Second International Symposium on Marine Propulsors
smp'11, Hamburg, Germany, June 2011

An application of paddlewheel propulsion to a high speed craft

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ABSTRACT

The paddle wheel has existed as a marine propulsor for in excess of two hundred years yet has been developed little over the last one hundred years. It could be concluded that the paddle has stagnated as a means of efficient propulsion for today's marine vessels. As technology develops in other areas, is it possible that paddle propulsion could find applications not yet considered? Has the paddle reached its maximum potential?

This paper challenges the assumption that paddle forms of propulsion are ineffective at high rotational speeds through two means: a report on a set of trials of a relatively high powered and high rotational speed paddle wheel fastened to a purpose built 8m long planing skiff which achieved speeds in excess of 32 knots; and results from previous experimental work on paddles, carried out in the Denny experiment tank at Dumbarton, Scotland, were non-dimensionalised, replotted and applied to predict and interpret the performance of the paddle operated on the planing skiff.

The paddle achieves propulsion through a relatively large swept area, thus achieving the potential for relatively high efficiency of propulsion. The present design operated with low immersion, up to about 75mm, but at lower immersions down to 5mm at higher speeds, in the wake behind an immersed transom stern. The paper reports on performance of this paddle and interprets the results in the light of the existing historical experimental results.

Keywords

High Speed, Paddlewheel, Propulsion

1 INTRODUCTION

Paddle propulsion dates back to the infancy of engineering design yet has for decades played only a small part in vessel propulsion throughout the world, primarily based on slow speed vessels constrained in draft, such as river barges. Historically, paddle wheel propulsion was used for slow speed vessels with large, heavy slow speed machinery. Very little work has been done regarding the application of paddle propulsion to high speed vessels.

Preliminary investigations centred around an alternative propulsion system for a high speed, variable displacement craft as a means to reduce thrust at high speeds to maintain a reasonable level of safety as the craft lifted dynamically from the water. Although the thrust is limited by ventilation of the paddles and propulsive efficiency needs to be monitored, there may be a potential use for paddles for propulsion of medium speed craft, perhaps at speeds where water jets become less efficient and propeller diameters too large for practical applications, shallow water applications especially.

Historical developments throughout the industrial revolution involving power production resulted in higher operational speeds of engines from steam through to combustion engines. These higher operational speed profiles along with the development of the screw propeller led to the rapid demise of paddle propulsion as a mainstream alternative for ocean going vessels. This demise could be attributed to the perceived lack of ability of paddles to account for changes in vessel draft, and the associated immersion issues relating to a vessel operating in a seaway (Carlton 2007). It could be said that the infamous tug of war in 1845 between the Royal Navy paddle wheeler *Alecto* and HMS *Rattler,* one of the first Royal Navy vessels to utilize screw propulsion which the latter won convincingly (Paine 2000), was the turning point for the demise of the paddle.

This demise in paddle propulsion, coupled with the dominance of screw propulsion at a time when model scale testing was growing rapidly, resulted in vast amounts of model test data from screw propeller experiments being collected all over the world. This process continues to grow to this day, whilst paddle propulsion has remained predominantly in the shadows. This shift in technology, coupled with the observation that paddles are more of a drag device than a lifting device, seems to present few positives for paddle propulsion.

The lack of effective experimental data prompted Volpich and Bridge (1955) to undertake extensive model experimentation on various configurations of paddles. The experimental testing was carried out at the Denny Tank in Dumbarton, Scotland, which is now part of the Scottish Maritime Museum. The testing was broken down into three phases and carried out across a three year time frame.

Phase one was a preliminary experiment involving a large 1.04m (3.4ft) wheel and a small 0.52m (1.7ft) wheel with both wheels having a fixed number of floats (paddles) at the same immersion. Phase one involved testing both a fixed float wheel in which the paddles were rigidly fixed to the wheel, and a feathering wheel where the paddle angle was changed during rotation to optimise the angle of entry and exit of the paddle with the water surface. The results achieved within the preliminary phase led the authors to believe that further work on feathering paddles would be the best course of action. These preliminary results presented feathering wheels as the most efficient over a greater working range, both in terms of advance velocities and wheel rpm. As a result, fixed float paddle wheels made no further progress within the experimental phases.

Phase two involved more detailed testing across a greater range of variables, including, but not limited to: variations in immersion level, number of floats, size of floats, and variations in star centre position of the feathering wheel. The final phase was to investigate the results achieved experimentally in terms of the correlation between model and ship.

Volpich and Bridge (1955) also noted that history has recorded vessels that had achieved a high level of propulsive efficiency using paddle propulsion. In the 1880's *Belle* type vessels which were 75m (246ft) in length were achieving propulsive efficiencies of almost 60% at 12 knots and 54% at 16 ¾ knots. This propulsive efficiency was a ratio measure of effective power over the installed power. With this in mind, it was felt that the demise of paddle propulsion was due to a number of issues related more with the inability to deal with large variations in vessel draft due to loading conditions. Another problem area involved operation within a seaway, which created problems with variations in thrust created along the length of the paddle. These variations created a vessel that had difficulty maintaining directional stability.

Little has been done since Volpich and Bridge's (1955) work to further investigate not only paddle propulsion, but to question or expand on their results. With modern engine developments creating more compact high speed engines, only Wray and Starrett (1970) analysed high speed paddle propulsion on a small scale model. Although the work of Wray and Starrett (1970) was very detailed in terms of thrust, torque, variations in immersion and paddle rpm, they themselves agree that the scale factor used may have induced scale distortions. Although the testing was carried out on a small scale (paddle was 0.13m (5in) in diameter by 0.13m (5in) wide), it was one of the only tests to be carried out with a simulated transom forward of the wheel mechanism. This is the only experimental verification able to be sourced that includes an immersed transom assembly as part of the testing assembly as was used in the current design.

More recent works include those done by Alexander (1999), who, whilst investigating propulsion options for a Landing Craft Amphibious vehicle, discovered the effects of both wakes and bow waves created by paddle wheels at high speeds. Alexander (1999) believed that the results achieved by his tests relating to wake effects correlate well with the areas of unexplained reduction in propulsion force of both the Volpich and Bridge (1955) and the Wray and Starrett (1970) results. Alexander (1999) investigated the response of paddle wheels in relation to a Froude No. to investigate both the effects of the stern wake at low speeds and the bow wave at high speeds. Just as a vessel has interactions with its wake, Alexander (1999) believed that the operation of a paddle wheel was also reliant on the interaction of the paddle wheel with its own wake at low vessel speeds. As speed increased, the main problem became the creation of a bow wave forward of the wheel, which caused a reduction in thrust at a certain speed. This speed was in terms of a relationship between paddle rpm and vessel speed. Alexander (1999) believed that if this bow wave could be eliminated, then the performance of the paddle wheel would be improved.

In order to investigate paddle propulsion, INCAT Tasmania built a prototype vessel with a paddle wheel mechanism equivalent in size to that tested experimentally by Volpich and Bridge (1955) but operating at much higher revolution speed. These trials challenge the assumption that paddle forms of propulsion are necessarily inefficient at high revolutionary speeds:

- The results from previous experimental work on paddles, carried out by Volpich and Bridge in the 50s (Volpich & Bridge 1955, 1956a, 1956b) in the Denny experiment tank at Dumbarton, Scotland, were non-dimensionalised and replotted. These results were applied to predict and interpret the performance of the paddle operated on the planing skiff.
- The paper reports on a set of trials of a relatively high powered and high rotational speed paddle wheel fastened to a purpose built 8m long planing skiff which achieved speeds in excess of 32 knots.

2 EXPERIMENTAL THRUST AND TORQUE COEFFICIENTS OF PADDLES

The work of Volpich and Bridge (1955) represents the only experimental results for a paddle wheel of significant scale. The larger wheel size of 1.04m (3.4 ft) diameter was one of the largest wheels ever to be tested in an experimental tank. Although the experiments were detailed, the authors presented only one plot of raw data of thrust against both advance velocity (Va) and paddle rpm. This plot presented thrust values for paddle rpm's from 0 up to 100 rpm across a range of advance velocities from 0-4.6 m/s (0 to 900 feet per minute) in 0.508 m/s

(100 ft per minute increments). These experimental results were fixed in terms of paddle immersion and the number of blades or floats. No raw experimental results were presented for torque, but a torque coefficient was presented. All the results presented are dimensional and so the results were replotted in a non-dimensional format by using coefficients used for cycloidal propulsors (Bose 2008):

$$K_T = \frac{T}{A\rho\Omega^2 R^2} \tag{1}$$

where T is thrust (N); A is wetted surface area (m^2); ρ is density (kg/m^3); Ω is paddle rotational speed (rad/s); R is radius of paddle (m).

$$K_Q = \frac{Q}{A\rho\Omega^2 R^3} \tag{2}$$

where Q is torque (N-m).

$$J = \frac{V_a \pi}{\Omega R} \tag{3}$$

where V_a is vessel speed (knots).

$$\eta = \frac{K_T J}{K_Q \pi} \tag{4}$$

where η is efficiency; K_T is thrust coefficient; K_Q is torque coefficient; J is advance coefficient.

The results of thrust and torque coefficients against advance coefficients are shown in Figures 9 and 10 (see end of paper), respectively. These plots show that the non-dimensional values of thrust and torque coefficients are not unique at a given advance coefficient, but vary with rate of revolutions and advance speed. It is also possible that other parameters influence the results, such as depth of immersion of the wheel or immersion as a ratio of diameter, and this is supported by the tests of Alexander (1999). The plots show that:

- thrust coefficients drop to zero as the advance coefficient approaches π (as would be expected since this would be the point at which the paddle circumferential speed reaches the advance speed);
- thrust and torque coefficients reduce as the revolutions of this paddle at this immersion increase and they appear to converge to a minimum value as the revolutions are increased;
- thrust and torque coefficient values converge as the advance coefficient approaches π; and
- there is a local minimum in the thrust and torque coefficient values in the mid-range of advance coefficient (values between 1.0-1.5) that may be due to interaction of the blades with waves made by the paddlewheel itself (Alexander 1999) or interaction between the blades and the water cavity left by the preceding blade.

3 PADDLE PROPELLED PLANING SKIFF

The project used the capabilities available within INCAT's production facilities at Derwent Park, Hobart, Tasmania, to manufacture an 8 metre prototype vessel to be used for paddle wheel testing. The specifications of the prototype vessel were generated by Revolution Design, INCAT's design department. Preliminary sizing was based on a decision to have a 1 metre diameter paddle wheel to compare against the Volpich and Bridge (1955) experimental test results shown in Table 1.

Table 1: Sizing Comparison

Paddle Characteristics	V & B Dimensions	INCAT Dimensions
Diameter	3.4ft (1.04m)	1m
Float Length	2.5ft (0.76m)	1.5m
Float Height	0.667ft (0.2m)	0.1m
Float Thickness	1/8in (3.2mm)	3mm
Immersion	0.5ft (155mm)	5-75mm
Material	Aluminium	Aluminium
Shape	Flat	Flat*

* Curved above fluid contact zone

Vessel size was calculated by Revolution Design based on previous experience of area required for helm station, powering and drive train configurations. Primary design drivers included fine entry bow with highly raked stem, minimal wetted surface area at high speed and a smooth water flow into the paddle mechanism. To achieve a smooth flow of fluid into the paddle mechanism in open water conditions a flat bottom transom configuration as shown in Figure 1 was adopted.

Figure 1: Skiff Hull Configuration

Vessel control was achieved through cable steering. Throttle control was achieved via a cable pull hand control, with engine rpm indicated via digital display in front of the operator. This digital tachometer was connected directly to the engine management computer system. Powering for the prototype was achieved through the use of a 3.8 litre V6 automotive engine, mounted transversely, coupled to an automotive 6 speed manual gearbox. Initial operation was carried out without the use of a clutch mechanism with the gear simply selected and the engine started. This system was changed in later iterations to include a clutch. Engine installation included a self contained cooling system allowing testing of the drive system on a hard stand. This drive was transferred to the paddle mechanism via 2 automotive 3.08:1 ratio differentials. This configuration allowed paddle speed variations as shown in Table 2.

Table 2: Drive Train Ratios

	4^{th} Gear 1.19:1	5^{th} Gear 1:1	6^{th} Gear 0.75:1
Engine RPM	6000	6000	6000
Gearbox RPM	5042	6000	8000
Diff 1 RPM	1637	1948	2597
Diff 2 RPM	531	632	843
Final Drive Paddle RPM	531	632	843
Paddle Tip Speed (m/s)	27.8	33.1	44.2
Paddle Tip Speed (knots)	54	64.3	85.9

In initial trials, porpoising instability occurred regardless of the level of immersion of the paddle wheel or speed of the vessel. To overcome this, after several design iterations, flat bottomed hull extensions were eventually installed either side of the paddle mechanism as shown in Figure 2. This figure also shows the steering system that was employed, showing the rudder shape to provide greater area at low speed and reduced area at high speed.

Figure 2: Hull extensions and rudder modifications

To allow for paddle wheel immersion variations, a hinged mount system was developed, including a jacking mechanism, to ensure simple variation of paddle immersion independent of the vessel itself as shown in Figure 3. This was achieved by pinning the entire propulsion system to a large cantilevered arm ensuring correct alignment of the drive system was maintained at all times relative to paddle wheel position.

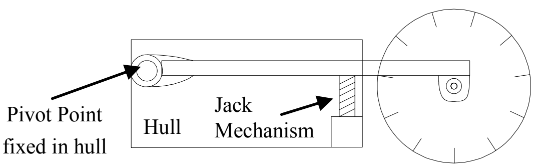

Figure 3: Paddle immersion mechanism

Initial testing at low levels of immersion presented problems in terms of paddle floats or blades impacting on the water surface causing large vibrations. It appeared that the large amount of power being applied to the paddle wheel mechanism was creating a large vertical force which was lifting the wheel clear of the water. This impact force varied from vibration through to physical bouncing of the entire drive mechanism depending on the application of engine power to the system. Smooth power application resulted in smooth operation whereas sudden changes in power caused a thrust breakdown from which it was difficult to recover.

The initial configurations relied on the weight of the power plant and drive train to prevent any backlash or whipping of the cantilever system. This proved ineffective in preventing the paddle wheel from experiencing vibration relating to paddle float impact with the surface of the water. To overcome this problem, the paddle wheel mechanism was clamped.

Clamping of the drive system resulted in positive results in terms of preventing the slamming of the paddle mechanism. However, the fix created another problem in terms of a method to vary the immersion of the paddle. To overcome this, a trim tab type mechanism was installed on the aft edge of the transom extending aft towards the paddle wheel mechanism. This trim tab could be modified via a jacking mechanism located within the vessel by the operator and it can be seen in Figure 4.

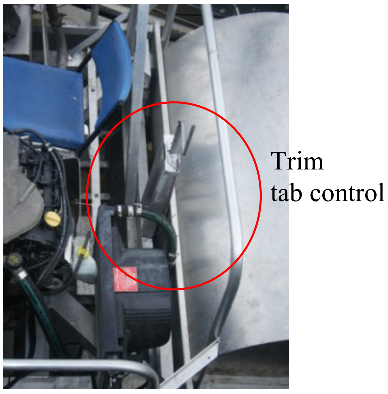

Figure 4: Cockpit showing trim tab control

In a further design iteration, the paddle floats were changed from a flat paddle type arrangement to a semi circular cupped arrangement as shown in Figure 5. The floats were cupped at the top to reduce the carry of water around the wheel during the rotation of the paddle wheel as it was observed that power loss was occurring due to the carry of fluid around inside the paddle mechanism when accelerating from a standing start. This resulted in a reduction of fountaining of fluid behind the wheel when operating at speed as can be seen in comparison between Figure 6 and Figure 7. A negative result of these cupped blades were increased upward forces as a result of deflected fluid being forced up into the cupped paddles thereby pushing the bow of the vessel down; this was solved through a further series of design iterations. The final cockpit layout is shown in Figure 8.

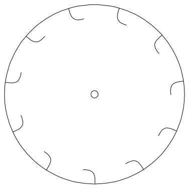

Figure 5: Profile view of cupped paddle floats

Figure 6: Flat float wake fountain

Figure 7: Cupped float wake fountain

Figure 8: Cockpit layout

4 RESULTS AND DISCUSSION

Time precluded comprehensive testing of the paddle skiff and so results consist only of revolutions versus speed attained. Speed was measured using a handheld GPS.

Table 3: Skiff performance on trials

Item	Run 1	Run 2	Run 3
Engine RPM	5800	5800	5800
Gear Ratio	5	5	5
Paddle RPM	611	611	611
Paddle Immersion	5mm	10mm	15mm
Paddle Tip Speed	62.2 knts	62.2 knts	62.2 knts
Skiff Speed	20 knts	32.8 knts	26 knts

In the trials, the paddle operated in the wake of a transom. The immersion of the paddle wheel was relatively low (5-20mm on a diameter of 1m) and the trials experience was that thrust increased as immersion reduced down to an immersion of 10mm, possibly because the thrust coefficient varied at different immersion levels.

The resistance of the skiff was estimated by using a variety of methods, within Formation Design's "Hullspeed" program including, Savitsky's planing method, and Blount's and Fox's method, all of which gave similar results. The results using Savitsky's method were used in the following predictions.

Figure 11 (see end of paper) shows the comparison between hull resistance and thrust across a speed range from 20 to 50 knots. The thrust has been predicted by using curve fits to the coefficient curves at both the maximum and minimum revolutions tested by Volpich and Bridge (1955). These curve fits of thrust coefficient, torque coefficient and efficiency are shown at these maximum and minimum revolutions in Figure 12 (see end of paper). No allowances were made for wake and thrust deduction fraction and hence the hull efficiency was taken as 1.0.

The comparison shows that if the thrust coefficient is at the value estimated from the tests at the maximum revolutions tested, then the skiff would reach a speed of 35 knots; if the thrust coefficients are a level estimated from the tests at minimum revolutions, then it is possible to attain a maximum speed of about 46 knots. In these predictions the immersion level is taken as 20mm.

5 CONCLUSIONS

This paper challenges the conventional assumption that paddle forms of propulsion are ineffective at high vessel and revolution speeds. The trials on a paddle propelled skiff have demonstrated that it is possible to propel a surface craft at speeds in excess of 30 knots by using a paddle as the propulsor. So, in answer to the questions posed in the abstract:

- Has the paddle reached its maximum potential? Almost certainly not.
- Is it possible that paddle propulsion could find applications not yet considered? Almost certainly yes.

The re-presentation of the data published by Volpich and Bridge (1955), and the performance of the paddle used on the planing skiff during trials have also highlighted how little we really know about the performance of paddles as propulsors. In order to achieve the maximum potential of paddles as propulsors for high speed craft, further work is needed and, in particular:

- experimental work on the performance of paddles at high revolution speeds;
- a comprehensive parametric and dimensional analysis to assess which factors are of primary influence on the thrust and torque and hence efficiency of a paddle wheel;
- a study of why it appears that a high speed paddle wheel at lower immersions has a higher thrust coefficient than one at deeper immersions;
- a study of how the interaction of the paddle blades with the free water surface causes variations in the thrust and torque coefficients of the paddle; and
- further practical trials of paddles as propulsors for high speed craft to assess their performance over time, including potential erosion and fatigue life.

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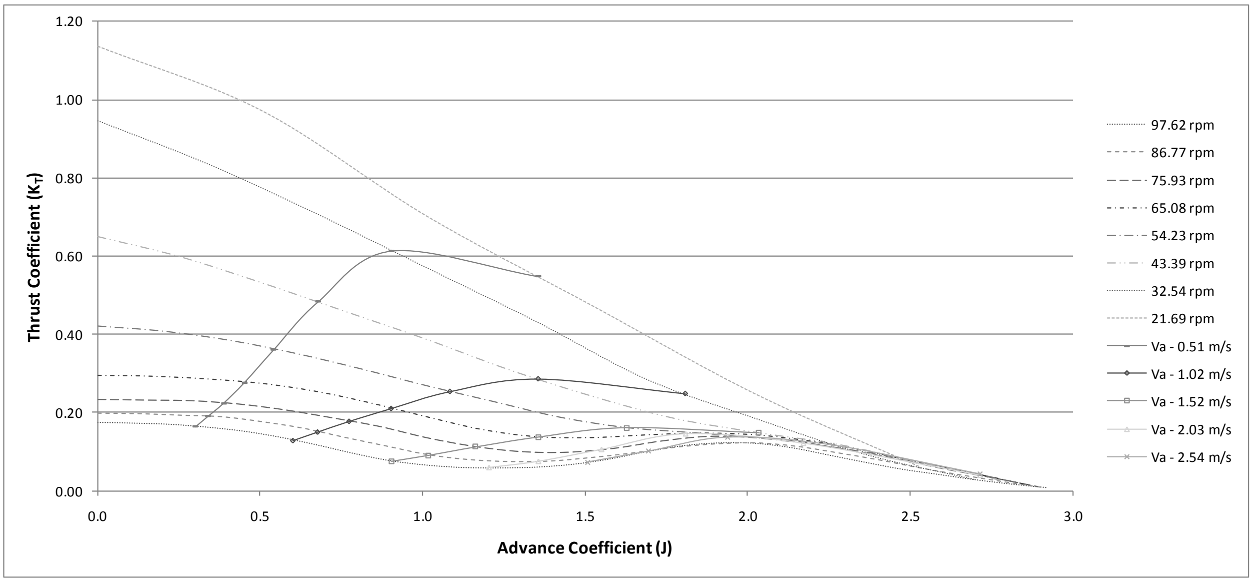

Figure 9: Bose (2008) thrust coefficient values for Volpich and Bridge (1955) tests at various paddle rpm and advance velocities

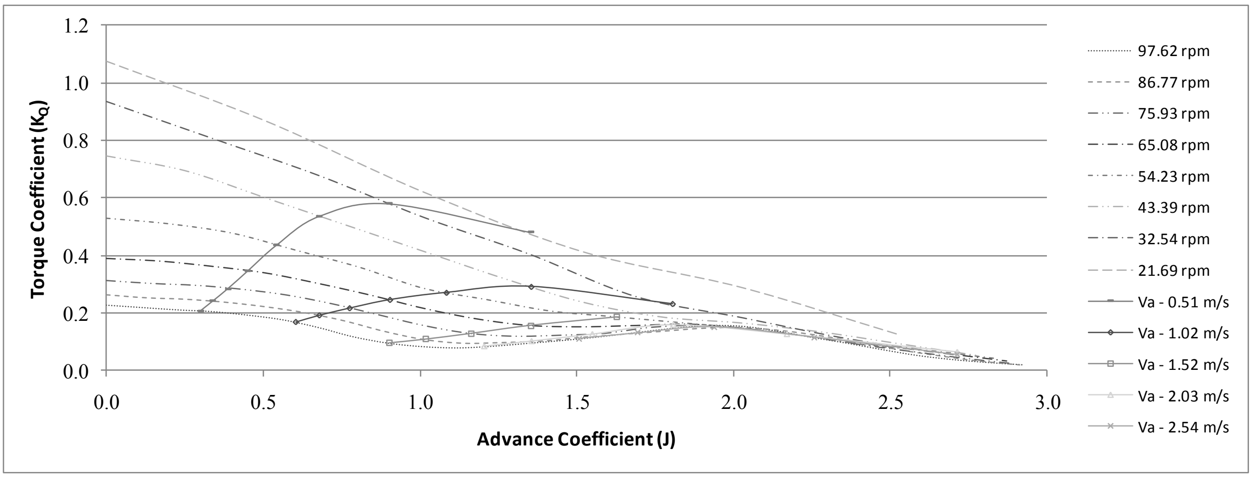

Figure 10: Bose (2008) torque coefficient values for Volpich and Bridge (1955) tests at various paddle rpm and advance velocities

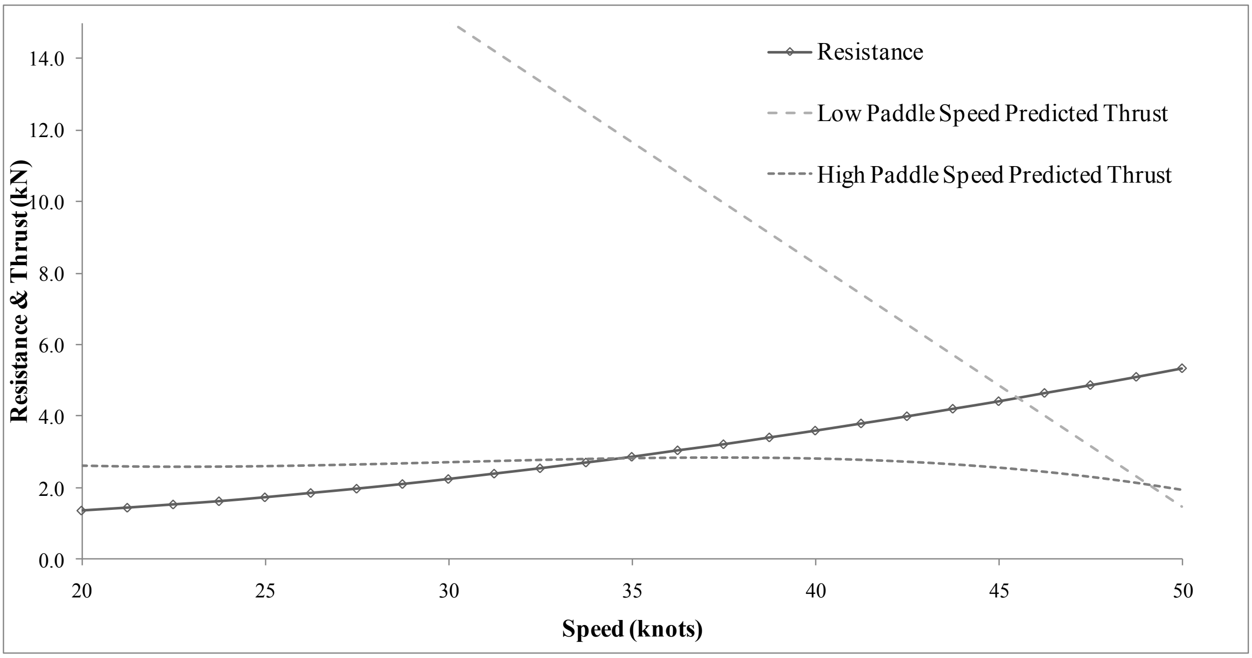

Figure 11: Thrust predictions for high and low paddle rpm

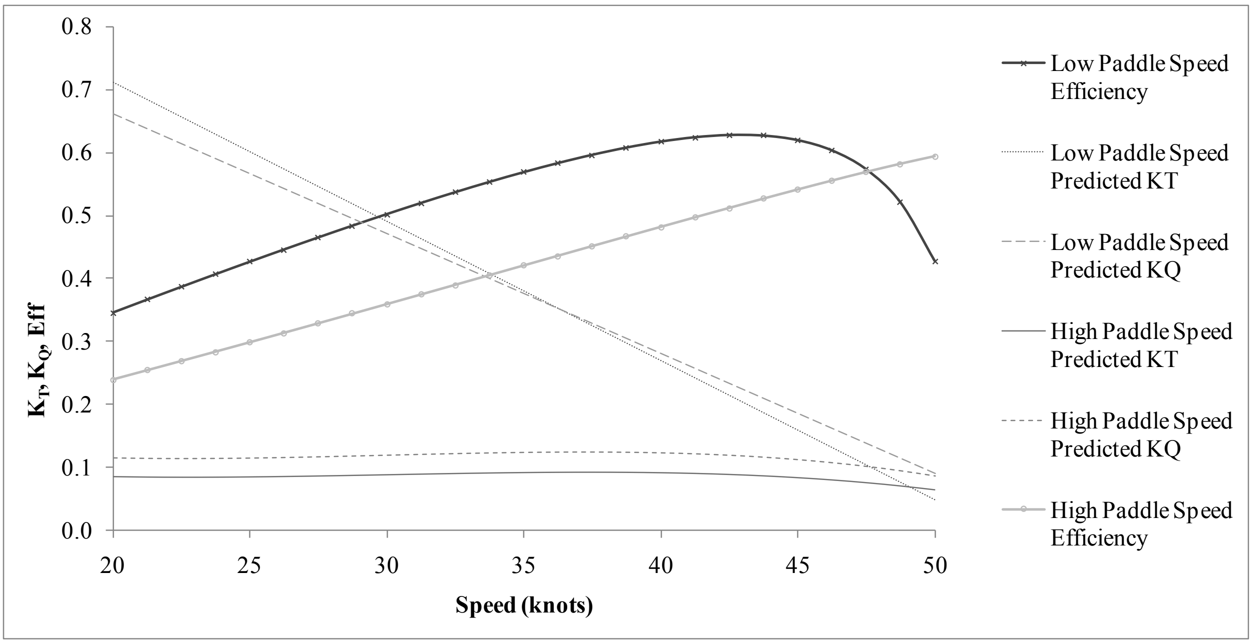

Figure 12: Thrust, torque coefficients and efficiency plots for both high and low paddle rpm operation