The Specialist Committee on Unconventional Propulsors

Final Report and Recommendations to the 22nd ITTC

1 MEMBERSHIP AND MEETINGS

The 21st ITTC appointed the Specialist Committee on Unconventional Propulsors with the following membership:

- Dr. Neil Bose, (Canada), Chair; Memorial University of Newfoundland, St. John's.
- Dr. Michael Billet (USA), Secretary; Applied Research Laboratory - Penn State, State College, PA.
- Dr. Poul Andersen (Denmark); Technical University of Denmark, Lyngby.
- Dr. Mehmet Atlar (UK); University of Newcastle upon Tyne, Newcastle upon Tyne.
- Mr. Christian Dugué (France); Bassin d'Essais des Carènes, Val de Reuil.
- Dr.Ing. Marco Ferrando (Italy); Università degli Studi di Genova, Genova.
- Dr. Wenhao Qian (China); Marine Design and Research Institute of China, Shanghai.
- Dr. Yide Shen (Belgium); University of Liège, Liège.

At the first meeting of the Committee, Dr. Michael Billet was elected Secretary of the Committee. Meetings were held as follows:

- University of Liège, Belgium, February 1997.
- Memorial University of Newfoundland, Canada, September 1997.
- Università degli Studi di Genova, Italy, February 1998.
- Bassin d'Essais des Carènes, France, September 1998.

2 TASK SET FROM THE 21ST ITTC

Develop guidelines for carrying out propulsion tests and extrapolating the results to full scale for propellers with ducts, partial ducts, pre- and post-swirl devices, tip plates and z-drives.

3 INTRODUCTION

In order to address extrapolation methodology for unconventional propulsors, the Committee first reviewed known extrapolation methods currently in use for ships fitted with different types of these devices. A summary of these reviews is given in section 4 of this report. Section 5 then describes the main extrapolation methods in further detail.

Many of the extrapolation methods presently in use for powering performance prediction of ships fitted with unconventional propulsors are based on modifications of the ITTC 1978 method. The ITTC 1978 formulation was developed primarily for single-screw open-propeller ships and established an extrapolation methodology using interaction parameters such as wake and thrust deduction fraction as well as Reynolds number based friction corrections. Organisations have not only modified the ITTC 1978 approach for open propellers based on experience, but more importantly in the present context, developed new methodology for each particular type of unconventional propulsor treated by using these methods. However, on the horizon are developing unsteady RANS codes, new testing procedures for towing tanks, application of large high-Reynolds number water tunnels, etc., that can more rigorously address these issues. Also, there is still a great need for reliable full scale unconventional propulsor performance data to validate any methodology.

Devices added to the blade tips, such as bands, tip fins, bulbs, etc., and those added to the propeller hub, such as boss cap fins, form a class of unconventional propulsor that can utilise relatively reliably the ITTC 1978 extrapolation methodology. However, the effect of Reynolds number on this methodology needs to be resolved as devices that locally modify the flow field to control cavitation or improve efficiency are often tested at relatively low Reynolds number in the towing tank. The more extreme cases where the propulsor can be considered part of the ship hull, such as integrated ducted propulsors, form a group of unconventional propulsors where the ITTC 1978 methodology is clearly not adequate. In some cases, unconventional propulsors involving pre- and post-swirl vanes, ducts, propeller pods, etc., have been treated for extrapolation purposes by using a modified ITTC 1978 methodology. In other cases new methods have been developed for extrapolation and it is expected that the most reliable methods to be developed in the future will make use of new methods.

Section 6 discusses the rationale behind the Committee's proposals for the future development of more general extrapolation methods for powering prediction of ships fitted with unconventional propulsors. Section 7 gives the Committee's Conclusions and Recommendations.

4. MODEL SCALE AND FULL SCALE RESULTS OF VARIOUS EXTRAPOLATION METHODS FOR UNCONVENTIONAL PROPULSORS

4.1 Propeller Boss Cap Fins (PBCF)

Recent activities concerning effective boss cap designs to improve the efficiency and cavitation performance of marine screw propellers have focused on the device known as "**P**ropeller **B**oss **C**ap **F**ins" (**PBCF**). This was proposed by a group of Japanese inventors, Ogura *et al.* (1988), as a means of increasing the efficiency of a ship screw propeller. Almost all the research and development work on PBCF have been reported through a series of technical articles published by the technical departments of the Mitsui OSK Lines Ltd., West Japan Fluid Engineering Laboratory Co. Ltd and the Mikado Propeller Co Ltd. who are the joint patent holders of the device.

Ouchi (1988) and Ouchi *et al.* (1988 and 1989) presented early research and development work on PBCF involving detailed model tests, flow visualisations and the first full-scale measurements on the 44,979 GT PCC "Mercury Ace". These investigations reported a gain of 3-7% for the propeller efficiency through the "reverse **POT**" (**P**ropeller **O**pen **T**est) in a circulating water channel, 2 to 2.3 % gain in the propulsive efficiency based on self-propulsion tests and finally a gain of 4% for the power output on the actual vessel. Further investigations were also reported by Ouchi (1989), Ouchi & Tamashima (1989) and Ouchi *et al* (1989,1990 and 1992) including 3-D LDV measurements in the model propeller slipstream, full-scale measurements with 11 vessels and cavitation and noise investigations. The comparative analysis of the sea trial results of 11 ships and their models indicated considerable scale effect between the model and actual measurements such that the efficiency gain at full scale could be two or three times that at the model scale. This was mainly attributed to laminar flow around the boss cap and possible laminar flow separation at the back of the fins. Finally a suitable scale factor, which could be determined from a large number of tests on models and actual ships in the same manner as the roughness allowance in the resistance of ships, was recommended to compensate for the scale effect.

Ouchi (1988) and Ouchi *et al.* (1990) also reported on various other aspects of PBCF associated with the extrapolation of the powering performance. They indicated that the presence of a rudder significantly reduces the strength of the hub vortex and hence the gain in propeller efficiency due to PBCF can be reduced by 10-30%. Self-propulsion tests to investigate the propulsive efficiency by PBCF indicated that the efficiency gain obtained from the self-propulsion tests were similar to the gain obtained from the "reverse POT" tests with the rudder. The effectiveness of the PBCF was reported to be hardly affected by the hull wake suggesting that most of the efficiency gain was due to the propeller flow itself, but it was important to take into account the effect of the rudder. The thrust identity analysis of the self-propulsion tests, in which the PBCF was considered as an appendage to the propeller, demonstrated that the effect of the PBCF primarily appears as an increase in the relative–rotative efficiency due to the reduction in the propeller torque due to the PBCF.

In order to shed more light on the function of PBCF, Gearhart & McBride (1989) carried out a detailed flow analysis and power measurements with PBCF. The experimental analysis of the efficiency gain from PBCF utilised the results of LDV measurements in the ARL Pennstate water tunnel with a model propeller behind a hull including its rudder. This concluded that 2 % out of a total 6% efficiency gain was due to thrust increase while the remaining 4% was associated with torque decrease. It was anticipated that the total gain would be somewhat reduced due to the frictional drag of the fins. Indeed thrust and torque measurements with the same model showed 5% gain in efficiency with the PBCF confirming the earlier predictions independently. It was also claimed that it was not apparent from the respective studies that the PBCF type device had to contain the same number of vanes as the number of the propeller blades to achieve the max efficiency gain as claimed in the PBCF patent.

As far as extrapolation methods are concerned, no particular procedure dedicated to PBCF has been reported in the above references. However, it appears that the patent holding companies have a well-established procedure to quantify the power saving and hence the efficiency by the use of PBCF. This procedure is based upon the reverse POT set-up in their circulating water channel and the experience gained over a considerable number of full-scale measurements. Moreover, private communication with the West Japan Fluid Engineering Laboratory Co. Ltd. by this committee has confirmed that the standard extrapolation technique for the performance prediction is based upon the ITTC 1978 procedure, as briefly reviewed in Section 5.6.

Atlar and Patience (1998) gave a detailed review of activities on various boss cap designs as well as on PBCF. Amongst them is a recent innovative concept, which is known as the **H**ub **V**ortex **V**ane (HVV), jointly developed by the Potsdam Model Basin, SVA and SCHOTTEL. The HVV is a small vane propeller fixed to the tip of a cone shaped boss cap within a limiting radial boundary where the tangential velocities due to the hub vortex are greater than those of the propeller. In this range the small vane propeller diverts the high tangential velocities in the direction of the jet, thereby generating additional thrust. It is claimed that this mechanism is different from the mechanism for PBCF where the fins are located usually beyond this limiting radius and the fins themselves do not generate thrust. Unlike PBCF, the number of the vanes of HVV is greater than that of the propeller blade. In their product briefs, Potsdam Model Basin(SVA) (1995) and SCHOTTEL (1995), and in a detailed technical report by Schulze (1995), the results of LDV measurements and cavitation details with HVV models are presented. These publications report remarkable reductions in the hub vortex cavitation by the HVV as well as a successful application of HVV on the full scale claiming an increase of 3% in the propeller efficiency.

Atlar and Patience (1998) also reported on a series of experimental investigations with various boss caps, which were done in the Emerson Cavitation Tunnel at Newcastle University, over the last decade. These involved comparative efficiency and cavitation performance measurements with different boss caps through the reverse open water tests in the presence of no rudder and later on with a rudder for some caps, Atlar *et al* (1998). The cylindrical boss caps tested were fitted with: fins similar to PBCF; fins with end plates; fins as extension to the propeller blades as **T**railing **E**dge **F**laps (**TEF**); an accelerating duct; a decelerating duct and **Bo**ss **S**lots (or holes) (**BoS**). The latter are designed to increase the pressure in the hub vortex core. The comparison of the performances for these caps displayed comparable efficiency savings relative to a standard cone shape type except for the cap fitted with the decelerating duct which displayed a large amount of efficiency loss. The BoS type cap displayed a superior suppression of the hub vortex cavitation, but also at the cost of a considerable efficiency loss. No particular mention has been made of extrapolation issues in these investigations.

4.2 Tip Fins

End-plate and other such propellers that have modified blade tips, have various names like tip-fin, Kappel, winglet, etc., propellers. The special geometries are in general only limited modifications of the propeller. For this reason the model testing and scaling of results to full scale can in principle be done in the same way as for a conventional propeller. In this section a review of model testing methods and scaling of the results from such propellers is given. As far as end-plate and similar propellers are concerned the literature often deals with the propulsors separately, i.e. if and how they improve their efficiency, whereas scaling and power prediction are not treated. Here, only the latest references or those related to scaling and prediction are given.

Sparenberg & de Vries (1987) by using linear optimisation theory designed and tested a 3-bladed end-plate propeller. Open water tests were carried out in two different series for two different rates of revolutions. Correspondingly, the Reynolds numbers (with respect to chord length and relative inflow velocity at the 0.85R section) varied. There were clearly differences in the measured open-water curves (K_Q and η, but only small differences in K_T) reflecting the influence of Reynolds number. Over a range of advance ratios, including the design value, the efficiency was higher for the high Reynolds number, as expected. The tests also included measurements with the end-plate propeller working behind the model of a tanker, making it possible to present the propeller efficiency in the behind condition.

Model testing of an optimally designed propeller with two-sided shifted end plates on the blades was described by de Jong *et al.* (1992). The tests consisted of open water tests and cavitation tests. The former were carried out at a rate of rotation sufficiently high to avoid laminar-flow effects on the propeller blades which was checked by further experiments. In the cavitation tests uniform inflow was considered. The interest was focused on the end plate propeller and on comparisons with the same propeller without end plates and with a reference propeller. For that reason scale effects were not considered in greater detail than as outlined above, nor were interaction between propeller and ship or aspects of power prediction dealt with.

For the special end-plate propeller known as the CLT propeller, Hollstein *et al.* (1997) outline the design of such a propeller for a bulk carrier and compare this with data for a sister ship fitted with a conventional propeller. According to them no model tests were carried out, although resistance, open-water and self-propulsion test results for a conventional propeller are available for the designer of the CLT propeller. This is because "...the interpolation procedure to be applied to test results with CLT propellers request certain unexpected corrections only known by (the designer) and hence not available for model basins. This makes it advisable to rely for the predictions of full-scale performance of the CLT propeller on the basis of direct calculations". In their paper they show calculated open-water characteristics of their CLT propeller as well as the characteristics of the conventional propeller. They also give trial trip results for the ships with conventional and CLT propellers.

Table 1 Propulsion coefficients, power and rate of revolutions obtained by model experiments and three different extrapolations and by theoretical predictions (Andersen 1996, table 3).

V = 9.03 m/s	DMI extrapolation		ITTC-78 Extrapolation		DMI-ITTC extrapolation		Design*	
	Tip-fin	Conv.	Tip-fin	Conv.	Tip-fin	Conv.	Tip-fin	Conv.
w	0.30	0.27	0.27	0.26	0.30	0.27	0.26	0.26
t	0.16	0.16	0.16	0.16	0.16	0.16	-	-
η_R	1.00	1.02	1.00	1.02	1.00	1.02	1.00	1.00
η_H	1.21	1.16	1.16	1.14	1.21	1.16	-	-
η_0	0.63	0.62	0.68	0.66	0.67	0.65	-	-
η_D	0.75	0.73	0.79	0.76	0.81	0.77	-	-
$\frac{\eta_{D\,tip\ fin}}{\eta_{D\,conventional}}$	1.037	1	1.038	1	1.047	1	-	-
P_D [kW]	6720	6980	6410	6650	6270	6570	6850*	7260*
n [1/s]	1.95	1.99	1.98	2.01	1.95	1.99	1.97	2.01
C_{Th}	1.58	1.49	1.46	1.42	1.58	1.49	1.55*	1.55*

* Design includes an increase in resistance (allowance) of 9 per cent

For the tip-fin or Kappel propeller with integrated fins in the tip region (bent blade tips) Andersen (1996) carried out a comparative study with the conventional propeller actually fitted on the ship as a reference. The model tests consisted of traditional tests, i.e. in addition to resistance tests for the ship model, open-water, self-propulsion and cavitation tests for both conventional and tip-fin propeller models. Three different methods were used for making the full-scale power prediction on the basis of model test results. One method was due to the model basin (Danish Maritime Institute) where a correlation allowance and a correction to the wake fraction coefficient was applied. The second method followed the ITTC 1978 with the exception that a more detailed scaling of the influence of friction on the open-water propeller characteristics was carried out. This procedure was based on the flat plate frictional resistance coefficient. The third method combined the two others in applying the procedure of the model basin (correlation allowance and correction to the wake) and the correction to the propeller open-water characteristics. The three methods gave different required power, demonstrating the need for rational procedures for such estimations (see table 1.).

4.3 Surface Piercing Propellers

Surface piercing propellers (SPP) can be included into the unconventional propulsor family because of their particular mode of operation. However, system geometry and components for surface piercing propulsion is quite a conventional concept for it includes only the classical propeller-shaft-rudder layout.

Current extrapolation methods are focused on two items:

1) extrapolation to full scale of the model open water performance;
2) extrapolation to full scale of the propeller hull interaction (wake fraction, thrust deduction and relative rotative efficiency).

Comments on item 1). This extrapolation procedure should take into account the influence of two additional parameters, namely the Froude number, F_n, and the Weber number. The influence of the Froude number is relevant as the propeller acts at the interface between air and water much like hulls. The Weber number is a ratio between inertial and surface tension forces. Its influence can be easily foreseen for an SPP, which continually pierces the water surface.

The first comments on the role of F_n are due to Shiba (1953). In his investigation he pointed out that gravity affects the shape of the air cavity through the Bernoulli equation that can be enforced at the boundary between water and the atmosphere vented cavity. Accordingly, the influence of F_n based on the diameter of the propeller as the length parameter and nD as the speed, vanished when the air cavity approached its ultimate form, i.e. for Froude numbers greater than 3. Hadler and Hecker (1968) indirectly concurred with this hypothesis, but they calculated the Froude number using the total inflow velocity and the immersion of the shaft as the length parameter. Finally, Olofsson (1996) acknowledged an influence of F_{nD}, (calculated with the diameter as the length parameter), but he set to 4 the limiting value beyond which the influence disappears. Summarizing, it is widely recognized that the Froude number does affect the behavior of SPPs, and all authors have suggested the existence of a threshold value that limits this influence. The same general agreement has not been reached on the minimum value that must be attained to avoid scaling problems. This is due to the different kind of Froude numbers that have been used. Further research is required to identify the minimum value of F_n which must be achieved during open water tests. Nevertheless, provided that open water tests are performed at F_n beyond the threshold value, the Froude number identity can be avoided during the tests without affecting the full scale performance of the propeller.

According to Shiba (1953), surface tension plays its role when the propeller is about to be fully ventilated. Complete ventilation is a rather sudden phenomenon that can be correlated to a certain value of J called the critical advance coefficient J_{CR}. The critical advance coefficient can roughly be located in the middle of the transition region and the sudden drop of K_T and K_Q identify its position. Indeed Shiba found a correlation between Wn and J_{CR} for a single propeller. Later work on this matter (Ferrando and Scamardella 1996) acknowledged the influence of the Weber number on the location of J_{CR}, and the existence of a threshold value of W_n beyond which its influence disappears. Unfortunately there is not sufficient evidence to suggest the existence of a unique threshold value. Further research on this subject is therefore strongly needed. Anyway, provided that open water tests are performed at W_n beyond the threshold value, the Weber number identity can be avoided during the tests without affecting the full scale performance of the propeller.

As far as is known, provided that open water tests are performed in agreement with the above requirements, the performance of SPPs can be scaled in the same fashion as conventional propellers, i.e. by applying a Reynolds number correction only.

Comments on item 2). The propeller is usually located quite far from the hull (i.e. 1 to 2 diameters away) and the projection in the vertical plane of the immersed hull surface near the propeller is small. As a consequence, the value of the thrust deduction factor for SPPs is negligible or zero. Furthermore, from a theoretical point of view the phenomenon of augmented resistance for this kind of propulsion can be treated with the same physical model used for conventional propellers because no additional thrust affecting devices are involved. Hence the same scaling procedure adopted for conventional propulsion can apply also to SPPs.

Surface piercing propulsion is mainly employed on planing or semi-displacement craft, exploiting the ventilation of the transom. As SPPs operate far behind the hull the contribution of the potential wake is small. On the other hand, the viscous wake cannot be neglected, because part of the propeller lies in the boundary layer of the hull. Considering that surface piercing propulsion does not include any additional wake modifying device and that the physics of the wake production process is the same as in the case of conventional propulsion there are no obstacles to using the same scaling procedure as that used in the case of conventional propeller arrangements.

As regards relative rotative efficiency there is no particular reason that prevents the application of traditional scaling methods.

Setting aside the preceding considerations, almost all applications of surface propulsion are in the pleasure craft trade. Generally, the budget for the boat design does not allow for big expenditures like extensive tank testing or high standard sea trials. There is not a single known case study in the open literature providing data for extrapolation purposes.

In some favourable circumstances EHP is determined by means of towing tests, while for the great majority of applications an effective horse power estimate is performed using Savitsky's (1964) equations or something similar. In both these instances there is not enough data for a true extrapolation of the propeller-hull interaction. The absence of sea trial results prevents the development of a reliable full scale extrapolation.

4.4 Oscillating Propulsors

Oscillating propulsors are still at the research and development stage. Only very few actual applications have been built and tested at full scale and all of these have been for small craft. Examples are the small boat driven by an oscillating propulsor and outboard motor designs built by Isshiki *et al.* (1987), the experiments done by a group led by M. Triantafyllou at MIT, and various oscillating propulsor designs used on human powered craft (e.g. Bennett 1996), including examples on human powered submersibles (Skidmore *et al.* 1989). As a result, extrapolation methods for powering prediction for oscillating propulsors are rare.

Other propulsors that operate with a similar propulsive mechanism to the oscillating propulsor are trochoidal propellers (a type of cycloidal propeller), the most recent example of which is the Whale Tail Wheel (Anon. 1998) being fitted to the inland waterways vessel "Ludwina". Such propulsors have been studied at model scale (e.g. Manen 1973, Bose and Lai 1989, Riijarvi *et al.* 1994). A special type of oscillating propulsor, that might be described as a hybrid between a conventional and an oscillating propulsor, is a propeller with cyclic pitch control (e.g. Gabriel and Atlar, 1998).

Estimates of the full scale performance of ships fitted with oscillating propulsors have been made by Lai (1990) and Yamaguchi and Bose (1994). These have been based on conventional approaches to powering prediction using estimates of the hull efficiency to account for propulsor/hull interaction. Yamaguchi and Bose (1994) assumed neligible hull/propulsor interaction leading to a hull efficiency of one. The rationale for this estimate was that as an oscillating propulsor design normally has relatively light loading, they are usually large devices with relatively large swept area and large span relative to the ship beam and/or draft. In view of this much of the propulsor is not in close proximity to the hull and the hull/propulsor interaction was assumed to be small. For the ship under consideration, this was a conservative estimate as the hull efficiency was above one for the comparable conventional propeller system. Lai (1990), on the other hand, assumed similar values of the wake and thrust deduction fraction as the conventional propeller system for the oscillating propulsor proposal when considering the powering performance of four ships. This was done in the absence of more detailed data as the hull efficiencies for the four vessels varied from 0.92 to 1.12.

4.5 Pre and Post-Swirl Devices

Since the 1970s the development and application of hydrodynamic energy-saving devices (ESD) have been demonstrated all over the world. Tables 2-4 show energy savings from pre- and post-swirl devices. The propeller inflow compensate nozzles developed by Schneekluth have been installed on more than 600 ships (Schneekluth, 1986). Powering performance prediction for full-scale ships with pre- and post-swirl devices have been made based on model test results. For most of these the ITTC 1978 method and the two dimensional Froude method have been used to predict the full-scale performance for ships having a large range of block coefficients. At MARIC several different prediction methods based on a modified ITTC 1978 method have been used to predict full-scale performance from ship model test results. These predictions have been compared with the results from sea trials or from statistical sailing data.

Table 2 Pre-swirl energy-saving devices.

Name of pre-swirl device	Energy-saving mechanism	Energy-saving rate % model test/trial
Reaction fin	PRI	4-8 / 4-9
Novel integrated duct	IPI	4-5 / 4-6
Inflow compensate nozzle	IPI, AFS	6-11 / 8
Simplified compensate nozzle	IPI, AFS	4-9 / 4-9
Wake-adapted duct	IPI, AFS	4-9 / 4-9
Hydrodynamic fins	IPI	4-6 / 3-9
Fore-propeller hydrodynamic fin sector	IPI	4-7 / 4-7
Thrust shaft brackets	PRI	4.5-12 / 5-8
Sheathed shaft bracket	PRI	5-8 / 5-8
Hydrodynamic partition plate	IPI, AFS	2-3.7 / 2-3.7
Aperture fin	PRI	2-4 / 2-4
Stern-appended fin	IPI, AFS	3 / 3
Flettner rotor at stern post	IPI, AFS	8 /
Fore-propeller vane-wheel	IPI, AFS	3-4 /

Since 1984 MARIC has developed more than ten ESDs such as the composite device of simplified compensate nozzle (SCN) and Costa propulsion bulb (CPB), thrust shaft brackets (TSB), fore-propeller hydrodynamic fin sector (FPHFS) etc. (Qian *et al.*, 1992a, Zhou *et al.*, 1990, Qian *et al.*, 1992b). Eight different types have been put into use on more than 200 ships over the range from 500 to 150,000 dwt with an annual 50,000 ton fuel-savings. In all cases a modified prediction method based on the ITTC 1978 method has been used in the prediction of the full-scale performance from the ship model test results. Practice indicates that the energy-saving rate (ESR) or the predicted speed of the ships installed with ESDs correlate well with the full scale data.

Table 3 Post-swirl energy-saving devices.

Name of post-swirl devices	Energy-saving mechanism	Energy-saving rate % model test/trial
Fixed guide vane after propeller	RRE	4-6 / 5
Vane-wheel	RRE	5-15 / 6-12
Rudder-appended thrust fins	RRE	4-5 / 4-8
Reaction rudder (asym. rudder)	RRE	2-4 / 2-4
COSTA propulsion bulb	DVP	2-4 / 2-4
Propeller cap fins	DEP	2-5 / 2-5
Eddy-eliminating composite propeller	DEP	5.1 / 8.7
Integrated duct-vane-wheel	RRE	5-10 /
Rudder-appended Flettner rotor	PAT	10 /

Table 4 Pre and post-swirl energy-saving devices.

Name of pre and post-swirl devices	Energy-saving mechanism	Energy-saving rate % model test/trial
Hydrodynamic fins & guide vane-wheel	IPI, RRE	/ 9-18
COSTA propulsion bulb & rudder-appended thrust fins	DVP, RRE	4-14 / 4-7.4
Duct-thrust fins after propeller	DVP, RRE	4 / 4-5
COSTA propulsion bulb & simplified compensate nozzle	IPI, AFS, DVP	6-12 / 4-12
Assembly propulsion devices	IPI, RRE	5-15 / 5-15
Composite energy-saving technique	IPI, RRE	8-14 / 8-14

PRI – give a pre-rotation to the propeller inflow
IPI – improve propeller inflow
AFS – alleviate flow separation
RRE – recover rotational energy from downstream
DVP – decrease viscous loss after propeller cap
DEP – decrease eddy after propeller cap
PAT – produce additional thrust

4.6 Ducted Propulsors

Some methods for the powering performance of ships fitted with ducted propellers are described in section 5.4. Included here are some observations from a set of tests done over a range of Reynolds numbers which include, in particular, some trends at higher than normally tested values of Reynolds number.

Extensive propulsion related tests were recently done on a tugboat model at Bassin d'Essais des Carènes. This tug-boat was a twin-screw vessel with two ducted propulsors. A rudder was placed closely behind each shroud (about 1/3 of the chord of the rudder and less than 1/2 the length of the shrouds behind) so that the interaction between rudder and propulsor had the potential to be strong. The slope of the aft end of the boat was high, so that the propulsors were working in inclined flow conditions. The distance between the shroud and the hull was small which created the possibility of detached flow in some conditions and strong interaction between the hull and propulsors. Due to restricted space, the propulsors were highly loaded.

The model was about 4.5 m long, equipped with a propeller of about 20 cm diameter. The speeds tested, based on Froude number scaling, were about 1.5 m/s for the transit speed and about 0.5 m/s for high towing conditions. Open water, resistance, self-propulsion and varying load tests at all advance ratios were performed, including measurements of the axial thrust of one shroud. The boat was a twin screw vessel, which implies that the prediction and extrapolation methods based on open water, resistance and self propulsion tests should not be affected by the problems of high wake as would be expected on single-screw ships.

Additional tests were done to investigate some problems related to prediction and extrapolation for ducted propellers and these have relevance also for some other types of unconventional propulsors. These were:

- towing conditions at low speed (from 0.2 to 0.4 m/s) to detect any scale effects due to low Reynolds number, especially on the shrouds;
- self propulsion tests without rudders (transit and towing), to separate effects from

interaction of the propulsor with the hull and the propulsor with the rudder;

- resistance tests of the hull plus the two shrouds, without rudders and without propellers;
- resistance tests of the shroud alone without the propeller ("open water test").

The drag of the shroud in the "open water" test (figure 1) clearly shows a separation phenomenon at low speed and a transition speed. A speed of at least 2 m/s is necessary to get stable values away from this transition. This problem seems to be avoided in the behind condition where the values are different.

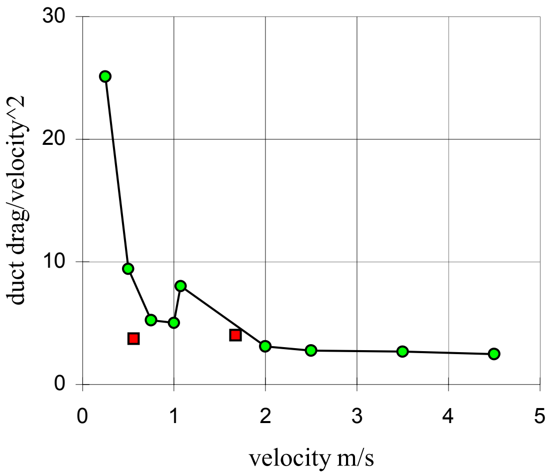

Figure 1. Shroud drag. The circles denote the drag from the "open water test"; the squares are the values found from the resistance of the hull tested with and without the shrouds fitted.

The tests of resistance with the shrouds and without propellers showed that the shroud drag was about 25% of the total. Comparison of these values with those obtained from the open water test on the shrouds, showed that the interaction term was between 1 and 1.5% (the total drag of the hull plus shrouds was 1% to 1.5 % higher than the value obtained from the addition of the naked hull resistance and open water drag of the shrouds). This interaction term is very small despite the geometry.

The open water test of the whole propulsor shows a Reynolds number effect on K_T and K_Q that affects the propeller characteristics in the speed range up to 4.5 m/s. The extrapolation of the characteristics of the propeller at the Froude speed (about 0.5 and 1.5 m/s for towing and transit) would not give the characteristics of the propeller at the higher speed tested.

The plot of K_T of the propeller (K_T obtained with the shroud but not including the thrust of the shroud) versus K_Q (figure 2) for the open water tests and the two conditions of load varying tests (with and without rudders) shows that over the whole range of advance ratio (from bollard pull to almost no thrust) the three curves are almost identical. This indicates that the relative rotative efficiency is very close to 1 with a precision of less than 1%. The dispersion between the curves is due to precision uncertainty in the measurements, the importance of which becomes relatively smaller for points close to bollard.

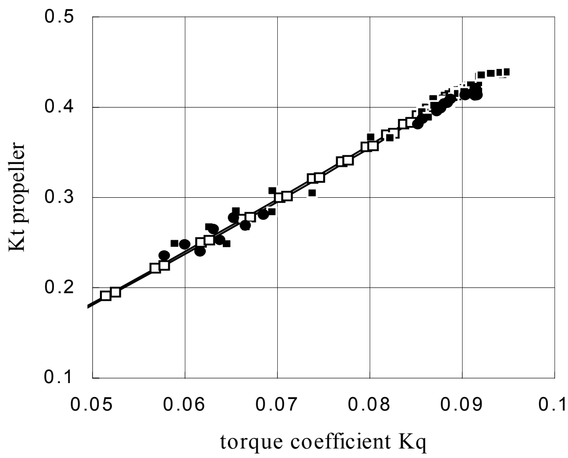

Figure 2. Thrust coefficient of the propeller versus torque coefficient. The open squares are open water values; the solid squares are the behind condition with the rudder; and the solid circles are the behind condition without the rudder.

Nevertheless, looking at the detail of the curves in figure 2 for high loads, there is an important difference between the curves. Although the torque coefficient plots against advance coefficient are not shown here, the torque coefficient in the bollard condition for the open water and behind condition (K_Q = 0.0915) was the same with a precision of less than 0.2 % for both propellers. However, K_Q becomes 0.0945 in the bollard condition with rudders (an increase of 3%). Also, the K_T of the propeller shows little change from 0.422 in open water to 0.414 in the behind condition without rudder (a decrease of 2%), while there is an increase to 0.438 (plus 4%) with the rudder. Clearly, the rudder has a "postswirl" effect on the propeller rotor, increasing the thrust by 6% and the torque by 3% between the two behind conditions. As a result of this postswirl effect, thrust is increased more than torque leading to an increase in efficiency. These arguments together with figure 2 show that the effect of the rudder on the relative rotative efficiency is far more important than the difference between operating in open water and the behind condition.

The plots against advance coefficient (see figure 4 for thrust coefficient against advance coefficient) show also that a K_T identity for the behind condition with rudder, based on the rotor alone is meaningless for low advance ratio: K_T changes from 0.438 in the bollard pull condition to 0.426 for J_S = 0.3 while K_T in open water has a maximum value of 0.422. A K_T identity without rudder is also problematic as a value of K_T = 0.414 is obtained in open water for J_0 = 0.25 (i.e. less than the open water value and leading to a negative wake). A K_Q identity could be done in this latter case, as the bollard K_Q is the same in the open condition and the condition without rudder, but the slope of the K_Q versus J_S is too small to be precise at least for small J. When a plot was drawn (not presented here) showing K_T of the propeller versus K_Q, for the open water condition and behind condition only, but with additional points from propulsion tests at smaller Reynolds number (simulating towing at full scale speeds of 2 to 4 knots), there was a big discrepancy between some of these last points due to scale effects and/or large errors in measurement of very small forces.

The two plots of K_T of the duct versus K_Q and K_T of duct versus K_T of the propeller (the latter only is plotted in figure 3) for open water and behind condition with and without the rudder shows again an effect of the rudder.

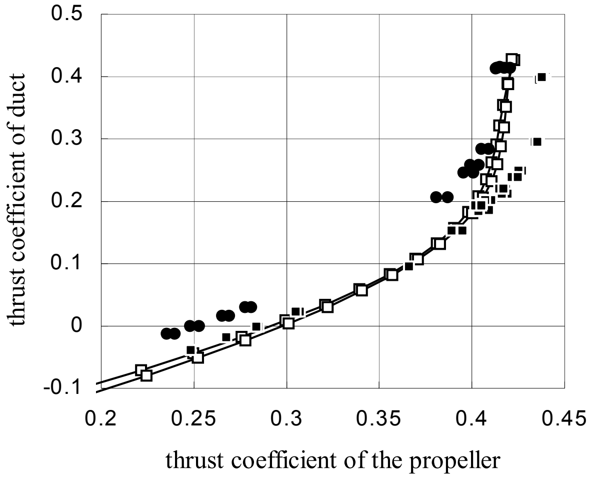

Figure 3. Thrust coefficient of the duct versus thrust coefficient of the propeller. The open squares are open water values; the solid squares are the behind condition with the rudder; and the solid circles are the behind condition without the rudder.

Direct comparison betwww.kriso.re.kr/ITTC/index.htmleen the open water and behind condition with rudder gives a confusing picture which at first sight seems to show that there is no effect on the shroud at transit speeds as a result of operating in the hull wake and in an inclined flow, but that the effect at towing speeds is large. This is a false conclusion since comparing the open water results with the behind condition curve without the rudder shows a shift in the thrust of the shroud. However, the thrust values at bollard for these two conditions are within 1%. Everything appears as if the shroud is "seeing" a flow having less speed than the propeller rotor itself, that is to say as if the wake fraction "seen" by the shroud is a little bit larger than that "seen" by the propeller rotor. Is this due to a boundary layer effect of the hull on the pressure distribution of the shroud? Direct comparison of the open water results with the behind condition with rudder gives then a confusing picture simply due to the fact that for the same flow around and in the shroud, the rudder gives a shift on thrust and torque of the propeller rotor (postswirl effect) which is an effect as large as the effect of the hull on the shroud. But the postswirl effect doesn't change the flow around the shroud as the bollard thrust of the shroud indicates.

These latter points have further implications on the identification of wake fraction from the thrust identity. From the plots of thrust coefficient against advance coefficient (one of which is shown in figure 4) the following can be seen. Comparing open water and behind condition with rudder gives a wake fraction of about 10% for J around 0.3-0.4 (towing condition). However, in the case without rudder, wake fraction is below 3%. The value of 1-w is surprisingly more than 1 in this region of advance ratio. There seems to be a difference of regime between these low J values and the region of transit.

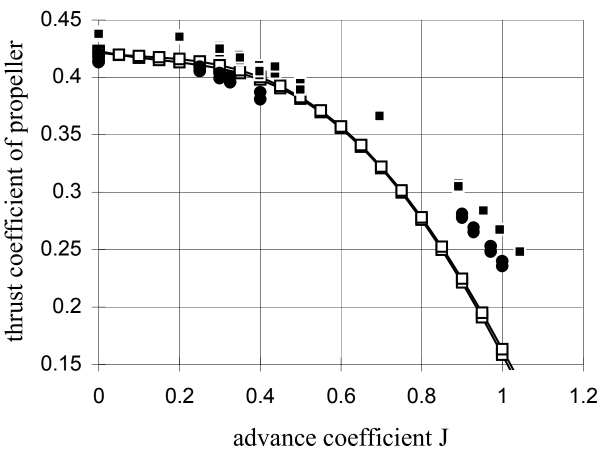

Figure 4. Thrust coefficient of propeller versus advance coefficient. The open squares are open water values; the solid squares are the behind condition with the rudder; and the solid circles are the behind condition without the rudder.

In the region of transit speeds, the effect of the rudders is about 0.04-0.05 on the advance coefficient of the rotor. The K_T identity based upon the total K_T may work because of a cancelling effect between the rudder and the shroud.

4.7 Partial Ducts

Wake equalizing ducts were developed by Schneekluth (1986). They consist of two nozzle-shaped half ring ducts which are installed on both sides of the stern ahead of the propeller. Their diameters are about the same as the radius of the propeller and their chord is smaller than the diameter. Sometimes, only one duct is fitted to the stern on one side of the propeller.

The wake behind single-screw ships is non-homogeneous (i.e. there are very small velocities at the top of the propeller disc) and it is assumed that improving the homogeneity of the wake will improve the propulsion efficiency (essentially the open water propeller efficiency). Optimising the angle of the partial duct to the stern under load conditions (accelerating the flow in the top part of the disc and slowing it in the lower part) is said to improve the homogeneity of the wake.

Power savings of 5 to 10 % have been reported (Schneekluth 1986), despite the fact that this appendage creates an additional viscous drag. On the other hand, this partial duct can create a thrust and reduce eventual separation.

The extrapolation of powering performance of wake equalising ducts fitted on large single-screw ships are a typical example of the difficulty of scaling data from model tests of short appendages. The reason is that these devices are usually fitted to very large ships, which means a very low towing speed in the tank to comply with Froude number scaling of the full scale ship speed. For a given ship speed, Froude number scaling imposes a reduction of the model speed for a smaller scale model; whereas the maximum model size remains basically the same for each facility.

Reynolds effects can act in the opposite way:

- separation which occurs on the model may not occur, or may occur over a smaller region, at full scale, hence a partial duct effective at reducing this separation on the model may be less effective at full scale;

- the boundary layer is relatively far smaller at full scale than on the model, hence the size and inclination of the duct optimised for the model may not be optimal on the full scale;

- friction of the partial duct is exaggerated on the model due to the very low Reynolds number on such small appendages (values of Reynolds number on the appendages may be about 50,000 only).

Friesch and Johannsen (1994) did tests in a large cavitation tunnel (HYKAT) on one model of the size normally used in model towing tank tests at speeds from 3.4 to 5.5 m/s in order to investigate these Reynolds number effects. As published results over a large Reynolds number range are rare, some of the implications of this work are described; however the Committee cautions that published records from many more tests of this type are needed for different types of unconventional propulsor in order to reliably identify true trends. The equivalent Froude speed for this model was about 1.2 m/s and comparisons were made with towing tank tests done at this speed. Their results are as follows:

- The wake equalising duct did not reduce the drag of the ship (without the propeller).

- There was a reduction of power required for propulsion, but the behaviour of it was very Reynolds number dependant: from 2.5 % at Froude speed; to 9.6% at 3.5 m/s; to only 6% at 5.5 m/s.

- The propeller characteristics were affected very little by the presence of the ducts (slight increase of thrust and torque coefficients), but, alarmingly for the testing community, they became stable for speeds higher than 3.5 m/s only (at which speed the torque coefficient was reduced by 20% compared with the value obtained at Froude speed for a comparable thrust coefficient). This shows that open water tests can be used, but that Reynolds number effects on the characteristics of the propeller may be underestimated even for a conventional propeller.

- The propeller open water efficiency actually decreased slightly with the duct for this model, in contradiction to the principle of operation of wake-equalising ducts.

- The wake fraction and thrust deduction fraction changed slowly, but did not reach a constant value over the range of speed tested. In these tests the ducts reduced suction and increased wake so then the increase in propulsive efficiency came from an increase in hull efficiency.

In conclusion, partial ducts may result in energy saving at full scale, but this was not, and probably cannot be proven by model tests at Froude speed by use of the present testing procedure. These tests done at higher Reynolds number in a large cavitation tunnel showed an erratic behaviour of the energy saving with test speed.

No conclusion could be drawn from the Froude scale speed tests as they were affected by unrealistic levels of separation and other Reynolds number effects. Although not necessarily true for all partial ducts, the savings indicated in this set of model tests were apparently due to increased hull efficiency; the open-water efficiency seemed to decrease a bit and the wake was not more homogenous. Tests where separation is occurring on the hull without the propeller and where this is reduced/ suppressed with the propeller, can lead to an underestimation of suction (thrust deduction) and an overestimation of wake fraction (i.e. an overestimation of hull efficiency) when analysed using ITTC 1978 methods.

More generally, it is difficult to reliably predict full scale performance of this type of device from model tests at equivalent Froude speed in the towing tank and even tests at higher speeds in the cavitation tunnel can give uncertain trends. To address these issues, and to bridge the Reynolds number range, further reliable correlations are needed between: tests done in large cavitation tunnels at even higher speeds; towing tank tests done for some equivalent speeds at the lower end of the cavitation tunnel range; and full scale trials results.

Tests performed on a large tanker in the GTH at the Bassin d'Essais des Carènes at full speed (11 m/s) with partial ducts optimised in model tests at Froude speed showed a completely different effect of the duct on the wake in the two situations. This implies that optimisation of the inclination of the duct should be done from the results of high Reynolds number tests. Propulsive tests showed no energy saving for the high Reynolds number tests. Specifically:

- towing tank tests showed energy savings of 1-2% depending on the propeller tested and speed simulated;

- there was a difference in nature of the wake (without the partial duct) between tests at the Froude speed and those at higher Reynolds number, in particular the occurrence of a region of high vorticity in the region where the duct was to be fitted;

- by subtracting the wakes, the effect of the partial duct appeared to be completely different from the two tests at differing speeds, the smoothing effect being cancelled at higher Reynolds number;

- pressure fluctuations were improved with the duct fitted and reproduced in the higher Reynolds number tests similar signals to the full scale.

Reynolds number effects in tests performed at speeds of about 1 m/s may have more effect on the predictions than not using the Froude scaling identity, especially for ship testing at very small Froude numbers.

4.8 Z - Drives (Podded Propulsors)

An early investigation into the power prediction of vessels fitted with podded drives was done by Rains *et al.* (1981). They presented a semi-empirical approach to estimate the total drag of a podded drive and comparative power estimations. The investigation also included a model test programme with a DD-963 class destroyer model fitted with 3 different drive systems: a conventional twin drive; a non-azimuthing twin tractor; and a pusher type podded propulsor. The tractor type drive has its propeller ahead of the pod while this is reversed in the pusher type. The model tests involved resistance tests with and without the propulsion appendages, open water tests of the propeller in isolation, which was the same for all three propulsion systems, and self-propulsion tests. In the latter tests, only torque for each propeller was measured and the propulsive coefficients analysed appear to be based on the torque identity method. Although the performance comparison of the three propulsion systems was presented in terms of the appendage drag, effective power and delivered power on the full scale, no specific reference was made as to how they were extrapolated to full-scale.

Minsaas (1988) investigated a tractor type non-azimuthing Z-drive, with a steering flap at the tail of the drive strut, as a promising alternative to water jet propulsion for speeds up to 55 knots. Neglecting the cavitation and propeller induced drag, semi-empirical formulae were given to calculate the viscous drag of the unit due to the body, the strut within the slipstream and strut outside the slipstream. In these formulae emphasis was placed on the considerable scale effect on the resistance estimation as well as on its validity for only smooth surfaces. Later Halstensen and Leivdal (1990) described the development and full scale application of this type of podded drive, called "SpeedZ", to Fjellstrand catamarans "Sleipner" and "Draupner" and to Westamarin SES catamarans "Super Swede" and "Super Dane". The study reported on the open water tests done with sub cavitating and partially cavitating propellers fitted to the propulsion unit in the cavitation tunnel at MARINTEK, as well as towing tank tests with the naked hull. Although no details of the power performance prediction were discussed, the complexity of the resistance prediction at full scale due to the effect of the propeller was reported. From this study it appears that the full-scale estimations were made based on the naked hull resistance tests from the towing tank and the efficiency of the total propulsion unit obtained from the cavitation tunnel. The latter included a resistance correction for the full scale.

A recent study on the power performance aspects of azimuthing podded drives was presented by Kurimo *et al.* (1998) on the development of two 14 MW Azipod units for the cruise ship “Elation”. The study reported on an extensive model test programme done in a wind tunnel, towing tank and cavitation tunnel at three institutions. The experiments done in the VTT towing tank involved open water tests with a stock propeller, hull resistance tests and self-propulsion tests. The purpose of the open water tests was threefold: to determine the interaction coefficients between the propeller and the unit; to compare the performance of two pusher and one tractor type unit; and to provide a basis for the analyses of the propulsive coefficients. The open water tests were done using a special dynamometer that could separately measure the thrust of the propeller and resistance of the Azipod housing. Based on these tests and separate wind tunnel experiments, which involved resistance measurements of various components of the propeller housing, a tractor type Azipod was optimised. The resistance tests were performed by using an earlier hull model which had appendages for conventional twin screw drive. However these appendages were removed and the optimised Azipod housing was fitted. These tests were performed with and without the housing such that the resistance of the housing could be obtained. In the self-propulsion tests, the total thrust of the each Azipod unit was measured. It was not possible to measure the propeller thrust and the resistance of the pod separately due to the small scale of the model. The measured delivered power indicated a saving of 3-4% over the same hull driven by conventional twin screws.

The final group of tests, which were performed at MARIN, involved open water tests, hull resistance tests with the same hull and self-propulsion tests. However, the propeller used was a new design developed by the Krylov Ship Research Institute which was customised to the Azipod and hull. Two sets of open water tests were done with this propeller, with and without the Azipod housing. The calculated net efficiency of the Azipod in the open water tests, which was subject to low Reynolds number effects, was found to be slightly lower than that of the propeller alone at the corresponding advance coefficient. The hull resistance tests were done without the Azipod units. Following the self-propulsion tests a value of 5-7% of power saving was predicted on the full-scale when compared with the original vessel fitted with a conventional propulsion system. Although no details were given, it was indicated that the prediction was carried out according to MARIN’s “Standard” method for thrusters.

In this study, the use of a special dynamometer for the open water tests highlighted the importance of the complex interaction between the Azipod housing and the propeller for a tractor type unit when the propeller thrust and the resistance of the pod housing were measured separately. Therefore it was argued that the adopted scale effect correction methods used in the extrapolation of the thruster propulsion test results often concentrate on the scaling of the measured “resistance” of the housing. This may be acceptable for the pusher type Azipod, but not for the tractor type due to the complex interaction between the propeller and pod housing which results in an increased local pressure field behind the working propeller. Therefore, emphasis was placed on the gap between the Azipod housing and the propeller since low gap sizes would increase the resistance of the housing and the measured thrust of the propeller, if they were separately measured, although the net thrust of the azipod unit would be the same. Based on this argument, it was claimed that the difference between the total thrust of the unit and the thrust of the propeller in the tractor type of unit should not be related to Reynolds number dependent viscous effects alone. Also, more detailed methods for the extrapolation of test results from tractor type devices are recommended to be developed.

Mewis (1998) has discussed recent developments in large podded drives with a specific emphasis on the range of their efficiency improvements and difficulties in power performance prediction. This relates mainly to the tractor types, a factor which was also pointed out by Kurimo (1998). A part of the difficulties has been associated with the different configurations used in open water tests and the treatment of the thrust force measured during these tests. In order to demonstrate these difficulties, open water tests of a tractor type Azipod drive have been carried out in three different configurations as reported by the Propulsion Committee of this Conference.

Kurimo (1998) presented the results of sea trials with the cruise ship “Elation” which were carried out in the Gulf of Finland in December 1997. The trial results involved the presentation of speed measurements, cavitation observations, pressure pulse measurements and manoeuvring tests. The trials indicated that at the full power of 2 x 14MW, the vessel achieved approximately 0.55 knots greater speed than the mean value of the corresponding speeds of the six previous sister ships, which have conventional twin-screw diesel electric drives. Despite this, the scatter between the predictions from the different model basins, which used different extrapolation methods, was large and designers were cautioned to treat test results with great care to avoid optimistic expectations.

5 DESCRIPTION OF EXTRAPOLATION METHODS AND TEST PROCEDURES

5.1 Momentum Methods

The propulsion of a ship or marine vehicle represents one of the earliest applications of inviscid blade design theory and, as a result, introduces the problem of scaling. An extrapolation method developed by the ITTC community has been widely used for commercial ships having an open screw propeller to predict full-scale powering performance. The ITTC 1978 methodology uses resistance tests of the ship model, open-water tests of the model open propeller, and model-scale self-propulsion tests, to determine interaction coefficients, which are then scaled to predict full-scale performance. However, this methodology and the scaling of its interaction coefficients are based on particular commercial ship databases. The direct application of this methodology to complex propulsors such as ducted propellers or pump jets, water jets and various novel forms of open screw propellers continues to be an unresolved issue.

Any propulsor configuration and its performance are highly dependent upon the hydrodynamic characteristics of the ship hull/marine vehicle. Energy caused distortions are present in the ingested flow due to the skin friction drag of the ship hull/marine vehicle and upstream appendages. It is in this environment that wake-adapted complex propulsors excel. Complex propulsors provide more design options with which to meet the many additional performance requirements in these complex wakes. These new requirements include, but are not limited to: 1. minimization of unsteady forces, 2. reduction in radiated noise, 3. improved efficiency for heavily loaded propellers, 4. reduction of cavitation and elimination of specific cavitation types, 5. improved off-design performance, and 6. enhanced ship-maneuvering characteristics.

The principles employed and the problems encountered in designing either an open screw propeller or a complex propulsor are similar. However, the design methods employed are different for the two cases. First of all, the design of a wake-adapted complex propulsor having a duct and/or multi-blade rows treats the propulsor as a unit. Therefore, its model-scale testing procedure and scaling methodology will be different than that developed for an open screw propeller. One present-day method of designing complex propulsors includes a combination of a vortex lifting-surface method with a computational fluid dynamics based through-flow analysis method. These two methods allow for hull interactions, viscous effects and blade-to-blade row effects to be included as well as three-dimensional effects such as streamline curvature, radial pressure gradients, and secondary flow (Kerwin *et al.* (1994) and Schott (1996)).

Interest in multi-blade row propulsors has developed because of increasing efficiency demands for high-speed surface ships and submerged vehicles. The attractiveness of ducted propulsors for these applications is due to the ability to design a propulsor having a lower blade relative velocity and a higher efficiency than can be achieved by an open propeller in these applications. The similarities which exist between ducted propulsors and axial-flow compressors and liquid pumps have lead to the application of momentum analysis methods. Nowhere are different testing methodologies more evident between an open propeller and a complex multi-blade row marine propulsor than in reviewing a momentum based design methodology.

Early efforts in developing momentum based design methodology are given by Wislicenus (1960 and 1968), Henderson *et al.* (1967), and Bruce *et al.* (1974). This method determines the type, size, and design of any multiblade row propulsor from an optimisation of the mass flow through the propulsor to achieve design goals. An optimised blade-row spanwise-circulation distribution is obtained from minimising the energy losses in the propulsor and in the discharge jet. A detailed analysis of the flow field at various stations through the propulsor follows by solving the momentum, continuity and energy equations. The upstream boundary conditions are the mass flow rate, the momentum and the kinetic energy of the ingested flow.

A relationship to determine required rotor thrust is determined by integrating the momentum in the stream wise direction between a station located far upstream of the propulsor and a station located after the body and can be expressed as $T = \dot{m}\,(\overline{\Delta V})$ where T is the rotor thrust, $\dot{m}$ is the mass flow rate and $\overline{\Delta V}$ is the integrated change in stream wise momentum. An important assumption in this relationship is that free-stream static pressure exists at each station.

An incoming velocity profile near the plane of the propulsor and the predicted hull drag coefficient are necessary to start the design process. This has been traditionally obtained from model scale tests with corrections for variations in Reynolds number. In some cases, analytical predictions are also being used due to more reliable computational fluid dynamic procedures and their validation to specific geometries (Larsson *et al.* (1998), Arabshahi *et al.* (1998), Zierke *et al.* (1997) and Stern *et al.* (1996).

It is also necessary to determine the total drag of the hull, which includes increments due to appendages, etc. The flow field solution requires the calculation of the frictional drag on all the various propulsor components. It must be remembered that the pressure drag of the afterbody, which is initially included only as a bare-body drag coefficient, is substantially modified by the addition of the propulsor. However, this effect is initially estimated from the modified pressure distribution with the propulsor/hull combination.

It must be emphasised that this momentum-based design procedure is applied where there is incoming vorticity in the flow. The approach uses an inviscid calculation of the flow field and is coupled with an energy analysis through the propulsor. Therefore, all the energy losses through the propulsor are calculated and the efficiency of the rotor becomes the hydraulic efficiency, which is the ratio of the energy placed in the fluid to the shaft energy. Solutions to the flow field are obtained by using the Streamline Curvature Method (Treaster, 1969). Today, this has been extended to a RANS solution using computational fluid dynamics modeling.

This design procedure for complex propulsors does not require open water tests for extrapolation methodology to predict full-scale speed and rpm. In fact, testing of these wake-adapted complex propulsors in uniform flow significantly reduces propulsor efficiency and introduces corrections that are not well defined. Therefore, only model-scale propulsion test data is required for extrapolation to full scale. This model test is conducted over a range of over/under loading conditions. This is usually achieved by varying the rotor rpm at constant ship velocity. Resistance test data or calculations are only necessary for the design.

At these model scale conditions, the efficiency can be defined as

$$\eta_D = \frac{(R_{T_m} - F_D)V_m}{2\pi\, q_m\, n_m} = \frac{\text{effective power}}{\text{delivered power}}$$

and

$$F_D = \{(1+k)(C_{fm} - C_{fs}) - C_A\}\frac{1}{2}\rho_m\, S_m\, V_m^{\ 2}$$

From the model scale K_T and J_V curve, corrections are made to account for Reynolds number differences to full-scale first by skin friction corrections to the propulsor components ($\Delta K_T, \Delta K_Q$) and secondly, by corrections to the wake (Δn). It is very important to note that these corrections cannot account for separating flows.

In summary, the model scale propulsion test provides a K_T versus J_V curve. This curve must be corrected for Reynolds number effects both on the propulsor and on the hull which produces a wake scale effect. The weak point of this or any other extrapolation methodology remains the determination/prediction of the full scale ship drag which must include appendage drag and propulsor/hull interactions. Current attempts to do this utilise CFD procedures, large water tunnel tests, and an extended analysis of propulsion data.

5.2 Other General Methods

Other extrapolation methods are being developed using self-propulsion testing, the philosophy of which could be considered for unconventional propulsors. For these methods an open water test is not necessary, only a load varying self-propulsion test is required.

A modern MARIN method based on a form factor $(1+k)$ concept has been developed for some ship configurations for the extrapolation of propulsion test data to full scale. The form factor is the ratio of viscous resistance to the flat plate drag based on the ITTC 1957 formula. In general, this factor can be determined for each hull form using low rpm self-propulsion measurement data. The scale effect on resistance (F_D) is determined from the standard equation including an incremental resistance coefficient (C_A). The measured relationship between the thrust coefficient (K_T) and the apparent advance ratio (J_V) from the propulsion tests is corrected for both wake and propeller blade friction scale effects to predict power and rpm. This is effectively an ITTC 1978 method with the wake fraction found from a statistical method based on previous test results rather than directly through open water tests of the actual model components.

An extensive analysis of typical combatant/auxiliary Naval ship data has been conducted by the Naval Surface Warfare Center, Carderock Division, to validate model test extrapolation procedures (Karafiath, 1997). The developed prediction methodology uses a correlation allowance (C_A) to account for differences between ship and model roughness and for other variables that influence the powering prediction. Thus, the propulsion test is conducted at an overload condition with an added tow force (F_D) to overcome the effects of an additional frictional resistance and achieve equivalent thrust loading. This is accomplished by varying the propeller rpm at a constant velocity. One basic assumption of this method is that the efficiency of the model propeller is the same as that of the ship propeller; this is appropriate for small wake fractions. This is again an ITTC 1978 type method, but dispenses with the form factor, open water tests and other corrections.

Both of the above methods have limited application for unconventional propulsors since the previous test data and correlation coefficients, needed for the methods to work, do not exist.

5.3 Performance Prediction Methods For Ships With Pre-Swirl Stators

Van, Kim and Lee of KRISO (1993) proposed two alternative procedures for performance prediction for ships fitted with pre-swirl stators. The authors explicitly state that the two procedures, named A and B respectively, basically follow the ITTC 1978 method. In answer to a request for clarification Van pointed out that at their towing tank the form factor method is not used. Accordingly these procedures are not true variations of the ITTC 1978 correlation methodology as they follow the 2-D ITTC 1957 approach. The two procedures are discussed separately.

Method A. In this method, the propeller and the stator are considered as a propulsion system and are tested together. This assumption implies that in both open water and self-propulsion tests, the thrusts of propeller and stator are measured simultaneously and their sum is used as the thrust of the propulsion system. The hull resistance is scaled according to the ITTC 1957 method (i.e. a form factor is not used).

In his correspondence with the Committee, Van stated that the open water characteristics of the propeller are scaled following ITTC 1978 procedure while no scale effect correction is applied to the thrust/drag produced by the pre-swirl stator.

Van *et al.* (1993) state that this method provides a better full scale prediction than the ITTC 1978 procedure. This statement is supported by the close agreement between model and full scale predictions presented, that is not found by using the ITTC 1978 procedure directly.

It is well known that the ITTC 1978 correlation procedure fails to correctly scale the performance of unconventional propulsion systems, and this is due to two main causes. The first one is that whatever device is applied to the hull, it generally has a longitudinal dimension which is much less than that of the hull itself. Given the usual testing speeds, this produces a Reynolds number corresponding to a laminar flow on the device. The second cause is that the model hull has a boundary layer that differs from the full scale one both in thickness and in velocity distribution. Therefore, the interaction between the hull and the special device can seldom be correctly reproduced at model scale if it strongly modifies the flow around the hull.

In this particular case of the pre-swirl stator, we can assume that the flow around the stern is not overly affected. This can be argued following the principle of operation of the pre-swirl stator that is designed to produce a rotational speed component opposite to that induced by the propeller. For this reason, testing the stator together with the open propeller insures a higher Reynolds number on the stator itself due to the propeller-induced velocities. Probably this is the main reason of the better agreement of Method A performance prediction with the model results.

Method B. This procedure does not require the joint test of the stator and the propeller because the stator is tested with and considered part of the hull. On the other hand, it requires that a double set of resistance and selfpropulsions test are done with and without the stator.

The scaling process is again the two dimensional approach of the ITTC 1957 method with an exception made for the determination of the full-scale wake, which is performed by means of the following formula that closely resembles that suggested by the ITTC 1978 correlation procedure:

$$w_S = \left(t_{MO} + 0.04\right) + \left(w_{MO} - t_{MO} - 0.04\right)\frac{C_{FS} + C_A}{C_{FM}} + \left(w_{MS} - w_{MO}\right)$$

where:
w_S = ship wake
w_{MO} = model wake without stator
w_{MS} = model wake with stator
t_{MO} = model thrust deduction without stator.

while the standard ITTC 1978 ship wake is:

$$w_S = \left(t + 0.04\right) + \left(w_M - t - 0.04\right)\frac{\left(1+k\right)C_{FS} + \Delta C_F}{\left(1+k\right)C_{FM}}$$

The major difference compared with the ITTC 1978 formulation is the term ($w_{MS} - w_{MO}$). Since in the opinion of Van *et al.* (1993) the main effect of the stator is the increase of the angles of attack of the propeller blade sections, the stator action can be considered to be a mainly potential phenomenon. Thus, the difference in wakes with and without stator can be directly transferred to full scale.

If the flow on the hull is not overly affected by the presence of the stator, this assumption looks reasonable and this procedure is acceptable. The same would not be true in the case of other devices that accelerate or decelerate the flow on the hull, like ducts, partial ducts etc. Van *et al.* (1993) state also that Method B exhibits a good agreement with power savings obtained at model scale.

General remarks. Both of the proposed methods are basically 2-D procedures that can be regarded as variations of the ITTC 1957 correlation approach, but in principle these techniques could be used also with the ITTC 1978 3D correlation procedure.

From a theoretical point of view, the proposed methods are acceptable when applied to special propulsive devices that do not considerably alter the flow around the hull. In other words, these methods are suitable for scaling the performance of propulsive devices whose effect is mainly confined to altering the propeller inflow, without affecting the pressure field around the hull. Since these methods address the scaling problem only from the potential flow point of view, they are not suited to treat such devices that could produce a considerable variation of the pressure field acting on the hull. Actually, a hull pressure-modifying device could alter the characteristics of the flow around the model hull, e.g. the extent of laminar separation. In this case, the proposed methods will probably produce results as inaccurate as those of the unmodified ITTC 1978 procedure. The lack of sea trials results prevents a practical evaluation of the capability of these methods to correlate model test results with the actual performance of the ship. Thus, further effort is required to validate these methods.

From an experimental point of view, if the two proposed techniques are equally reliable, it appears that Method B would be preferable. Actually, the testing procedure related to Method B requires additional resistance and self-propulsion tests; this is its major drawback, but the procedure is straightforward and does not require any special test rig. On the contrary, an ad hoc test rig is necessary to simultaneously measure the thrust of the propeller and of the stator as required by Method A. The time consumption of this technique is comparable with that of standard tank practice, but the required special equipment would not be feasible or available in all of the towing tanks.

5.4 ITTC 1978 Modified Methods for Ducted Propellers.

Some ITTC 1978 modified methods have been proposed for extrapolation of powering performance of ducted propellers. Stierman (1984) presented three methods: where the nozzle is considered as an appendage; where the nozzle is treated as a part of the propulsion unit; or where the screw, nozzle and hull are treated as three interacting objects respectively.

Nozzle as an appendage of the hull. The nozzle is primarily considered as a flow regulator. The resistance test is done with the nozzle behind the hull and the open water test with the screw alone.

The thrust deduction fraction is defined as

$$t = \frac{T_P - R_{H+N}}{T_P}$$

where T_p is the propeller thrust and R_{H+N} is the resistance of the hull with nozzle. The wake fraction is found by propeller thrust identity

$$w = \frac{V - V_A}{V}$$

where V is the model speed and V_A is the advance velocity of the screw. The total resistance of the hull with nozzle is extrapolated by subtracting the estimated model nozzle resistance from the measured total resistance, scaling the resistance of the naked hull according to the ITTC 1978 method, and adding again the estimated full size nozzle resistance. It is assumed that the formula to estimate the nozzle resistance does not introduce too large errors because the nozzle resistance only amounts to a few percent of the total resistance. Note that this is in contrast to the tests described in section 4.6.

The scaling of the propeller characteristics is performed in exactly the same way as outlined in the ITTC 1978 method. The interaction coefficients are extrapolated either according to the ITTC 1978 or by the method of Stierman (1984). In the case of the open water screw test, the equivalent open water velocity in which the screw works in the nozzle must be known. The velocity change is expressed by a ΔJ correction

$$\Delta J = \frac{U_N}{nD} = J\frac{1-\tau}{2\tau}(1 + \sqrt{1 + \tau C_T})$$

where τ is the thrust ratio T_P/T_T. The purpose of the ΔJ correction is to find the correct subdivision of the wake fraction into a potential and a viscous part.

In this method, the influence of the nozzle on the screw is barely taken into account. A nozzle limits the radial outflow of a screw and therefore, the K_T/K_Q ratio will not be measured correctly during the open water test.

Nozzle as a part of the propulsion unit. Here the open water test is done with the screw + nozzle system and the resistance test is done with the naked hull. The thrust deduction fraction is defined using the naked hull resistance, and the total screw + nozzle thrust

$$t = \frac{T_P + T_N - R_H}{T_P + T_N}$$

In this case, V_A in the defining formula of the wake fraction is the entrance velocity into the screw + nozzle system. The naked hull resistance is extrapolated according to the ITTC 1978 guidelines. The scaling of the screw thrust and torque from the open water test is performed with the same ΔK_{TP} and ΔK_Q corrections as proposed by the ITTC 1978. The nozzle thrust coefficient K_{TN} should also be scaled. The resistance difference ΔK_{TN} is roughly estimated using a flat plate friction line and a formula according to Hoerner. The interaction coefficients are extrapolated using the ITTC 1978 method or by considering the effective power of the screw + nozzle system in their normal position and in a position far behind the hull.

The objections against the method are twofold:

(1) the nozzle thrust must necessarily be measured during the self-propulsion test;
(2) the action of the screw + nozzle system is different in open and behind conditions.

In the behind condition the nozzle produces a larger thrust due to the contracting inflow at the stern. For instance, in an open water condition $\tau = T_P/T_T = 0.90$ may be found, while $\tau = 0.70$ in the behind condition. Such an effect can be seen also on a twin screw ship even with no contracting inflow, perhaps due to boundary layer or inclined flow influences. Due to this discrepancy, it is incorrect to determine the wake fraction by using the total thrust identity axiom.

Screw, nozzle and hull as three interacting objects. To assess the interaction between the screw and the nozzle, an open water test is carried out with the ducted screw. The resistance test is performed with the nozzle behind the hull to take into account the interaction between the nozzle and the hull. The thrust deduction and wake fraction are defined as in method 1, by making use of the hull + nozzle resistance, the screw thrust measured during the self-propulsion test, and the entrance velocity V_A into the screw disk. The open water velocity V_A in which the screw acts is the sum of the translation velocity of the screw + nozzle system and the nozzle induced velocity. The screw must be out of the nozzle. The K_{TP}/K_Q ratio remains constant, but the entrance velocity is changed. This means a shift of the J-axis of the open water diagram with a ΔJ-correction, standing for the dimensionless nozzle induced velocity. The nozzle induced velocity can be calculated by momentum methods, and vortex ring or sheet methods.

The resistance of the hull and the nozzle is separately extrapolated, as described in method 1. The propeller characteristics are scaled using the well-known ΔK_T and ΔK_Q corrections.

5.5 The Modified Full-Scale Performance Prediction Method (MARIC Method e.g. Zhao *et al.,* 1988).

This method has been used to predict the performance of ships fitted with energy saving devices such as the simplified compensate nozzle; COSTA propulsion bulb; thrust shaft brackets; fore-propeller hydrodynamic fin sector; COSTA propulsion bulb and rudder-appended thrust fins; composite energy-saving technique; etc.

The size of MARIC's towing tank is 70*5*2.5m. The length of the geosim ship models were 3.5 to 4.5m. The diameters of single and twin-propeller models D_m are bigger than 0.12m and 0.11m respectively. The following method is used for the energy saving devices described in section 4.5 and tested at MARIC. The method utilises resistance, self-propulsion and open water tests. The open water test is done with the propeller alone; the resistance and self-propulsion tests are done with and without the energy saving devices. This testing procedure is described in the following paragraphs.

Resistance test. The resistance test of the single-screw ship model is usually done without the appendages such as bilge keel etc. Measurements are made of the total model resistance R_m, towing speed V_m and the water temperature t_m at the same loaded condition as the propulsion test. To get a reliable value of $(1+k)$ more data are taken in the low speed area than elsewhere. For the power prediction of the full-scale ship with the above mentioned energy saving devices, the resistance test of the ship model is usually done with and without the above mentioned energy saving devices to compare their influence on the resistance performance.

Open-water test. An open water test is done for the propeller model used in the propulsion test. In this test the propeller submergence is bigger than its diameter. The advance speeds vary from 0 to the case of zero thrust while the rotational speed is kept constant during the test. The results of the open-water test are used in the analysis of the self-propulsion test results. The R_n of the propeller model in the open-water test should reach the critical value $3*10^5$ as far as possible. For the power prediction of the full-scale ship with the above mentioned energy saving devices, the open-water test is usually carried out without the energy saving device.

Self-propulsion test. When performing the self-propulsion test, the model is tested at a minimum of 5 different speeds. At each speed the propelling forces Z_m can be varied by changing the propeller revolution rate to enable the skin-friction correction F_D to vary. The thrust produced by the propeller should meet the following condition:

$$T_m(1-t_m)+Z_m = R_m$$

The ship self-propulsion point is given by the following condition:

$$Z_m = F_D = \frac{1}{2}\rho_m S_m V_m{}^2\{C_{FM}(1+k)-C_{FS}(1+k)-\Delta C_F\}$$

The self-propulsion points can be calculated when V_m, n_m, T_m, Q_m, Z_m have been recorded. For the power prediction of the full-scale ship with the above mentioned energy saving devices, the self-propulsion test of the ship model is usually carried out with and without the energy saving device in order to compare their influence on the propulsion performance.

The ship model speed V_m is first corrected for blockage (a procedure following the Emerson method is used – see the Resistance Committee report of the 19th ITTC, 1990). The testing procedure of the ship model with the energy saving device is the same as that without the energy saving device. The wetted surface area of the energy saving device is not taken into account in the transformation of the results to the full scale.

The basic transformation. The following formulae can be derived from the kinematical and dynamical similarities

$$V_S = \lambda^{1/2}V_m$$

and

$$n_S = n_m / \lambda^{1/2}$$

The following equations can be easily derived:

$$P_{ES} = R_{TS}V_S \text{ and } R_{TS} = 0.5C_{TS}\rho_S V_S{}^2 S_S$$
$$P_{Dm} = 2\pi Q_m n_m \lambda^{3.5}\rho_S/\rho_m$$

When the ITTC 1978 method is used

$$C_{TS} = (1+k)C_{FS} + C_W + \Delta C_F$$

and

$$C_W = C_{Tm} - C_{Fm}(1+k)$$
$$\Delta C_F = \left[105(K_S/L_{WL})^{1/3} - 0.64\right]10^{-3}$$
$$K_S = 150\times10^{-6}\,\text{m}$$
$$C_{Tm} = R_m / \left[0.5\rho_m V_m{}^2 S_m\right]$$

and the friction coefficient is the ITTC 1957 line.

When the Froude method is used

$$C_{TS} = C_{FS} + C_R + \Delta C_{F\text{II}}$$
$$C_R = C_{Tm} - C_{Fm}$$

where, $\Delta C_{F\text{II}}$ is the resistance correction coefficient. It can be adjusted to keep C_{TS} calculated from both methods nearly the same. Normally it can be taken from the following empirical formulae:

$$10^3\Delta C_{F\text{II}} = 0.75 - 0.00352L_{WL}$$

or

$$P_E = (R_{Tm} - F_D)V_m\lambda^{3.5}\rho_S/\rho_m$$

where, F_D is the resistance correction of the self-propulsion point. It can be estimated from the following formulae:

$$F_D = 0.5\rho_m V_m^{\ 2} S_m \left[(1+k)C_{Fm} - (1+k)C_{FS} - \Delta C_F\right]$$
$$F_{D\text{II}} = 0.5\rho_m V_m^{\ 2} S_m \left[C_{Fm} - C_{FS} - \Delta C_{F\text{II}}\right]$$

Determination of the propulsion coefficient. The coefficients K_{Tm} and K_{Qm} are obtained from the self-propulsion test as follows

$$K_{Tm} = T_m / \left(\rho_m n_m^{\ 2} D_m^{\ 4}\right)$$
$$K_{Qm} = Q_m / \left(\rho_m n_m^{\ 2} D_m^{\ 5}\right)$$

Based on the thrust identity method J_{0m}, K_{Q0m} and η_{0m} can be read from the propeller model characteristics whose scale effect correction has been made with K_{Tm} as the input data. Following this, the propulsion coefficients can therefore be calculated as follows

$$\eta_{Rm} = K_{Q0m} / K_{Qm}$$
$$1 - w_m = J_{0m} n_m D_m / V_m$$
$$1 - t_m = (R_m - F_D) / T_m$$
$$\eta_{Hm} = (1 - t_m)/(1 - w_m)$$
$$\eta_{Dm}' = \eta_{0m}\eta_{Hm}\eta_{Rm}$$

where η_{Dm}' should be coincident with η_{Dm} (in the above equation $\eta_{Dm} = P_E / P_{Dm}$).

Basic transformation for twin-screw ships. With T_m and Q_m values for both propeller models as the input data, the power and the thrust deduction fraction can be calculated. For determining w_m, η_{0m}, and η_{Rm} the respective average values for both propeller models should be used.

Basic transformation for ships with asymmetrical stern form or twin-skeg form. For ships with a big tangential component in the propeller inflow, such as those with asymmetrical stern and twin-skeg forms, the ship's pre-rotating efficiency η_n will lead to a propulsion efficiency in the form

$$\eta_n = n_0 / n$$

where, n is the revolution rate of the propeller behind the asymmetrical stern and n_0 is the corresponding nominal revolution rate. This can be obtained by taking the average value between the revolution rates of right-hand and left-hand turning propeller models in the self-propulsion tests. Or it can be the revolution rate of this propeller model in the open-water test with V_A as its axial inflow speed when its thrust reaches the value in the following equation

$$\eta_D = \frac{RV_S}{2\pi nQ} = \frac{TV_A}{2\pi n_0 Q_0}\frac{n_0}{n}\frac{Q_0}{Q}\frac{(1-t)}{(1-w)} = \eta_0\eta_n\eta_R\eta_H$$

The prediction of the full-scale performance for single screw ships. The scale effect of the propulsion factors recorded in the self-propulsion test revised at MARIC by the following formulae

$$w_S = t_m + (w_m - t_m)\frac{\left[C_{FS}(1+k) + \Delta C_F\right]}{\left[C_{Fm}(1+k)\right]}$$
$$t_S = t_m - 0.08834 + 0.01262 L_{WLm}$$
$$\eta_{RS} = \eta_{Rm} + 0.08645 - 0.01236 L_{WLm}$$

J_{0s}, K_{Q0s} and η_{0S} can be read from the load coefficient of the full-scale propeller K_{TS}/J_S^2 chart with the K_{TS}/J_S^2 value as the input.

$$K_{TS} / J_S^{\ 2} = \frac{SC_{TS}}{2D^2(1-t_S)(1-w_S)^2}$$

The following can then be calculated

$$n_s = \frac{(1-w_S)V_S}{J_{0S} D}$$
$$P_{DS} = 2\pi\rho_S D^5 n_S^{\ 3} K_{Q0S} / \eta_{RS}$$
$$\eta_{HS} = (1-t_S)/(1-w_S)$$
$$\eta_{DS} = P_E / P_{DS} \quad \text{or}$$

$$\eta_{DS} = \eta_{0S}\eta_{RS}\eta_{HS}$$

The additional increment of the bilge keel and the air resistance of the superstructure over the water level can be estimated as 4-6% of P_{DS}. The rate n_S is proportional to $(1.04\text{-}1.06)^{1/3}$. The values of C_P and C_N obtained from this method are close to 1, so P_D, n_s and V_S can be used directly to predict the full-scale performance.

For the prediction of the full-scale performance for twin-screw ships with shaft brackets

$$w_S = w_m$$
$$t_s = t_m$$
$$\eta_{RS} = \eta_{Rm}$$

For twin-skeg ships w_m should be corrected based on experience. The full-scale performance prediction method and its procedure are the same as those for single-screw ships.

5.6 Propeller Boss Cap Fins (PBCF)

Although no particular extrapolation method specific to PBCF has been reported in the open literature, it appears that the PBCF patent holding companies have a well established procedure to quantify the efficiency gain by the use of PBCF. This procedure is based upon the so-called "reverse **POT**" (**P**ropeller **O**pen **T**ests) set-up and experience gained over a considerable number of full-scale measurements. Furthermore, private communication by this committee with the West Japan Fluid Engineering Laboratory Co. Ltd has confirmed that the extrapolation for ship powering performance is based upon the ITTC 1978 procedure, Nishimoto (1998). These are briefly reviewed in the following.

In the reverse POT configuration the propeller is placed downstream of the open water propeller boat in order to provide a free flow field for the development of the hub vortex. Since the presence of a rudder would weaken the swirl in the propeller slipstream, it is essential to place the rudder behind the propeller in these tests. By using this configuration, the measurements of thrust and torque are taken for the propeller with and without the PBCF. Based on the measured values of the torque, it is possible to estimate the delivered power and hence the ratio of the delivered power with and without the PBCF. The measurement of this ratio will be the gain (improvement rate) in propeller efficiency ($\Delta\eta$pm) due to the PBCF in the model scale.

In the POT set up one should bear in mind that the presence of the boat housing and the rudder behind the propeller will affect the propeller advance velocity. Therefore the necessary correction should be made in the advance coefficient in terms of the wake fraction caused by the boat housing and rudder, and comparison of the efficiencies should be made at the corrected advance coefficient for the same thrust.

By considering the scale effects mentioned earlier in Section 4.1, the propeller efficiency gain in the model scale ($\Delta\eta$pm) is related to the improvement rate of the propulsion efficiency on the full-scale ($\Delta\eta$s) as shown, for example, by Ouchi & Tamashima (1989), in figure 5. This guidance implies that the power saving or efficiency gain expected at the full-scale will be 2 to 3 times greater than the model scale predictions.

In the above outlined procedure the propeller and PBCF are considered as a unit propulsor and the effect of the PBCF is included in the propeller open water characteristics. The prediction of the efficiency gain due to PBCF does not require self-propulsion tests, but relies heavily on the accumulated knowledge and experience gained with PBCF on the full-scale.

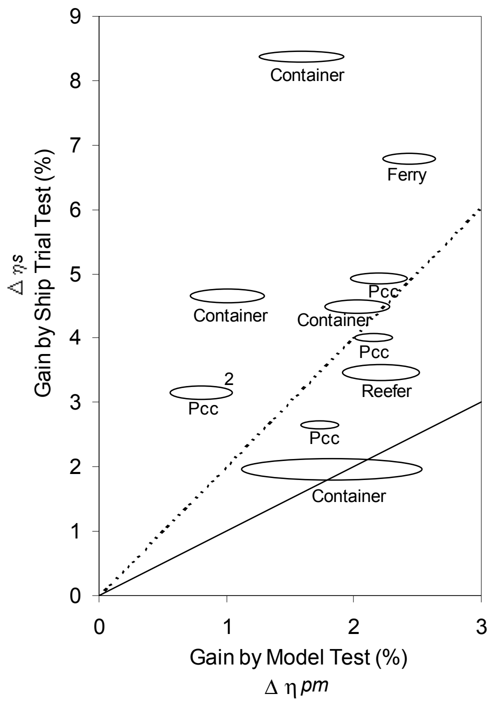

Figure 5. Correlation of Efficiency Gains by PBCF between Model and Actual Ship, Ouchi & Tamashima (1989)

An alternative to the above procedure, which makes use of self-propulsion tests, has been used by the West Japan Fluid Engineering Laboratory Co Ltd. and is also reported implicitly by Ouchi (1988). In this procedure the PBCF is considered to be an appendage and then the usual open water characteristics of the propeller are predicted from standard series data. The effect of the PBCF is included in the self-propulsion factors, which are obtained by means of the thrust identity method. The performance of the ship with PBCF is predicted based on the standard ITTC 1978 procedure.

5.7 ITTC 1978 Modified Method for Tip Fins

The only extrapolation method reported in detail in the literature for end plate and similar propellers is that reported by Andersen (1996). Since the tip fin propeller is only a slight modification of a conventional propeller the ITTC 1978 procedure has been followed with only a minor modification. The standard ITTC 1978 correction to the propeller characteristics due to difference in blade drag at model and full scale is developed for conventional propellers. Moreover, it relies on data for one representative profile only: at 0.75R. Hence due to differences in geometry, velocity and load distributions this method of scaling is not believed to be accurate enough for tip fin propellers. Instead the blade was subdivided into streamwise strips and the sectional drag estimated using the theoretically calculated velocities and a simple, flat-plate estimate of the frictional resistance which was dependent on mainly the local Reynolds number. To secure a fair comparison this procedure was applied to both the tip fin and the conventional comparator propellers. By this procedure the corrections turned out to be bigger for the tip fin propeller, i.e. it is more sensitive to scale effects. Unfortunately, no full scale tests have been done, so no confirmation of this scaling procedure can be made.

5.8 Extrapolation From an Extended Analysis of Self Propulsion Test Data

Under this category are grouped some methods that were developed with the aim of improving the powering prediction process by avoiding and replacing some of the assumptions inherent in their use. The common theme behind these methods is that the use of the towing resistance is considered meaningless or misleading when attempting to predict full scale power.

About ten years ago, some researchers (Holtrop 1990), started to make full scale powering predictions using self propulsion and open water tests as the sole source of experimental data. This was the first step in the direction of dispensing with towing resistance data, but this method was still consistent with traditional propulsion analysis since the model resistance had to be reconstructed from model self propulsion test results.

Briefly described here are two methods which fall into this category, but which have not yet been developed to a stage suitable for general application. These methods, however, have promise in the development of useful tools for the full scale powering prediction for vessels equipped with unconventional propulsors.

Iannone (1997a, 1997b), starting from the ITTC 1978 methodology framework, introduced a new propulsive analysis through the separation of the self propulsion flow into its viscous and potential components. To this end only open water and British self propulsion tests are required. In particular, to evaluate the viscous component of the self propulsion flow the innovative concept of *self propulsion form factor* K_{SP} is introduced.

In the first version of this methodology the self propulsion form factor was derived from model thrust measurements at low speed during self propulsion tests. This procedure, however, produced rather uncertain values of K_{SP}, due to the small values of measured thrust and the unavoidable presence of laminar flow or separation.

Iannone's investigations (1997b) revealed that a sort of thrust deduction phenomena occurs during self propulsion tests. Furthermore it was appreciated that the self propulsion form factor at service conditions depends not only on hull trim at speed, but also on speed itself (Iannone 1998). As a consequence the prediction method was refined (Iannone 1998) by determining the thrust deduction factor and the self propulsion form factor by tests at the service speed. Moreover a new data reduction method was proposed (Iannone 1998), leading to different relationships among propulsive characteristics measured during British self propulsion tests.

After some satisfactory applications to single and twin screw hulls, Iannone claims that the method, though still in a refinement phase, is promising for its suitability to be used as an extrapolation tool for full scale powering predictions for unconventional propulsors.

The second method described in this section, Schmiechen's (1991) "rational theory", is a method for ship powering analysis based on results from model self-propulsion load varying tests alone. Stand alone resistance and propulsor open water tests are avoided.

Two overload tests are done at the same steady speed, but different values of the overload. Care must be taken to ensure that the speed is steady to avoid significant acceleration/inertia forces in the longitudinal momentum equation. The following parameters are measured for the two tests: speed, shaft rotational speed, shaft thrust, torque and the towing force. These are designated

$$\begin{matrix} V_1 & N_1 & T_1 & Q_1 & F_1 \\ V_2 & N_2 & T_2 & Q_2 & F_2 \end{matrix}$$

The thrust, T, and torque Q_P, are assumed to vary as a quadratic function of the rotational speed N such that

$$T = T_O N^2 + T_H NV$$
$$Q_p = Q_{P0} N^2 + Q_{PH} NV$$

which is equivalent to them being linear functions of the ship advance coefficient J_H. The subscripted terms on the RHS of these equations are found from the experimentally measured values as follows

$$T_0 = \frac{T_1 N_2 V_2 - T_2 N_1 V_1}{N_1^2 N_2 V_2 - N_2^2 N_1 V_1}$$
$$T_H = \frac{N_1^2 T_2 - N_2^2 T_1}{N_1^2 N_2 V_2 - N_2^2 N_1 V_1}$$
$$Q_{P0} = \frac{Q_1 N_2 V_2 - Q_2 N_1 V_1}{N_1^2 N_2 V_2 - N_2^2 N_1 V_1}$$
$$Q_{PH} = \frac{N_1^2 Q_2 - N_2^2 Q_1}{N_1^2 N_2 V_2 - N_2^2 N_1 V_1}$$

From the steady state form of the longitudinal momentum equation

$$0 = T_E + F - R = T(1-t) + F - R$$

and by assuming that the thrust deduction fraction can be modelled as a linear function of the ship advance coefficient

$$t = t_H J_H$$

the value of t_H is found as

$$t_H = \frac{T_2 + F_2 - T_1 - F_1}{\left(\frac{T_2 V_2}{DN_2} - \frac{T_1 - V_1}{DN_1}\right)}$$

where D is the propeller diameter. The resistance is found from

$$R = T_1\left(1 - \frac{t_H V_1}{DN_1}\right) + F_1$$

and the advance coefficients of the two steady states are found as

$$J_{H1} = \frac{V_1}{DN_1}; J_{H2} = \frac{V_2}{DN_2}$$

The following coefficients are then found based on Schmiechen's definitions

$$K_{T0} = \frac{T_0}{\rho D^4}; \quad K_{TH} = \frac{T_H}{\rho D^3}$$
$$K_{QP0} = \frac{Q_{P0}}{\rho D^5}; \quad K_{QPH} = \frac{Q_{PH}}{\rho D^4}$$

Schmiechen then assumes that the power coefficient representing the power losses of the propeller, the difference between the shaft power supplied to the propeller and the jet power of the propeller in coefficient form

$$K_{PL} = K_{PP} - K_{PJ}$$

can be represented by a quadratic equation in the advance coefficient of the propeller, J_P.

$$K_{PL} = K_{PLPO} + K_{PLPI} J_p + K_{PLP2} \frac{J_p^{\,2}}{2}$$

The coefficients of this parabola are found in the following sequence from the coefficients already calculated:

$$K_{PPO} = 2\pi K_{QPO}$$

$$K_{PPH} = 2\pi K_{QPH}$$

$$K_{PLPO} = K_{PPO} - \left(\frac{2}{\pi}\right)^{1/2} K_{TO}^{\;3/2}$$

$$K_{PLP1} = K_{PPH} - \left(\frac{2}{\pi}\right)^{1/2} \frac{3}{2} K_{TO}^{\;1/2} K_{TH} - \frac{K_{T0}}{2}$$

To obtain K_{PLP2} use is made of the zero thrust condition when the equation

$$K_T = K_{T0} + K_{TH} J_{HT} = 0$$

yields in sequence

$$J_{HT} = \frac{K_{TO}}{K_{TH}}$$

$$K_{PLT} = K_{PPO} + K_{PPH} J_{HT}$$

and a cubic equation for the propeller advance coefficient in the zero thrust condition

$$J_{PT} = \frac{K_{PPH}}{K_{TH}} - \frac{2}{\pi}\frac{P_{PLP1}}{J_{PT}} + \frac{4}{\pi}\frac{(K_{PLT} - K_{PLPO})}{J_{PT}^{\;2}}$$

As J_{PT} occurs on both sides of this equation, a solution here is to assume a value for J_{PT} on the RHS and calculate a new value for J_{PT} on the LHS. This procedure can be iterated until the RHS and LHS values fall within a pre-set limit. From this the coefficient K_{PLP2} can be found from

$$K_{PLP2} = \frac{2}{J_{PT}^2}(K_{PLT} - K_{PLPO} - K_{PLP1}J_{PT})$$

Using the above basis, the propulsive performance of the vessel can be found over a range of ship advance coefficients, J_H, which can be assumed. In sequence the thrust and power coefficients are found as

$$K_T = K_{TO} + K_{TH}J_H$$
$$K_{PP} = K_{PPO} + K_{PPH}J_H$$

An iteration is then necessary to obtain the working value of the propeller advance coefficient at this ship advance coefficient and this is done using the loss parabola.

$$K_{PL} = K_{PLPO} + K_{PLP1}J_P + \frac{K_{PLP2}J_P{}^2}{2}$$
$$K_{PJ} = K_{PP} - K_{PL}$$
$$J_P = \frac{K_{PJ}}{K_T} - \frac{2}{\pi}\frac{K_T{}^2}{K_{PJ}}$$

In this procedure also, a value of J_p is assumed and a new value is calculated; the process is iterated until the desired accuracy is achieved.

From here the following propulsive coefficients are obtained

$$K_{QP} = \frac{K_{pp}}{2\pi}$$
$$K_{QL} = \frac{K_{PL}}{2\pi}$$

wake fraction

$$w = 1 - \frac{J_P}{J_H}$$

thrust loading coefficient

$$C_P = \frac{8}{\pi}\frac{K_T}{J_P{}^2}$$

various parameters in Schmiechen's " thrust deduction theorem"

$$\tau = (1 + C_T)^{1/2} - 1$$

thrust deduction fraction $t = t_H J_H$

$$\chi = \frac{\left[(1+\tau)t - \frac{\tau t^2}{2}\right]}{(1-t)}$$

energy wake

$$\frac{w_e}{w} = 1 - \frac{\chi(1-w)}{w}$$

and the various propulsive efficiencies:

propeller efficiency

$$\eta_{TP} = \frac{K_T J_P}{K_{PP}}$$

jet efficiency

$$\eta_{TJ} = \frac{2}{1 + (1 + C_T)^{1/2}}$$

pump efficiency

$$\eta_{JP} = \frac{\eta_{TP}}{\eta_{TJ}}$$

effective thrust coefficient

$$C_E = \frac{K_T(1-t)}{J_H{}^2}\left(= \frac{T_E}{\rho D^2 V^2}\right)$$

hull efficiency

$$\eta_{ET} = \frac{(1-t)}{(1-w)}$$

total propulsive efficiency

$$\eta_{EP} = \eta_{ET}\eta_{TP}$$

configuration efficiency

$$\eta_{EJ} = \frac{\eta_{EP}}{\eta_{JP}}\left(= \frac{P_E}{P_J} \right)$$

energy wake

$$w_e = w - \chi\,(1\text{-}w)$$

and normalized pressure level in the wake $C_p = (1-w)^2(\chi^2 + 2\chi)$.

Although Schmiechen has not done a whole set of evaluations for a series of ships, he has presented comparisons between model and full scale parameters for detailed tests done on the *Meteor*. For this vessel, the following parameters were found to be similar for the model and full scale: K_T, K_{QL}, η_{JP}, C_E, η_{EP} and η_{EJ}. Small variations between model and full scale occurred in: K_Q, w, η_{TJ}, η_{TP} , C_P, η_{ET}, C_E, η_{EJ} and t.

5.9 Z - Drives (Podded propulsors)

Allied with the recent upsurge in the number of applications of podded drives, there is growing concern with the differences in testing procedures and extrapolation methods being used for these propulsors, particularly for tractor type units. The extrapolation methods in use have not been published in the open literature. Also there is a lack of full-scale data for these systems due their short application history. The following outline approaches to the testing and extrapolation for podded propulsors have been suggested.

In the first approach the housing of the azimuthing unit and the propeller is considered as a whole and the propulsive coefficients are related to the interaction between the whole propulsion unit and the hull. In this case the system's net thrust must include the drag forces on the housing. The resistance test with the hull is performed without the propulsion unit fitted.

In the second approach, the azimuthing unit is considered as an open propeller in isolation and the propulsive coefficients are related to the interaction between the propeller and the hull appended with the housing. Therefore the propeller thrust and torque will not include the effect of the drag forces on the housing and the resistance test must be performed with the housing unit fitted, but without the propeller in place.

In both of the above approaches the problem arises as to how to scale the drag of the housing unit due to the low Reynolds number at model scale which results in relatively large drag forces. In the first approach, a scale effect correction can be applied to the model K_T values obtained from an open water test with the complete unit where the system's net thrust is measured. In the second approach, this correction can be applied to the appended hull resistance based on the resistance results. The final power prediction is expected to be the same for both cases. However, the total propulsive efficiency will be different due to the difference in the resistance for the naked and appended hull. This is actually a matter of definition of propulsive efficiency. In some cases a stock propulsor unit may be used instead of a geometrically similar model of the actual propulsor.

In a third approach, open water tests of the podded drive as a whole unit are also done. However, the thrust and torque measurements are taken at the shaft excluding the drag effects of the housing and are corrected to the full-scale. The resistance tests with the model hull are done without the podded drive while the resistance of the drive housing and the scale effects are estimated numerically.

In a fourth approach, in addition to resistance and self-propulsion tests, two open water tests: one with the propeller alone; the other with the propeller plus the housing together (whole unit), are used for the evaluation of the podded drive. In this approach, the podded drive is assumed to be an appendage and its drag is converted to the full-scale without any correction for scale effects. The thrust deduction fraction is calculated from the system's (net) thrust and the total resistance with the pods. The wake fraction is obtained based on a K_Q identity and no correction is made to the full scale. The open water test results for the whole podded drive are corrected for the difference in Reynold's number between the open water test and the self-propulsion test. Further corrections are also made to the open water test results for the full scale by using ΔK_T and ΔK_Q values based on open water test results with the propeller alone.

6 DISCUSSION OF EXTRAPOLATION METHODOLOGIES

In section 4, this report has given an overview of methods of extrapolation used in the past for different types of unconventional propulsor. Problems arising during testing and extrapolation have been highlighted in some cases. In section 5, these methods of extrapolation, together with other candidate extrapolation methodologies, have been discussed in further detail.

To date, many extrapolations for unconventional propulsors have used methods heavily based on the ITTC 1978 methodology. However, the modifications to the ITTC 1978 method proposed and used for many of these unconventional propulsors are different: i.e. there are as many or more modifications to the ITTC 1978 method as there are types of unconventional propulsor. This situation is less than ideal as there is always a question as to whether any power saving predicted is a fundamental characteristic of the device or a function of the testing, analysis and extrapolation method in use.

As a result of its work, the Committee feels that where there is weak interaction between the unconventional propulsor and the ship hull, then methods developed on the core of the ITTC 1978 approach can give levels of accuracy for extrapolation of full scale ship powering performance that are of the same magnitude as those obtained for conventional propulsors. Weak interaction occurs with devices such as tip fin and tip plated propellers, propeller boss cap fins, and other devices that are only a small modification to a conventional propeller. Effective modifications do need to be made to account for differences in blade frictional drag resulting from differences in Reynolds number between model and full scale and a method to do this for tip fin propellers is referred to in section 5.7. However, where there is strong interaction between the device and the hull, then methods based on the ITTC 1978 approach are not adequate. Strong interaction occurs with all types of ducted and partially ducted propellers, pre- and post-swirl devices, z-drives, etc. The reason for this is that these latter unconventional propulsors have large, complex and strongly modifying effects on the flow around the hull. In addition, the exact physical mechanisms by which some of these devices interact with the hull is not always clear. As a result, the testing and analysis leading to extrapolation does not lend itself to being broken down into the pieces designated by the ITTC 1978 procedure (i.e. separate resistance, open water propulsor, and self-propulsion tests and their analysis). Most of these interactions between propulsor and hull are strongly Reynolds number dependant.

Benchmark work is necessary to investigate these flow phenomena and to identify trends in powering performance as the Reynolds number is increased. Examples of some of the types of test that are necessary are described in sections 4.6 and 4.7 for models fitted with ducts and partial ducts. To do this work it is necessary to use the largest facilities available and to avoid strict adherence to Froude scale speeds. New test techniques need to be developed in some cases. A heavy reliance needs to be made on self-propulsion tests with geometrically scaled models of the unconventional propulsor and hull arrangements. Use can and should also be made of RANS-type CFD calculations to investigate trends in certain detailed flow behaviour as the Reynolds number of a particular set up is increased.

To scale or extrapolate powering performance, it is necessary to know the trends in performance as Reynolds number is increased and this can be found from these specially designed and perhaps expensive tests. Full scale trials results from ships fitted with unconventional propulsors are needed for the same cases that have been studied in depth at model scale. Once this base of knowledge is established, more routine work might be possible following a more simple test and extrapolation procedure.

In carrying out propulsion tests for unconventional propulsors, the Committee recommends the use of extensive load varying tests done at high values of Reynolds number. In working towards a guideline for extrapolation, the Committee recommends that the data from these tests be used in combination with an analysis using one or more of: a momentum analysis, as described in outline in section 5.1; the results from RANS-type CFD calculations; and/or an extended analysis of self-propulsion tests, some perhaps rather simplistic examples of which are described in section 5.8. The aim is to avoid assumptions made in the conventional ITTC 1978 type analysis which are known to be less than realistic. An example of one of these is the use of a propeller open water test (often done at one Reynolds number) with results from the propeller in the behind condition (often done at a different Reynolds number) to identify a wake fraction. A second example is the empirical methods used for the scaling of that wake fraction from model to full scale.

A true momentum analysis would identify streamtubes passing from ahead of to behind the propulsor unit. Detailed measurements of the flow velocities would be made at these locations and these would be used to identify the thrust of the device. (Note that this is analogous to the method of analysis proposed for waterjets by the Waterjets Group of the 21st ITTC.) The thrust deduction effect could be found from an integration of local pressures over the afterbody. CFD methods would be used to supplement velocity values from actual test results and to investigate trends at higher Reynolds numbers. In contrast, the reason for looking at an extended analysis of self-propulsion test data is with the plan that powering performance information can be extracted from these tests in a more macro manner (than by using the integrations referred to above), but by avoiding as far as possible the less realistic aspects of the assumptions present in conventional methods of extrapolation. The Committee recommends that work is continued on the development of new general methods for extrapolation of the powering performance of ships fitted with unconventional propulsors along these lines.

The published performance of unconventional propulsor systems is sometimes clouded for a number of reasons. Often the advantage of fitting an unconventional propulsor design is not actual energy saving, but the suppression of cavitation or the reduction of unsteady pressure pulses and vibration. There may be no efficiency gain in some situations. In addition, unconventional propulsors can be fitted to ships where the initial design of the conventional propeller is not optimum. This latter leads to an apparently large propulsive efficiency gain with the device, whereas a proportion of that gain is the result of more focussed design effort being placed on the propulsion design in general.

7 DRAFT CONCLUSIONS AND RECOMMENDATIONS

General technical conclusions

- The exact physical mechanism by which some unconventional propulsors interact with the hull is not clear.

- Extrapolation of full scale powering for ships fitted with unconventional propulsors may be developed from extensive load varying tests and in addition one or more of momentum analysis, CFD computations, or an extended analysis of self propulsion test data.

- Use of extrapolation methods for unconventional propulsors based on modifications to the ITTC 1978 method show similar levels of variation of the powering prediction found between methods as the level of the power saving expected with the device. In other words, the accuracy of these extrapolation methods is in most cases of the same order of magnitude as the level of power saving of the device under analysis.

- There is a shortage of accurate data from full scale trials supporting extrapolation predictions made for unconventional propulsors.

- Extrapolations cannot be reliably made of self-propulsion test data if flow separation which occurs on the unconventional propulsor and/or the ship hull is not scaled correctly.
- It is recognised that most methods of extrapolation currently in use are modifications to a greater or lesser degree of the ITTC 1978 method. It is recommended that the ITTC 1978 method is only used with caution as a guideline for extrapolation of model test results to full scale ship powering prediction for unconventional propulsors.

- Extrapolation methods for unconventional propulsors based on modifications of the ITTC 1978 method are expected to give powering predictions to the same level of accuracy as that achieved for conventional propulsors when applied to ships fitted with tip fin and tip plated propellers, propeller boss cap fins, surface piercing propellers and other devices which are modifications of a conventional propeller.

<u>Recommendations to the Conference</u>

- For powering prediction it is recommended that ship models fitted with unconventional propulsors, such as propellers with ducts, partial ducts, pre- and post- swirl devices, z-drives, etc., should be tested as a unit and not broken down into component tests of the hull, propulsor and rotor and stator components.
- Extrapolation methods of full scale powering for unconventional propulsors should be done using self-propulsion load varying tests of the geometrically similar ship model and geometrically similar propulsor.

- It is recommended that for self-propulsion tests with unconventional propulsors the effect of the rudder on the propulsion system be considered due to the influence of the downstream swirl on the rudder.

- To reduce/ eliminate the scaling of flow separation effects during self-propulsion tests these tests should be done at higher levels of Reynolds number than can be achieved by rigid adherence to Froude number scaling.

<u>Recommendations for future work</u>

- It is recommended that work is continued on the development of new general methods for the extrapolation of model test results for unconventional propulsors. The extrapolation methods should be validated against extensive model tests and full scale trials.

- Extensive tests for unconventional propulsors over a wide range of Reynolds numbers have shown that some usual assumptions of trends in performance are false. There is a need for detailed tests over a wide range of Reynolds numbers to be done on ship models fitted with each different type of unconventional propulsor. Where possible such test results should be compared with CFD computations and full scale trials.

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The Specialist Committee on Unconventional Propulsors

Committee Chair: Dr. Neil Bose, MUN
Session Chair: Dr. U. Grazioli, INSEAN

I Discussions

Contribution to the Discussion of the Report of the 22nd ITTC Specialist Committee on Unconventional Propulsors

by Pengfei Liu, Institute for Marine Dynamics

This report demonstrates an excellent job by the Committee on such a difficult and comprehensive subject in such a short time. It is a pity that this Specialist Committee will not continue. Continuation of the work by this Committee is important, because the methods of predicting, both numerically and experimentally, the performance capabilities of oscillating propulsors, tip fins on screw propellers and others are still not yet fully developed. There is a lot of remaining work to be done.

A bigger database covering more aspects of the performance of energy saving devices, and alternatives or modifications to the conventional propulsors would be valuable. Experiment data for calibration and validation of computer codes for unconventional propulsors is very scarce. For example, it is difficult to find experimental data on tip fin or end plate propellers. Such data need to include not only the propeller shaft thrust and torque coefficients but the blade pressure distribution as well. As stated in the report, the development of oscillating foil propulsion systems is in its infancy. Little experiment data on pressure fluctuation on the oscillating foil surface has been published. Such data would be invaluable for calibration for CFD work. There is also a need for some research and development work on control mechanisms for full scale oscillating propulsors, but this is outside of the scope of the ITTC.

Building on the success of the workshop held by the Propulsion Committee to compare RANS and panel methods to predict propeller performance I suggest a workshop on CFD modeling of propellers operating in ducts. Many issues could be addressed in such a workshop. Both experimental and CFD predictions of unsteady pressure distribution on the inner surface of a duct are difficult to find in the literature, although some authors have presented the steady pressure distribution inside the duct without experimental verification. Benchmark data including measurements of pressure fluctuation on the duct's surface would be very useful for calibration and verification of panel methods and RANS codes. A workshop would focus efforts in this field.

Oscillating Propulsors

by M.X. van Rijsbergen (MARIN)

In the discussion on extrapolation of oscillating propulsors, the report of the committee concentrates on propulsor-hull interaction. MARIN would however like to ask attention for scale effects on the oscillating propulsor itself.

MARIN has been thoroughly involved in the development of the first prototype Whale Tail Wheel (see e.g. van Manen et al.,1996). This is a cycloidal propulsor which can operate in a trochoidal blade motion. From open water tests at scale factor 2 and several rotation rates, it appeared that mainly the torque coefficient is considerably sensitive to the Reynolds number (see the figure below).

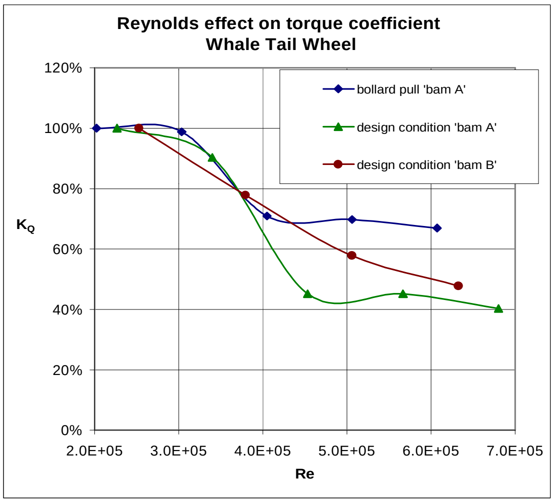
The Reynolds number is defined as:

$$Re = \frac{V_R c}{\nu}$$

where V_R is the average resultant velocity relative to the blade, c is the chord length and ν is the kinematic viscosity of water. The torque coefficient is defined as:

$$K_Q = \frac{Q}{\rho n^2 D^4 L}$$

where Q is the torque delivered by the two main shafts (corrected for friction), ρ is the density of water, n is the rotation rate, D is the diameter of the wheel and L is the span of the blades. The torque coefficient is normalised relative to the value at the lowest Reynolds number for each condition and blade angle motion (designated 'bam A' or 'bam B').



The figure above shows that the torque coefficient decreases with 30 to 60% for Reynolds numbers in the range from $2 \cdot 10^5$ to $7 \cdot 10^5$, dependent on the blade angle motion and the condition. The torque coefficient of the wheel is dependent on both the lift and drag coefficients of the blades.

The observed decrease in torque coefficient of the Whale Tail Wheel occurs in the same region of Reynolds numbers where a significant decrease in drag coefficient of streamlined foils has been reported (see e.g. Hoerner, 1965).

Trochoidal propellers can operate at the condition where at some point in the blade path, the entrance velocity to the blade equals zero or practically zero, causing extremely low Reynolds numbers. Special precautions during testing and for extrapolation of results may have to be taken. Little is known on this issue.

Due to the sometimes large angles of attack of the flow on the blade, in combination with low Reynolds numbers, leading edge separation may occur, which is also likely to be prone to scale effects.

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Discussion

by Hassan Ghassemi, Institute for Marine Dynamics

Of the many different unconventional propulsors considered by the Committee, podded propulsors are of increasing practical interest at the present time. Due to their better maneuverability, regular inflow to the propeller, reduction of cavitation and vibration, compactness, different types of podded propulsion systems (pulling, pushing and shuttle) are becoming popular in current construction using electrical propulsion in place of conventional propulsion systems.

Some recent computational research on a pulling type of podded propulsion system at the Institute for Marine Dynamics (IMD) * raised issues concerning the best approach to take in making reliable predictions. For example, the drag of the strut/pod may be affected by the downstream propeller wake and its modeling. Thus it is important to find a reasonable vortex wake model to calculate the drag of strut/pod in the propeller slipstream using a panel method. By carrying out experiments, Halstansen and Leivdal showed that strut/pod drag is compensated by recovery of rotational energy in the slipstream. What method does the Committee recommend for the strut/pod drag of the pulling type of podded propulsion in the vortex wake generated by the propeller?

The hydrodynamics of different types of podded propulsion systems (pushing, pulling and shuttle) may change the overall hydrodynamic performance of the vessel. Can the Committee comment on these differences and how they may influence the approach to making performance predictions?

- Ghassemi, H. and Allievi A. 1999: Fluid Analysis and Hydrodynamic Performance of Conventional and Podded Propulsion Systems, Oceanic Engineering International, Vol. 3 No. 2.

Discussion

by Jaakko V. Pylkkanen (VTT)

First of all I would like to express my appreciation to the Committee for its competent report.

(a) The fourth general technical conclusion is: “There is a shortage of accurate data from full scale trails supporting extrapolation predictions made for unconventional propulsors”. My question is: Was any such data available to the Committee; did the committee validate extrapolation procedures with full-scale data?

(b) The fourth recommendation to the conference is: “To reduce the scaling of flow separation effects, self-propulsion tests should be done at higher levels of Reynolds number than can be achieved by adherence to Froude number scaling.” My comment is: In this context turbulence stimulators should account for this. I would like to have recommendations for the use of turbulence stimulators included in the committee report.

Pod-propulsion, model measurement

by Friedrick Mewis, HSVA

First I would like to give my thanks to the committee for it’s really good work. In chapter 5.9 (page 240) are described very well the possible methods for using open water test results for podded drives.

Some factors are mentioned which affect the measured values of propeller thrust, propeller torque and unit thrust (system thrust) and additionally four different approaches to extrapolation methods are described.

Unfortunately, the estimation of the propeller thrust in the case of pulling pods is very difficult and the correct measurement is impossible. The reason for that fact is the gap between the propeller hub and the pod housing. The pressure in this gap influences to a substantial level the measured propeller thrust. This substantial inner force in the gap is a result of the high pressure behind the working propeller, the pressure change due to the stagnation point of the pod shaft coupled with the large gap area. The width of the gap, which can vary in different model tests, and in different towing tanks, influences the measured thrust too.

The error can be up to 10% dependent on the propeller loading, the pod-geometry, the width of the gap and other reasons. The measured unit thrust is not affected by this problem because the force in the gap is an inner force of this unit. My proposal is to use the measured unit-thrust as the basis for the estimation of the so called "small figures" like thrust deduction fraction, effective wake fraction, propulsor efficiency… in pod propulsion only.

II Committee Replies

Reply to P. Liu

First of all, we would like to thank Dr. Liu for his very supportive comments on the Committee's efforts and his discussion. The most difficult problem for the Committee was to identify reliable model/full-scale data for any unconventional propulsor. In almost every case these databases are incomplete for the calibration and validation of computer codes; however, it is very important to note that only model-scale data exists for most unconventional propulsors. We could not identify any experimental database that includes details of the local steady/unsteady pressure and velocity fields associated with the unconventional propulsor. In most cases the unconventional propulsor was designed to improve powering thus only nominal powering measurements were made. At this time the Committee could not identify a specific reference that presents unsteady duct pressure measurement data.

We agree that a workshop on CFD modeling of propellers operating in ducts is needed. These types of unconventional propulsor are becoming more common and experiments are being conducted in water tunnels to measure the flowfield.

Reply to M.X. van Rijsbergen

Having been involved to some extent [Riijarvi et al. 1994] with testing and computational work on trochoidal propellers, we agree that Reynolds number scale effects are very important in the extrapolation of open water and self propulsion data on these propellers. As the discussor explains, large angles of attack do occur on the blades of a trochoidal propeller, both during each revolution and at low advance ratios of operation. Work has shown, in fact, a sort of critical advance ratio below which stall effectively reduces both the thrust and torque coefficients of the propeller.

A multiple stream tube theory [Bose, 1987] using experimental lift/drag coefficients at different Reynolds numbers for the foil sections in a quasi steady manner can be used to predict the performance of a trochoidal propeller in a way that models this behaviour and this critical advance coefficient, at least in a qualitative way. A more accurate method would require experimental characteristics for the foil sections during dynamic stall from oscillating foil tests for the appropriate conditions and over a range of Reynolds numbers.

References:

Riijarvi, T., Li, J., Veitch, B.J., Bose, N., 1994, Experimental performance and comparison of performance prediction methods for a trochoidal propeller model, International Shipbuilding Progress, Vol. 41, No. 426, pp. 113-136.

Bose, N., 1987, Rotary foil propellers, Papers of the Ship Research Institute, Tokyo, Japan, Vol. 24, No. 5, pp. 45-67.

Reply to H. Ghassemi, IMD

The committee would like to thank Dr. Ghassemi for his questions.

Although the methods for predicting the drag of the pod housing is not the main concern of this committee, Dr. Ghassemi is asking for the committee's opinion on this issue. He is particularly interested in the pulling (or tractor) type in which case the pod housing will be in the wake of complex propeller flow.

As reported in Section 4.8 of this committee's report, studies due to Rains et al. (1981) and Minsaas (1988) include some information on the drag characteristics of pusher and tractor type propulsors. In the method for the tractor type, Minsaas neglects the effect of cavitation and propeller induced drag, and provides a semi-empirical formulae for the viscous drag of the housing components depending upon whether the component is in or outside the propeller's slipstream.

On the other hand, amongst the reported computational methods, one can mention studies e.g. Dumez (1997) (reported in Le magazine du Bassin d'Essais des Carenes) and Korpus et al (1998) (reported in PRADS `98). The former of these studies is based on the panel technique while the later utilized a RANS technique. There is also a recent useful work reported by Vartdal et al (1999) (published in the CFD `99 workshop, Norway) comparing these two procedures on a tractor type pod drive.

In the light of this information, the committee feels that panel methods are adequate to model the potential flow effects taking into account the influence of the propeller's vortex wake. However, the major contribution due to the viscous effects still remains as the problem which will require complex boundary layer and even separated flow models. For these, the committee recommends use of RANS techniques, as successfully applied by Korpus et al (1998), including the effect of the propeller.

As far as the performance prediction of different types of pods is concerned: whether it is a pusher or a tractor type, both configurations are unconventional, the latter being less conventional and hence more complex to handle.

However, if the performance prediction is based on model test procedures, it would not make much difference to look for fundamental differences in the approaches depending on the type of the propulsor. This is because the current procedures, which are not in the public domain although they are outlined in Section 5.9 based on the best of our knowledge, utilize standard tests of the resistance open water and self-propulsion. In any case, these tests will be performed for both types in a similar fashion, although the scale effect correction for the pod drag (due to low Rn) will be more complex for the tractor type.

However, our recommendation to the ITTC as outlined in Section 7, is that the extrapolation of podded propulsors should be done based on a self-propulsion test which includes the hull, pod and propeller as unit and not by breaking the test up into components such as open water, resistance etc.

References:

Dumez, F-X, 1997. PODS, some encouraging conclusions, Le magazine du Bassin d'Essais des Carenes, No. 7, January.

Korpus, R. et al., 1998. Hydrodynamic design of integrated propulsor/stern concepts by Reynolds – Averaged Navier – Stokes techniques. PRADS '98, The Hague

Vartdal, L., et al, 1999. On the use of CFD Methods in Developing the Azipull Podded Propulsion System, CFD Conference '99, Ulsteinrik, Norway

Minsaas, K.J., 1988. Propulsion Systems for High Speed Marine Craft, MARINTEK A/S Report, May issue.

Reply to J. Pylkkanen

The following briefly summaries the verbal reply given at the conference.

(a) Where full scale data was available, estimates of energy efficiency gains were made using this data and the results presented in the report reflect this. An example of this is included in the efficiencies quoted in tables 2.4.

(b) The effectiveness of turbulence stimulators in controlling separation is extremely uncertain and cannot take the place of tests done at increased levels of the Reynolds number.

Reply to F. Mewis

The following briefly summarises the verbal reply given at the conference.

We thank the discusser for his comments and we agree that the gap between propeller and pod in tractor units has a great influence on the measured thrust values of the propeller.
Again we recommend self propulsion tests to be used on their own as the basis of extrapolation of full scale power. The intermediate values in this process, such as thrust deduction fraction, wake, etc., may not have the same meaning, or values, as in the powering prediction for conventional propulsion systems.