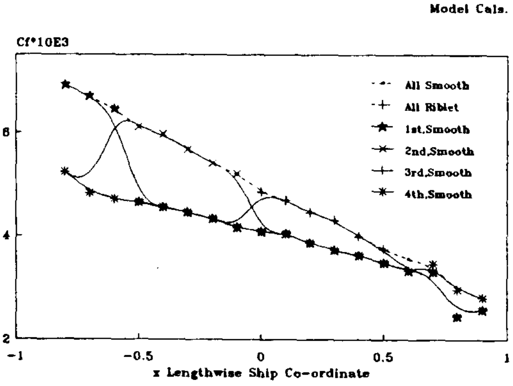

Fig. 8. Effect of zonal hull smooth on girth averaged wall friction coefficient.

ing upon the separated flow at the stern. Nevertheless, the results could serve as a handy indicator of a prevailing trend.

Comparisons with Experimental Datas

Now the new modification version was used to recalculate the TBL on a full size ship whose slender hull had the same shape as the tested model. Link is here made with the sea trials referred to by Nichlson and Lewkowicz(1983), Okuno, Lewkowicz and Nicholson(1985) and Okuno and Lewkowicz(1987).

For those measurements taken the present calculations offer a much improved and are assembled in Fig. 9 in all cases.

The reasons why old version yields so very much poorer results(beyond hitherto encountered diffrences) was not investigated in depth but is thought to have been mainly due to the much higher Reynolds number for the full size hull. At those conditions the riblet effects relationship as a governing equation could steer better the overall TBL solution. For the same reason the correction with ΔH due to riblet effects might also be more effective.

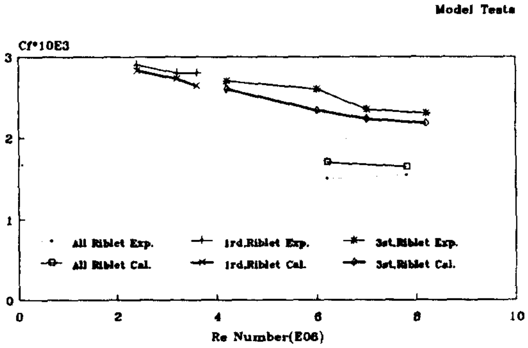

Fig. 9. Comparison of model tests with present calculations.

Finally, when testing hull models and transferring the laboratory data to the full size counterpart, the right technique of scaling becomes of some practical importance. Sasajima and Himeno(1965) proposed a simple formula linking the surface riblet induced decrease in the hull friction coefficient for the model with that for the full size ship. The formula reads :

$$\Delta C_{fs}/\Delta C_{fm} = (C_{fs}/C_{fm})^{n} \qquad (17)$$

and is valid if the riblet Reynolds number $(Uh^{+}/\nu) < 10^{3}$ is similar for the model and the ship. The authors found that $n=2$ yields the best correlation for Eq.(17). The formula was checked experimatally by Baba & Tokunaga(1980) who tested it on two large hull models of similar riblet Reynolds number$(Uh^{+}/\nu)=410$ but of different size. Their hull Re did not change much, i. e. $2.2 \cdot 10^{6} < Re < 9.3 \cdot 10^{6}$. Watanabe et al. (1969) applied the formula to a ribleted model $150 < Uh^{+}/\nu < 760$ and $4.5 \cdot 10^{7} < Re < 7.6 \cdot 10^{7}$ in oder to estimate the riblet friction resistance on a full size ship ('Lucy Ashton') for which $1.52 \cdot 10^{9} < Re < 2.53 \cdot 10^{9}$, but could not actually verify the overall validity of the formula.

The present authors used modification to check if the Sasajima-Himeno formula agrees with the Liverpool University model tests and those for a corresponding full size ship. For the two comparison cases the same riblet function -that for the model tests- was arbitrarily assumed and the riblet Reynolds number set at $Uh^{+}/\nu=400$ for the ship and the model. Computations were performed for a series of ship Reynolds numbers: $10^7 < Re < 10^{10}$. Now, $n = n(Re)$ was abinitio anticipated and Figure 11. gives evidence to this. Whilst $n = 2$(Sasajima and Himeno (1965)) holds around $Re = 10^7$ and for $Re > 10^9$, the present calculations seem to reveal that for $1.1 \cdot 10^7 < Re < 10^9$, n tends to slightly higher values reaching even $1.9 < n < 2.1$.

Conclusion

(1) A calculational method has been presented for boundary layer calculations on smooth and riblet surfaces. An example of this method applied to the estimation of skin friction decrease for different surface conditions has been added. It was our aim to show that this method can be used as a tool for hull form designers and ship owners.

(2) Further modifications of program were implemented to the Kim and Lewkowicz(1992) program for predicting ship-hull turbulent boundary layers aiming at a more realistic representation of the hull riblet effects.

(3) The modifications proved quite successful in calculating more accurately the wall friction coefficient measured on a smooth and local ribleted slender hull model in a wind-tunnel.

(4) It was demonstrated that, disregarding the complex effect upon the BL transition, for zonally smooth hull the most important contribution to the overall hull friction drag comes from the first quarter (counting from bow) of the hull wetted area. The contribution progressively diminishes as successive hull quarter areas fore to aft are individually considered. This analysis is unable to take into account the role of stern separation and other thereto related effects.

(5) An important model-to-ship scaling formula was devised in Himeno & Sasajima(1967) making it possible to estimate approximately the riblet correction to the hull friction resistance coefficient of a full size ship hull from its geometrically similar ribleted model counterpart. The validity of this simple formula was checked and show that, although valid in its original form for some hull Reynolds numbers, it may need to be adjusted if $1.1 \cdot 10^7 < Re < 10^9$.

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