JAAP WIND

HUB SIZE SELECTION CRITERIA FOR CONTROLLABLE PITCH PROPELLERS AS A MEANS TO ENSURE SYSTEMS INTREGRITY

EDITOR'S NOTE: This paper is a modified version of presentations recently made by the Author for naval architects and navy specialists engaged in design and operation of ships with gas turbine-CPP propulsion, both in Europe and North America. Nomenclature used by the Author throughout the text will be found in the APPENDIX.

THE AUTHOR

is a citizen of the Netherlands. He studied Aircraft Engineering and Mechanical Engineering at the Delft Institute of Technology from which he received his MS degree in 1955. He began his career as a Research Engineer in the field of Nuclear Engineering, employed by VMF-Stork, and he holds several U.S. Patents relating to ultra-centrifuges for the separation of Uranium. He was next engaged in design of control gear for radio astronomy aerials and operating gear for bridges and huge floodgates in the Scheldt and Rhine Delta. Later he became involved in the control of diesel engines and controllable pitch propellers and in 1967 joined LIPS Propeller Works where he developed CP propeller mechanisms, solved the problem of high-pressure oil supply to large diameter marine shafts, and made an improvement for high-speed propeller nozzles. As Head of the Design Department he was responsible for the design of large controllable pitch propellers for fast Container Ships and Naval Ships with gas turbine drive. Mr. Wind has authored several technical publications among which are "Controllable Pitch Propellers, Their Principles and Mechanisms" and "The Development of CP Propeller Systems." At the present time he is a Corporate Engineering Consultant to the Propeller Division of LIPS B. V.

ABSTRACT

In a CP propeller the size of the hub largely influences the reliability of the system in operation. Accurate choice of the hub-size provides the best means to prevent the system from structural propeller failures and is therefore the best method to ensure reliability and optimized performance.

Recent failures of high-power CP propellers, of which one Navy propeller had a serious mission failure, could have been avoided if a larger hub had been adopted.

The paper contains two tools, available also to non-specialists. One allows clear recognition of the technical degree of difficulty of all types of propellers and the other is a scale to determine the degree of load of each CP propeller hub. Information is given concerning the influence of the hub/diameter ratio on the hydrodynamic performance.

INTRODUCTION

ACCORDING TO AN OLD STORY which has come down from the past, Julius Caesar had a fighting chariot of outstanding design and construction. He set out with his chariot for one of his famous campaigns through the North and West of Europe and civilized *seventeen* Gallican and Germanic tribes. Having completed his campaign, he returned to Rome. Just at the moment that he re-entered the gates of Rome, his fighting chariot disintegrated completely to dust. No part of it was left: wheels, pole, crossbar, front, and floor; everything turned to dust in one split second.

This fighting chariot is referred to as being the most ideal model of design and construction [1]. No part of it was stronger than the rest; no part of it was stronger than needed. There was no weakest spot in the system. Thus the system disintegrated fully and at the very right moment; just after completely having fulfilled its mission. Finally, because there were no remnants, it solved its own environmental pollution problem.

The famous American Author OLIVER WENDELL HOLMES has written a verse of practically the same purport. In "The Deacon's Masterpiece" he introduces a one-horse shay, built according to such logical principles that it had completely uniform strength in all its parts and components. The shay survives generations of children and grandchildren and finally, at the end of a lifetime of a hundred years, it goes to pieces all at once, completely, just as bubbles do when they burst [2].

The wisdom which comes from these two legends is that in a good engineering system, whatever it is, the strength and the lifetime of each part must be tuned perfectly in harmony with all the other components. No one part is allowed to have a safety factor deviating too much from the others. This rule is used by the Author when investigating a number of CP propeller applications and analyzing the causes of failure.

GROWTH OF CP PROPELLERS

In 1948 RUPP [3] gave a review of the state-of-the-art with respect to controllable pitch propellers, the largest unit being suitable for a few thousand horsepower. CP propellers have grown in size and power ever since. More than five thousand units have been put into service and have been applied to practically all types of ships. An impressive number of larger CP propellers in the range of 15 to 35 MW are in regular operation on Naval and Merchant Ships. The most powerful is claimed to be a unit of 34 MW (46,000 HP), installed in the *Australian Emblem,* a single-screw Container Ship, built for the Australian National Lines [4]. The necessity to install a

CP propeller on this ship is said to be found in her multi-diesel engine drive. The majority of high-power CP propellers, however, is associated with the application of unidirectional aircraft-derivative gas turbines [5][6][7].

Figure 1 illustrates a large type CP propeller system that is currently being manufactured for use in Naval and Merchant Ships.

The extent of growth of CP propellers since 1950 is shown in Figure 2. This chart displays *two* curves: one for Naval and one for Merchant Ships. Both curves have typical long-term growth characteristics, with breakthroughs followed by periods of relative inactivity. The waves are caused by the fact that a period of breakthrough usually brings about a number of unexpected events, shortcomings, defects, and other unforseen trouble that impede further growth for some time thereafter. The waves in the Merchant Curve are less distinct and follow more rapidly in succession than those in the Naval Curve. This is explained by the leadtime from project stage to first field experience being much shorter with Merchant Ships. Also the process is better controlled since the CPP manufacturers have more freedom of taking their own engineering decisions. Merchant Ship CP propellers have been developed to a higher power than Naval Ship CP propellers, which is somewhat unusual as compared to the situation with monobloc propellers.

The growth of power has to be considered as a typical improvement of the state-of-the-art. See Figure 3, where the power P is shown as a function of time: $P = P(t)$. Approximation of the input effort to achieve this improvement is given by the area under the curve:

$$E_{i1,2} = k \int_{t_1}^{t_2} P(t)dt$$

This implies that for further development of CP propellers to still higher power, an appreciably greater engineering input will be required. This justifies a warning, in so far that for the application of CP propellers in the highest power range, the specification requirements of the hydrodynamic performance must be carefully weighed and adjusted within the limits of the present state-of-the-art.

FAILURE INCIDENTS

Figure 2 also lists the names of ships that have brought CP propellers to a higher level of power. Most of these

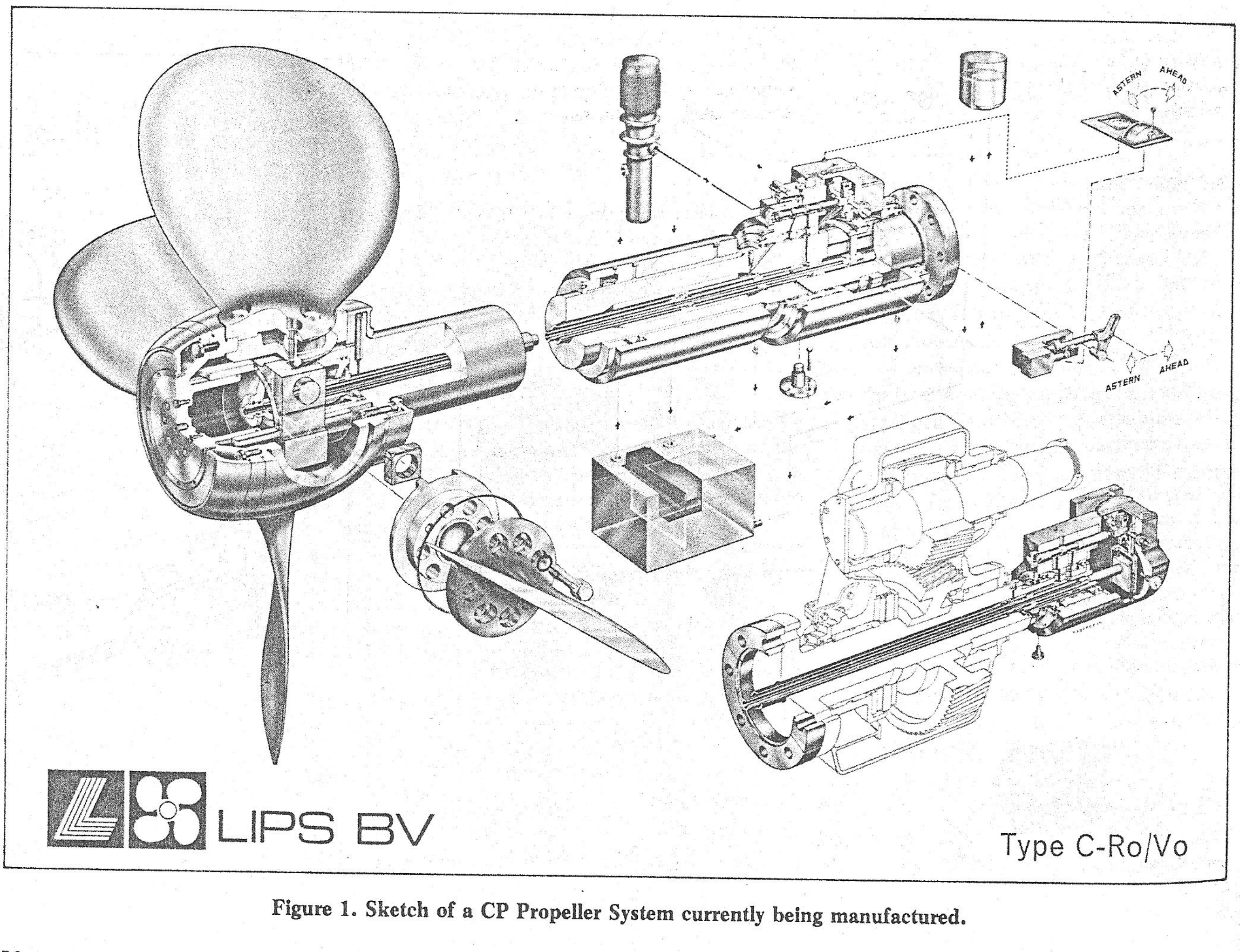


Figure 1. Sketch of a CP Propeller System currently being manufactured.

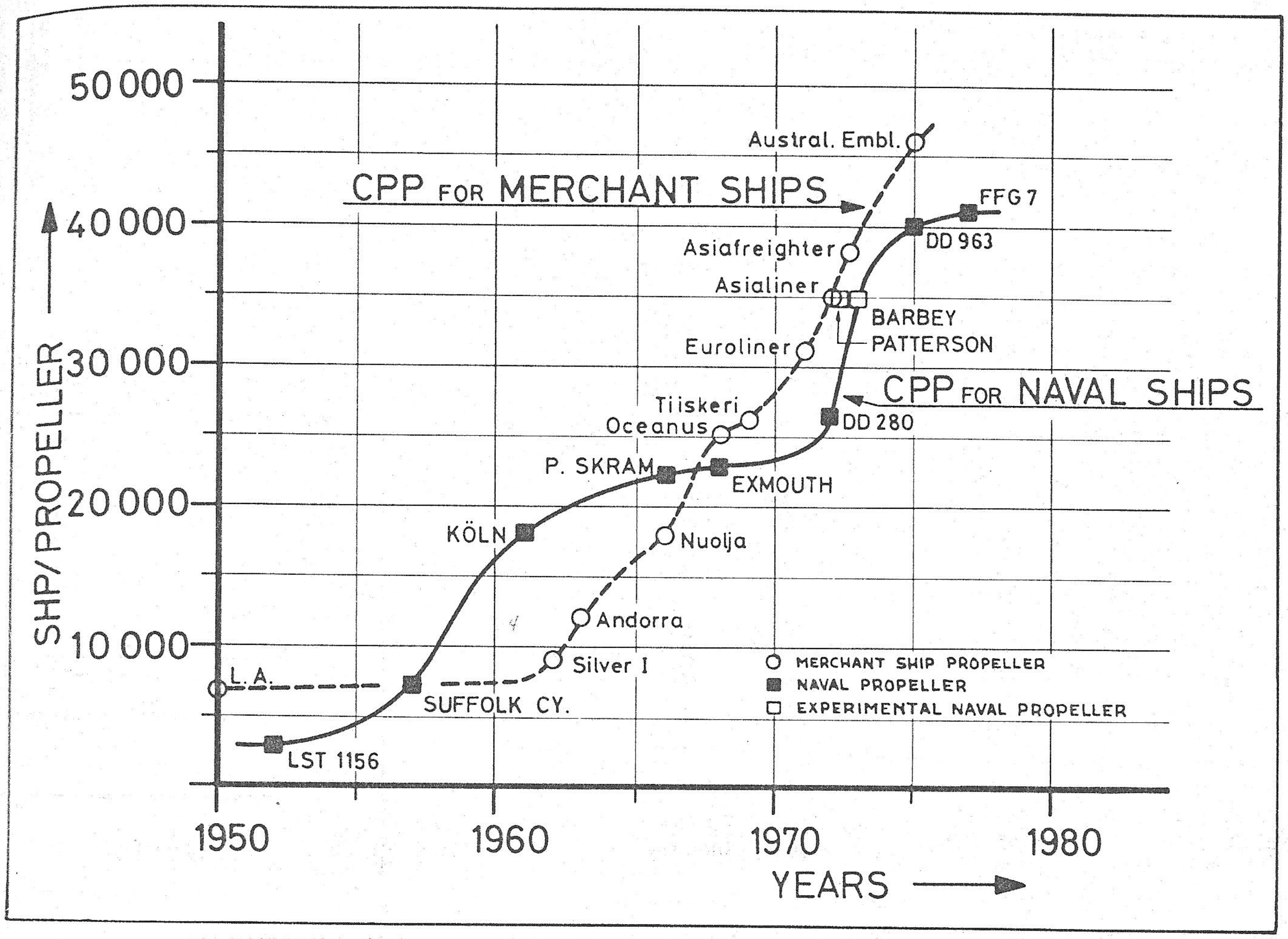


Figure 2. Chart of CP Propeller Growth for Merchant and Naval Ships.

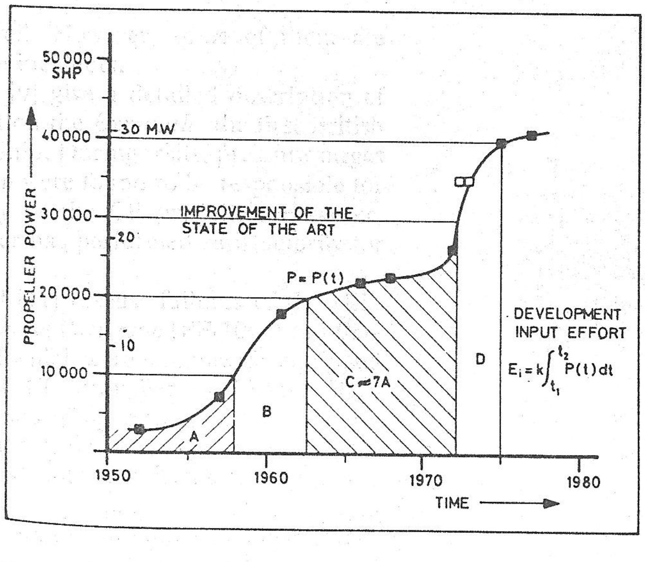


Figure 3. Approximation of the Input Effort in Man-Hours in order to develop CP Propellers to larger power is given by the area under the Growing Curve. Further development will require an appreciably higher engineering effort.

propellers perform well. However, some of them are associated with failure incidents.

References [8] and [9] give a detailed description of problems encountered on the *Exmouth,* the first British all-gas turbine Naval Ship. During trials, pressure surges in the hydraulic system were found to be responsible for serious pitch runaway of the CP propeller. However, since then the propeller has performed satisfactorily for two years.

References [10] and [11] discuss failures of the U.S. NAVY KNOX Class Frigates *Patterson* (FF-1061) and *Barbey* (FF-1088), both of which were temporarily equipped with an experimental CP propeller of 35,000 SHP. *Patterson,* after two years of operation, suffered from a failure of the control valve, which rendered the CP propeller inoperative. Most notorious is the failure of the *Barbey.* In August 1974 after 2,000 hours of service all five blades separated from the hub during a crash astern maneuver. This failure caused a great deal of concern and doubt by naval authorities about the feasibility of high-power CP propellers for NAVY ships. A number of tasks were allotted to technical institutes, laboratories, and naval architects to analyze the failure and to investigate materials in order to formulate rules for structural improvement. The adequacy of the 40,000 SHP CP propellers of the SPRUANCE Class (DD-963) and PERRY Class (FFG-7) was discussed. It was revealed that blade bolts and blade carriers of these propellers had unsatisfactory fatigue life. "Crash-Stop" maneuvers on these ships appear to be prohibited, which is rather astonishing. A "Crash-Stop," though being of great commotion to the ship, is *not* the most critical load condition to the propeller since full power is absorbed at the relatively low thrust developed. If a CP propeller cannot survive a "Crash-Stop," it certainly will be unable to survive years of normal fatigue load.

For Merchant Ships *Lloyd's Register of Shipping* contains a file of data on 1,255 CP propellers fitted on ships between 1965 and 1974 [12]. An analysis of hub defects has been made in terms of engine powers transmitted, and there is, according to Lloyd's, no doubt that the incidence rate rises rapidly with increased power to be transmitted by the propeller. The nature of hub failures observed is described by DET NORSKE VERITAS, and in a Circular Letter instructions are given to Surveyors to pay special attention during survey to a number of seven different types of *possible* hub damages [13].

For this reason LIPS has a very careful design approach for the hub size selection, and the following *two* significant tools must be mentioned in this respect: a) Recognition of the technical *degree of difficulty* of the propeller, and b) The use of a scale to determine the *degree of load* of a CP propeller hub (load factor method).

The *first* tool is open to everybody. The *second* one is a specialist's tool, but the results are available to non-specialists as well. Discussion of these two tools is the main subject of this paper.

RECOGNITION OF THE DEGREE OF DIFFICULTY

Propellers are not all the same: some are known as "easy propellers," easy to design and easy to manufacture, but others are more difficult. Thus, the first step is to find out what sort of propeller one has to deal with.

Two parameters are useful to this purpose: one indicating the propeller size and the other the propeller load. It is clear that more difficulties are associated with making a large propeller than with making a smaller one. Big propellers need more careful design. More risk and money is involved; material properties are harder to maintain in larger sections; casting, machining, testing, handling, and transportation require more effort. Thus the size is of first importance. We prefer to measure size in terms of power rather than using the physical dimension of diameter. This means that we have to express the Propeller Power Magnitude, P, in [SHP], [Watt], or [Joule/sec]. The last unit displays the picture of energy-flow through the propeller.

It is then logical to take the *density* of that *power* as the other parameter in order to determine the degree of difficulty. A higher power in the same area leads to higher structural loads which makes the propeller mechanically more critical: Propeller Power Density L = $\frac{P}{\frac{1}{4}\pi D^2}$ expressed in [HP/m²] or [Joule/m²sec]. Here it it must be mentioned that the hydrodynamic concept of load — $C_T = \frac{T}{\frac{1}{2}\rho Ve^2 \times \frac{1}{4}\pi D^2}$ — is not of any value when looking to the mechanical loading forces working on the propeller. The C_T value is strongly influenced by the speed and therefore is more an expression of the load of the water than of the propeller. C_T ranges from 0.5 to ∞ are adversely high for "easy propellers," such as tugboats, and low for difficult high speed propellers. Therefore, this parameter *cannot* be used to analyze the degree of difficulty of the propeller structurally.

The parameters *power magnitude* and *power density* have been taken as the coordinates in the diagram of Figure 4. Groups of ships have been plotted: Fishing Boats, Tugs, Ferries, Tankers, and Container Ships as examples of Merchant Ships, and Frigates, Destroyers, Cruisers, and Aircraft Carriers as naval vessels. Most Merchant Ship propellers have a power density *below* 1 MJ/m²sec, while naval propellers are in the range of 1 to 2 MJ/m²sec. This shows already that naval propellers tend to have a higher technical degree of difficulty than Merchant Ship propellers.

Propeller diameters appear in this diagram as straight lines through the origin: the range being from fast Patrol Boats having small, high area ratio propellers of 1m to Tanker propellers with diameters up to 10m with a smaller blade area ratio.

The *technical degree of difficulty* is now defined in circular areas around the origin, indicating *five* different classes of which Class 1 is the easiest and Class 5 the most difficult. The origin is the absolute point-of-ease. Close to it are fishery and tugboat propellers. Circle 3 approaches the limit of CP feasibility with the present state-of-the-art. Beyond that limit today only fixed-pitch propellers will be found. Names of some ships with famous propeller applications, CP as well as FP, are indicated.

This diagram leads to a quick recognition of the technical degree of difficulty of each propeller to be considered. As early as the definition phase of a propulsion system, the risks can be evaluated and the requirements of the performance can be specified in harmony with the degree of difficulty.

THE HUB LOAD FACTOR

A controllable pitch propeller system consists of *five* basic elements: PROPELLER, SHAFT, OIL DISTRIBUTION BOX, HYDRAULICS, and CONTROLS. These can be arranged in a group of rotating parts and a stationary group as shown in Figure 5. The rotating parts are large and heavy, but of rather uncomplicated principles [14]. Reliability here is a matter of strength, i.e., keeping the stresses below the allowable values. The stationary parts are smaller in their physical dimensions, but considered as a system they are much more complex. Here reliability is not merely a matter of strength but depends also upon numerous other factors such as leakage, lubrication, dirt, jamming, heat balance, wear, process control, et cetera.

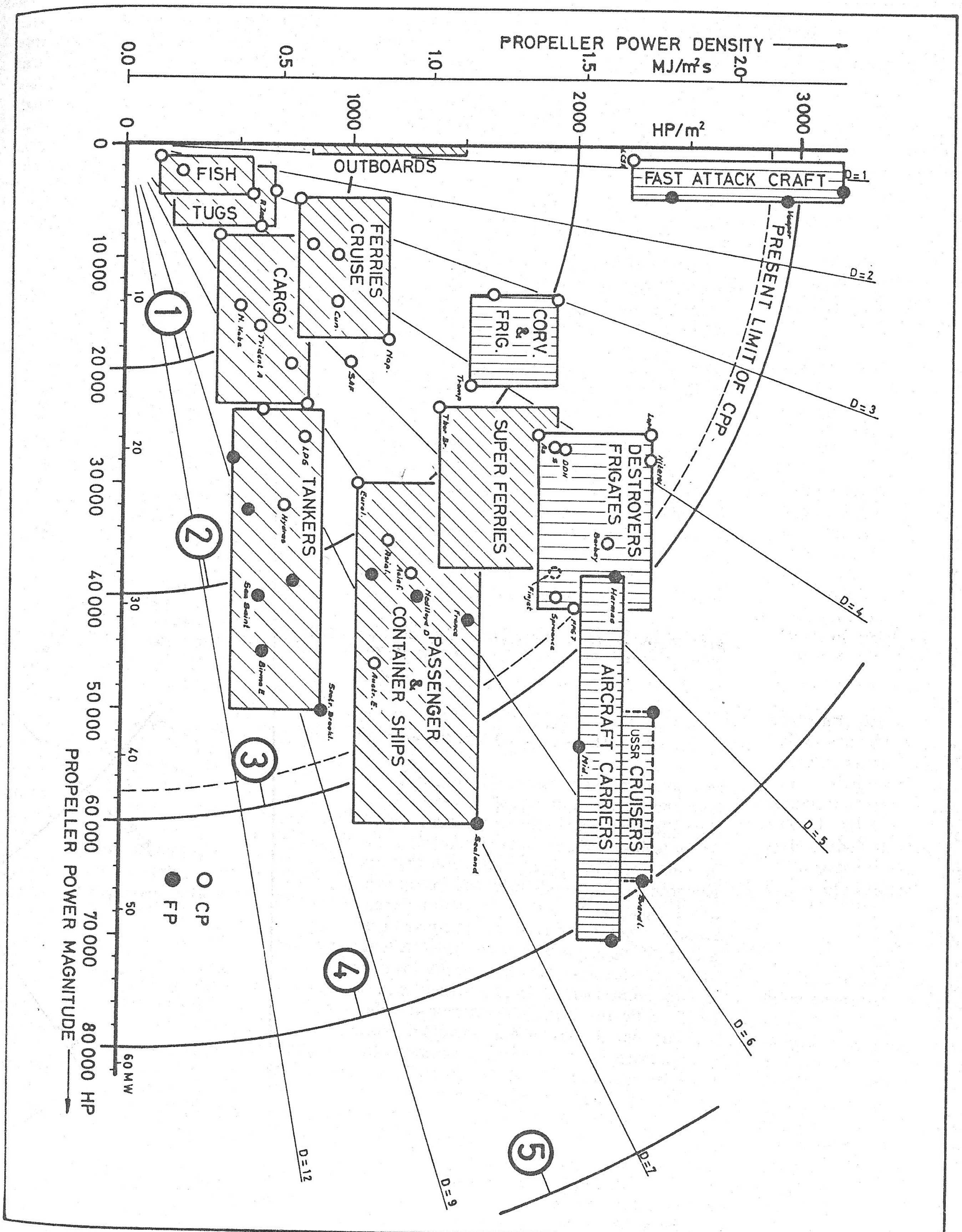


Figure 4. Propellers arranged in different Classes, each Class indicating the Technical Degree of Difficulty of the propeller. It appears that the Technical Degree of Difficulty coincides with the type of ship.

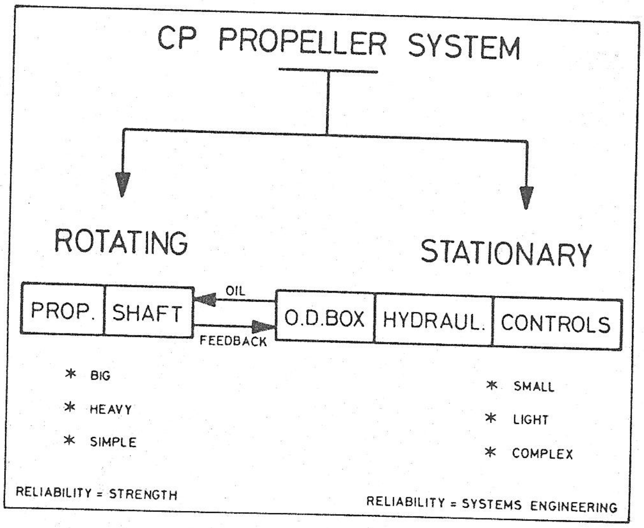


Figure 5. Main Parts of a Controllable Pitch Propeller System.

This paper deals with the reliability of the propeller which consists of a hub and blades. The physical function of the hub is to hold the blades in the correct position to the shaft. This holding to the shaft is associated with forces because the system is exposed to engine power which is transmitted as torque by the shaft. The torque induces forces on the blades: the thrust "T," the tangential force, "TG," and, since the system rotates, a centrifugal force, "C." See Figure 6, the *thrust* being the most dominating.

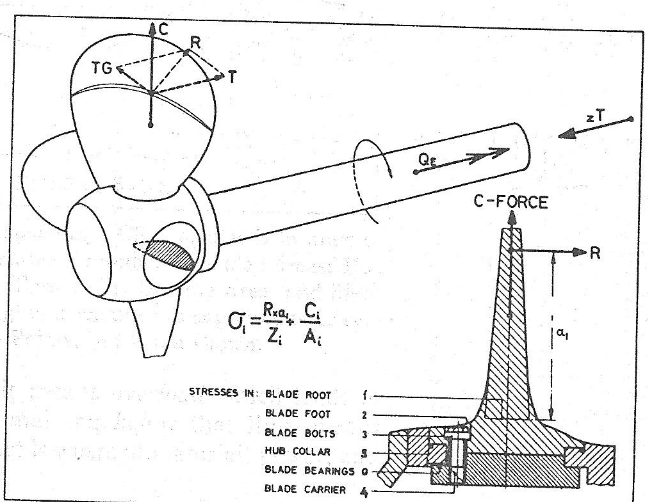


Figure 6. Torque, Forces, and Stresses in Shaft and CP Propeller Blade and Hub.

These forces cause stresses in blades, blade root, blade foot, bolts, blade carriers, and hub body. Once the forces have been defined by power and speed, the level of the stresses is only determined by the amount of material available in the hub. The quantity of this material is not infinite, but is restricted by the size of the hub. Figure 7 shows that the available space must be distributed over a number of mechanical areas: blade bearings, bolts and blade root being restricted in the blade foot, and blade carrier and hub bridges inside the hub. Meanwhile space must be retained for the pitch changing mechanism (not shown in Figure 7). Each of the areas has a limit of allowable stress. It is assumed that CP propellers have been optimized in design by their manufacturers through the years, so that the various areas have given approximately the same strength safety factor. If one of these areas is loaded to its maximum, the hub is loaded to the maximum limit.

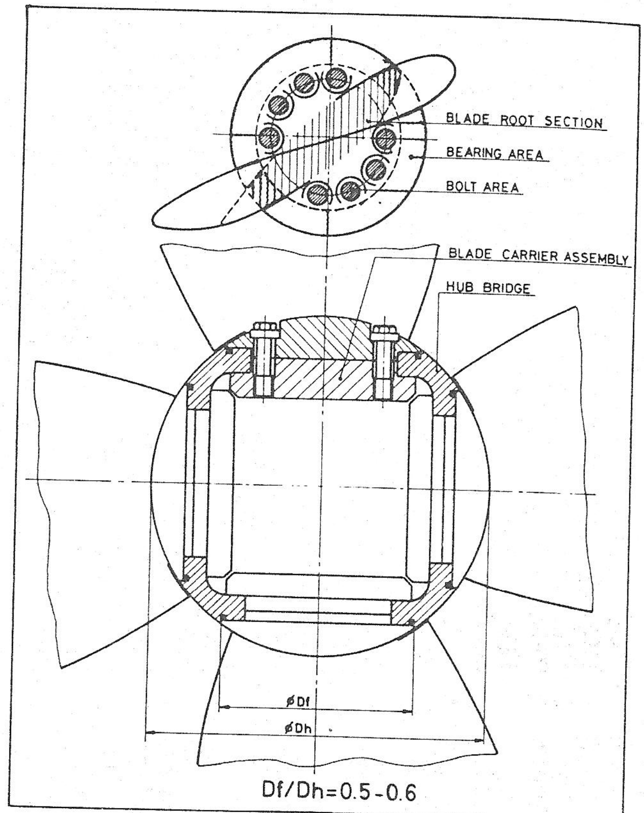


Figure 7. The available space in a CP Propeller Hub must be distributed harmonically over a number of loaded areas: Hub Bridges, Blade Carriers, Blade Bolts, Bearing Area, and Blade Root Section. Pitch changing mechanism is supposed to be centralized under the Blade Palms, but is not shown.

Exceeding that limit means *overload* which leads to risks and failures. Remaining *below* that limit means *underload,* which in fact is waste of material, power, and cost.

The goal is to load the hub to its maximum.

The *hub load factor,* "LF," now is defined as the ratio of the actual hub load to the maximum allowable load for that hub. The load limit of a hub can be indicated as the maximum moment that may be introduced to the blade port. Figure 8 gives the relation of the maximum root moment, "QBF," as a function of the size of hub for four-bladed CP propellers and for five-bladed CP propellers.

The concept of *hub load factor* now provides a scale to determine the degree of load of each propeller hub, irrespective of the type of manufacture. It presents a useful means to predict whether a certain hub application can be expected to be able to operate safely or not.

CP Propellers

Maximum blade root moment

Four bladed CP propellers: Lips hub designation	Hub diam. Dn [m]	Max. root moment QBF [kNm/blade]	Five bladed CP propellers: Lips hub designation	Hub diam. Dn [m]	Max. root moment QBF [kNm/blade]
4C07	0.7	27			
08	0.8	40			
09	0.8	57			
10	1.0	78			
4C11	1.1	105			
12	1.2	135	5C12	1.2	90
13	1.3	170	13	1.3	115
14	1.4	215	14	1.4	145
15	1.5	270	15	1.5	175
4C16	1.6	320	5C16	1.6	215
17	1.7	380	17	1.7	260
18	1.8	460			
19	1.9	530			
20	2.0	620	5C20	2.0	420
4C21	2.1	720	5C21	2.1	485
22	2.2	830	22	2.2	560
23	2.3	940	23	2.3	640
24	2.4	1070	24	2.4	725

$QBF = (f(D_h)$

$\text{Load-factor} = \frac{QB1}{QBF} \times 100$

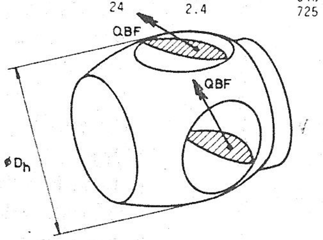


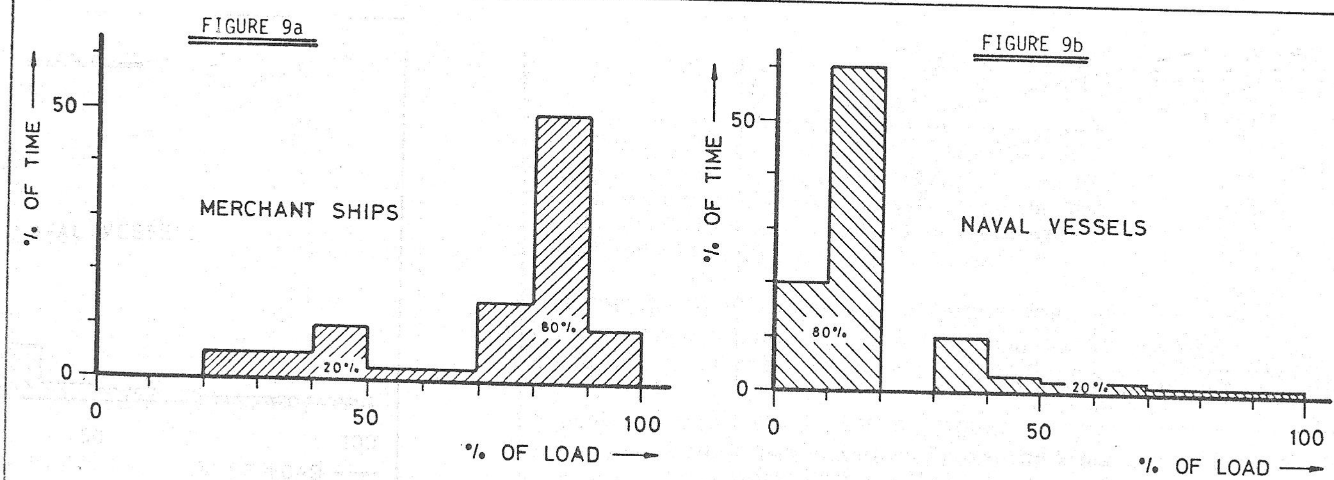
Figure 8. Each Hub has a maximum Blade Root Moment being the Limit of Load to be introduced to the Blade Port.

Load Limit for CP Propellers

The maximum propeller hub load, as defined in the previous paragraph, must be seen as the limit for Merchant Ship propellers having to fulfill free running duties. A Merchant Ship operates most of its time close to full power conditions. A typical mission profile of a Merchant Ship is shown in Figure 9a. Considering the present state-of-the-art, it is recommended that a *load factor* of 100 percent be the maximum value for these propellers.

For ships that have to operate frequently at conditions close to bollard circumstances ($J = 0$), such as Tugs, Trawlers, Dredgers, an extra safety margin must be taken. Also Ice Class propellers must have an extra safety margin since the propeller hub must be able to withstand the higher ice torque during ice-crushing and milling. This means that the load factor has to remain far below 100 percent, largely depending on the class of ice-strengthening required.

Naval Ships have a mission profile completely different from Merchant Ships as is shown in Figure 9b. Most naval vessels operate only a small portion of their life at full power and 80 percent of their life time at less than 20 percent power. Since this largely influences fatigue life, a higher hub load factor may be adopted. Based upon LIPS' knowledge and experience, it is recommended that the limit for safe operation of Navy propellers be 150 percent.



	HUB LOAD FACTOR
MERCHANT SHIPS	LF ≤ 100%
ICE CLASS SHIPS	LF ≪100%
NAVAL VESSELS	LF ≤ 150%

Figure 9. Hub Load in proportion to Mission Profile. Merchant (9a) and Naval (9b) Ships have different mission spectra which leads to the adoption of different maximum allowable Hub Load Factor Values as shown.

Figure 10 shows the statistical distribution of the hub load factors of 338 controllable pitch propellers, all of which are of the C-Ro/Vo type. (Figure 1) Merchant Ship propellers are found in a band close below 100 percent. The scatter is caused by the fact that hubs are only available in discrete sizes. From one hub-size to another there is a step resulting in approximately 30 percent load factor difference which explains why the width of the band is from 70 to 100 percent. It can be seen that there are only a few exceptions outside this range.

Ice Class propellers in Figure 10 have a hub load factor distinctly below 100 percent and are recognized in *three* clusters: heavy, medium, and light.

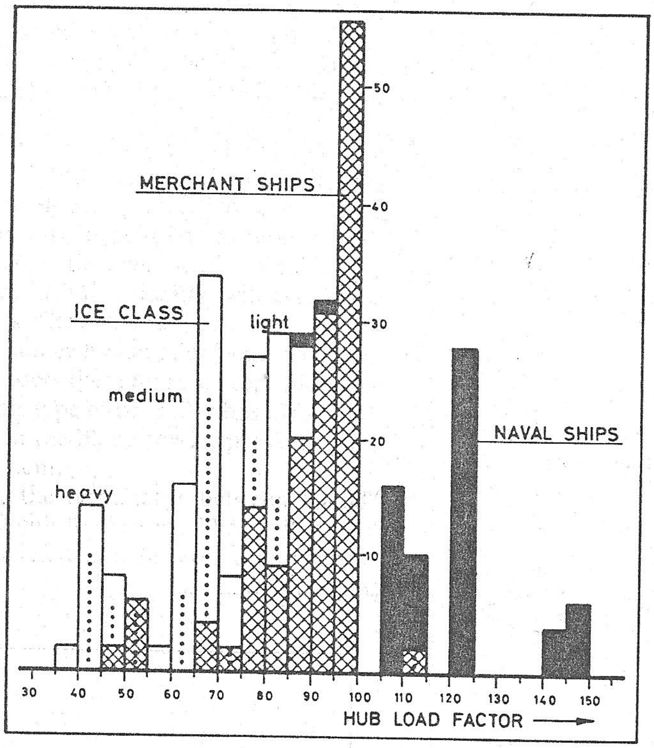


Figure 10. Statistical distribution of Hub Load Factors of 338 LIPS CP Propellers: 117 Ice Class, 156 Merchant Ship, and 65 Naval Ship propellers.

The Naval Ships area is characterized by more scatter in the hub load factor values. This must be explained from the fact that *two* different methods of blade design technique for high-speed propellers are in use. One, applying very fine tip sections, allows for a high load factor of the hub. Propeller blades, designed according to the other method with heavy tip sections, require a lower hub load factor because of the greater risk of damage to the blade port and hub under shocks due to explosion or grounding.

The hub load factor method has already been in use for more than ten years at LIPS as an aid in the selection of CP propeller hub size. Detailed calculation of the stresses in the hub, as indicated in Figure 6, occasionally can lead to the selection of a smaller or larger hub. The load factor scale has proved to be one of the most effective *failure prevention* tools.

Correlation of Failures and Load Factor Value

Hub failures — such as cracks and fractures in blade foot, blade bolts, blade carriers, or hub parts — fortunately have been kept out of LIPS' experience, thanks to hub size selection by means of the load factor philosophy.

Nevertheless, a number of CP propeller failures are known. These failures are a confirmation of the foregoing theory. TABLE 1 shows the field experience of 20 controllable pitch propellers — 13 Merchant and 7 Navy — in relation to their hub load factor. It can be recognized that the frequency of failures increases at higher hub load factors: BELOW 100 PERCENT — no hub defects, well loaded; 100 TO 110 PERCENT — occasional defects, critical; and BEYOND 110 PERCENT — general defects, overloaded.

This relates to Merchant Ship propellers. See also Figure 11. Cases 6 and 7 could lead to the idea that the load factor is an extremely sharp tool. However, this is *not* true. A probability distribution has to be expected: not all objects subjected to the *same* load will fail at the *same* time. Differences in hub principle will also cause dispersion in the results. Cases 6 and 7 concern a type of CP propeller having a somewhat smaller blade foot than other makes, which renders them more susceptible in the 100 percent region. This type of propeller has also a sudden change of section in the blade carrier presenting the weakest spot in the system.

In the Navy sector, the terminator between "good" and "bad" is more difficult to observe. The available experience is rather restricted and failures are sporadic. Case 19, *Spruance* becomes marginal, and *Barbey* is

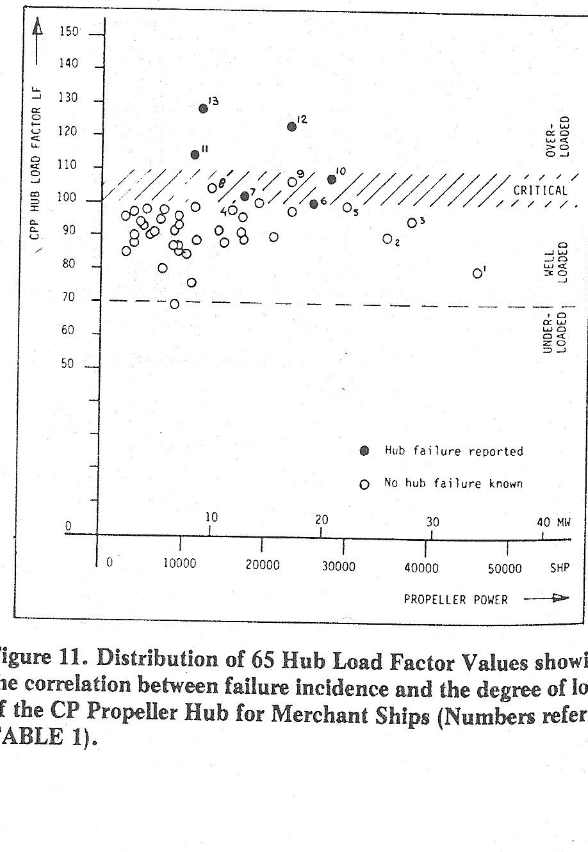


Figure 11. Distribution of 65 Hub Load Factor Values showing the correlation between failure incidence and the degree of load of the CP Propeller Hub for Merchant Ships (Numbers refer to TABLE 1).

TABLE 1

MERCHANT SHIPS

Name of Ship	Propeller Power [SHP]	Hub Size [mm]	Hub Load Factor	Field Experience
1. Australian Emblem 1975	46,000	2,400	80%	No hub defects known.
2. Asialiner 1972	35,000	1,940	90%	6 Years; no hub defects.
3. Asiafreighter 1972	38,000	1,940	95%	5½ Years; no hub defects.
4. Trident Amsterdam 1969	16,000	1,680	98%	8 Years; no hub defects. 4 ships
5. Euroliner 1971	30,000	1,830	99%	7 Years; no hub defects.
6. Australian Enterprise 1969	26,000	1,750	100%	Broken bolts and blade foot.
7. Nedlloyd Nagoya 1971	17,500	1,700	102%	5 Years; broken blade bolts, fatigue in blade carriers, wear in brgs. 4 ships same.
8. Cunard Adventurer 1971	13,500	1,150	104%	No hub defects known.
9. Conoco Britannia 1972	23,200	2,000	106%	No hub defects known.
10. Nihon 1972	28,200	1,940	107%	Cracks in blade foot.
11. Fernland 1969	11,400	1,440	114%	Cracks in hub bridge.
12. Snow Flake 1973	23,000	1,700	123%	3 Years; loss of oil, broken bolts, blade carrier fatigue, wear in brgs. 8 ships same.
13. Suffren 1966	14,300	1,440	128%	8 Years; broken blade foot and bolts. 2 ships.
NAVAL & COAST GUARD VESSELS				
14. Hamilton Class 715 1967	20,000	1,250	108%	No hub defects known.
15. DDH 280 1972	26,500	1,410	114%	5 Years; no hub defects. 4 ships
16. Hamilton Class 719 1968	20,000	1,230	115%	No hub defects known.
17. Lupo Class 1977	25,000	1,200	122%	No hub defects.
18. Tromp Class 1975	21,000	1,200	144%	2 Years; no hub defects. 2 ships
19. Spruance (DD-963) 1975	40,000	1,555	172%	Minor defects; no "Crash-Stop."
20. Barbey (FF-1088) 1973	35,000	1,320	235%	Broken blade bolts; loss of 5 blades in "Crash-Stop."

shown to have an extraordinarily high hub load factor. Failure of this propeller would not have been an unforseen event had the load factor scale been consulted (See Figure 12). Therefore, we do not support the statement that the failure would *not* have been predicted by any of the current design techniques [15].

From the investigation program conducted by the Naval Ship Engineering Center (NAVSEC) it is known that the failure is believed to have been caused by material deficiencies and design defects. No thought was given to the idea that regardless of material and design defects, the *lack* of *adequate quantity* of material already would *doom the propeller to failure.* Moreover the propeller is characterized by a complete disharmony in the various stress areas of the hub. This indicates a foreign input in the hub design process, as shown in TABLE 2, where the stresses in various hub parts are presented, calculated according to a standard LIPS method.

TABLE 2

	Barbey stresses		(Allowable)
Blade root	6,930	underloaded	(9,000) N/cm²
Blade bolts	21,500		(9,000) N/cm²
Crank ring	15,950	seriously overloaded	(6,000) N/cm²
Hub collar	11,600		(6,000) N/cm²

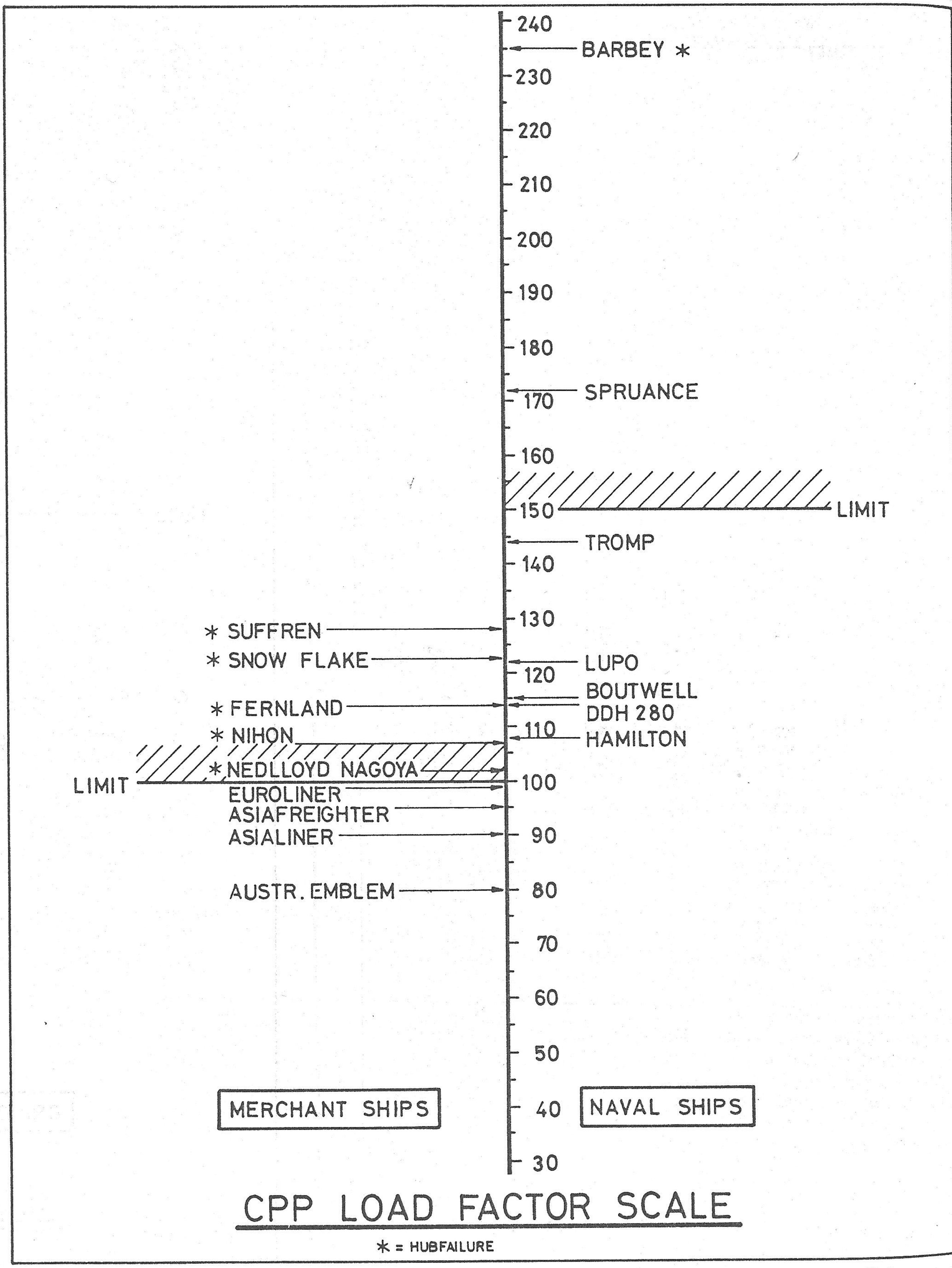


Figure 12. The Hub Load Factor Scale — A Useful Tool to Detect the Degree of Load of each CP Propeller Hub.

A more general comparison is given in TABLE 3. Obviously the blades have been safeguarded at the cost of the surrounding areas, which is not in agreement with the principle of harmony mentioned in the introduction of this paper. It is also in contrast to the well-known pyramidal strength concept as generally accepted for CP propellers. Blade bolts, crank ring, and hub collar of the *Barbey* propeller have been loaded with extreme stresses, 2 to 2.5 times as high as allowed. Irrespective of the material properties these parts would have failed in fatigue sooner or later. The fact that the *Barbey* propeller failed so soon, in a relatively light "Crash-Stop" condition, becomes clear when comparing the *Barbey* hub details with those of the HAMILTON Class, fitted with a similar but smaller hub of the same manufacturer [16]. It is seen that in deviating from the standard design some specious modifications have been introduced — increased fillet radii in the blade carriers and shorter blade bolts for underway blade replacement as examples — which in fact have badly aggravated the susceptibility to failure.

If a larger hub size of normal unmodified design had been adopted, the propeller would not have failed at all. It is clear that the *Barbey* propeller failed due to an accumulation of faulty design decisions. The *Barbey* propeller failure, therefore, may not be claimed as a fair reason to be concerned about the suitability of high-powered CP propellers on Naval Ships in general.

Influence of Hub Size on Hydrodynamic Performance

We have seen that increasing the hub size leads to a reduction of the load factor and thus to improvement of the reliability of the propeller. There is, in fact, a minimum hub size required for safe operation of the propeller. For highly-powered propellers this means the necessity of a hub-diameter ratio which is beyond the traditional horizon of hydrodynamicists. It must be emphasized that once the decision has been taken in favor of a CP propeller, safety of the propeller cannot be considered as being less important than the hydrodynamic performance. With a larger hub not only an increased weight but also a certain loss in efficiency must be accepted which does not mean that the propeller efficiency has run completely out of control.

The loss in efficiency due to hub-diameter ratios outside the normal range has been investigated. KONING [17], assuming that a larger hub results in a loss of disc area, recommended the following approximation for a hub ratio larger than 0.20:

$$\text{Efficiency correction } \frac{\eta_0'}{\eta_0} = \frac{1 - (D_h/D)^2}{0.96}$$

The factor 0.96 in the formula is based on a smallest hub-diameter ratio of 0.20 which means that *four* per-

TABLE 3

COMPARISON OF STRESSES

In the following TABLE a comparison is made of the stresses in the main parts of four different types of CP propellers for war-ships.

Name of Ship Class		*Spruance* DD-963	— (DD 963)	*Barbey* DE-1088	*Tromp* GM Frigate	Allowable
Propeller diameter	[inch]*	204	204	180	165.4	
Hub designation		155.5 S/5	LIPS 5C16N	D132/5	LIPS 4C12N	
Hub diameter	[inch]*	61.2	63	52	47.2	
Power	[SHP]*	40,000	40,000	35,000	20,000	
Shaft speed	[RPM]*	168	168	240	230	
Thrust	[lbs]*	278,000	278,000	273,000	160,000	
Blade root stress $_1$	[psi]*	11,700	11,700	9,850	11,100	12,800
Blade bolt stress $_3$	[psi]*	14,100	13,000	30,700	11,300	12,800
Mean bearing pressure $_0$	[psi]*	4,300	3,550	5,650	3,000	3,550
Stress in flange of crankpin ring (blade carrier) $_4$	[psi]*	15,600	8,500	22,700	8,050	8,500
Stress in bearing ring (hub collar) $_5$	[psi]*	(26,300)	8,300	16,500	6,300	8,500
General hub load factor LF	[%]*	172	155	235	144	150

NOTE: *The Units in this TABLE are non-metric in deviation from the APPENDIX.

cent of the disc area always is considered to be lost. According to BAKER [18] the process of hub losses is more complex. Three separate effects must be taken into account: 1) a higher flow velocity around the hub, 2) increased frictional rub and diminution of the effective thrust capacity, and 3) aspect ratio of the blades. BAKER carried out tests with hub-diameter ratios 0.192, 0.25, 0.317, and 0.395. Corresponding propeller efficiencies were measured and plotted in a K_T — J Diagram.

Figure 13 shows that the results obtained by BAKER are a conformation of the KONING rule, KONING being somewhat pessimistic for hub-diameter ratios in excess of 0.30.

When applying this to the case of the *Barbey,* we find that an increase of hub size from 1,320mm to 1,500mm would reduce the load factor from 235% to 147%. This would have guaranteed safe operation of the propeller. The hub-diameter ratio would be raised from 0.290 to 0.328, and according to KONING, when compared to the efficiency of the smaller hub that failed, the correction on the efficiency would amount to:

$$\frac{\eta_0'}{\eta_0} = \frac{1 - 0.328^2}{1 - 0.290^2} = 0.974$$

The loss in speed consequently would be less than 0.15 knots.

This shows clearly that only a small amount of efficiency loss has to be accepted to ensure fully reliable and safe operation of the controllable pitch propeller without any restraint in use or mission — which clearly advocates a "trade-off" in favor of safety.

CONCLUSIONS

1) Propeller power magnitude and propeller power density provide two useful parameters with which the

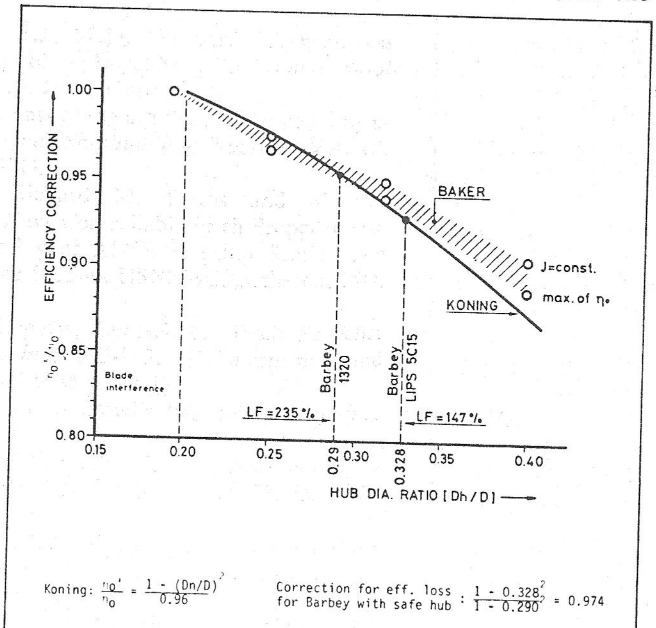


Figure 13. Propeller Efficiency Losses as a function of the Hub Diameter Ratio.

technical degree of difficulty of each propeller clearly can be recognized.

2) A Propeller Hub Load Scale provides an important design tool to ensure reliable and safe operation of controllable pitch propellers without restrictions.

3) Analysis of hub failures of controllable pitch propellers in operation, according to a LIPS engineering standard, reveals a clear condition of overload of the propeller hubs in all cases investigated.

4) Since a minimum hub size is required for safe operation of a controllable pitch propeller, hydrodynamic considerations should *not* lead to the adoption of a mechanically inadequate hub size.

5) Further development of controllable pitch propellers to a higher level of power is considered to be feasible provided that safety of the propeller is not made secondary to performance, cost, or weight.

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APPENDIX

NOMENCLATURE			UNIT
A	=	blade root section	[m²]
a_i	=	moment arm (i = 1,2,3...)	[m]
C_i	=	centrifugal force per propeller blade (i = 1,2,3...)	[N]
C_T	=	hydrodynamic propeller load coefficient	
D	=	propeller diameter	[m]
D_h	=	hub diameter	[m]
D_f	=	blade foot diameter	[m]
E_i	=	input effort	
J	=	advance ratio	
L	=	propeller power density	[J/m²sec]
LF	=	hub load factor	
n	=	rotational propeller speed	[sec⁻¹]
P	=	propeller power magnitude	[W]
T	=	propeller thrust force per blade	[N]
t	=	time	[sec]
TG	=	tangential force per blade	[N]
V_e	=	entrance velocity	[m/sec]
QBF	=	max. blade root moment	[Nm]
QB_i	=	bending moment (i = 1,2,3...)	[Nm]
Q_E	=	engine torque in propeller shaft	[Nm]
R	=	resulting loading force (R = T + TG)	[N]
z	=	number of blades	
Z_i	=	section modulus (i = 1,2,3...)	[m³]
$\eta 0$	=	open water efficiency	
$\eta 0'$	=	open water efficiency corrected for larger hubs	
$\sigma 0$	=	bearing pressure	[N/cm²]
$\sigma 1$	=	blade root stress	[N/cm²]
$\sigma 2$	=	blade foot stress	[N/cm²]
$\sigma 3$	=	stress in blade bolts	[N/cm²]
$\sigma 4$	=	stress in blade carrier flange	[N/cm²]
$\sigma 5$	=	stress in hub collar	[N/cm²]