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The problem of propeller design for high ice class transportation ships

A.V. Pustoshny, G.K. Darchiev, I.G. Frolova.
Krylov State Research Centre, St. Petersburg, Russia

ABSTRACT

Development of large-capacity merchant vessels of high ice classes for Arctic has raised a range of challenges for propeller designers. Usual propeller design approaches are often not applicable for high ice-class merchant vessels.

In particular, the requirements regarding vibration levels related to propeller cavitation, which are traditionally specified for merchant ships, cannot be directly implemented in case of propellers designed for operation in ice because general measures to reduce cavitation like skew or tip unloading prove to be either not feasible or limited by the requirement to ensure safe propeller operation in ice conditions. The measures against cavitation are also restricted by special blade section profiles used for ice propellers. Some methodological problems in recording cavitation during model tests of ice propellers were also identified.

The paper examines the above-mentioned aspects of propeller design.

It is shown that the above issues can be resolved only when both ship designers and ship owners are fully aware and take account of these problems. In the situation when the minimum level of power delivered to propeller is governed by safety-in-ice considerations, it is critical to ensure correct specification of ship speeds as well as cavitation requirements for open water operation. Inappropriate choice of ship's service speed, which actually sets design conditions for propeller, will cause not only cavitation problems but also economic losses because propeller designers will have to meet the prescribed requirements by revising the propeller geometry.

INTRODUCTION

Large capacity merchant vessels of high ice classes are intended to carry goods from Arctic to European and Japanese ports. Such lines include hundreds of miles distance covered by ice and thousands of miles in open waters. It have raised a range of challenges for ship designers including: requirement to modify strength standards worked out in the XX century for relatively small ice-going vessels; higher hydrodynamic resistance of ice-class ships in open water primarily related to elimination of bow bulb; maneuverability issues of large-size vessels in ice (partly resolved by implementation of azimuthing thrusters as main propulsors).

Some of these problems were expected to be solved by so-called DAS (double acting ship) concept invented in Finland. According to this concept the vessels equipped with electric podded propulsors should sail bow-first in open water and stern-first in ice. However, from the Russian experience with 3 series of heavy-tonnage merchant DAS of different types it is seen that the concept evolves not exactly as it was initially envisaged by its inventors. In deviation from the original declarations of inventors high ice class DAS are built without bow bulb to enable both bow- and stern-first operation in ice, the vessels require additional strengthening of stern (according to presently discussed Lloyd's opinion stern and propulsor should be one ice class higher) and special maneuvering tactics in ice.

The strength issues of DAS propulsion system have been successfully solved (with no reported accidents), but as it was found by our analysis the specified blade section thickness is significantly higher than it is required for the given ship ice class.

Still the challenges of propeller design for high ice class merchant vessels are not restricted to strength considerations. It is seen that in recent years shipowners request and shipyards accept the same requirements for ice propellers as for propellers designed to operate in ice-free waters. First of all it concerns specifications related to cavitation, in particular pressure fluctuation levels on hull induced by propeller. Advent of Arctic cruise liners has complicated the situation even more because in this case the noise and vibration requirements have long been the absolute priority for shipowners.

This study analyzes the ways how these requirements can be taken into account in the propeller design process.

1 CONSTRAINTS FOR IMPROVEMENT OF ICE PROPELLERS

Traditionally, the main requirement in design of propellers for high ice class ships (earlier these were mainly logistic supply vessels, icebreakers and specialist vessels designed solely for Arctic operations) was to provide the maximum (specification) bollard pull to achieve the target ice going capability, as well as the requirement to assure proper blades strength in ice in compliance with the ice class notation. These criteria were met based on the rules of the appropriated classification society and verification of propeller strength in ice, in particular by finite element analysis.

As a rule, hydrodynamic characteristics for transit mode were not strictly regulated but determined only to predict ship propulsion performance in open water.

For high ice class merchant vessels operating on routes with long open-water distances the efficiency considerations have become quite important. However in this case we have to reckon with considerable constraints typical of ice propellers:

- minimum thickness of ice propeller blades specified by classification societies is much larger than that of propeller blades designed for ice-free operation, while for estimation each 10% increasing of blade thickness inevitably reduces the efficiency in ice-free running conditions by 0.6-0.8%;

- increased water resistance of ice-going vessels results in higher hydrodynamic loads and, respectively, in lower efficiency of ice propellers.

A traditional method of raising propeller efficiency by minimizing blade area ratio is also constrained in case of ice propellers. The lowest limit of blade ratio is determined by the requirement for bollard pull mode to avoid thrust break cavitation at full power consumption. Such mode is usually verified experimentally in vacuum tanks or cavitation tunnels of large cross-section where models can be tested in bollard pull mode without interference of propeller-induced velocities. In some cases the blade area ratio is increased based on blade strength considerations in an effort to keep relative blade section thickness within reasonable limits.

The upper limit of blade area ratio is set at 0.7-0.75, as recommended by ice technology experts, to make it easy for ice blocks to pass between blades as well as to minimize ice interaction with several blades simultaneously.

Thus, the blade area ratio of ice propellers, as a rule, can be varied only within a limited range of 0.6-0.75.

Let us analyse which options are available for propeller designers to reduce cavitation arising in open water.

Until now the issues related to cavitation inception on ice propellers have not been sufficiently studied. Ice propellers are made of high-strength stainless steel, and in bollard pull conditions there is a prevailing tip vortex that does not cause erosion. Thus, erosion issues were left out of consideration here.

Vibrations related to cavitation on propeller blades of icebreakers and ice-going vessels were also not considered as the first priority problem, since propeller vibration and noise as well as noise due to hull/ice interaction are considered as an inevitable trouble during operation in ice-covered waters. The ship speed to be used for vibration regulation of icebreakers in open water is still a debated subject.

However in view of cavitation requirements introduced for modern merchant vessels and rather high speeds of these vessels (to be discussed in more detail below) it should be reasonable to examine if the measures proven for fighting cavitation on open-water merchant vessels could also be applied to ice-going ships.

- Cavitation mitigation by appropriate choice of blade area ratio is a rather limited measure in case of ice propellers, as was noted above.

- Propellers intended for ice operations are featuring special blade section profiles designed for ice milling conditions. According to the Russian Maritime Register of Shipping, the blade thickness at 5% chord length from leading and trailing edges is subject to regulation for strength reasons, which gives rise to additional problems in terms of blade hydrodynamic performance. Blade section outlines of ice propellers can be described as “more blunt” in comparison with conventional hydrodynamic blade sections. There is only a limited room for modification of these blade profiles for cavitation improvement because the first priority of ice propellers is strength.

- Implementation of tip unloading is also quite limited. These constraints in the rules of the Russian Register are explained by the requirement to ensure some positive angle of attack for blade tip sections at ice milling. In case of large tip unloading there is a risk of flat blade/ice interaction or negative angle of attack in ice milling. It may result in blade tip damage or bending. Multiple cases of blade tip bending in skewed propellers reported for merchant ships in Russia suggested that the blade skew should be limited for ice-class propellers (in Russian icebreakers propeller blades were always of symmetric outlines). In particular, Ref. [1] recommends restriction of the skew angle of reversible CP propellers to 5 degrees in case of icebreakers and high ice classes Arc 7-Arc 9, and to 10 degrees for ships of Arc4-Arc 6 classes with strength verification of higher than 5 degree cases using finite-element analysis. Also, it should be taken into account that skewed blades even in case of lower ice class ships are prone to damage risks, especially in reversing modes. For non-reversible propellers (CPP) these skew angle constraints could be made somewhat tougher for all ice classes.

From the analysis it seen that there are very limited opportunities to adjust cavitation characteristics of ice propellers by means of geometry modifications. Introduction of pulling propulsion pods slightly improves the situation because with this arrangement the propeller inflow has low non-uniformity. An exception is middle propellers in triple propeller systems where interaction between keel box wake and static head pressure before the strut of pulling pod may result in a rather significant local area of decelerated flow and increased cavitation.

Thus, designers have limited opportunities to improve cavitation performance of high ice class propellers.

2 DETERMINATION OF SPECIFICATION SPEED FOR REGULATION OF CAVITATION CHARACTERISTICS.

In order to work out propeller cavitation requirements for ice class vessels it is necessary to choose and justify the ship speed for which these requirements are formulated. In case of other than ice class vessels the cavitation performance is specified for the ship speed attained at 80-85% MCR (maximum continuous rating) with allowance of sea margin. As a rule, this is also the design propeller point for which propulsion requirements are formulated, and the propeller geometry is chosen to minimize cavitation in this mode. However, this speed specification procedure is doubtful for ships of high and medium ice classes (starting from Arc 5 or 1A Super) because national classification societies regulate minimum propulsion power on board depending on the ice class. It should be noted that unfortunately the international Polar Classes (PC) is an exception with no minimum power requirements still figured out.

According to Table 1, which is based on the inputs from FSA Rules and estimates published by MAN DIESEL, Ref. [2], the minimum power of even medium ice class 1A Super ships must be at least 60-70% higher than that of non-ice class vessels, while for ships of about 100000 dwt the power ratio reaches 2.2.

Table 1. Comparison of MCR for non-ice-class medium bulkers and tankers with the minimum power of the same type 1A Super ships

Bulk carrier						
Type	Handysize	Handymax	Panamax	Cap-size	Cap-size	Capsize
Dw (th.t)	30	50	70	100	150	200
Pmin(1As) /Pnorm	1.69	1.71	1.52	2.20	1.66	1.74
Tankers						
Type	Handy size	Handy max	Pana max	Afra max	Suez max	
Dw (th. t)	30	50	70	100	150	
Pmin(1As) /Pnorm	1.63	1.70	1.40	2.03	1.9	

The minimum power formulas for ice class vessels are different in different classification society's rules. Table 2 shows how the minimum power determined as per RS and ABS rules changes between 1A Super class and two higher ice classes. In spite of a wide variations between the minimum power requirements for high ice classes (especially starting from Arc 7) it can be noted that even following to "moderate" relationship suggested by the Russian Register the minimum power requirement increases by 15-20% for each next class upward.

It is of principle importance that the minimum power specified for ice class vessels includes "ice margin" to cater only for safety operation in ice. This margin may be as much as 50 - 80% of propulsion power in non-ice-class ships considered as sufficient for operation in open water. Even if we take into account the need to increase the power of ice-going vessels by about 20-30% due to a greater hydrodynamic resistance of ice-class hull lines, the ice margin of a high ice class vessel will be about 30-50% as compared with the power of non-ice-class vessel of similar type.

Table 2. Comparison of minimum power requirement increment for higher than 1A Super ice classes as per RS and ABS formulas.

Russian Register (RS)				
Displacement	RS class	Arc 5 (1A Super)	Arc 6	Arc 7
100000 t	Pmin/Pmin(Arc5)	1	1.16	1.32
30000 t	Pmin/Pmin(Arc5)	1	1.24	1.48
ABS				
Displacement	ABS class	A1 (1ASuper)	A2	A3
100000т	Pmin/Pmin(A1)	1	1.27	2.18
30000 т	Pmin/Pmin(A1)	1	1.2	2.13

In the recent years there is a trend to assign the specification speed of high ice class vessels in open water based on the total propulsion power, including the ice margin mentioned above. When cavitation requirements are specified for the speed assigned in this way, the propeller designer has to meet these requirements at about 1.5 times higher propeller loading and at a higher speed as compared to non-ice-going vessel (making use of about 50% ice margin power). At the same time practically all traditional measures of improving propeller cavitation performance by adjustment of propeller geometry are not allowed.

It should be noted that inevitably it will be very uneconomical to operate ice-class hulls at this speed, and shipowners, as a rule, reduce the service speed in open water. Thus, the actual operating speed of ships is quite different from the specification speed verified during sea trials. Fig.1 shows the speed versus power curve for an Arc 7 tanker, indicating that the economic speed is achieved at 37% of available total power. Even if we take into account that usually the specification power is reduced with respect to MCR by 15% sea margin, still about 50% power is used as the ice margin on this Arc 7 ship.

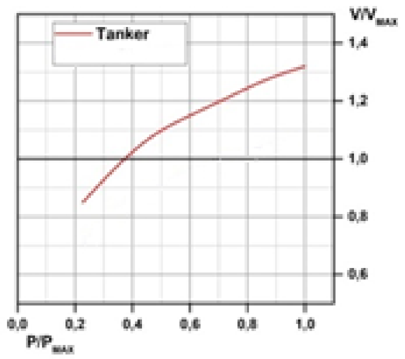


Fig.1. Speed and power ratios for ships of high ice classes (Arc 7).

3 VERIFICATION OF CAVITATION CHARACTERISTICS FOR ICE-CLASS PROPELLERS.

Cavitation performance of propellers is tested both on models and in full-scale conditions. Model tests include cavitation observations and recording of pressure fluctuations on hull near cavitating propeller, while in full scale conditions the pressure fluctuation values are checked. Recently, it has also come into practice to analyse a wide spectrum vibration and noise within the hull spaces in a wide range.

The latest break-through in this area is application of boroscopic technologies for full-scale observation of propeller cavitation supported by high-speed video and computer data storage and processing tools. This kind of technology notably simplifies outboard deployment of optical instruments, make it possible to avoid viewports in the underwater hull replacing these with 1 or 2 small well-sealed penetration holes for boroscope, and enables researchers to observe cavitation in good daylight environment without special strobe lighting used in the past.

However, recognizing a significant progress in research technologies we should also note errors arising in cavitation tests. The analysis should be started from choosing specification modes for recording of parameters related to propeller cavitation in full scale.

According to the usual practice established by diesel-engine manufacturers, the specification power and rpm of propellers are assigned with allowance for the sea margin (i.e. for 85% MCR) with the propeller curve crossing 100% MCR point and corresponding to the maximum hydrodynamic resistance (hull fouling, stormy seas). At the same time it is recommended to design the propeller for the same speed of revolution, but based on the propeller curve corresponding to freshly painted hull and open water. It means that the propeller is designed for about 5-6% lighter conditions in terms of rpm, and 85% MCR is achieved for clean hull at the speed of revolution by 5-6% higher than the design rpm.

At the shipyard sea trials of ships with clean hull in calm sea the propeller advance ratio is about 5% lower than the design value, which influences the cavitation pattern. It is impossible to achieve the specification power with a clean hull at design propeller revolutions in such sea trials. Thus, following the established procedure for specification of the design mode, justifiable from the standpoint of propulsion performance and safe operation of diesel engines, we obviously run the propeller in off-design mode during shipyard cavitation sea trials.

The same “sea margin” system is applied to ice class merchant vessels with electric propulsion. Apparently, it can be explained by required power margins to support safe operation of diesel generators, though the electric drive is much more flexible to overloading.

This error in specification of the propeller mode is automatically passed to test modes of cavitation performance in model tests adding up to uncertainty inherent to test methods used in cavitation tunnels or vacuum tanks.

The most important of these errors is caused by inevitable discrepancy between model and full scale velocity fields. Today, modern CFD tools enable us to predict quite accurately the wake of a full-size ship. However, it is not practically feasible to simulate this wake in model conditions. Therefore, researchers of different laboratories employ various artificial measures. E.g., cavitation tests at MARIN’s depressurized tank are performed at somewhat higher than design advance ratio to take into account the speed scale effect at the 12 o’clock blade position.

A related aspect is errors in simulation of propeller/hull interaction coefficients. Strictly speaking, even if we manage to simulate a close to full-scale nominal wake, it does not guarantee that the right wake fraction at running propeller is obtained. Wake fraction values obtained experimentally for a single-shaft merchant vessel are compared below with RANS computations by a team of KSRC researchers led by M.P. Lobachev:

Self-propelled tests, model …..…… 0.24

Computation of flow area near propeller based on the wake field measured on model …… 0.50

Nominal wake (average velocity over propeller disk measured on model).. ..………………...…0.51

Calculation of flow around hull including propeller operation ……………………..…...…….0.25

As it is seen from the above comparison, the correct propeller mode with the wake fraction of 0.25 matching the self-propelled model test data was obtained only after complete flow calculations for the entire hull with running propeller by RANS method. It highlights the importance of taking into account the interaction between propeller and flow around hull, which is impossible to model physically using various artificial flow simulation techniques during cavitation tunnel tests (shortened models, flow liners, honeycombs).

In propeller model tests there is a wide Reynolds number discrepancy for flow around blades. In particular, it applies to tank tests where propeller revolutions have to match the self-propulsion test mode, the same being about two times lower than the maximum rpm set during cavitation tunnel tests. Therefore, compared to full scale conditions a larger area of model blades feature laminar flow patterns, which influences both propulsion and cavitation characteristics of propellers. For this reason an important issue is flow turbulization on blades, in particular for depressurized tank tests. According to ITTC only 3 laboratories use artificial roughness on blades in cavitation tests. MARIN tank applies roughness strips near leading edge. SSPA cavitation tunnel applies special paints on blades. This artificial roughness, apart from turbulence, may cause flashes of cavitation behind each roughness element. Given a margin blade area ratio traditionally included in the propeller design, this cavitation quickly disappears once the maximum pressure reduction near the leading edge is gone. However, when the pressure reduction area extending over the most blade chord length is close to the critical cavitation inception values, it may provoke significant cavitation over the entire suction side and considerable distortion of the cavitation bucket. (see Fig.2,3).

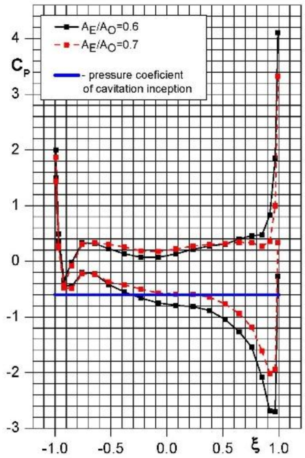


Fig.2 Pressure distribution over blade outline at various blade area ratios

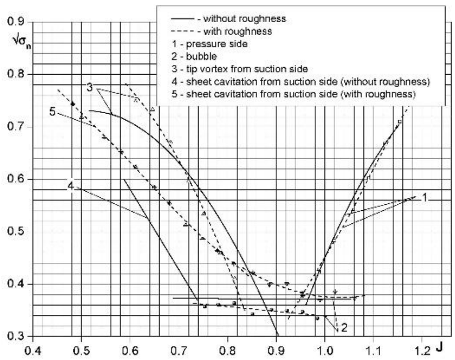


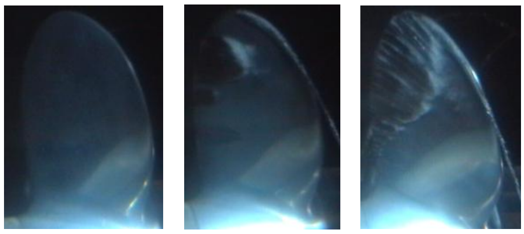
Fig.3. Distortion of the cavitation bucket due to artificial roughness applied on blades of passenger ship propeller

Cycles of tests were performed on both ice and non-ice class propellers to study how roughness influence cavitation processes. The test scope included tests without roughness, as well as tests with a 30 μm roughness strip on leading edge at 2.5% chord length and with a roughness 3 μm at the same location.

Fig.4 shows cavitation patterns versus roughness size on an ice propeller under the same conditions. Fig.5 illustrates cavitation development versus advance ratio with and without roughness.

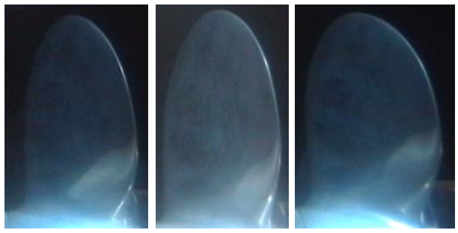
A study with various types of roughness has shown that in the case of ice propellers distortions of the cavitation bucket noticeably depend on the roughness size (Fig.6).

As it is seen from the data given above, cavitation patterns are significantly altered by roughness, in particular at low advance ratios. For the ice class propeller the left-hand branch of the cavitation bucket was noticeably changed. Thus, it is noted that roughness parameters strongly influence both the

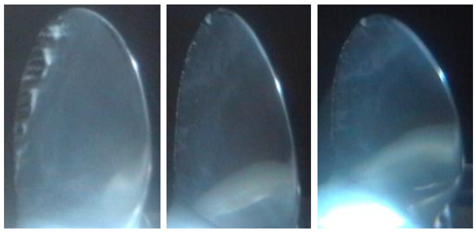


(a) (b) (c)

Fig.4. Cavitation on propeller blades at J=0.512 $\sqrt{\sigma_n}$ =0.409. (a) – without roughness; (b) – with roughness μ=3μ m; (c) – with roughness μ=30 μm.



Without roughness



With roughness μ=30 μm

Fig.5. Propeller blade cavitation at $\sqrt{\sigma_n}$ =0.470 for different advance ratio J=0.539; 0.763; 0.906.

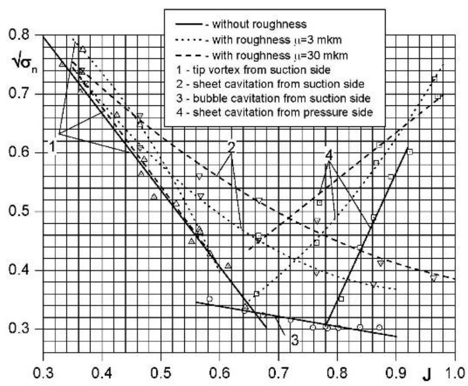
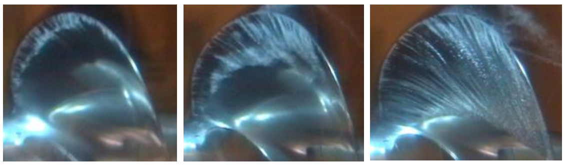


Fig.6. Effect of leading edge roughness on cavitation bucket of ice propeller.

cavitation pattern and cavitation inception of the ice propeller.

However, absolutely similar tests on a non-ice class propeller with tip relief gave surprisingly different results. Fig.7 shows cavitation development in function of the cavitation number for such propeller with and without roughness. It is notable that neither cavitation pattern nor cavitation inception bucket (Fig.8) depends on the size or shape of the applied roughness (3 and 30 μm strip with a width of 2.5 chord lengths).

(a)



(b)

(c)

Fig.7. Non ice propeller blade cavitation at J=0.700 and $\sqrt{\sigma_n}$ =0.600; 0.520;0.420. (a) – without roughness; (c) – with roughness μ=3 μm; (c) – with roughness μ=30 μm.

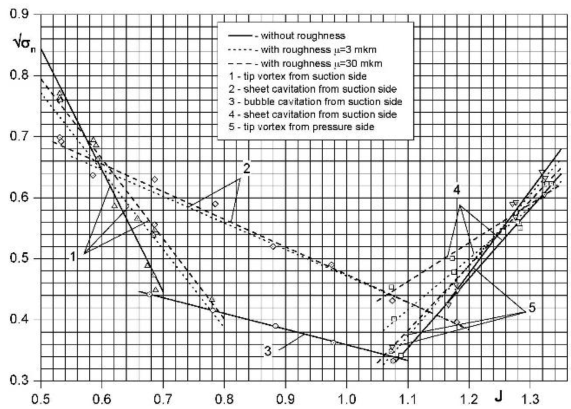
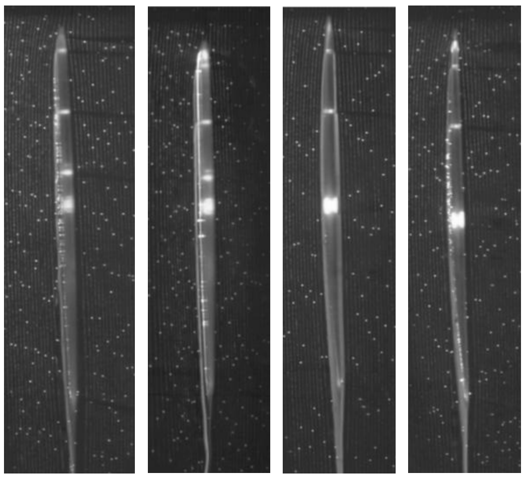


Fig.8. Effect of leading edge roughness on cavitation bucket of non-ice class propeller.

Thus, in case of applied roughness it was found that ice propellers designed within ice class constraints using ice propeller design practices as well as non-ice class propellers with tip unloading are featuring, along with traditional relationships of cavitation in function of advance ratio and cavitation number, considerable dependence of model cavitation test data on both the propeller type (ice or non-ice class) and roughness parameters.

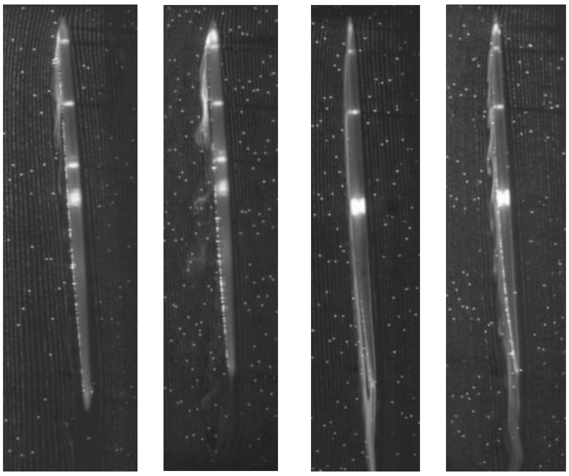
The reasons of such discrepancies are not yet understood and no indications or relevant relationships are found in publications of research centers using artificial roughness techniques. It can be presumed that for an ice propeller designed in an effort to increase its efficiency for extra 2-3% by neglecting blade area ratio margins all implications of test method assumptions are compounded, and it is extremely difficult to correctly predict such effects.

A study with visualization of laminar flow around flat foils with and without roughness was carried out for a more detailed analysis of the leading edge roughness effect on flow characteristics. Flow visualization was done by paint (injected through special holes in foils) and hydrogen bubbles (generated by wire electrodes arranged across the test section). The IK-82 ice blade sections were taken, which are used by KSRC in design for ice class vessels. The chord length was chosen L=150 mm, maximum width - 4.54 mm, maximum camber - 1.47 mm. NACA 66-mod series with similar maximum camber and thickness was chosen to represent non-ice class case. Figs 9 – 11 present the test results obtained for these sections at 2^o and 4^o angles of attack, respectively.



(a) (b) (c) (d)

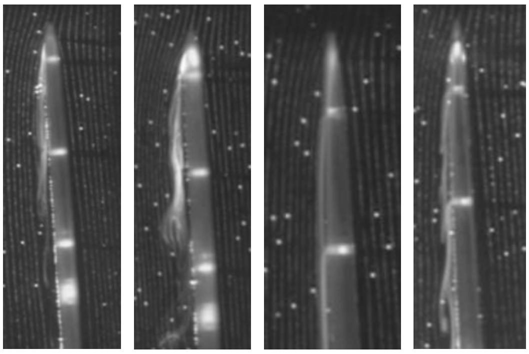
Fig.9. Test results of blade sections with and without roughness at 2^o attack angle; (a), (b) – ice blade section without roughness and with roughness at leading edge, respectively; (c), (d) –NACA 66-mod section without roughness and with roughness at leading edge, respectively



(a) (b) (c) (d)

Fig.10. Test results of blade sections with and without roughness at 4^o attack angle; (a), (b) – ice blade section without roughness and with roughness at leading edge, respectively; (c), (d) –NACA 66-mod section without roughness and with roughness at leading edge, respectively

The laminar flow patterns shown here demonstrate two things. First, flow patterns around ice blade sections are different from flow patterns around NACA sections starting from tip (as expected, in case of blunt ice sections the streamlines have a more sharp angles of deflection in way of the tip as compared with NACA sections). Roughness has practically no effect on flow pattern around the NACA section, but it changes the flow pattern around the entire ice blade giving rise to unsteady vortex structures on the suction side some distance away from the tip, which are moving along the blade. In the aft part of the ice blade a noticeable return flow is observed.



(a) (b) (c) (d)

Fig.11. Test results of blade sections with and without roughness at 4^o attack angle, view of leading edge area; (a), (b) – ice blade section without roughness and with roughness, respectively; (c), (d) –NACA 66-mod section without roughness and with roughness, respectively

Summarizing results of the test above, the following point may be figured out.

Two types of cavitation tests are known – studies of developed cavitation and studies to determine cavitation inception (bucket) diagram. In view of the identified roughness effects each of these two types of studies need to be addressed separately. When cavitation inception diagram is determined based on tests behind hull, we can talk only about cavitation inception of a given propeller behind a given hull model (i.e., about a particular non-uniformity). It should be noted that because cavitation inception is highly sensitive to changes in advance ratio, cavitation number and propeller type all the above-mentioned errors will significantly affect the results. The fully developed cavitation is less sensitive to the said errors, and the blade roughness application techniques (which are strictly speaking well established for simulation of developed cavitation patterns in full scale conditions) may serve as an indication of cavity sizes on blades.

Krylov State Research Centre has four decades of experience in propeller design and cavitation predictions based on experimental determination of the cavitation inception diagrams in cavitation tunnels for model propellers in open water (without hull) using the hull wake estimated for full scale conditions. This approach make it possible to avoid errors related to simulation of full scale wake in cavitation test facilities as well as to run propeller model at maximum speed to maximize flow turbulization.

According to the Russian practice, operation of blades in non-uniform wake are usually characterized by a) local cavitation number calculated for a given blade position (water column height) and local inflow velocity at a given point; b) instantaneous advance ratio at a given point of the propeller disk in full-scale conditions. In the propeller design the pitch and camber of different blade sections are chosen to ensure that in any point of the propeller disk the blade mode (characterized by instantaneous advance ratio and cavitation number) would be kept within the cavitation bucket (Fig. 12).

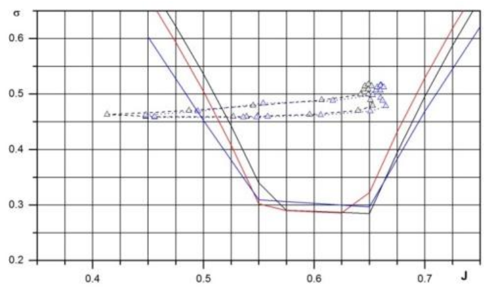


Fig. 12. Typical distribution of instantaneous advance ratios over one blade turn (different cavitation buckets correspond to cavitation at different propeller radii).

If the wake non-uniformity is large and a range of propeller operating points cannot be kept within the bucket, it is preferable to shift the operating range to lower advance coefficients, i.e. allowing suction side cavitation. Nevertheless, the main propeller design tactics is to position the cavitation bucket as uniform as possible with respect to the instantaneous advance ratio range.

When the shipowner or shipyard rely on the data obtained from cavitation tests in model wake without consideration of scale effect and require that the propeller designer should achieve the best result in the model wake, it means that the designer has to considerably revise the bucket position. The propeller is obviously designed incorrectly with respect to the full-scale conditions, i.e. overdesigned to avoid cavitation (non-optimum combination of tip unloading, pitch distribution and camber for full scale conditions). It always results in efficiency losses. It means that if the customer fully relies on model cavitation test data he will run higher fuel costs during ship lifetime. Considering that the minimum relative flow speed for the 12 o'clock blade position is about 0.1-0.15 higher in full scale as compared to model conditions, if the propeller is designed for a model wake it is required to shift the design advance ratio by about 0.05-0.07, which translates into efficiency losses in the design mode of approximately 0.015-0.02 (2-2.5%). Naturally, the method of model open water cavitation tests with allowance of the results of full scale wake behind hull, which is used in Russia, is not free from criticism, however it has been well validated by numerous full-scale observations of propeller cavitation in the Soviet period, and make it possible to take into account the scale effect in a well-informed manner. However, it should be recognized that the blade roughness test method used at European laboratories may give more accurate predictions for developed cavitation at high speeds. As for the tests intended to determine the cavitation inception bucket, being a critical input for propeller design, artificial blade roughness introduce uncertainties, which may alter the cavitation bucket and lead to incorrect propeller design with all commercial implications for shipowners.

CONCLUSION

The following conclusions can be figured out from the above studies:

1. Requirements for ice-class ship propellers should be formulated taking into consideration significant ice-class constraints, which are of the first priority for safety considerations, in particular for high ice classes.

2. Shipowners should assign and shipyards accept the specification speeds keeping in mind that the minimum power requirements for propulsion plants are determined for a given ice class based on safety-in-ice considerations. Specification of propulsion and cavitation characteristics for speeds attainable with ice margins may result in impracticable cavitation requirements and lower energy efficiency of ships. Specifications should be worked out for valid ship service speeds.

3. Advent of large-capacity merchant vessels and introduction of energy efficiency and cavitation requirements for these vessels induce us in a number of cases to revise design and model test technologies for ice propellers. In this situation it is necessary to set right priorities for ice-class propeller requirements.

REFERENCES

1. A.V. Pustoshny. Problems of propulsion performance of merchant ships. KSRC Publication. SPb, 2016. (in Russian)

2. MAN DIESEL and Turbo (Copyright MAN Diesel and Turbo 5510-0140-00 web Aug.2013)

Comment from Dr. B. Actas
The author has presented interesting insight into propeller design for ice class vessels, but the presented experiments are for bollard pull condition at a basin. From our experience the presence of ice in close proximity of propeller significantly effect on cavitation induced noise, vibration and potentially erosion.
Author's closure
Operation in ice is extremely hard mode, rather unsteady because of permanent change of ice surrounding, rate of wake blockage, etc. It is in regards of full scale. But as for experimental methods – ice propeller design and investigations mainly performed for bollard pull mode because it is the heaviest hydrodynamic mode (may be except reversal mode, which for ice condition is not so crucial due to low operational ship speed). So, we consider that bollard pull allows investigation of main problems, especially taking into account low ship speed at ice operation.
Question of S. Turkmen
How ice was simulated in the cavitation tunnel.
Author's closure
We do not simulate ice in cavitation tunnel. We know such attempt from publication of M.Atlar (GB), who simulate propeller blockage by ice.
Question from V.S.Carltin
In your lecture you mentioned many aspects for ships transporting the Northern Sea Rout. Does shallow water become an issue for example in Sanikov Straute, as the ship resistance will increase significantly.
Author's closure
First of all, for ships specially designed with allowance of shallow water operation, there are strong and unfavorable restrictions on propeller diameter at design stage. Sometimes it force to design 3-4 propeller scheme, otherwise it is impossible to utilize power needed for ice operation by propellers of restricted diameter.
Also it is true – in shallow water resistance increased, but it is not main problem at shallow water as the ship speed is relatively low. More important especially at operation near the platforms is the problem to avoid propeller – sea bottom interaction. But usually the strength of ice class propellers is enough even for that.

Questions from Yin Lu Young
1.The vibration characteristics of ice class propellers are critical due to powerful impacts. In particular, it is important to lock at how in the wake in ice conditions affect the model pattern and coefficients.
2.Further coupled fluid and structural analysis should be performed for ice class propellers to understand the serious concentration and defect patterns. Such analysis is particularly needed for reverse operation in ice.
Author's closure
If I understand true, these questions are also in regards of model simulation. These is common interest from all participants of written discussion.
My position based on 17-years' experience of work in ITTC committees is that it is impossible to simulate experimentally or by calculations the rather random conditions of propeller in wake in ice operations. Even in open water ITTC experts note a lot of uncertainty for experiment and their scaling. In propeller – ice interaction with impact of ice on wake the flow pattern as well as forces on the blades and propeller are rather random and even on averaged value, the scattering of results will be very high. So, in ice we cannot obtain so exact values as in open water experiments. We can speak only on some probabilistic results. That's why the approaches and accuracy applied for ship in open water in order to determine exact specification values (of speed, force on blades, vibration etc.) are not acceptable for high ice class ships – accuracy of any kind of simulation for ice conditions will be at times lower than for non-ice conditions. The fact that ice class investigations require alternative methodology for investigation is one of main topic of my presentation.