Journal of Sports Sciences, July 2011; 29(10): 1059–1069

Realistic evaluation of hull performance for rowing shells, canoes, and kayaks in unsteady flow

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(Accepted 28 March 2011)

Abstract

In this study, we investigated the effect of hull dynamics in shallow water on the hydrodynamic performance of rowing shells as well as canoes and kayaks. An approach was developed to generate data in a towing tank using a test rig capable of reproducing realistic speed profiles. The impact of unsteady shallow-water effects on wave-making resistance was examined via experimental measurements on a benchmark hull. The data generated were used to explore the validity of a computational approach developed to predict unsteady shallow-water wave resistance. Comparison of measured and predicted results showed that the computational approach correctly predicted complex unsteady wave-resistance phenomena at low oscillation frequency and speed, but that total resistance was substantially under-predicted at moderate oscillation frequency and speed. It was postulated that this discrepancy arose from unsteady viscous effects. This was investigated via hot-film measurements for a full-scale single scull in unsteady flow in both towing-tank and field-trial conditions. Results suggested a strong link between acceleration and turbulence and demonstrated that the measured real-world viscous-flow behaviour could be successfully reproduced in the tank. Thus a suitable tank-test approach could provide a reliable guide to hull performance characterization in unsteady flow.

Keywords: *Rowing shell, canoe, kayak, hull performance, hydrodynamics, unsteady flow*

Introduction

Background and literature review

In boat-based sports, sailing has long led the way in the application of physical testing, in test tanks, wind tunnels and at full-scale, as well as computational analysis, driven especially by the high budgets of America's Cup yacht design. In rowing, canoeing, and kayaking, the use of both computational hydrodynamics and physical testing in performance assessment has been more limited.

Tuck and Lazauskas (1996) and Lazauskas (1998) performed steady-speed thin-ship (inviscid) computational studies of rowing shells. Scragg and Nelson (1993) used a steady-speed inviscid wave-resistance code, including shallow-water effects, to predict the performance and design of two hulls. More recently, Formaggia and colleagues (Formaggia, Miglio, Mola, & Montano, 2009; Formaggia, Miglio, Mola, & Parolini, 2008) computed the effects of heave and pitch motions on resistance using a potential-flow approach, and later utilized this in a sophisticated dynamic model of the rower–hull–fluid system. Berton and colleagues (Berton, Alessandrini, Barré, & Kobus, 2007) presented results for an unsteady viscous computational fluid dynamics (CFD) approach. Other researchers (e.g. Wellicome, 1967) have used steady-speed tank tests as an aid to the development of improved hull forms for rowing shells; many other tank-test studies carried out remain commercially confidential.

The application of these techniques to canoes and kayaks has been more limited. Lazuaskas and Tuck (1996) applied the steady-speed thin-ship approach to explore optimal hull forms for racing kayaks; Lazauskas and Winters (1997) compared the performance of optimal hull forms and some real designs. Bugalski (2009) documents the history of canoe hull-form development, and outlines a detailed technical

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ISSN 0264-0414 print/ISSN 1466-447X online © 2011 Taylor & Francis
DOI: 10.1080/02640414.2011.576691

program implemented in support of the design of Plastex canoes, including tank testing and CFD applications.

As hull designs evolve, available gains diminish, and increased demands are placed upon the accuracy of both experimental and computational approaches. Nonetheless, the extremely small winning margins still justify the extraction of every last possible improvement. In the Beijing Olympics, over the 14 rowing events, 18 crews were within 0.5% of mean speed of the gold medal-winning crews in their event, from as low as fourth place, while 33 were within 1%.

Consequently, effects that might have previously been considered too small or too challenging to model may need to be considered, even where inclusion of these effects requires novel approaches. Two such effects are explored here: the impact of shallow water, and the effect of unsteady variation in speed through the stroke.

Effect of water depth

The key parameter in characterizing the effect of water depth on resistance is the *depth Froude Number*, $Fr_h = U/\sqrt{gh}$, where U is boat speed, g is the gravitational constant, and h is water depth. If $Fr_h \leq 0.5$, results are similar to deep water. As the boat approaches the *critical* speed ($Fr_h = 1.0$), wavelengths, wave heights, and wave resistance all increase. Indeed, for this reason, high-speed ferries normally avoid operating in a depth Froude number range of 0.8–1.2. For *supercritical* ($Fr_h > 1.0$) speeds, the transverse components of the wave pattern disappear and the wave resistance may be reduced compared with the critical value. Faltinsen (2005) gives a detailed discussion of the effect of water depth on wave patterns and wave resistance.

On a rowing lake with depth of 3.0 m, the critical speed is around 5.4 $m \cdot s^{-1}$; many elite rowers will be travelling at this speed at some point in their stroke cycle. Hence it is important to be able to account for the effects of shallow water both experimentally and computationally in a first-principles approach to hull design.

Effect of unsteady speed

The surge acceleration of a rowing shell can be substantial. Figure 1, re-plotted from Kleshnev (2002), shows acceleration for a men's rowing pair at a rate of 35 strokes $\cdot$ min^{-1}, plotted against a proportion of the stroke period $T = 1.71$ s. The maximum deceleration here is over 1 g, occurring in the "catch" phase of the stroke. Assuming a mean speed of 5.0 $m \cdot s^{-1}$ (equivalent to a medal time for a rowing pair in Beijing), the associated speed variation and distance travelled can be found by time integration. The range of the speed variation is almost 50% of the mean value; in water 3.0 m deep, the depth Froude Number would vary from 0.65 to 1.09.

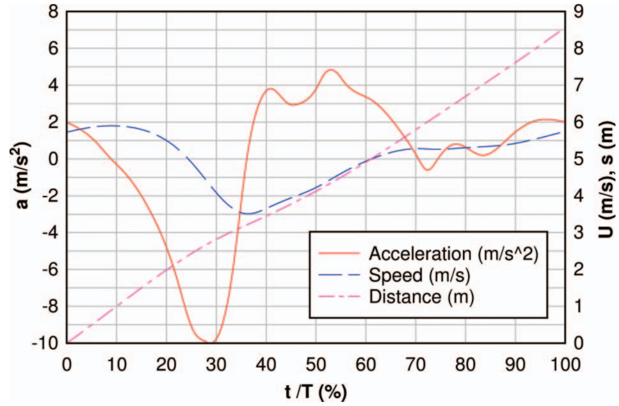

Figure 1. Measured surge acceleration and resulting speed and distance (acceleration re-plotted from Kleshnev, 2002).

The variation in speed modifies the resistance in two key ways. First, the waves generated by the boat, and the associated wave-making resistance, will change. These changes will be more pronounced in shallow water, especially close to the critical speed. Second, the boundary layer around the hull will be affected, leading to changes in the viscous resistance; these changes are less likely to be sensitive to water depth.

Aim and objectives

Our aim in this study was to contribute to the understanding of the effect of unsteady hull dynamics in shallow water on the wave-making and viscous resistance of rowing shells, canoes, and kayaks by developing an experimental approach to generate realistic physical test data in laboratory conditions, and by utilizing a computational approach to predict unsteady shallow-water wave resistance.

The objectives of the study were:

- to design and build a test rig capable of reproducing realistic speed profiles in the towing tank;
- to use the rig to explore the impact of unsteady shallow-water effects on wave-making resistance via experimental measurements on a benchmark hull;
- to use the data generated to explore the validity of a computational approach to the prediction of unsteady wave resistance;
- to examine the impact of unsteady speed on viscous flow around the hull in real-world and laboratory conditions;

- to demonstrate that the measured real-world viscous flow behaviour can be successfully reproduced in the tank and thus that a tank-test approach can provide a reliable guide to viscous-flow performance characterization.

Development of test rig

The test rig was designed to be installed in the towing tank at the Kelvin Hydrodynamics Laboratory in Glasgow, Scotland. The tank has dimensions of 76.0 × 4.57 × 2.5 m, with a water depth of up to 2.3 m. The main towing carriage can be used to generate unsteady motion; however, its peak acceleration is limited to around 0.8 $m \cdot s^{-2}$, or less than 10% of the peak value shown in Figure 1.

In the present study, the main towing carriage was used to generate the mean speed, and a sub-carriage was mounted on the main carriage to generate the surging motion. The specification of the sub-carriage required careful consideration of full-scale behaviour and appropriate similarity (scaling) conditions. The data from Figure 1 were used in the first instance to outline the specifications.

The requirements for a full-scale pair was obtained by subtracting the mean speed, and the distance travelled at mean speed, from the corresponding instantaneous values of speed and distance, to give the perturbation speed and the excursion required for the sub-carriage, as shown in Figure 2.

To scale wave effects correctly, the Froude Number based on length, $Fr = U/\sqrt{gL}$, was kept constant between model- and full-scale. Here U is the speed and L is waterline length. Under Froude scaling, accelerations were identical at model- and full-scale; model-scale speed was reduced as the square-root of the scale factor. For a rowing pair, with length 10.25 m, mass 195 kg, mean speed 5.0 $m \cdot s^{-1}$, and stroke rate of 35 strokes $\cdot$ min^{-1} in water 3.0 m deep, at a scale of 1:2, the model would be 5.125 m long, with mean speed 3.54 $m \cdot s^{-1}$ and stroke frequency of 0.82 Hz, in water 1.5 m deep. After allowing for acceleration and deceleration of the main carriage, this would yield around 10 cycles at a steady mean speed. However, the total displacement of the model would be only around 25 kg, so a lightweight model hull would be required. Using the data from Figure 2, the model-scale perturbation speed would vary from -1.05 to $+0.64$ $m \cdot s^{-1}$ and the excursion from -0.12 to $+0.14$ m.

A digitally controlled, electrically driven actuator available from a previous project, with maximum travel of 1 m, speed of 2 $m \cdot s^{-1}$, acceleration of 20 $m \cdot s^{-2}$, and force of 20 kN was found to be adequate. The actuator drove a sub-carriage approximately 2.0 × 1.0 m on which the standard towing system was mounted (see Figure 3). Pre-calculated data points specified carriage position at each moment in time through one cycle; the cycle was repeated to generate periodic motion. The complete test set-up for shallow water is shown in Figure 4.

Only the surging motion of the boat was controlled in the system. For rowing shells, fore-and-aft movement of the athletes and the surging acceleration of

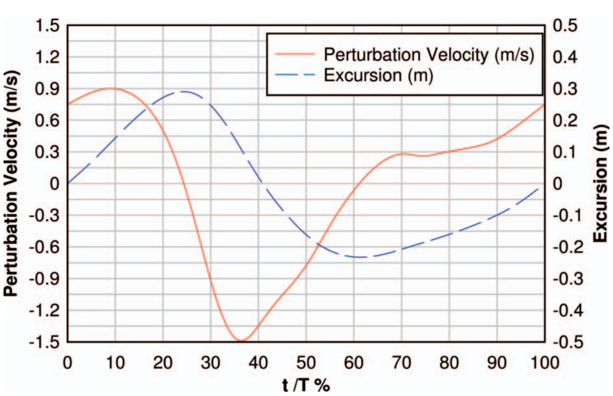

Figure 2. Perturbation velocity and excursion for full-scale rowing pair.

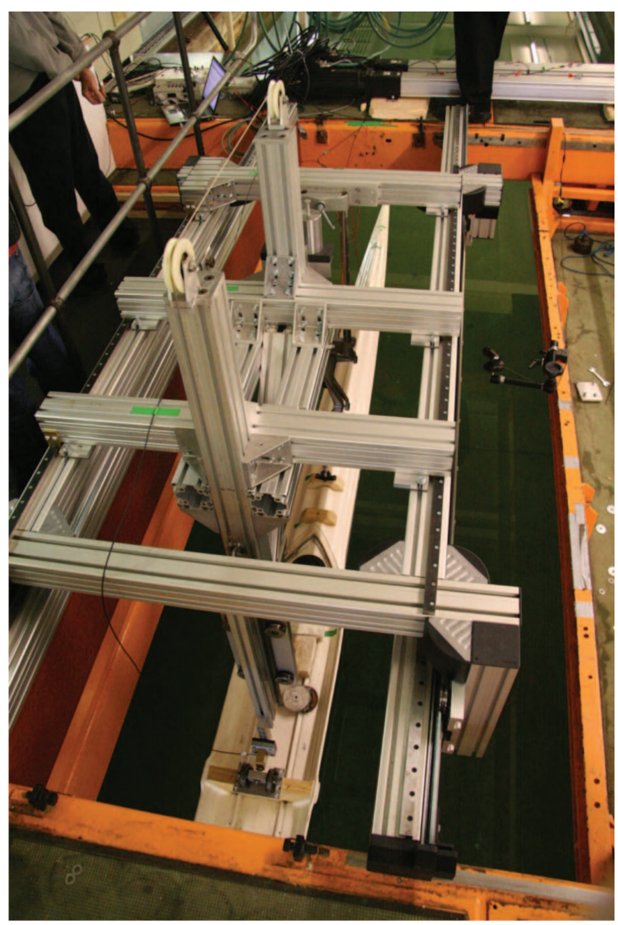

Figure 3. Sub-carriage set-up with single scull.

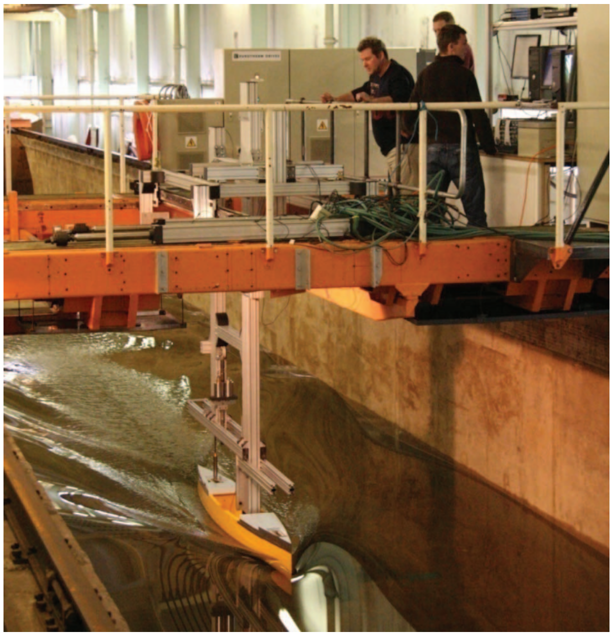

Figure 4. Shallow-water testing of Wigley hull.

the boat lead to a pitching motion, while vertical acceleration of the athletes and oars leads to a heaving motion. In the current system, these motions were not controlled; nonetheless, the boat could heave and pitch freely due to the varying hydrodynamic forces.

Where the focus of testing is on measurement of unsteady hydrodynamic forces, inertial forces become extremely important. These are typically an order of magnitude larger than the steady hydrodynamic forces, and possibly two orders of magnitude larger than unsteady effects. Hence the force measurement system had to be highly sensitive, linear, and repeatable, and both the acceleration and the hull mass had to be measured extremely precisely.

Tank testing of benchmark hull in shallow water

The first set of tank tests explored the effect of unsteady wave-making resistance in shallow water, using a well-known benchmark design, the *Wigley* hull, constructed with length $L = 3.0$ m, beam $B = 0.3$ m, and draught $T = 0.1875$ m. The increased beam-to-length ratio of the Wigley hull compared with a rowing shell exaggerated the wave effects, making interpretation of the results more straightforward. The unsteady speed took the form:

$$U(t) = \overline{U} + \widehat{U} \sin \omega t$$

The mean velocities, $\bar{U}$, perturbation velocity amplitudes, $\hat{U}$, and frequencies ω were varied.

In parallel with the experimental study, an unsteady inviscid thin-ship computer code was developed to predict the time history of the wave-making resistance in water of any depth. The code took advantage of the simple formulae describing the parabolic waterlines and sections of the Wigley hull to reduce computational effort in this highly numerically intensive calculation. Details of the hull form and the theoretical basis of the unsteady wave resistance code are given elsewhere (Doctors, Day, & Clelland, 2010).

Figure 5a shows a typical measured speed profile from the tests, plotted against the time non-dimensionalized with period, showing steady acceleration followed by steady speed prior to the smooth sinusoidal variation. Figure 5b shows one comparison of measured and predicted time histories of wave resistance R_W (non-dimensionalized with model weight W), plotted against non-dimensional distance, s/L, where s is the distance travelled in metres. The mean Froude number was $\overline{Fr} = \overline{U}/\sqrt{gL} = 0.3$, the amplitude of oscillation of the Froude Number $\widehat{Fr} = \widehat{U}/\sqrt{gL} = 0.06$, and the mean depth Froude Number $\overline{Fr}^h = 1.0$.

The curve marked "Expt" shows the unsteady wave resistance, calculated from experimental data for total resistance using a quasi-steady approximation for viscous resistance, in which the instantaneous viscous resistance was estimated from the instantaneous speed using a standard established relationship between steady speed and steady viscous resistance. The relationship adopted is known to give good predictions of steady resistance for slender ships over a wide range of speeds. The curve marked "US" is the computational prediction for unsteady wave resistance. Finally, the curve marked "QS" is the predicted quasi-steady wave resistance, calculated from the variation of steady wave resistance with steady speed, as predicted by a conventional steady thin-ship wave-resistance code.

This plot illustrates some of the challenges of shallow-water oscillatory testing: the oscillations in the wave resistance curve grew as the model progressed along the tank. This behaviour was correctly predicted by the unsteady code, while the quasi-steady approach, based on the steady code, yielded extremely poor prediction of the time history, dramatically underestimating the peaks of the resistance curve, and exhibiting oscillations that were not present in either experimental or computational unsteady data.

Figure 5c shows the root-mean-square wave resistance plotted against the frequency parameter $\tau = U\omega/g$ (where ω is the oscillation frequency in rad $\cdot$ s^{-1}). This parameter indicates the ratio between the forward speed of the vessel and the phase speed of the waves generated by the oscillation (in deep water). The value at $\tau = 0$ indicates the corresponding steady-speed value. It can be seen that over much of this range, the unsteady value was substantially higher than the steady-speed value. The

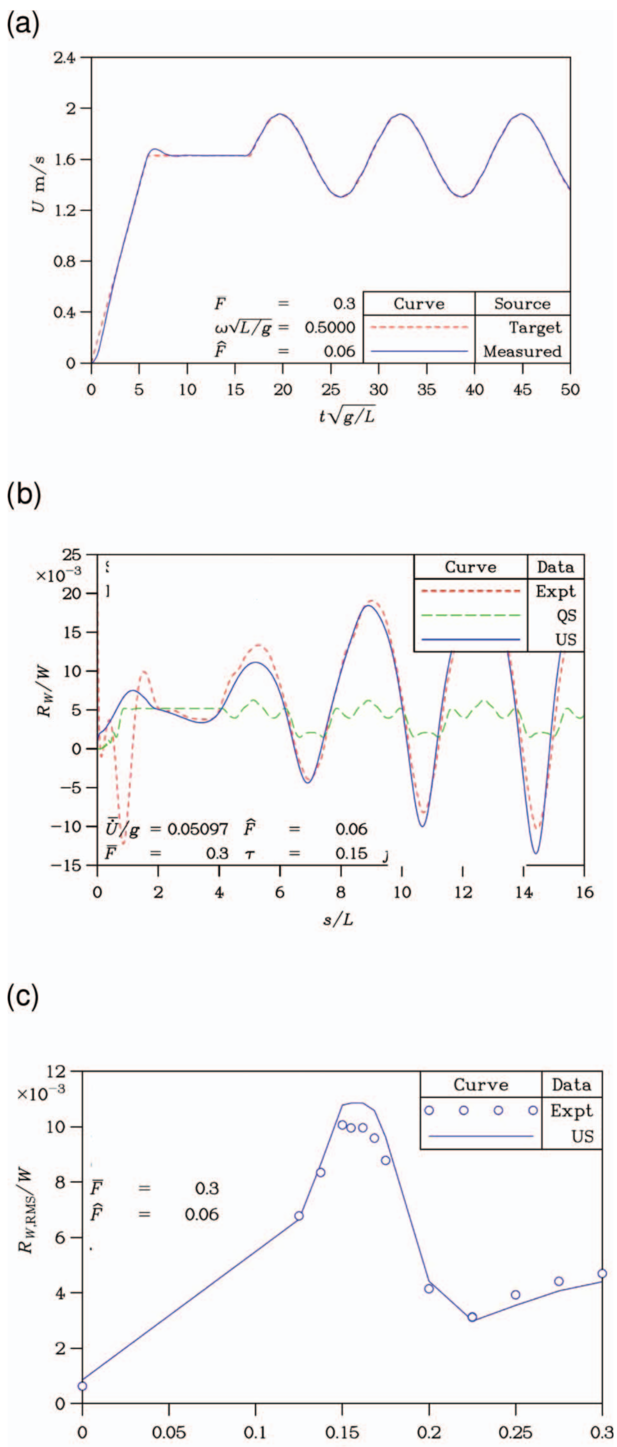

Figure 5. Selected results for benchmark tests at low speed and low frequency. (a) Typical speed profile. (b) Measured and predicted wave resistance. (c) Measured and predicted root-mean-square (RMS) wave resistance.

substantial "hump" in the graph around $\tau = 0.16$ was well predicted by the theory.

In general, good agreement was found at low mean speeds and oscillation frequencies between the unsteady shallow-water wave-resistance computations and the values derived from tank tests. It could thus be inferred that the effects of shallow water on unsteady wave-making resistance was correctly predicted using the computational approach in these conditions.

Since the tank-derived values of wave resistance relied on the quasi-steady approximation to frictional resistance, it can also be inferred that in these conditions the viscous resistance was well predicted by this approximation. In contrast, there was very poor agreement between tank-derived values for unsteady wave resistance and predicted quasi-steady approximation for wave resistance.

However, subsequent tests at higher mean speeds and with higher values of the frequency parameter, closer to those experienced in rowing and/or kayaking races, did not show such good agreement. The data for Figure 6 were obtained with $\overline{F}\mathrm{r} = 0.5$, $\widehat{F}\mathrm{r} = 0.1$, and $Fr_h = 1.0$; it can be seen that the trends were poorly predicted for $\tau > 0.7$. Much higher values than this of the frequency parameter are found in rowing races; for example, the data shown in Figure 1 correspond to $\overline{F}\mathrm{r} = 0.498$, $\widehat{F}\mathrm{r} = 0.09(+)$, $\widehat{F}\mathrm{r} = 0.15(-)$, and $\tau = 1.86$. At these higher frequencies, the measured resistance was found to be substantially greater than that predicted using the computational approach, suggesting that the approach was breaking down in conditions relevant to rowing.

The unsteady wave-resistance calculation, however, made no assumptions about speed or frequency except a common linearization that wave steepness was small. Hence, the approach should in principle also have behaved well at higher speeds and frequencies, unless wave behaviour changed dramatically in some way. This could have resulted from

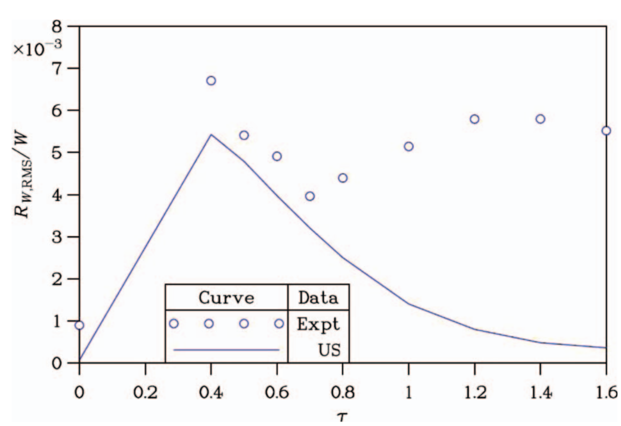

Figure 6. Root-mean-square (RMS) wave resistance for Wigley hull at moderate speed and frequency.

wave-breaking; however, video recordings showed no evidence of this.

A more likely corollary is that the quasi-steady approximation for the viscous resistance was failing in these conditions, and that viscous resistance increased substantially with higher speed and frequency. One possible contributor to this was the influence of acceleration on turbulence in the boundary layer and in particular on the transition between laminar and turbulent flow. Predicting the location of the laminar–turbulent transition from first principles is a hugely challenging problem in ship resistance prediction even in steady flow; in unsteady flows of the type of interest here, there is virtually no information available.

The location of laminar–turbulent transition is known to be of great practical relevance in hull design optimization; designing bow shapes to delay transition is a key strategy for resistance reduction in yachts, and has been extensively investigated by America's Cup technical teams. A preliminary indication of the importance of the location of laminar–turbulent transition in the present context was given by steady-speed tests with transition "forced" at different locations by a girth-wise line of small studs fitted to the surface of the hull. For the single scull used here, the steady resistance between 2.5 and 4.0 $m \cdot s^{-1}$ was found to be around 0.5% higher on average with studs located 400 mm from the bow compared with the studs located 600 mm from the bow.

The second phase of the study thus focused on unsteady effects on laminar–turbulent transition. As well as providing insight into unsteady effects on viscous flow, transition provided a useful metric for the comparison of laboratory and field-trial data for validation purposes. Total hull resistance would have been the ideal choice, but was impractical due to the challenges associated with the measurement of hull resistance in the field with suitable accuracy.

Field measurement of viscous flow

A series of field trials was carried out with the twin objectives of establishing realistic speed profiles for reproduction in the tank, and providing field measurements against which the test-tank data could be validated to demonstrate that realistic viscous flow could be created in the absence of the athlete.

A single scull was chosen for these trials because it could be tested at full-scale in the tank, hence avoiding scaling issues for this preliminary study. Three series of field trials were carried out, in varying conditions and locations, allowing progressive refinement of the systems, and also allowing the rower to become accustomed to the reduced stability of the hull resulting from the installed equipment.

The scull was fitted with conventional hot-film anemometry gauges (Dantec Dynamics Ltd., Bristol, UK) in a number of different locations. In the early sets of tests, several gauges were set up to identify suitable locations on the hull (see Figure 7a); as runs progressed, forward gauges were removed to allow undisturbed flow to gauges further aft. In the final set of tests, one gauge was located on each side in the best positions identified to ensure no interference between the gauges.

Motions were recorded using an integrated system designed for logging race-car data (VBox 3i, Racelogic, Buckingham, UK); this comprised GPS to capture mean speed, accelerometers to obtain surge and pitch motions, and a portable data logger including analogue inputs used here to gather the hot-film data. The data logger and hot-film amplifiers were mounted in a waterproof box aft of the foot stretcher, as shown in Figure 7b.

Several runs were made during each set of trials; each run included some "cruising" strokes, some "racing" strokes, and also a "coast-down" period, in which the scull decelerated smoothly in a natural manner. A sample of measured motion data from the field trials is shown in Figure 8.

The present study focused on the flow behaviour at the faster stroke rate; a representative cycle was chosen with maximum speed of 4.4 $m \cdot s^{-1}$. The time history from the trials motion was then used to create an input file for the sub-carriage drive system. The resulting time history of position is shown in Figure 9.

Towing-tank measurement of viscous flow

The second set of tank tests also focused on the measurement of turbulence near the bow of a full-scale single scull. The scull used was similar, but not identical, to that used in the field trials. Hot-film gauges were applied in positions similar to those used in the final set of field trials, at 400 mm and 600 mm aft of the bow.

A hot-film signal can be characterized as consisting of four main components: a DC signal that varies non-linearly with speed; a DC signal that is higher for turbulent flow than for laminar flow; an AC signal representing flow turbulence; and intermittency when the flow is sometimes laminar and sometimes turbulent.

A series of runs was first carried out at steady speed to test the hot-film measurements. The data were filtered with a low-pass digital filter with a cut-off of 20 Hz to remove electrical noise. Figure 10 shows a time history of a typical run.

The non-linear variation of hot-film signal with speed can be seen between 1 and 3 s. Jumps of around 0.5 V in the hot-film signals can be observed at just

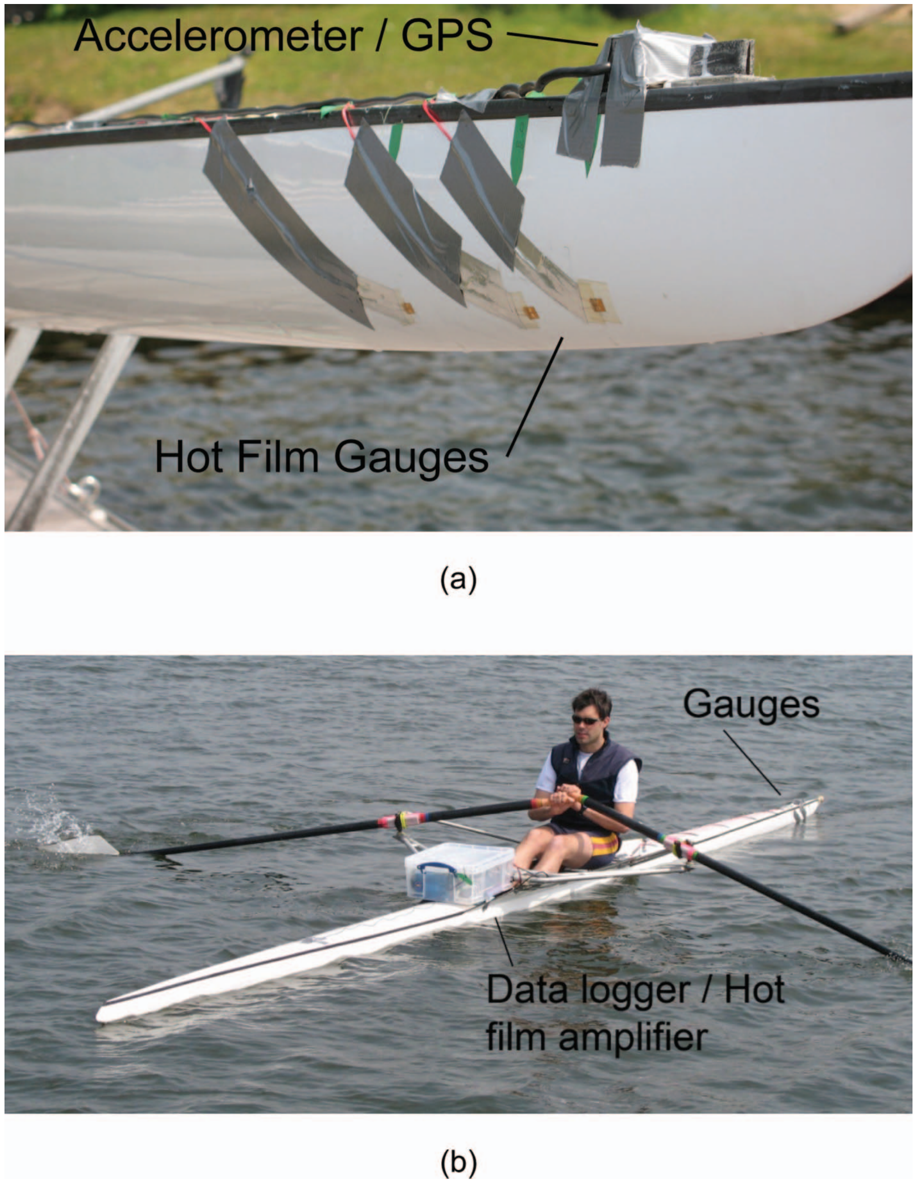

Figure 7. Instrumented single scull: (a) Detail of hot-film gauges installed on bow; (b) General layout of instrumentation and data logging systems.

after 3 s for the aft gauge and after 4 s for the forward gauge, indicating laminar–turbulent transition. Once the speed reached a constant value, the signal level dropped as the flow re-laminarized; occasional bursts of turbulence were still observed on the aft gauge where the Reynolds Number was higher. This plot indicates the influence of even simple and smooth acceleration patterns on transition.

To confirm the impact of turbulent flow on output signal, one run was carried out with a small wire attached forward of the forward gauge to force transition. The jump in signal was similar to that observed with natural transition.

A series of oscillatory runs was then carried out reproducing the field-trial motions. Figure 11 shows data from a run at mean speed comparable to the field trials. These data were filtered, but otherwise unprocessed; zero offsets were not removed. It can be seen that the signal displayed variations due both to the changes in speed and transition from laminar to turbulent flow.

An attempt was then made to separate the effect on output signal of speed variation from the effect of transition. Using the data from constant-speed runs, calibration curves for each of the hot-film gauges were derived, and used with the instantaneous speed data to generate a quasi-steady approximation to the speed-related component of the signal. This quasi-steady approximation was subtracted from the total signal.

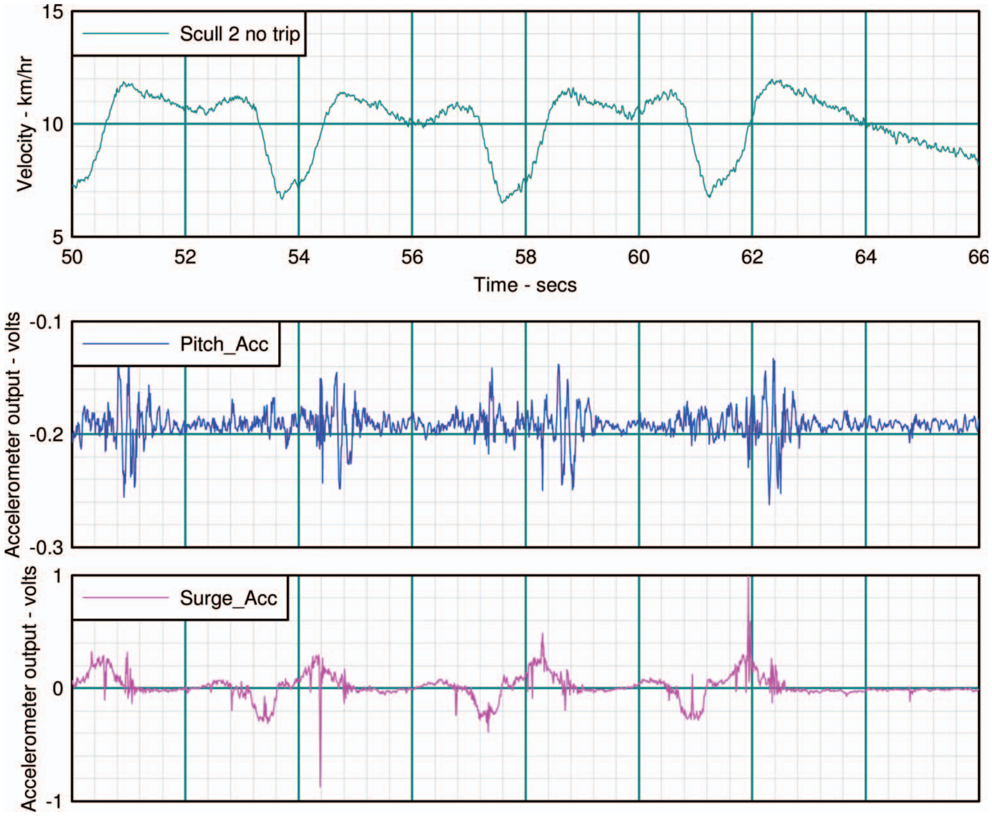

Figure 8. Typical results from field trial motion measurements.

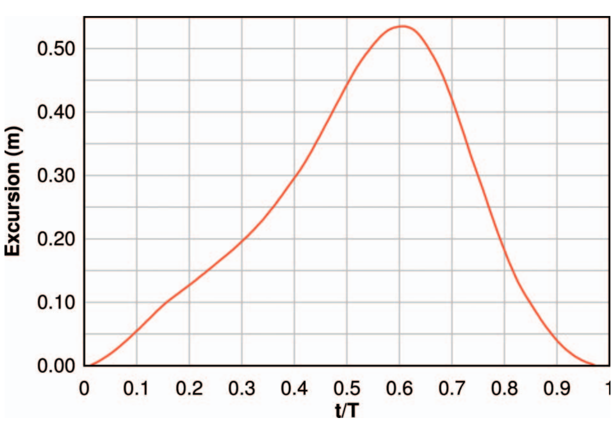

Figure 9. Carriage excursion data derived from field trials.

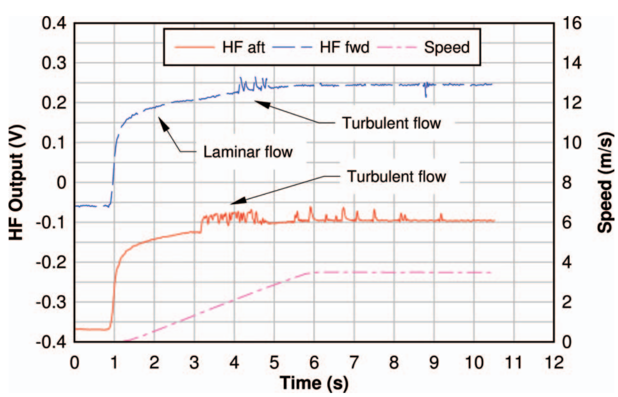

Figure 10. Typical time history of hot-film output in steady-speed tank test.

The remainder could be regarded as an estimate of the unsteady component of the signal – that is, the part related to flow acceleration. The impact of this process is shown in Figure 12 along with non-dimensionalized acceleration data indicating the phase of the signal. It can be seen that the estimated unsteady components were close to zero when the acceleration was small, indicating that the decomposition of signal into quasi-steady and unsteady components was largely successful.

The results show a marked relationship between acceleration and turbulence: the unsteady component clearly peaked on both gauges at peak deceleration, indicating that rapid deceleration was triggering transition; as might be expected, the turbulence lasted longer on the aft gauge at higher Reynolds Number. A secondary peak appeared regularly on the aft gauge near the secondary local minimum of the acceleration. The pattern of the unsteady component, although complex in form, appeared strongly periodic in nature, with features repeating over several cycles. The increased turbulence suggested that viscous resistance would be higher than in a comparable steady-flow situation. It was found that this unsteady component was relatively insensitive to the mean speed; reducing mean speed to $3.0\ \mathrm{m \cdot s^{-1}}$ was found to have little impact on the shape of the curve.

Since the emphasis for the validation was on the unsteadiness of the flow, it seemed reasonable to adopt this unsteady component as a metric to compare the influence of acceleration on viscous flow in tank and field trials.

Comparison of unsteady flow in laboratory and field-trial data

A similar approach was taken to analyse the trials data. Calibration curves were estimated from coast-down data, and used to calculate the unsteady component of the hot-film output for comparison with the tank data.

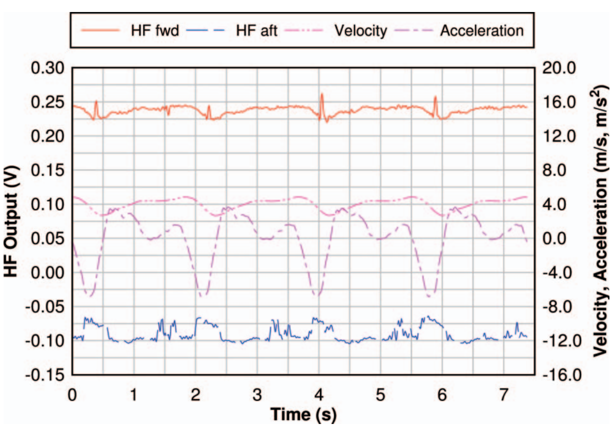

Figure 11. Typical run: Fast rowing pattern, mean speed $= 4.0\ \mathrm{m \cdot s^{-1}}$.

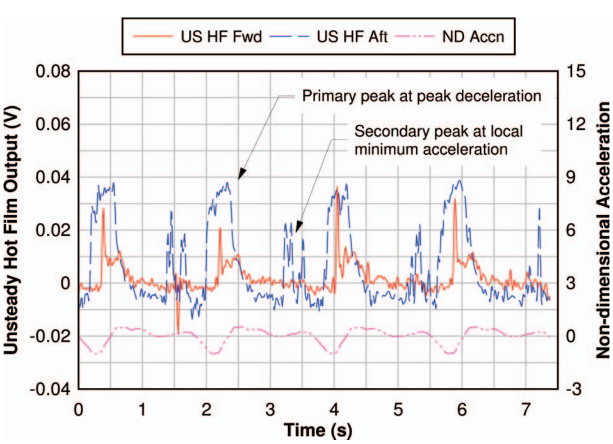

Figure 12. Unsteady component: Fast rowing pattern, mean speed $= 4.0\ \mathrm{m \cdot s^{-1}}$.

Several sections of trials data were identified as similar to the tank data in terms of velocity and acceleration. Key comparisons for one section of field data are shown in Table I; Figure 13 shows the corresponding perturbation velocity and acceleration, non-dimensionalized with respect to their maximum absolute values. The time scales of both signals were non-dimensionalized with respect to stroke periods to aid comparison. It can be seen that the field-trial and laboratory velocity signals were very similar, although small details of acceleration differed slightly.

Figure 14 shows the results for the gauges located 600 mm from the bow for these sections of the time histories. The field-trial hot-film data were offset for

Table I. Velocity and acceleration parameters for chosen comparison data.

Parameter	Laboratory	Field trial	Difference (%)
Mean velocity ($\mathrm{m \cdot s^{-1}}$)	4.00	4.16	4
Peak unsteady positive perturbation velocity ($\mathrm{m \cdot s^{-1}}$)	0.84	0.87	4
Peak unsteady negative perturbation velocity ($\mathrm{m \cdot s^{-1}}$)	−1.33	−1.37	3
Peak negative acceleration ($\mathrm{m \cdot s^{-2}}$)	−6.85	−6.11	−11
Peak positive acceleration ($\mathrm{m \cdot s^{-2}}$)	3.65	3.87	6
Stroke period (s)	1.83	2.03	11
Stroke rate ($\mathrm{strokes \cdot min^{-1}}$)	32.8	29.6	−9

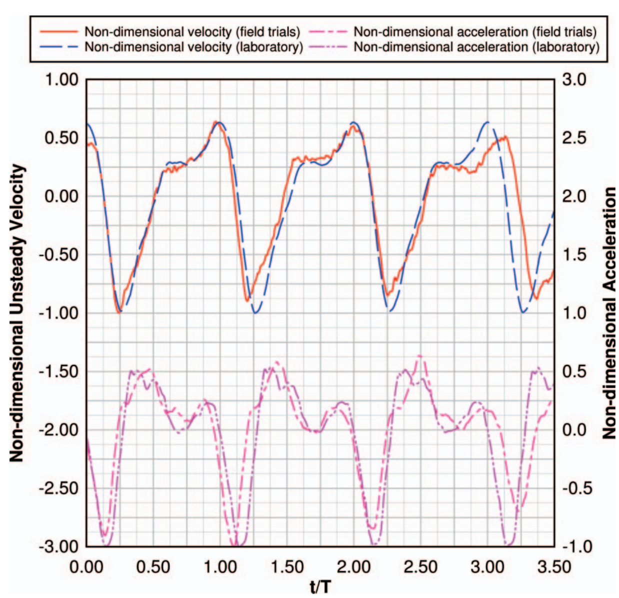

Figure 13. Comparison of velocity and acceleration data between field and tank for chosen section.

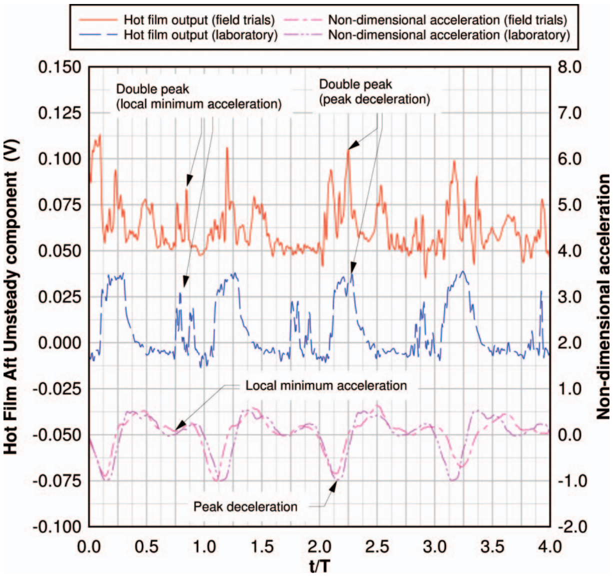

Figure 14. Comparison of field trial data with tank data for aft gauge.

clarity and also scaled to account for differences in amplifier settings between field and tank tests. This did not affect the validity of the comparison since the focus here was on the variation of signal with time rather than the absolute magnitude of the signal.

It can be seen that the key features of the signal were largely comparable between laboratory and field data in spite of the minor differences between the hulls used. The field-trials data exhibited more variability than the data from the laboratory. This could be expected for three reasons: background turbulence was likely to be higher in the field trials; stroke-to-stroke variations were greater; and, finally, the impact of athlete movement on heave and pitch would be variable in the field trials.

Both data sets showed a large periodic double peak, suggesting onset of turbulent flow at peak deceleration (e.g. $t/T \approx 2.125$–2.250). The relative magnitude of the two peaks varied rather more in the trials data than in the laboratory data. Both data sets also showed a second smaller set of periodic double peaks that corresponded to a local minimum acceleration (e.g. $t/T \approx 0.75$); these peaks were slightly more variable in the trials data, failing to appear in the second stroke shown. The only periodic feature in the trials data that did not appear in the laboratory data was a third peak, which appeared to occur near the maximum positive acceleration (e.g. $t/T \approx 0.5$). Examination of motion data did not suggest any particular cause for this.

Nonetheless, the general character of the signals was unquestionably similar in most respects. Thus it is proposed that the testing methodology correctly recreated the complex mechanisms of unsteady viscous flow in the towing tank.

Conclusions

We have described the development of a test rig for generating realistic oscillatory speed profiles for a test hull in a towing tank. The test rig has been used to identify the presence of some complex unsteady shallow-water wave-resistance phenomena. The mean unsteady wave resistance was shown to be considerably higher than the comparable steady-state value in some cases.

The computational study showed that at low speed and low frequency the unsteady resistance for a benchmark model hull was well predicted by a combination of unsteady wave-resistance and quasi-steady viscous-resistance models. However, results also indicated that the quasi-steady approximation for viscous forces was not valid at higher speeds and frequencies, and that unsteady viscous resistance was higher than predicted by the quasi-steady approximation. This suggests that accurate computational prediction of the unsteady total resistance at higher speeds and frequencies presents substantial challenges.

The test rig was used to identify the impact of unsteady effects on laminar–turbulent transition in both laboratory and field-trial conditions. In both cases, turbulence was shown to be strongly related to acceleration through the stroke cycle. Comparison of tank test results with field-trial measurements showed that the unsteady viscous flow phenomena identified in the real-world measurements were also present in the tank. Hence it was concluded that the rig generated plausible unsteady viscous flow phenomena in the test tank, and thus the tank tests could reliably be used to investigate improved designs.

The use of a test rig that replicates the real-world hydrodynamics of rowing shells as well as canoes and kayaks allows a number of opportunities for performance improvements. The approach could be used to assess designs directly, or to validate CFD calculations. Designs intended to reduce resistance can be evaluated in realistic conditions in a controllable and repeatable environment, allowing measurement of the flow characteristics and the dynamic forces.

A study of the effect of the unsteady speed profile on the unsteady resistance for realistic rowing conditions is planned. A more generic study utilizing a thin flat plate to examine in detail the impact of unsteady speed on viscous resistance is also planned. Finally, it is intended to generalize the computer code to allow prediction of resistance of any slender hull form.

However, there are still two key questions to be addressed. In the present study, a single scull was used because the size and speed of the single allowed full-scale tests to be carried out within the speed limitations of the test tank. Even so, only a small number of oscillation cycles was possible at full speed. To test faster hulls, scale models would be required. Froude similarity would then lead to lower model-scale speed, and higher model-scale frequency, and hence more oscillations in the scope of the tank, but further validation would be required to understand the scaling of the unsteady viscous effects.

Finally, to complete the accuracy of the modelling, it would also be desirable to build a mechanism to replicate the complete six-degree-of-freedom motions. It is likely that heave and pitch would be dominant in rowing applications in which power is applied in a symmetrical fashion, while roll and yaw would also be important in canoe/kayak applications due to the asymmetry of the power application.

Acknowledgements

This work was funded by the UK Engineering and Physical Sciences Research Council (EPSRC) under the "Achieving Gold" programme grants EP/F006284/1 and EP/F00625X/1. Prof. Doctors' participation in the benchmark study was also supported by EPSRC under grant EP/F019998/1. We are indebted to Charles Keay and the technical staff of the Kelvin Hydrodynamics Lab for constructing the test rig, and assisting with all aspects of the tank-test programme

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