ON THE STRENGTH AND VIBRATION ANALYSIS OF A 8000 DWT CONTAINER SHIP PROPELLER

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ABSTRACT

This paper presents the investigation on the strength and vibration analysis of a propeller design for a 8000 deadweight container ship which is based on the B_p - δ series. Such analysis is important since the significant role of the propeller to convert the greater part of the power from an engine into thrust force to propel a ship and ensure that resonants of propeller induced vibration does not coincide with the main hull vibration. Various methods of analysis is studied in order to compare the results obtained and to justify that such method comply with the classification societies.

1.0 INTRODUCTION

A marine propeller is a propulsion device, which converts the greater part of the power from an engine into thrust force to propel a ship. The propeller is the most common form of marine propulsion device. Therefore, it is essential to design the propeller correctly. Nevertheless, two major criteria which often being neglected by naval architects when designing the propeller is the strength and vibration aspect.

The strength of a marine propeller depends significantly on the blade thickness. The determination of the blade thickness of a propeller is an important aspect of the design. Its affects primarily the resistance of the propeller to failure and damage. Its affect quite substantially the inertia, weight and thus the price. It affects to a relative minor degree the efficiency, power absorption and cavitation characteristics. Weight, price, cavitation and efficiency are all favourably influenced by reduced thickness and thus the highest allowable mean stress is used consistent with the achievement of all objectives.

In any ship, the major sources of vibration excitation are usually the main engine and the propulsion train. Where the main engine is of a rotating type, the propeller will probably be responsible for any significant vibration of the main hull. Therefore, it is important to assess the amplitudes of vibration excited by the propeller. Though these vibrations cannot be eliminated but it can be minimised. To achieve this, ensure that resonants of propeller induced vibration does not coincide with the main hull vibration.

This paper will discuss the strength and vibration analysis of the propeller using various theories and compared those values that has been suggested by some classification societies.

2.0 BASIS SHIP AND MAIN PROPELLER DIMENSIONS

The design calculation is based on a 8000 dwt Container Ship as shown in Fig. 1. The principle particular of the ship are as follows:

<u>Main Particulars of Ship</u>

Length Overall	L_{OA}	123.500 m
Length between perpendicular	L_{BP}	115.450 m
Breadth moulded	B	20.800 m

Depth moulded	D_{MLD}	10.800 m
Designed Draught	H	6.500 m
Loaded displacement	D	11151.00 tonnes
Service Speed	V	15.00 knots
Propeller Diameter	D	3.90 m
Number of Propeller		1
Number of blade	Z	4

Coefficients/Ratios

Block coefficient	C_B	0.705
Prismatic Coefficient	C_P	0.745
Midship coefficient	C_M	0.947
Waterplane coefficient	C_{WL}	0.831
LCB (% aft of midship)		0.932
LCF (% aft of midship)		2.075

Main Particulars of Propeller

Propeller diameter	Dia	3.900 m
Propeller boss diameter	Db	0.780 m
Blade area ratio	BAR	0.740 m
Angle of rake	Rake	0.0 degrees
Length of blade section at 0.6R		1.580 m
Max. thickness at centre of shaft		0.176 m

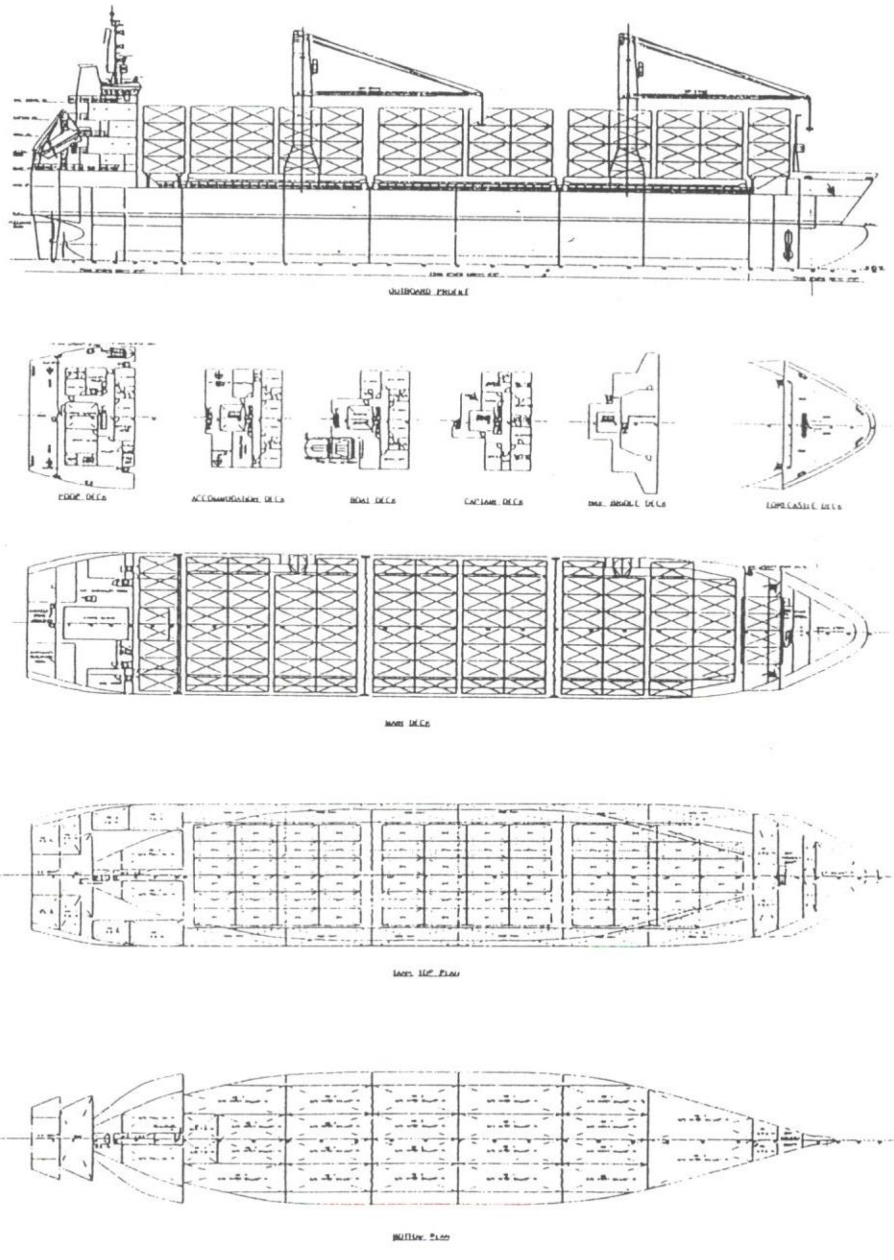

Fig. 1 General Arrangement of 8000 Dwt Container Ship

3.0 PROPELLER BLADE STRENGTH CALCULATIONS

The calculation of the maximum mean stress of the propeller is usually based on the the maximum tensile stress on the blade face at a prescribed position of the propeller radius. However, prior to these calculations, it is necessary to first determine the geometry of the propeller.

3.1 Propeller Dimensions

The offset for a 4 blade B Series propeller are given in Tables 1 and 2. The result of he designed propeller using Series charts and polynomial equation are shown in Tables 3 and 4. A propeller drawing, based on Series charts, is shown in Fig. 2.

In making estimates for weight, moment of inertia and blade stresses, values of the geometrical properties of blade sections are required. These comprise the ection area (A_S), distance of centroid from face chord $(\bar{y})$, distance of centroid rom leading edge (h), moment inertia about axis parallel to face chord (I_N) and moment of inertia about axis normal to face chord (I_P), as shown in Fig. 3. The ormulation of these geometrical properties are given as by:

$$A_S = 0.700ct \tag{1}$$

$$\bar{y} = 0.463t \tag{2}$$

$$\bar{h} = 0.455c \tag{3}$$

$$I_N = 0.042ct^2 \tag{4}$$

$$I_P = 0.040c^3t \tag{5}$$

Applying the above equations, the calculation of the propeller blade weight shown in Table 5 for series charts.

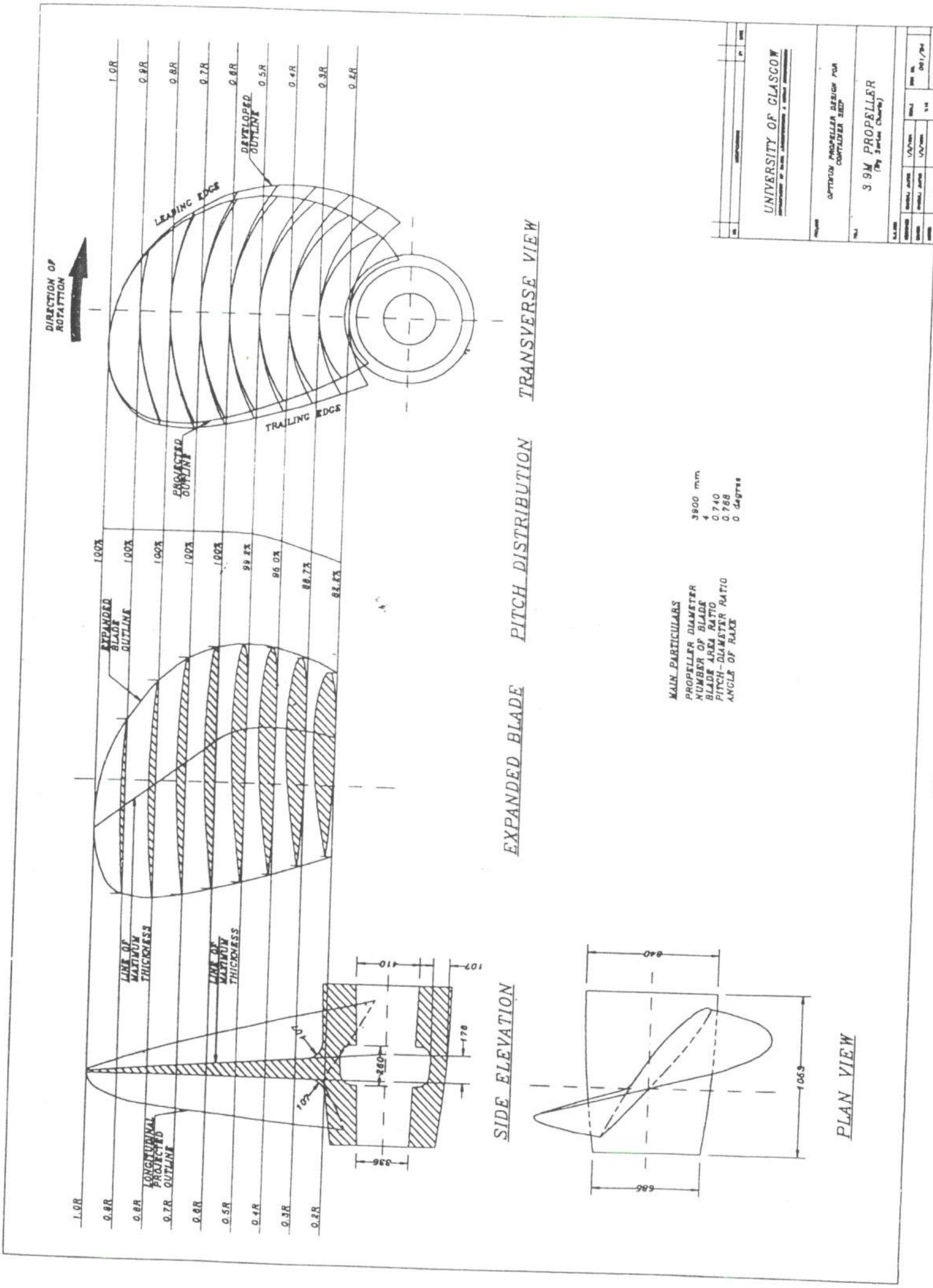

Fig. 2 Propeller Drawing

Table 1 Table of Ordinates of the B – Series Propeller (Distance of Ordinates from Maximum Thickness)

	From maximum thickness to trailing edge (%)					From maximum thickness to leading edge (%)						
r/R	100.00	80.00	60.00	40.00	20.00	20.00	40.00	60.00	80.00	90.00	95.00	100.00
Ordinates for the back												
0.20	-	53.35	72.65	86.90	96.45	98.60	94.50	87.00	74.40	64.35	56.95	-
0.30	-	50.95	71.60	86.80	96.80	98.40	94.00	85.80	72.50	62.65	54.90	-
0.40	-	47.70	70.25	86.55	97.00	98.20	93.25	84.30	70.40	60.15	52.20	-
0.50	-	43.40	68.40	86.10	96.95	98.10	92.40	82.30	67.70	56.80	48.60	-
0.60	-	40.20	67.15	85.40	96.80	98.10	91.25	79.35	63.60	52.20	43.35	-
0.70	-	39.40	66.90	84.90	96.65	97.60	88.80	74.90	57.00	44.20	35.00	-
0.80	-	40.95	67.80	85.30	96.70	97.00	85.30	68.70	48.25	34.55	25.45	-
0.90	-	45.15	70.00	87.00	97.00	97.00	87.00	70.00	45.15	30.10	22.00	-
0.95	-	44.80	72.00	88.00	97.20	97.20	88.80	72.00	44.80	29.50	21.60	-
Ordinates for the face												
0.20	30.00	18.20	10.90	5.45	1.55	0.45	2.30	5.90	13.45	20.30	26.20	40.00
0.30	25.35	12.20	5.80	1.70	-	0.05	1.30	4.60	10.85	16.55	22.20	37.55
0.40	17.85	6.20	1.50	-	-	-	0.30	2.65	7.80	12.50	17.90	34.50
0.50	9.70	1.75	-	-	-	-	-	0.70	4.30	8.45	13.30	30.40
0.60	3.10	-	-	-	-	-	-	-	0.80	4.45	8.40	24.50
0.70	-	-	-	-	-	-	-	-	-	0.40	2.45	16.50
0.80	-	-	-	-	-	-	-	-	-	-	-	7.40

Note: The percentages of the ordinates relate to the maximum thickness of the corresponding sections, the curve of thickness being assumed to be rectilinear. The connecting lines of the points at which set-back and back intersect, cut each other at 0.15R.

Table 2 Dimensions of the Four-Bladed Screws, Types B4, B4.55 and B4.70

	r/R	0.20	0.30	0.40	0.50	0.60	0.70	0.80	0.90	1.00	
Length of the blade sections as percentages of the maximum length of the blade sections at 0.6R	From centre line to trailing edge	29.18	33.32	37.30	40.78	43.92	46.68	48.37	47.00	20.14	Length of blade section at 0.6R = 0.2187D if Ae/Ao = 0.40
	From centre line to leading edge	46.90	52.64	56.32	57.60	56.08	51.40	41.63	25.35		Generally C0.6 = 0.5475x(Ae/Ao)
	Total length	76.08	85.96	93.62	98.38	100.00	98.08	90.00	72.35		
Blade thickness ratio as percentage of the diameter		3.66	3.24	2.82	2.40	1.98	1.56	1.14	0.72	0.30	Maximum thickness at centre of shaft = 0.045D
Distance of maximum thickness from leading edge as percentage of length of the sections		35.00	35.00	35.00	35.50	38.90	44.30	47.90	50.00		

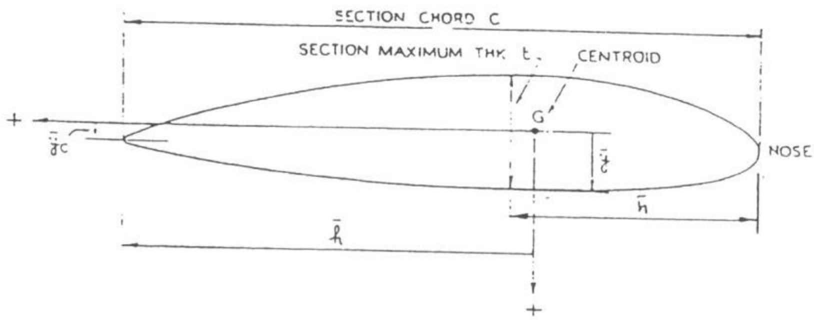

Fig. 3 Geometrical Properties of the Blade Sections

Table 3 Dimensions of 4-Bladed Screw

	r/R	0.20	0.25	0.30	0.40	0.50	0.60	0.70	0.80	0.90	0.95	1.00
Length of the blade sections	From centre line to trailing edge	461	494	526	589	644	694	738	764	743	319	318
	From centre line to leading edge	741	786	832	890	910	886	812	658	401	-	-
	Total length	1202	1280	1358	1479	1554	1580	1550	1422	1143	-	-
Blade thickness		143	135	126	110	94	77	61	44	29	12	12
Distance of maximum thickness from leading edge		421	448	475	518	552	615	687	681	572	-	-

Note All dimensions are in millimetres

Table 4 Table of Ordinates of the B – Series Propeller (Distance of Ordinates from Maximum Thickness)

From maximum thickness to trailing edge (mm)						From maximum thickness to leading edge (mm)							
r/R	100	80	60	40	20	20	40	60	80	90	95	100	r/R
Ordinates for the back													
0.20	-	76	104	124	138	141	135	124	106	92	81	-	0.20
0.30	-	64	90	110	122	124	119	108	92	79	69	-	0.30
0.40	-	52	77	95	107	108	103	93	77	66	57	-	0.40
0.50	-	41	64	81	91	92	86	77	63	53	45	-	0.50
0.60	-	31	52	66	75	76	70	61	49	40	33	-	0.60
0.70	-	24	41	52	59	59	54	46	35	27	21	-	0.70
0.80	-	18	30	38	43	43	38	31	21	15	11	-	0.80
0.90	-	13	20	25	28	28	25	20	13	9	6	-	0.90
0.95	-	5	9	11	12	12	11	9	5	4	3	-	0.95
Ordinates for the face													
0.20	43	26	16	8	2	1	3	8	19	29	37	57	0.20
0.30	32	15	7	2	-	0	2	6	14	21	28	47	0.30
0.40	20	7	2	-	-	-	0	3	9	14	20	38	0.40
0.50	2	9	-	-	-	-	-	1	4	8	12	28	0.50
0.60	4	-	-	-	-	-	-	-	1	3	6	19	0.60
0.70	-	-	-	-	-	-	-	-	-	0	1	10	0.70
0.80	-	-	-	-	-	-	-	-	-	-	-	3	0.80

Note All dimensions are in millimetres

Table 5 Determination of Blade Centroid and Weight

Main particulars

Propeller diameter	D	3.9 m	P/D ratio		0.768
Rotational speed	N	210 rpm	Material density	Density	8300 kg/m^3
Number of blade	Z	4	Type of material		Manganese Bronze

r/R	thk	chord	Section	S.M.	lever	f(V)	f(MV)	Distance of c.g from		2nd moment of area	
	t	c	Area					Face chord	Lead edge	parallell to	normal to
								y	h	face chord	face chord
	m	m	m^2					(m)	(m)	(m^4)	(m^4)
1.00	0.012	0.000	0.000	1	1.950	0.000	0.000	0.006	0.000	0.000E+00	0.000E+00
0.90	0.029	1.143	0.023	4	1.755	0.093	0.163	0.013	0.520	1.171E-06	1.732E-03
0.80	0.044	1.422	0.044	2	1.560	0.088	0.137	0.020	0.647	5.088E-06	5.061E-03
0.70	0.061	1.550	0.066	4	1.365	0.265	0.361	0.028	0.705	1.478E-05	9.086E-03
0.60	0.077	1.580	0.085	2	1.170	0.170	0.199	0.036	0.719	3.030E-05	1.215E-02
0.50	0.094	1.554	0.102	4	0.975	0.409	0.399	0.044	0.707	5.421E-05	1.411E-02
0.40	0.110	1.479	0.114	2	0.780	0.228	0.178	0.051	0.673	8.268E-05	1.423E-02
0.30	0.126	1.358	0.120	4	0.585	0.479	0.280	0.058	0.618	1.141E-04	1.262E-02
0.20	0.143	1.202	0.120	1	0.390	0.120	0.047	0.066	0.547	1.476E-04	9.934E-03
						1.852	1.764				

Volume per blade	**0.120 m^3**
Weight per blade	**1 tunnes**
Moment of volume	**0.115 m^4**
Centroid	**0.953 m**

3.2 Blade Strength Calculations

The blade strength calculations are done based on *Taylor*'s method, *Keryser* and *Arnoldus*'s method and compared with those provided by the classification societies. It should be borne that the designed propeller has zero rake. The propeller is assumed to have sufficient hull clearances. Therefore, there is no necessity to have a rake angle.

3.2.1 *Taylor's Method*

D.W. Taylor derived formula of determining the maximum compressive stress and tensile stress of a propeller blade. The strength calculation is carried out at section 0.2R. The maximum compressive stress occurring at the position of maximum thickness is:

$$S_C = \frac{C_1 P_1}{ND^3} \frac{1}{cb\tau^2} \tag{6}$$

The values of coefficient C_1 can be read off from Fig. 4. The product cb can also be given as:

$$cb = \frac{l_{0.2R}}{l_m} \frac{l_m}{D} = \frac{l_{0.2R}}{D} \tag{7}$$

The maximum tensile stress which in general, is smaller in absolute value than the maximum compressive stress and is given by;

$$S_T = S_C \left(0.666 + 1.17 \frac{L}{C} \frac{s}{l} \right) \tag{8}$$

The factor $1.17 \frac{L}{C} \frac{s}{l}$ can be obtained from Table 6.

Equations 6 and 8 for calculating the maximum stress take no account of those due to centrifugal force, since the rake angle is zero. If the propeller is raked aft, these extra stresses may become considerable magnitude and cannot be ignored. According to *Taylor*, the extra compressive stress due to centrifugal action is greatest at the position of maximum ordinate on the back of the blade element and the greatest extra tensile stress occurs at trailing edge. The computation of these extra compressive and tensile stresses due to the centrifugal force is mentioned in [1].

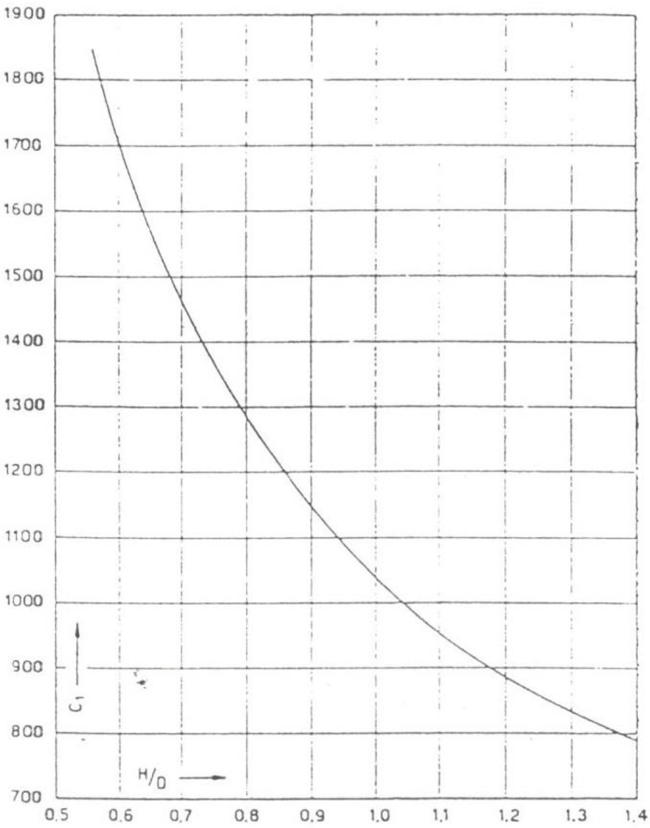

Fig. 4 Relation Between Coefficient C_1 and Pitch Ratio [19]

Table 6 Values of 1.17 *L/C* for Blade Elements at 0.2*R* [19]

Pitch Ratio	0.6	0.7	0.8	0.9	1.0	1.1	1.2	1.3
1.17L/C	0.650	0.710	0.754	0.784	0.804	0.817	0.823	0.20

Table 7 Summary of Stresses at 0.2*R* using *Taylor*'s Method

Compressive stress		Tensile stress	
Kg/cm^2	Lb/in^2	Kg/cm^2	Lb/in^2
515	7327	388	5524

Applying Eqs. 6 and 8, summary of the maximum stress occurs at $0.20R$ is shown in Table 7. The limitation in applying this method is that the maximum pitch diameter ratio is 1.4 for determining C_1 value and 1.3 in determining $1.17L/C$ value.

3.2.2 *Keyser & Aarnoldus's Method*

In reference 11, the moments M_{bs} and M_{bt}, as shown in Fig. 4, may be written as:

$$M_{bs} = f_s S_Z R \tag{9}$$

$$M_{bt} = f_t T_Z R \tag{10}$$

Hence,

$$M_b = M_{bs} \cos\alpha + M_{bt} \sin\alpha \tag{11}$$

The trust and torque force distributions are largely governed by the radial wake distribution behind the vessel and by thc pitch distribution of the propeller. These can be categories as follows:

a. Propellers with constant pitch working in a homogeneous velocity field.
b. Propellers with constant pitch working in unequal velocity field.
c. Propellers with variable pitch working in an unequal velocity field (pitch reduction approximately 20% towards the boss)

For a twin screw vessels, it fall into the first category. Whilst, single screw vessels fall into the last two categories. The coefficients of f_s and f_t in Eqns. 9 and 10 is given in Table 8 for the three categories.

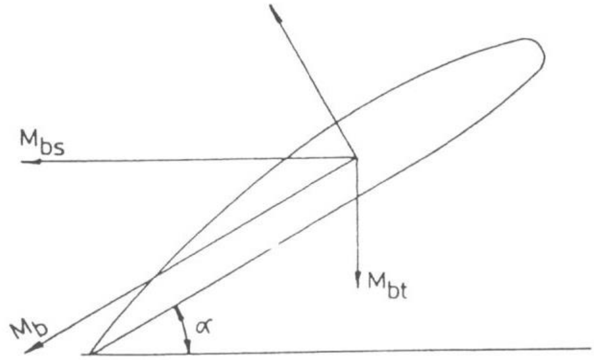

Fig. 5 Decomposition of Bending Moments

Table 8 Factors for the Determination of Bending Moments

r/R	Homogeneous field Constant pitch		Variable field Variable pitch		Variable field Constant pitch	
	f_s	f_t	f_s	f_t	f_s	f_t
0.2	0.481	0.423	0.464	0.406	0.444	0.378
0.3	0.384	0.326	0.364	0.309	0.348	0.283
0.4	0.291	0.238	0.273	0.223	0.257	0.202
0.5	0.205	0.164	0.191	0.149	0.176	0.136
0.6	0.130	0.103	0.120	0.0899	0.108	0.0830
0.7	0.0700	0.0555	0.0644	0.0464	0.0575	0.0440
0.8	0.0300	0.0235	0.0254	0.0182	0.0225	0.0175
0.9	0.0080	0.0060	0.0048	0.0032	0.0040	0.0030
$r_{f_{0.2}}$		0.623R		0.606R		0.578R

The compressive stress in a section of a propeller blade nearly always larger than the tensile stress. The calculation of the compressive stress is given by:

$$\sigma = \frac{M_b}{\alpha_{wd} s^2 l \cos^2 \varepsilon} \tag{12}$$

where α_{wd} is the coefficient of section modulus for maximum compressive stress or maximum tensile stress with a straight neutral axis or coefficient of section modulus for maximum tensile and compressive stress with s curved neutral line as shown in Fig. 6.

In applying Eqns. 9 and 12, summary of the maximum stresses occurs at 0.20R in shown in Table 9.

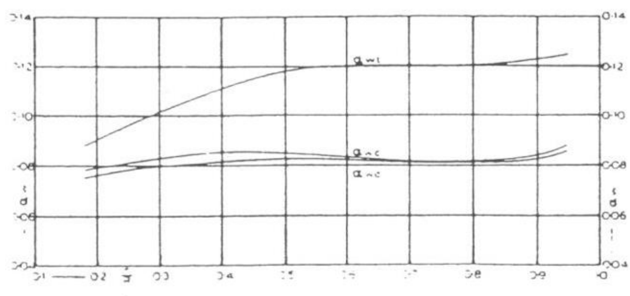

Fig. 6 α_w Coefficient [11]

Table 9 Stresses at 0.2R by *Keryer* and *Arnoldus*'s Method

Compressive stress		Tensile stress	
Kg/cm^2	Lb/in^2	Kg/cm^2	Lb/in^2
539	7664	455	6471

3.2.3 Classification Rules

Despite the calculations as mentioned in earlier sections, it is necessary to satisfy the Classification Society's requirement. In general, these requirements are based on minimum thickness at 0.25*R* and 0.6*R* for a fixed propeller. A summary of the formulation by the Classification Societies is shown in Table 10.

The results of minimum thickness of the designed propeller at 0.25R and 0.6R based in the formulations is given in Table 11.

Table 10 Extract of Classification Societies Guidance

Classification Society	Minimum thickness at 0.25R & 0.60R
Lloyd's Register of Shipping	$t_r = \dfrac{KCA}{EFULN} + 100\sqrt{\dfrac{3150\,MP}{EFRULN}}$
America Bureau of Shipping	$t_r = S\left[289\sqrt{\dfrac{AH}{C_n CRN}} \pm \left(\dfrac{C_n}{C_S}\right)\left(\dfrac{BK}{4C}\right)\right]$
Nippon Kaiji Kyokai	$t_r = \sqrt{\dfrac{K_1}{K_2}\dfrac{H}{ZNI}SW}$

Table 11 Comparison of Minimum Blade Thickness at 0.25*R* and 0.6*R*

Actual thickness		By LRS		By ABS		By NK	
0.25R	0.6R	0.25R	0.6R	0.25R	0.6R	0.25R	0.6R
135	77	133	68	84	76	99	54

4.0 PROPELLER INDUCED VIBRATION

Three distinguished methods will be used to determine the main hull vibrations which include *Todd*'s, *British Maritime Technology* (BMT) Design and *Erich Danckwardt*'s formulae.

4.1 *Todd*'s Formula

The first of these empirical formulae to be commonly used was that due to *Schlick* [18]. It is modified form of the ordinary beam formula;

$$N = \phi\sqrt{\frac{I}{\Delta L^3}} \tag{13}$$

where, $I = C_2 BD^3$

Todd proposed to replace I by the expression BD^3 and let the value of C_2 be merged in an overall coefficient,

$$N = \beta\sqrt{\frac{BD^3}{\Delta_1 L^3}} \tag{14}$$

An empirical formulae of Cargo Ship for 2 node vertical vibration is given as;

$$N_{2V} = 46750\sqrt{\frac{BD^3}{\Delta_1 L^3}} + 25 \tag{15}$$

where $\Delta_1 = \Delta\left(1.2 + \frac{B}{3H}\right)$

It should be highlighted that *Todd*'s formula is in imperial units.

For higher nodes of vertical and horizontal vibration, the relationship with 2 node vertical vibration is given by:

Vertical Vibration	Horizontal Vibration
$N_{3V} = 2N_{2V}$	$N_{2H} = 1.5N_{2V}$
$N_{4V} = 3N_{2V}$	$N_{3H} = 2N_{2H}$
$N_{5V} = 4N_{2V}$	$N_{4H} = 3N_{2H}$

Average limits of the above frequencies are as follows;

2 node vertical	1.5%	2 nodes horizontal	2.5%
3 node vertical	3.0%	3 nodes horizontal	9.0%
4 node vertical	8.0-10.0%	4 nodes horizontal	8.0-10.0%

4.2 *BMT* Design Formula

BMT design formula derived from measured natural frequency data for warship are represented in a graphical format with appropriate base function. An example of this graph is shown in Fig. 7, where only vertical node frequencies have been considered, as the horizontal node frequencies do not have significant effect on hull vibration. The base function is given by;

$$x = \sqrt{\frac{I_V}{\Delta_W L^3}} \tag{16}$$

where, I_V = second moment of area amidships

$$\Delta_{1V} = \Delta\left(1.2 + \frac{B}{3H}\right) \tag{17}$$

These formula are in metric units.

4.3 *Erich Danckardt* Formula

Erich Danckwardt presented an empirical formula for determining hull frequency based on *Todd*'s formula. He simplifies the formulation for determining vertical second moment of area with coefficient C_V, for different type of merchant ship. Two node frequency is given by:

$$N_{2V} = 200000\sqrt{\frac{I_V}{M_{\Delta 1} L^3}} + 28 \tag{18}$$

where, $I_V = C_V B D^3$

$C_V = 1.11 \, x \, 10^{-3}$ for container ship

$$M_{\Delta 1} = M_{\Delta} + \left(0.2 + \frac{1}{3}\frac{B}{H}\right) M_{\Delta} \tag{19}$$

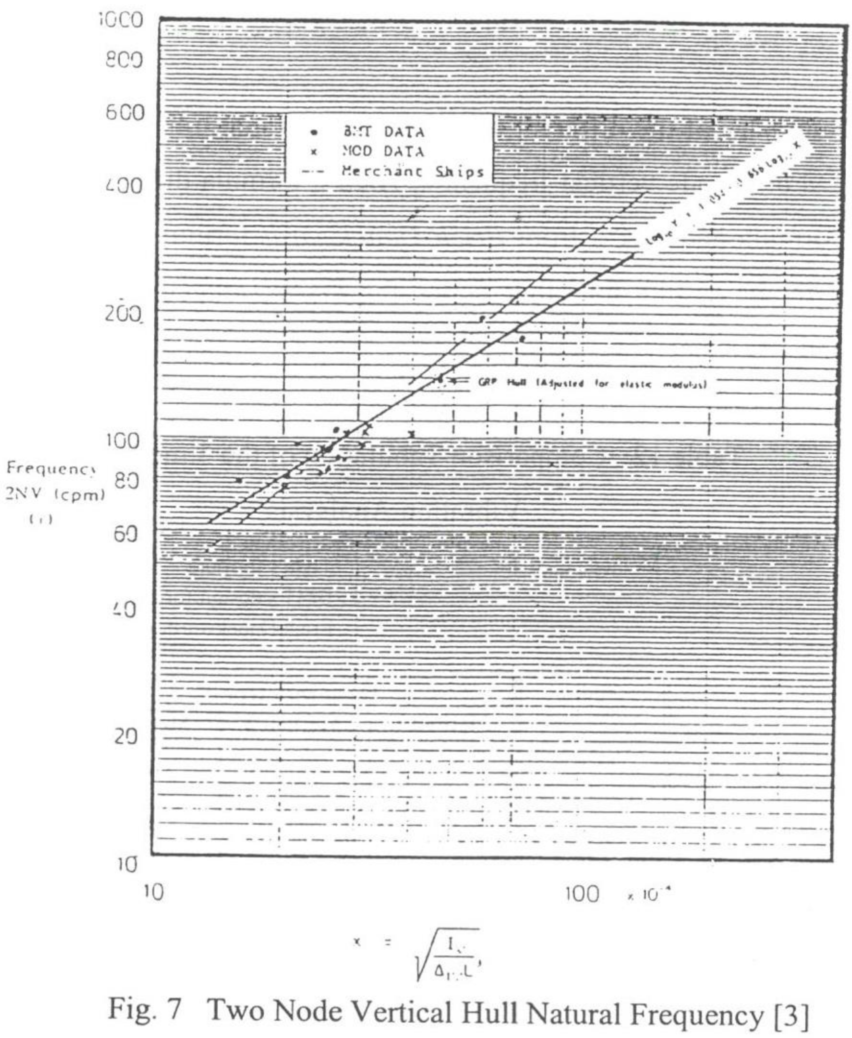

Fig. 7 Two Node Vertical Hull Natural Frequency [3]

Higher nodes of vertical and horizontal frequency are given as;

Vertical vibration	Horizontal vibration
$N_{3V} = 1.80 N_{2V}$	$N_{2H} = 1.50 N_{2V}$
$N_{4V} = 2.60 N_{2V}$	$N_{3H} = 3.10 N_{2V}$
$N_{5V} = 3.25 N_{2V}$	$N_{4H} = 4.75 N_{2V}$

Frequency bands for the above nodes are:

2 node	± 2.5%
3 node	± 5.0%
4 node	± 7.5%
5 node	± 10.0%

Summary of the different nodes of vertical and hull frequencies is shown in Table 12. For 2 node vertical frequency, *Todd*'s formula differ by + 7% from *BMT* Design formula and –64% from *Erich Danckwardt* formula. In fact, there is wide variation in the results for other nodes for *Erich Danckwardt* as compared with *Todd*'s and *BMT*'s.

Table 12 Comparison Tabulation of the 3 methods of Vibration Analsysis

Methods used are

1. Todd's formula
2. BMT's design formula
3. Erich Danckwardt formula

Nodes	Frequency (cpm)		
	Method 1	Method 2	Method 3
Vertical			
2	92	98	33
3	184	183	60
4	275	255	87
5	367	330	108
Horizontal			
2	138	-	50
3	275	-	103
4	413	-	158
5	-	-	213

5.0 DISCUSSIONS

5.1 Strength Analysis

According to Taylor, the minimum tensile and compressive stresses for manganese bronze propeller is 6000 lbs/inch^2. From the results in *Section* 3, it is found that the designed propeller stresses are greater than the minimum requirement as shown in Tables 7 and 9. The compressive stress is approximately 23-27% greater than minimum stress. And the tensile stress is approximately 7-8% greater than minimum stress. However, it is observed that by *Taylor*'s method, as shown in Table 7, the tensile stress is 8% less than the minimum stress. This implies that blade thickness need to be increased in order to meet the minimum stress requirement if calculations are done by *Taylor*'s method.

The thickness of the designed propeller as found in section 8.2 at 0.25R and 0.60R also exceed the minimum thickness as specified by the Classification Societies as shown in Table 11. According to the Series Charts method, *Lloyd*'s Rule seem to give a optimistic value of 1% difference of thickness at 0.25R and 0.60R. But *ABS* and *NK* are rather pessimistic with value of 3% difference of thickness at 0.25R and 0.60R.

5.2 Vibration Analysis

Careful scrutiny of *Erich Danckwardt* formula reveals that there is an error in the overall coefficient in Eqn. 18. Since this equation is based on *Todd*'s formulation, the overall coefficient should be:

From *Todd*'s formula:

$$f = c\sqrt{\frac{I}{L^3 M}} = c\sqrt{\frac{ft^4}{ft^3 t}} = c\sqrt{\frac{ft}{t}} = 46750\sqrt{\frac{ft}{t}}$$

Converting imperial units to metric

$$f = 46750\sqrt{\frac{0.3048}{1.016}}\ x\ \sqrt{\frac{m}{t}} = 25606\sqrt{\frac{m}{t}} \tag{20}$$

Hence, the overall constant for *Erich Danckwardt* formula should be 25606 instead of 200000.

A graph showing frequency of hull vibration versus propeller shaft revolutions for the designed ship is shown in Fig. 8.

The propeller blade frequency is 840 rpm at the propeller shaft revolution as shown in Fig. 8. This means that propeller resonants are clearly away from hull resonants. Hence, the designed propeller has minimum implication to hull vibration.

For these exercise, Todd's formula provides a simplified and good estimate for determining the hull vibration frequency at the preliminary design stage.

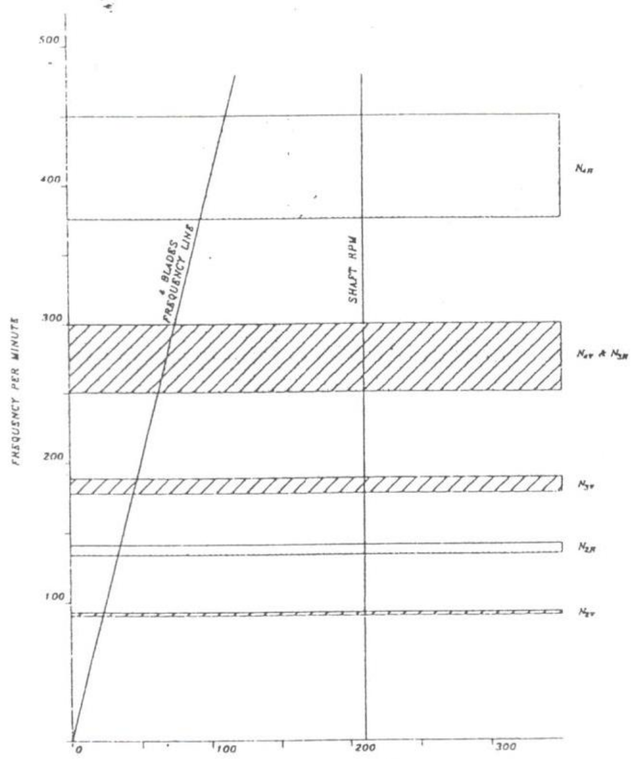

Fig. 8 Frequency of Hull Vibration vs Propeller Shaft Rpm (*Todd*'s Method)

6.0 CONCLUSION

A marine propeller not only should have a shape that engine power is converted into trust at an efficiency as high as possible, but it should also be capable of sustaining the attending loads without fracture. This implies the possibilities of the stresses in the propeller blades being calculated, and these stresses should not exceed a certain maximum value. The same implies to the vibration induced by the propeller. From a designer's point of view, during the preliminary design stage, it is necessary to use a simple formula but yet accurate results to calculate the strength and vibration of the required propeller.

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