An Investigation on Optimization of Containership Building Design and its Effect on Hull Natural Frequency

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This paper is focused on the design optimization and verification of dimensions of containership for carrying maximum number of TEU with minimum initial cost carrying capacity per TEU. The natural frequency of vibration of the hull structure has been estimated for various containerships by an analytical approach and the comparison has been made with results obtained using FEM. The comparison of the propeller blade stresses at the blade sections of the two different containership has been carried out and presented in the paper. The paper is concluded with comparison of STX Korean Co. forthcoming large containership dimensions and its TEU capacity with the optimized designed containership of the present work with respect to minimum ICS and ICCTEU cost.

KEY WORDS

Design optimization; Frictional resistance; Fuel consumption; Hull frequency; Second mode vibration; Stresses in blades.

NOMENCLATURE

c/D -- Blade section chord to propeller diameter
F_{2v} -- Second mode of vertical vibration, cpm
F_{2h} -- Second mode of horizontal vibration, cpm
F_{3v} -- Third mode of vertical vibration, cpm
F_{4v} -- Forth mode of vertical vibration, cpm
t_0 -- Blade thickness extrapolated to zero radius
t_1 -- Blade thickness at blade tip
t/D -- Maximum thickness of blade section to propeller diameter
r/R -- Blade section radius to propeller radius
S_c-- Compressive stress of blade due to thrust and torque, MN/m^2
S_t -- Tensile stress of blade due to thrust and torque, MN/m^2
S'_c -- Compressive stress blade due to centrifugal force, MN/m^2
S'_t -- Tensile stress of blade due to centrifugal force, MN/m^2
β -- Todd's constant = 8×10^4 --- 11×10^4
λ -- Wavelength, mts
Δ_v -- Virtual mass, tones

INTRODUCTION

At present, the world's largest container ship is owned by Emma Maersk having LBP of 397m with a capacity of 13,500TEU. Thus there is a scope to study such large containership building design and its analysis of frequencies due to periodic vibrations from engine and propeller systems. Volcy and Nakayama (1976) have studied the free and forced axial and lateral vibrations of line shafting. The result of lateral vibrations of line shafting have given natural frequencies of F_I= 662 cpm and F_{II}=951 cpm laying out of resonance hence no danger of appearance of active forced vibrations resonator. The forced vibrations of tail shaft was also of normal amplitudes for line shafting of 15 – 25 x 10^{-3} m in way of aft bush. The axial vibrations have the natural frequencies of F_1= 753 cpm, F_2 = 1685 cpm and F3= 2365 cpm falled into the resonance with two lowest propeller excited frequencies of $F_{n=4}$ = 346 cpm and $F_{n=8}$= 692 cpm with amplitude of 3—4 x 0^4 m hence no danger for oil film. They have also suggested to provide the additional steel walls between engine room and accommodation space, cementing on floors, thicker and heavier insulations etc to keep the the noise level at the lowest level in accommodations. Volcy (1978) concluded that the rapid growth in dimensions of ships and outputs of their propulsions plants, together with the introduction of powerful theoretical means allowing the decrease of scantlings, have led to – increase of flexibility of steel work of ships and increase of stiffnesses of their line shaftings. He also presented the detail analysis of problems related to flexibility of steel work (hull girder, double bottom of engine rooms and the influences exerted on them by outside shelling steel work, pillars, superstructures as well as foundations of thrust blocks) and stiffness of line shaftings as well as consecutive repercussions of Diesel engine crank shafts and bull gear assemblies with respective repercussions on them of thrust block. He emphasizes more on mutual interdependence between static and dynamic phenomena affecting the ship. Volcy (1984) have analysed the development in ship and marine engine building. The reasons for serious damages to propulsion plants due to incompatibility between stiffness of line shafting and flexibility of steel work and also vibrations in ship due to the presence of active and passive forced vibration resonators was studied. The author has measured i) 3-4 lowest frequencies and mode forms of vertical vibrations of hull girder, in connection with the presence of eventual active and passive resonators ii) local vibrations in the superstructures and the engine room iii)vertical, transverse, torsional and longitudinal vibrations of the line shafting, propulsion apparatus and iv) fluctuation of hydrodynamic pressures. The frequencies have been checked experimentally and the comparison have been made between obtained values of 8.66Hz and 8.58Hz and partial model, global model of the whole ship and exciter tests. Volcy (1986)also analysed the past development in shipbuilding and presented the consequences of the past development leading to the philosophy of simultaneous treatment of -static and dynamic interaction between machinery and hull, free and forced vibrations appearing on ships. He also determined experimentally the natural frequencies of transverse vibrations of main engine and superstructures vibrations as well as its reinforcements. It was concluded that the existence of a peak of resonance of the transverse vibrations of the main engine at the frequency F= 585 cpm, mode of vertical vibrations of the hull was F=580 cpm where as the natural frequency of vertical vibrations of the superstructure was noted F= 585 cpm.

This paper deals with the design optimization of large container ship for high dwt capacity with reduction in hull vibration with respect to the minimum initial cost of ship (ICS) and initial cost of carrying capacity (ICCTEU). Since the blades of propeller are responsible for the vibration and noise in ship hence the blade geometry and blade stresses has also been studied. If the service speed of the ship increases the frictional resistance offered by sea is also increases, which results in demand of more power from the engine to overcome the resistance. The analysis has been made in selection of number of blades for the propeller with suitable diameter of propeller in order to reduce the vibration because of cavitation. This paper is more focused on the determination and verification of dimensions of the box ship suggested by STX Korean Company. The comparison has been made for the TEU carrying capacity of STX Korean Co. ship with the presented optimized designed container ship.
(Clark 2005, Dr. Barrass 2004,Taggart 1980).

CONTAINERSHIP BUILDING DESIGN (AN ANALYTICAL APPROACH)

STX Korean Co. has proposed new design for container ship to carry 22000TEU. The present paper is more focused on design optimization of container ship to enhanced the TEU capacity and also to reduce the initial cost of ship per TEU.

Considering the LBP= 460m, B= 60m, D=30m, H=24m, C_b= 0.75, C_d=0.96

Displacement , W = L x B x H x C_b x ρ
= 460 x 60 x 24 x 0.75 x 1.025
= 509220 tonnes (1)

But, Light weight of ship = [Lwt. Part - A + Lwt. Part – B]
Where, Lwt Part– A=[Weight of steel structure, beams, hatch covers and double bottom plates etc]
and Lwt.Part--B= [Weight of Engine, EO, Super structures and Machinery etc.]

But Light weight Part -A = [Weight of steel Hull]
= [Volume of steel plates to built a ship x density of steel]
= (4622 + 10% 4622) x 7.80
= 39656 MT (2)

Table1. Light weight Part – B

Weight . MT			Total Weight, MT
Engine + Machinery	Boiler + E&OF	Super structure + Engine room	
4500	1100	400	**6000**

Now,
Light weight Part-B = 6000 tonnes (3)
Hence from eqs. (2) and (3),
Light weight of ship = [Lwt. Part - A + Lwt. Part – B]
= [39656 + 6000]
= 45656 tonnes (4)

Also, Dead weight, dwt = [Dwt Part - A + Dwt. Part - B]
Where Dwt.Part–A= [Weight HFO+MDO+LO+FW + Ballast]
and Dwt Part – B = [Load of Containers]

Table 2. Dead weight Part – A

Item	Volume m^3	Density T/ m^3	Weight MT	No. of tank
Heavy fuel oil conus. Per day (1100 MT / day for Max.V_s=28kt) Journey Max. 20days	24444.44	0.90	22000	10
Marine diesel oil	2565.90	0.88	2258	1
Lubricating Oil	2566.03	0.742	1904	1
Fresh water	5130	1	5130	2
Ballast water	26709	1.025	27377	10
		Total	**58669**	**24**

Here, Dead weight Part – A = 58669 MT

Table 3.Dead weight Part – B

	No. of Containers	Average Capacity, MT	Total weight, MT
On Deck	10726 TEU	16.63	178374
On Hold	13617 TEU	16.63	226451
Total	24343 TEU	16.63	404825

And Dead weight Part-B = 404825 MT
Since available no. of tanks in double bottom = 30 Nos.
Size of each tank = 450 x 57 x 3 = 2565 m^3
Therefore, Dead weight , dwt = 58669 + 404825
= 463564 tonnes (5)
But,
Dead weight , dwt = Displacement(W) – Light weight (Lwt)
= 509220 - 45656
= 463564 tonnes (6)
The equations (5) and (6) denoting the exact matching of the dead weight capacity of containership.

OPTIMIZATION OF SHIP BUILDING DESIGN OF CONTAINERSHIP

Optimization means finding the best solution from a limited or unlimited number of choices. There are two methods of approaching optimization problems -
Direct search approach: The solutions are generated by varying parameters either systematically in certain steps or randomly. Systematic variation soon becomes prohibitively time consuming as the number of varied variables increases. Random searches are then employed, but these are still inefficient for problems with many design variables.
Steepness approach: The solutions are generated using some information on the local steepness (in various directions) of the function to be optimized. When the steepness in all directions is zero, the estimate for the optimum is found. This approach is more efficient in many cases. Most optimization methods in ship design are based on steepness approaches because they are so efficient for smooth functions.
In order to undertake the optimization in ship building design , initial costs may be minimized.
Initial costs i.e. Building costs can be roughly classified into ,a) Direct labor costs. b) Direct material costs (including services bought). c) Overhead costs.
For optimization, the production costs are divided into --
Variable- dependent costs – Costs which depend on the ship's form a) Cost of hull b) Cost of propulsion unit c) Other variation-dependent costs, e.g. Hatchway, pipes etc.
Variable – independent costs – Costs which are the same for every variant, e.g. Navigation equipment, living quarters, etc. (Scheneekluth 2004).
The present paper includes the C programming for design optimization of ship for minimizing the initial cost, (Ref. Appendix-1).

EVALUATION OF NUMBER OF CONTAINERS ON FREEBOARD DECK AND HOLD

Considering the optimized LBP = 460 m for maximum carrying capacity of TEU.
Actual volume of each TEU = 6.06 x 2.42 x 2.42
= 35.48 m^3
Now increase in base area of each TEU with clearance between two TEU along the length and breadth of ship,

TEU_{BA} = 6.15 x 2.51
= 15.43 m^2

Number of TEU on Hold

The total hold volume of the ship is divided into three zones. Considering ***Zone - AB*** as amidship zone***, Zone- BC*** is towards the bow and ***Zone - DA'*** is towards the stern of the ship.

Zone - AB

Available length = [LBP – Engine Room length – Bow length – Stern length]
= [460 – 50 - 20 - 13]
= 382 m
Along length 382m, No. of holds are as,
= [382 / ((6.15 x 4) + 1.5}]
= 14.65 i.e. 14 Nos (Max.)
Here the space left between two holds is 1.5 m.
Since each hold is having 4 Bays, 21 Rows and 11 Tiers, Therefore,
No. of Containers in one hold would be,
4 x 21 x 11 = 924 TEU
and No. of TEUs in Zone - AB = 924 x 14 = 12936 TEU (7)

Zone – BC

Available length = 13 m
Therefore only three bays are available with 8Tiers along the breadth as follows,
3^{rd} Bay →→ 19 x 8 = 152 TEU
2^{nd} Bay →→ 17 x 8 = 136 TEU
1^{st} Bay →→ 15 x 8 = 120 TEU
Thus, no. of TEU's in Zone BC = 408 TEU (8)

Zone DA'

Available length = 15 m
Therefore only two bays are available with 7Tiers along the breadth as follows,
2^{nd} Bay →→ 20 x 7 = 140 TEU
1^{st} Bay →→ 19 x 7 = 133 TEU
Thus, No. of TEU's in Zone DA' = 273 TEU (9)
Therefore the total no. of TEU's on HOLD ,(Ref. eq. 7,8 and 9),
TEUH = 12936 + 408 + 273
= 13617 TEU (10)

Number of TEU on Deck

Since the hatch covers are not provided for this newly designed container ship, therefore all the three zones in hold can be extracted in Tiers only.
Available length = [Overall Length – Super Structure length – Bow length – Stern length]
= [485 – 20 - 12 - 8]
= 444 m

Zone – AB

Along length 382mfrom super structure,
The no. of TEU's are as,
(14 x 4) x (21) x (8) = 9408 TEU (11)

Zone – BC

Available length = 13 m

Therefore only three bays are available with 8 Tiers along the breadth as follows,

3^{rd} Bay →→ 19 x 7 = 133 TEU
2^{nd} Bay →→ 17 x 6 = 102 TEU
1^{st} Bay →→ 15 x 5 = 75 TEU
Thus, no. of TEU's in Zone BC = 310 TEU (12)

Zone DA'

Available length = 49 m, But length 34m is available on the top of the engine room and besides the superstructure to accommodate the additional TEU. Therefore seven bays are available with 7 Tiers along the breadth as follows,

7^{th} Bay →→ 21 x 7 = 147 TEU
6^{th} Bay →→ 21x 7 = 147 TEU
5^{th} Bay →→ 21 x 7 = 147 TEU
4^{th} Bay →→ 21 x 7 = 147 TEU
3^{rd} Bay →→ 21 x 7 = 147 TEU
2^{nd} Bay →→ 20 x 7 = 140 TEU
1^{st} Bay →→ 19 x 7 = 133 TEU
Thus,No. of TEU's in Zone DA' = 1008 TEU (13)

Therefore the total no. of TEU's on DECK (Ref. eqs.11-13),

TEUD = 9408 + 310 + 1008
= 10726 TEU (14)

Thus,

Gross no. of TEU's on HOLD and DECK are (Ref. eqs.10&14),

TEUH + TEUD = 13617 TEU + 10726 TEU
= 24343 TEU > 22000 TEU (15)

Thus it is confirmed from the eq.15 that, the STX Korean Co planned to record box ship of 460m x 60m x 30m with 22000TEU has been increased to 24343 TEU (Increased by 10.60%) by optimizing the dimensions of containership presented in the paper and it may be further increased to 12-16 % by increasing the no. of Tiers on Deck also.

COST ANALYSIS

The objective of the paper is to optimize the designed dimensions of the ship to carry maximum no. of TEU and also to minimize the initial cost of ship with respect to carrying capacity in US $ (Scheneekluth 2004).

Initial cost of ship , ICS = [(CTSH) (WSH) (0.7 / C_b)$^{0.5}$] + [(CTEO) (WEO)] + [(CTE) (WM)]

And Initial cost in carrying capacity per TEU,

ICCTEU = [ICS / (TEUH + TEUD)]

Criterion: To minimize the initial cost per carrier container

Where, CTSH -- Cost per ton steel hull , $/MT
WSH -- Weight of steel hull, MT
CTEO – Cost per ton E & O $/MT
WEO -- Weight of E & O, MT
CTE -- Cost per ton engine, $/MT
WM -- Weight of Machinery , MT
TEUH – No. of TEU on Hold
TEUD – No.of TEU on Deck

Table 4 represents the effect of containership dimensions on initial cost of ship and its TEU carrying capacity. It is observed that by reducing the breadth and depth dimension of ship, the TEU carrying capacity of ship is also reduces very sharply where as the initial cost of ship reduces gradually. The major and objectionable cost difference is observed for the ship with 460x55x28 dimensions. It is also observed that, the marginal increase in the initial cost of ship with 460x60x30 dimensions is preferable due to maximum TEU carrying capacity of ship. Hence it is recommended that one should prefer the container ship–D with the above said dimensions, which has less cost of ICCTEU too.

Table 4 Effect of Containership dimensions on initial cost of ship and TEU carrying capacity

	Container ship dimensions , m			
	L:B:D 460:60: 30 H =24	L:B:D 460:60: 28 H =22.4	L:B:D 460:55: 30 H =24	L:B:D 460:55: 28 H =22.4
Displacement, W , tonnes	509220	475272	466785	435666
Weight of Steel Hull Structure, MT	5084 x 7.8 = 39655	5008 x 7.8 = 39062	4993 x 7.8 = 38945	4932 x 7.8 = 38467
Total no. of TEU	24343	22720	22314	20827
ICS US $ Millions	270	265	265	263
ICCTEU US $ / TEU	1200	1166	1166	1262
TEU Capacity %	100	93.33	91.66	85.55
Recomme-ndations	Ö	**X**	**X**	**X**

FRICTIONAL RESISTANCE AND POWER REQUIRED BY SHIPS

When ship is moving in water, the following water resistance will be experienced by the ship. a) Frictional resistance b) Wave making resistance c) Eddy- making resistance d) Resistance due to wind and appendage. The froude has developed the equation for frictional resistance as, (Dr. Barrass 2004),

$$R_f = fAV^n$$

$$f = 0.417 + \frac{0.733}{L + 2.68}$$

$$A = 2.59\sqrt{WL}$$

$$P_{ne} = R_t V$$

$$P_e = P_{ne} + 10\% P_{ne}$$

$$Pt = Pe / h_{hull}$$

$$P_d = P_t / h_{prop}$$

$$P_b = P_d / h_{shaft}$$

$$P_i = P_b / h_{engine}$$

In the present scenario, the available engine has a maximum input power capacity of 90000 kW at 102 rpm with 16 cylinders in-line. The Input power required for newly designed containers C and D are tremendously high and the engine of that capacity is not available in the market. Thus there is a need to manufacture the engine of such capacity and requirement of power may be accommodate in either 20 cylinders or 25 cylinders with 11000kW or 8560 kW per cylinder respectively

Wake fraction and propeller diameter

$$w_f = \frac{C_b}{2} - 0.05$$

$$w_f = \frac{V_s - V_a}{V_s}$$

$$D_p = \frac{632.7(Pt)^{0.2}}{(RPM)^{0.6}}$$

The area of blade can be calculated as,

$$A_b = 0.709\frac{p}{4}D_p^{\,2}$$

Table 5 Frictional resistance and power requirement of ship

	Container Ship -A (LBP 290)	Container Ship -B (LBP 330)	Container Ship - C (LBP 360)	Container Ship - D (LBP 460)
Frictional Resistance R_f, kN	2271.79	2961.16	3562.22	5808.56
Total Resistance, R_t, kN	3786.32	4935.27	5937.03	9680.93
Naked effective power P_{ne}, kW	48696.34	63473.06	76356.88	124507.51
Effective power P_e, kW	53565.98	69820.36	83992.57	136958.26
Thrust power P_t, kW	54659.16	71245.27	85706.7	139753.32
Delivered power P_d, kW	72878.88	94993.69	114275.6	186337.77
Brake power P_b, kW	76714.61	99993.36	120290.11	196145.02
Input Power, P_i, kW	83385.45	108688.43	130750.12	213201.10

Table 5 represents the frictional resistance and power requirement of newly designed ships. It is observed that the frictional resistance and power requirement of ship increases with increase in dimensions of ship as well as increase in the dead weight capacity of ship. Since the optimized dimensions of containership are 460x60x30 (Ref. Table 4), therefore the selection of number of blades for its propeller may be decided by the second harmonics of the vertical and horizontal vibrations as follows,

a) For the propeller rpm = 102, with four blades,
First hormonics are 13.23 31.31 24.46 39.69
Second harmonics are 3.307 7.827 6.615 9.922

b) For the propeller rpm = 102, with five blades,
First hormonics are 13.23 31.31 24.46 39.69
Second harmonics are 2.646 6.262 5.292 7.978

c) For the propeller rpm = 102, with six blades,
First hormonics are 13.23 31.31 24.46 39.69
Second harmonics are 2.205 5.218 4.410 6.605

It is obvious that no synchronous of rpm of propeller with highest value of second hormonics of hull vibration with number of blades between 4 - 6. Thus the scope for resonance reduced if six number of blades for the propeller is preferred.

Table 6 Wake fraction and propeller details

(Dwt)→	Container Ship - A (100000t)	Container Ship - B (150000t)	Container Ship - C (200000t)	Container Ship - D (463565t)
Wake fraction, w_f	0.299	0.311	0.318	0.325
Velocity of advance, V_a	17.5	17.22	17.04	16.87
Theoretical speed, kt	26.76	28.25	29.75	32.71
Propeller	9.00	9.50	10.00	11.00

diameter, m				
Thrust on propeller, T_p kN	6067.12	8040.15	9775.71	8281.67
Thrust on propeller blade T_b, kN/m^2	137.36	163.72	184.88	124.49
Propeller pitch, P_p, m	8.1	8.55	9.00	9.90
No. of blades & Area of blade, m^2	6 44.16	6 49.10	6 53.87	6 66.52
Area of each blade, m^2	7.36	8.18	8.97	11.08

VIBRATION OF THE HULL STRUCTURE

The major role played by the propeller in induction of hull vibrations. Since the propeller includes the shaft system, propeller design, propeller cavitation and Rudder, propeller induced 57% hull vibration, The shaft system shared 30%, propeller cavitation 25%, propeller design 40% and Rudder 5% of the total propeller vibrations. A propeller produces its thrust by creating a difference between the pressures acting on the face and the back of the propeller blades, the pressure on the back of a blade section falling below the ambient pressure and the pressure on the face rising above it. If the pressure at any point on the back of the blade falls to the vapour pressure, the water at that point begins to cavitate. The propeller has tip, root, hub, leading edge, trailing edge, face and back cavitations. The classification according to nature of cavities and their appearance, sheet cavitation, spot cavitation, streak cavitation, cloud cavitation, bubble cavitation and vortex cavitation. The cavitation affects the nature of the flow around a propeller since the flow is no longer homogeneous. The formation of cavities has the effect of virtually altering the shape of the propeller blade sections, and as a result the thrust and , to a lesser extent, the torque of the propeller are reduced, and so also the propeller efficiency. The propeller induces ship hull vibration through the pressure fluctuations it produces when operating in a non-uniform wake. These pressure impulses are greatly magnified by the occurrence of cavitation. The propeller cavitation also causes high frequency vibration of the propeller blades and the surrounding structure, (GL Tech 2001, ABS 2006).

Modifications in propeller

The reduction of excitations generated by the propeller is possible only by improving the wake field by installations of a tunnel above the propeller and fins or wings. The advantage of installation of fins above the propeller and in the longitudinal plane of the ship is that the vibrations generated by the vortex built-up between the hull and the propeller blade in its top position can be efficiently reduced. The effect of fluctuations of hydrodynamic pressure can also be reduced by executing the well in the aft peak which will work as an air cushion damper by providing the overflow pipes, (Ghose 2004).

Detection and Detuning of forced vibration resonators by:

1 Crankshaft realignment

2.Installation of a damper on the free end of the crankshaft

3.Modification of the phase angle between crankshaft and the propeller

4.Strengthening of the foundations of the independent thrust bearing

5. Modification of the torsional system of the line shafting.

HULL NATURAL FREQUENCY OF NEWLY DESIGNED CONTAINER SHIP

Second mode of vertical vibration

Assuming that the ship's weight is uniformly supported along its length. The ship vibrates about the centre of gravity which remains stationary on the longitudinal neutral plane. There is no particle acceleration at the two nodes but particle acceleration increases with distance from these nodal points. If the support is uniformly distributed then the structure will vertically vibrate in such a way that its centre of gravity remains stationary so the basic mode of vertical vibration has two stationary points called nodes.

The second mode of vertical vibration of container ship is calculated using TODD's Eq. as,

$$F_{2v} = b\sqrt{\frac{BD^3}{\Delta_v L^3}} \tag{16}$$

where,

$$\Delta_V = W\left(\frac{B}{3D} + 1.2\right)$$

Second mode of horizontal vibration

A ship has natural horizontal modes of vibration about a vertical neutral plane. The hull is now bending about a vertical neutral plane and as the ship's beam is greater than the hull depth, the horizontal moment of inertia of the midship section will generally be greater than the vertical moment.

The second mode of horizontal vibration of container ship may be calculated by modifying the TODD's eq. as,

$$F_{2h} = b\sqrt{\frac{DB^3}{\Delta_v L^3}} \tag{17}$$

where,

$$\Delta_h = W\left(\frac{H}{3B} + 1.2\right)$$

Higher harmonic modes of free-free vertical vibrations

Higher modes of vertical vibration with more nodes can occur at higher frequencies, which are called the 1st ,2nd ,3rd harmonics etc. The following are the frequencies of Container –D of LBP= 460 m.

I) Second vertical mode,

Wavelength l_2 = L = 460 m and

Frequency F_{2v} = 13.23 cpm

II) Third vertical mode,

Wavelength l_3 = 2/3 L = 306.66 m and

Frequency F_{3v} = 2 x F_{2v} = 26.46 cpm

III) Forth vertical mode,

Wavelength l_4 = 1/2 L = 230 m and

Frequency F_{4v} = 3 x F_{2v} = 39.69 cpm

Theoretically, the number of harmonics is infinite but the amplitudes and energy levels in harmonics much higher than the forth mode are generally considered to be insignificant, particularly as the damping effect of the surrounding water increases with particle velocities that occur at the very high frequencies.

Higher harmonic modes of free – free horizontal vibrations

Horizontal vibrations are generally lower in amplitude than the vertical vibrations but of a frequency, as ship's hull are wider than they are deep so the midships moment of inertia will tend to be greater about a vertical neutral plane than the horizontal neutral plane, the F_{2h} value tends to be about 70% greater than F_{2v} .

Table 7 Effect of LBP on the higher harmonics of hull vibration

LBP (m)	F_{2v} cpm	F_{2h} cpm	F_{3v} cpm	F_{4v} cpm	Highest second harmonics, cpm (Hz)
A-290	17.3	55.58	34.61	51.91	9.263 (0.154)
B-330	14.04	53.31	28.07	42.11	8.885 (0.148)
C-360	12.48	51.29	24.96	37.44	8.548 (0.142)
D -460	13.23	31.31	26.46	39.69	5.218 (0.086)

The 2-noded hull vertical bending natural frequencies actually lie well below the dangerous exciting frequencies of diesel main engines and propeller. It is observed that the hull girder natural frequencies increase more or less linearly with node number from 2-noded value for first few modes.(Ref. Table 7) (Clark 2005,GL Tech 2001, ABS 2006).

ENGINE AND PROPELLER MATCHING FOR REDUCTION IN FUEL CONSUMPTION PER DAY

In the design of the propulsion system of ship, it is important to match the ship, the propeller and the propulsion machinery so that the propulsion system as whole operates in the optimum manner. The power-speed characteristics of the ship and propeller moreover change with the loading of the ship and the sea conditions, and with time as the hull and the propeller get progressively rougher due to fouling, corrosion and possibly cavitation erosion. In selecting the propulsion plant for ship and designing its propeller or propellers, it is necessary to ensure that the desired ship speed is achieved without overloading the engine or exceeding its rated rpm in the varying operating condition of the ship. If the engine and the propeller are not properly matched, the life of the engine may be reduced, maintenance costs may be higher and the fuel consumption may be greater. Since engine can be run at less than full power, it would appear that the problem of engine – propeller matching has a simple solution :

Design the propeller so that the propeller curve intersects the engine curves at the maximum rating of the engine.

The service margin i.e. the difference between the power required in the average service condition and that required in the fully loaded trial condition, depends upon mission profile of the ship. Service margin vary between 15-35 % depending upon the average weather conditions of the ship’s route. A large service margin ensures that the engine can be run at high rpm without being overload even when the hull and propeller becomes rough or the weather is bad. It is not possible to achieve the full engine power without exceeding the rated rpm. On the other hand, with a large service margin, the operating costs of the ship are reduced because of lower fuel consumption and reduced maintenance and replacement costs. There is thus an optimum service margin for each ship that will minimize its life cycle cost, (Ghose 2004).

In order to understand the engine propeller matching of rpm, considering an example of container ship of LBP = 290 m with dwt = 100000 tonnes.

The following engine specifications are selected for newly designed containerships, Make: WARTSILA RTA96C

Cylinder bore = 960mm, Piston stroke = 2500 mm, Speed = 102 rpm, Piston speed = 8.5 m/s , No. of cylinder = 14, BHP= 114800 bhp, Power = 84420kW, BMEP= 19.6 bar, BSFC= 171 g/kWh, Max. Torque = 5608312 lb/ft at 102 rpm, Total engine weight = 2300tons, Length = 89 ft, Height = 44 ft.

If engine is at continuous service rating of 85% maximum rated power and 95% rated rpm. The propeller rpm in the fully loaded trial condition at which the maximum rated power of the engine will be absorbed, if service margin of 20%is provided, can be calculated as,

For maximum continuous rating :

P_{bo} = 76714.61kW, N_o=102 rpm =1.7s^{-1}

Maximum rated torque, $T_o = P_{bo} / 2\pi N_o$

= 7182.06kN-m

For continuous service rating :

P_{b1} = 0.85 P_{bo} = 65207kW,

N_1 = 0.95N_o = 96.9rpm = 1.615 s^{-1}

For trial condition :

$P_{b\ trial} = kN^3 = P_{b1} / 1.2$

= 54339.50 kW

Here,

N_{trial} = N_1 = 96.9rpm = 1.615 s^{-1}

Therefore,

$k = P_{b\ trial} / N_{trial}{}^3$

= 12900.24 kWs^3

The value of N at which the maximum rated power of the engine will be absorbed is as follows,

$N^3 = P_{bo} / k$

= 5.946

Thus, N=108.7 rpm i.e. 106.5% of N_o, the rated rpm

For bad weather condition :

Due to the exceptionally bad weather condition, if the power demand of the propeller increases by 40% over that in the trial condition, and if 10% overloading of the engine over the maximum rated torque is permitted, then maximum propeller rpm can be calculated as,

$P_b = 1.4\ P_{btrial}$

= 1.4 x 12900.24 $(N)^3$ kW

= 18060.33 N^3 kW

and Torque, T = 1.10 T_o

= 1.10 x 7182.06

= kN-m

So that,

$T = P_b / 2\pi N$ = 7900.26

= 18060.33 $N^3 / 2\pi N$ = 7900.26

Thus ,

N =1.65 rps = 99.47 rpm

Therefore the propeller will be run at max. 99.47 rpm in bad weather condition.

Now, Here is a sample calculations for Fuel Consumption per day for Container ship of LBP=290 m,

$$F_c = \frac{W^{2/3}V^3}{FuelCons./day}$$

$$FuelConsumption/day = \frac{W^{2/3}V^3}{F_c}$$

$$FuelConsumption/day = \frac{(119047.61)^{2/3}}{120000}(25)^3 \quad (!8)$$

= 312.657 tonnes per day

(Here F_c – Fuel Coefficient = 120000 for diesel machinery)

It is obvious that, fuel consumption is directly proportional to cubic speed of ship and displaced volume of the ship. (Clark 2005, Ghose 2004)

Table 8 Effect of service speed on resistance, power and fuel Consumption for LBP – 360m

Service speed, kt →	V_s = 26	V_s = 27	V_s = 28	V_s = 29	V_s = 30
Frictional Resistance, kN	3826.17	4099.01	4380.29	4669.99	4968.05
Total Resistance, kN	6376.96	6831.68	7300.49	7783.32	8280.09
Naked effective Power, P_{ne} , kW	85295.39	94892.10	105159.58	116118.59	127789.44
Fuel Consumptions, MT / day	557.98	624.87	696.91	774.27	857.16

Table 9 Effect of service speed on resistance, power and fuel Consumption for LBP = 460m

Service speed, kt →	V_s = 26	V_s = 27	V_s = 28	V_s = 29	V_s = 30
Frictional Resistance, kN	1974.13	2114.49	2259.18	2408.18	2561.44
Total Resistance, kN	3290.22	3524.16	3765.31	4013.63	4269.08
Naked effective Power, P_{ne} , kW	44008.52	48950.58	54232.73	59878.90	65886.10
Fuel Consumptions, MT / day	925.72	1036.70	1156.21	1284.56	1422.09

STRESSES IN THE PROPELLER BLADE SECTION

The Taylor's method is incorporated for estimating the maximum compressive and tensile stresses in the root section of the propeller blade. The following are the assumptions made for determination of propeller blade strength. Each propeller blade is assumed to be a beam cantilevered to the boss. The thrust distribution along the propeller radius is linear. The maximum thickness of the blade also varies linearly with radius. The root section is at 0.2 R. The propeller efficiency is a linear function of the apparent slip in the normal operating condition. Based on the above assumptions, the maximum compressive and tensile stresses in the root section due to thrust and torque are given as,

$$S_C = \frac{C_O P_D}{ZnD^3 (c/D)(t_O/D)^2} \tag{19}$$

$$S_T = S_C (0.666 + C_1 \frac{t}{c}) \tag{20}$$

The additional compressive and tensile stresses due to centrifugal forces are given by,

$$S'_C = C_2 \rho_m n^2 D^2 \left(\frac{C_3 \tan e}{2(t_O/D)} - 1 \right) \tag{21}$$

$$S'_T = C_2 \rho_m n^2 D^2 \left(\frac{C_3 \tan e}{3(t_O/D)} + \frac{C_4 \tan e}{c_{\max}/D} + 1 \right) \tag{22}$$

Where C_0, C_1, C_2, C_3, C_4 --- Coefficients for Taylor's method
e - rake angle = 10^0
$D = D_p$ –Propeller diameter, m
Selecting blade section geometry of B-series propellers with skew angle 15^0 and rake angle 10^0
Table 10 for LBP=360m, represents the blade thickness and chord length with respect to r /R and it is observed that the blade thickness (t) and decreases with increase in the r/R value where as chord length (c) initially increases with r/R till r/R = 0.6 and decreases further till r/R =1.0. Table 11 represents maximum compressive and tensile stresses due to thrust and torque in the blade with respect to r/R. The additional compressive and tensile stresses due to centrifugal force are also calculated and tabulated. It is observed that both the stresses are initially decreases with r/R till 0.6 and further increases till r/R=1.0 . The reason could be the profile of the blade and the stresses are gets balanced from the root section to the tip of the blade. The additional compressive and tensile stresses due to centrifugal forces are having constant value with respect to the r/R.

Table 10 The propeller blade thickness at different sections for LBP = 360 m

r/R	**t/D**	**t**	**c/D**	**t/c**
0.2	0.0286	0.286	0.2147	0.1332
0.3	0.0254	0.254	0.2431	0.1045
0.4	0.0222	0.222	0.2648	0.0838
0.5	0.0190	0.190	0.2780	0.0684
0.6	0.0158	0.158	0.2825	0.0559
0.7	0.0126	0.126	0.2769	0.0455
0.8	0.0094	0.094	0.2544	0.0369
0.9	0.0062	0.062	0.2043	0.0303
1.0	0.0030	0.030	0.000	--

Table 11The propeller blade compressive and tensile stresses at different sections for LBP = 360 m

r/R	**S_c MN/m^2**	**S_t MN/m^2**	**S'_c MN/m^2**	**S'_t MN/m^2**
0.2	216.12	166.51	25.61	33.05
0.3	190.85	142.74	25.61	33.05
0.4	175.21	128.21	25.61	33.05
0.5	166.91	120.10	25.61	33.05
0.6	164.24	116.58	25.61	33.05
0.7	167.53	117.55	25.61	33.05
0.8	182.33	126.71	25.61	33.05
0.9	227.05	156.61	25.61	33.05
1.0	--	--	25.61	33.05

Table 12 for LBP = 460m, represents the similar pattern for the blade thickness and chord length with respect to r/R as shown in the Table 10. Table 13 represents again similar pattern for the compressive and tensile stresses due to thrust and torque with respect to r/R . The additional compressive and tensile stresses are also following the same pattern shown in the Table 11. The values of all the stresses are found to be more for LBP=460m in comparison with LBP=360m. This could be due to the increase in the propeller diameter from 10m for LBP=360m to 11m for LBP= 460m.

Table 12 The propeller blade thickness at different sections for LBP =460 m

r/R	**t/D**	**t**	**c/D**	**t/c**
0.2	0.03	0.33	0.2146	0.1332
0.3	0.03	0.33	0.2430	0.1045
0.4	0.02	0.22	0.2647	0.0838
0.5	0.02	0.22	0.2779	0.0684
0.6	0.02	0.22	0.2824	0.0559
0.7	0.01	0.11	0.2769	0.0455
0.8	0.01	0.11	0.2544	0.0369
0.9	0.01	0.11	0.2043	0.0303
1.0	0.000	0.00	0.000	--

Table 13 The propeller blade compressive and tensile stresses at different sections for LBP = 460 m

r/R	**S_c MN/m^2**	**S_t MN/m^2**	**S'_c MN/m^2**	**S'_t MN/m^2**
0.2	264.73	203.96	30.99	40.00
0.3	233.81	180.13	30.99	40.00
0.4	214.65	165.37	30.99	40.00
0.5	204.45	157.52	30.99	40.00
0.6	201.20	155.01	30.99	40.00
0.7	205.27	158.14	30.99	40.00
0.8	223.33	172.06	30.99	40.00
0.9	278.21	214.34	30.99	40.00
1.0	0.00	0.00	0.00	0.00

Flexural and Torsional Vibration due to Propeller Blade

The surrounding water also influences propeller blade vibration significantly: the mode shapes and frequencies of a propeller blade vibrating in air are quite different from those vibrating in water. Simple empirical formula proposed by Baker for estimating the frequencies of fundamental flexural (f_f)and torsional (f_t)blade vibration as,

$$f_f = [(\frac{0.305 t_o}{(R - r_o)^2})(\frac{C_o T_m g E}{C_m T_o r_m})^{0.5}]x0.65 \qquad (23)$$

= 15.36 Hz

$$f_t = [(\frac{0.92 t_{0.5}}{(R - r_o C_{0.5}} \frac{C_o (gG)^{0.5}}{C_m (r_m)^{0.5}})]x0.65 \qquad (24)$$

= 150.08 Hz

The blade frequencies in water are about 65% of the corresponding frequencies in air, (Ghose 2004).

FINITE ELEMENT ANALYSIS USING ANSYS 9.0

The hull of the ship has been modeled using shell-63 elements and the transverse sections or hull girder has been modeled using Beam4 3-D elastic beam elements.

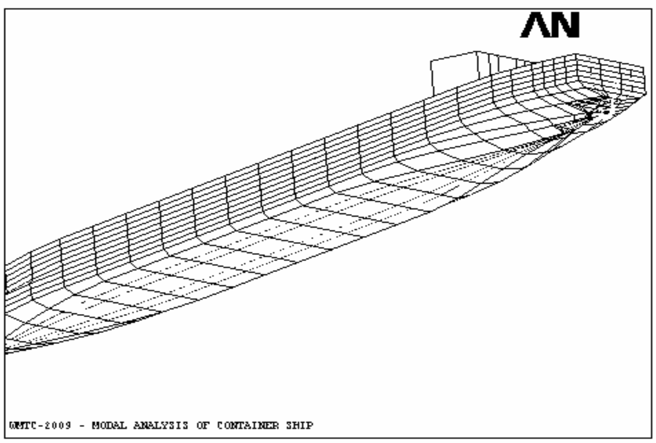

Fig. 2 3-D model of containership of LBP = 460m in Ansys

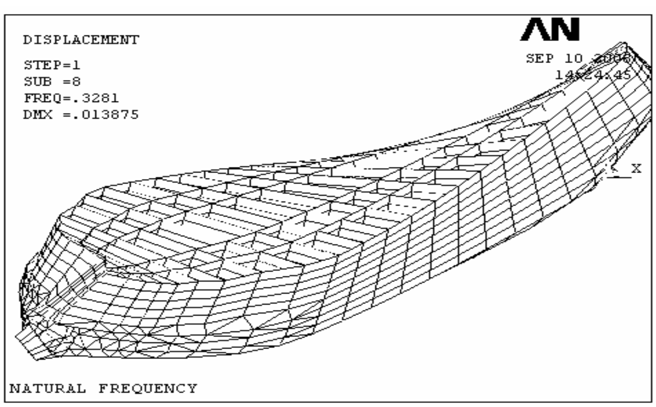

Fig. 3 Torsional vibration of containership of LBP = 460m

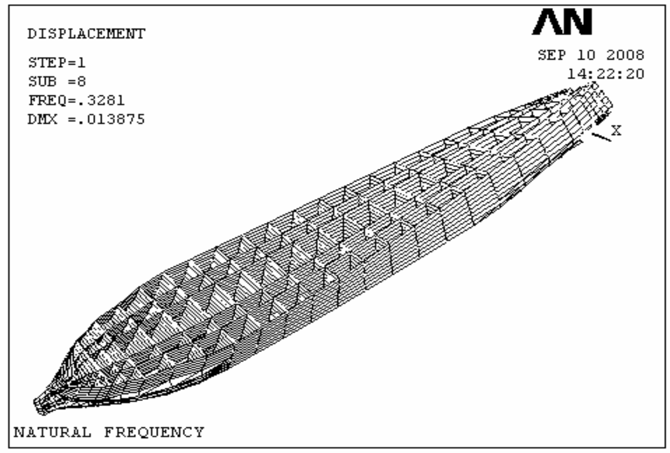

Fig. 4 Natural frequency of container ship of LBP =460m

The no. of DOF for Beam and Shell are 6 and 3 respectively. The plate thickness, Young's Modulus, Poisson's ratio and density of material are taken as 40 mm, 206.8 N/mm^2, 0.29 and 7850 kg/m^3. For the natural frequency analysis, the BLOCK LANCZOS method has been used in Ansys. 9.0.(Ref fig. 2-4)

STABILITY OF SHIPS

The stability of ship can also be checked by the initial transverse metacentre, which is the point of intersection of the lines of action of buoyancy force, when the ship is in the initial upright condition and subsequently heeled conditions, within small angles of heel. Using the relation , KM= KB + BM where KB= H/2, BM = B^2 / 12H , (Rhodes 2003).

To satisfy the IMO stability intact stability requirements the minimum GM for a ship is 0.15 m. Also KM > KG for stability.

Table 14 represents the KG,KM, BM, WSA, TPC and MCTE values for all the newly designed ships. It is observed that, if the dwt of ship is increase, the longitudinal and transverse metacentre of ship is also increase. The width and depth influences are also increases with increase in dwt of ships.

Table 14 Stability of the newly designed containerships

	Container A	Container B	Container C	Container D
KB , m	6.935	7.422	7.921	12.63
BM_T	10.209	14.45	17.59	13.98
KM_T	17.14	21.87	25.51	26.61
BM_L	443.41	536.25	599.27	611.77
KM_L	450.34	543.67	607.19	624.40
Width of Influence,	397.11	475.48	533.19	539.41
Depth of Influence	91.95	94.94	99.11	154.88

Water plane area, WPA, m^2	9641.54	13763.48	17330.08	22990.8
Tonnes per cm immersion, TPC	98.82	141.07	177.63	235.65
Moment to change trim one cm, MCTC, tm/cm	1715.98	2841.29	3949.45	6765.51

COMPOSITE MATERIAL FOR ENGINE FOUNDATION, ENGINE BULKHEAD, SUPER STRUCTURE AND PROPELLER

Weight saving composite materials became a very popular and has tremendous demand due to its high endurance, stability, damping and low weight.

Micromechanical and Macromechanical Analysis of Carbon-Epoxy Laminate

Generally a lamina is a thin layer of composite material, which is of the order of 0.125 mm. A laminate is constructed by stacking a number of such laminae in the direction of the lamina thickness. Mechanical structures made of these laminates, are subjected to various loads, such as bending and twisting. The design and analysis of such laminated structures demand the knowledge of the stresses and strains in the laminate .

The present study deals with the Micromechanical and Macromechanical analysis of Carbon-Epoxy laminate with the multilayered quasi - isotropic stacking sequence $[0^0/90^0/45^0/-45^0]_{2s;\,3s}$ where $_{2s;\,3s}$ are stacking sequences repeated two times and three times respectively before symmetry. The thickness of the layer selected for analysis is 0.218 mm. The ply orientations, symmetry and balance of laminate is shown in Fig.5.

Mathematical analysis using Classical Lamination Theory

The basic assumptions in the classical lamination theory are:

1. Fibers are uniformly distributed throughout the matrix.
2. Perfect bonding exists between fibers and matrix and matrix is free of voids.
3. Applied loads are either parallel or normal to the fiber direction.
4. The lamina is in stress-free state initially.
5. Both fibers and matrix behave as linearly elastic materials.

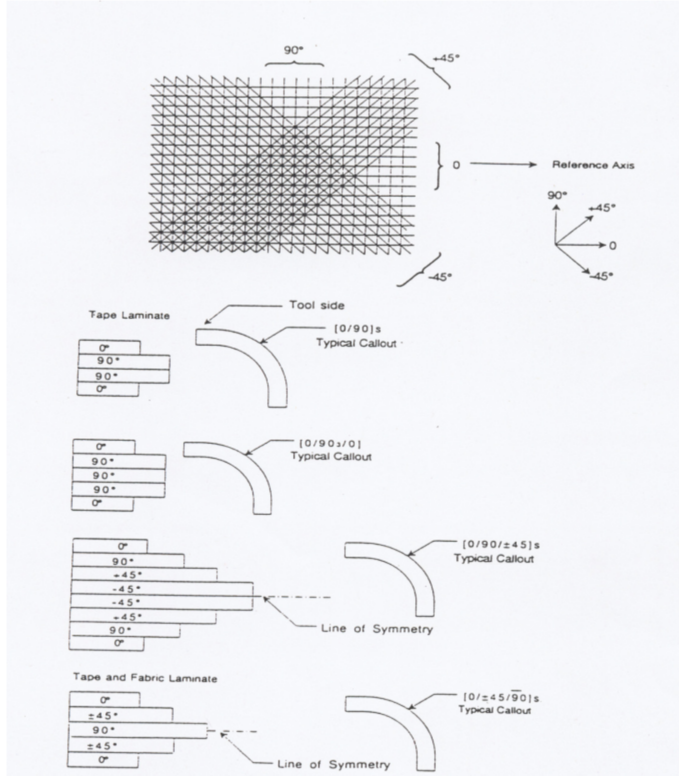

Fig. 5 Ply Orientations, Symmetry and Balance

Classical Lamination Theory

The classical Lamination theory is useful in calculating stresses and strains in each lamina of a thin laminated structure. Beginning with the stiffness matrix of each lamina, the step-by-step procedure in lamination theory includes calculation of stiffness matrices for the laminate,

calculation of mid-plane strains and curvature for the laminate

Material used

For the present study the material used for the composite laminate is in the following proportions,

Fiber – Carbon HM (60 % by weight)

Matrix – Epoxy Resin (40 % by weight)

Let,

'f' be the suffix for fibers

'm' be the suffix for matrix.

Young's Modulus of fiber, E_f = 230 GPa

Fiber Density, $\rho_f = 1.9$ gm/cm^3

Poisson's Ratio for Fiber, μ_f = 0.3

Poisson's Ratio, μ_m = 0.4

Young's Modulus for Epoxy Matrix, E_m = 2.41 GPa

Matrix density, $\rho_m = 1.1$ gm/cm^3

To calculate fiber volume fraction & matrix volume fraction,

The fiber weight fraction, $w_f = 0.6$

Therefore matrix weight fraction = w_m

$w_m = 1 - w_f = 0.4$

Fiber volume fraction (v_f) can be calculated as,

$$v_f = \frac{w_f / \rho_f}{\dfrac{w_f}{\rho_f} + \dfrac{1-w_f}{\rho_m}}$$
$$= 0.467$$
$$= 46.7\% \qquad (25)$$

and matrix volume fraction (v_m) can be calculated as,

v_m = 1- 0.467 (26)

= 0.533

= 53.33 %

Therefore the composite density (ρ_c) is,

ρ_c = 1 / {(w_f/ ρ_f) + (1- w_f) / ρ_m } (27)

= 1.478 gm / cm^3

Elastic properties

Assuming perfect bonding between fibers and matrix,

$e_f = e_m = e_c$ (28)

where,

e_f = Longitudinal strains in fiber

e_m = Longitudinal strains in matrix

e_c = Longitudinal strains in composite

Since both fiber and matrix are elastic, the respective stress can be calculated as

$s_f = E_f e_c$ and $s_m = E_m e_c$ (29)

Here $E_f >> E_m$, we conclude that the fiber stress S_f is always greater than the matrix stress S_m.The total tensile force P applied on the composite lamina is shared by the fiber and matrix so that

P = $P_f + P_m$ (30)

$s_c A_c = s_f A_f + s_f A_m$ (31)

Where s_c = average tensile stress in the composite

A_c = area of the composite

$A_c = A_f + A_m$

$V_f = A_f / A_c$

$V_m = A_m / A_c$

Thus,

$s_c = s_f V_f + s_m V_m$ (32)

$s_c = s_f V_f + s_m (1-V_f)$

The longitudinal modulus for the composite can be rewrite as,

E_{11} = E_x = E_f * V_f + E_m (1- V_f) (33)

= ($230x10^9$ x 0.467) + ($2.41x10^9$) x (1- 0.467)

= 108.7 GPa

E_{22} = E_y = (E_f * E_m) / (E_f * V_m + E_m V_f) (34)

= (230x2.41) / (230x0.533 + 2.41x0.467)

= 4.48 GPa

The following equations are used to calculate the elastic properties of an angle ply lamina in which continuous fiber are aligned at an angle θ with the positive X direction .

$\mu_{12} = \mu_{xy} = \mu_m V_m$ (35)

= (0.3 x 0.467) + (0.4 x 0.533)

= 0.3533

$\mu_{21} = \mu_{yx} = (E_{11} / E_{22}) * \mu_{xy}$ (36)

= (4.48 / 108.7) * 0.3533

= 0.0145

Shear modulus G is related to tensile modulus E as

$$G = \frac{E}{2(1+m)} \qquad (37)$$

$$\therefore G_f = \frac{E_f}{2(1+m_f)} = \frac{230}{2\,(1+0.3)} \qquad (38)$$

G_f = 104.5 G Pa

$$G_m = \frac{E_m}{2(1+m_m)} = \frac{2.41}{2(1+0.4)} \qquad (39)$$

G_m = 0.86 GPa

$$G_{LT=} \quad G_{12} = \frac{G_f . G_m}{G_f . V_m + G_m V_f} \qquad (40)$$

$$= \frac{104.5 \times 0.86}{(104.5 \times 0.533) + (0.86 \times 0.467)}$$

$G_{12} = G_{LT} = G_{xy}$ = 1.6 GPa

The following equations are used to calculate the elastic properties of an angle ply lamina in which continuous fiber are aligned at an angle ? with the positive direction. (Jones 1975)

$$\frac{1}{Exx} = \frac{Cos^4 q}{E_{11}} + \frac{Sin^4 q}{E_{22}} + \frac{1}{4}\left(\frac{1}{G_{12}} - \frac{2m_{12}}{E_{11}}\right) Sin^2 2q \qquad (41)$$

$$\frac{1}{Eyy} = \frac{Sin^4 q}{E_{11}} + \frac{Cos^4 q}{E_{22}} + \frac{1}{4}\left(\frac{1}{G_{12}} - \frac{2m_{12}}{E_{11}}\right) Sin^2 2q \qquad (42)$$

$$\frac{1}{G_{xy}} = \frac{1}{E_{11}} + \frac{2m_{12}}{E_{11}} + \frac{1}{E_{22}} - \left(\frac{1}{E_{11}} + \frac{2m_{12}}{E_{11}} + \frac{1}{E_{22}} - \frac{1}{G_{12}}\right) Cos^2 2q \qquad (43)$$

$$m_{xy} = E_{11}\left[\frac{m_{12}}{E_{11}} - \frac{1}{4}\left(\frac{1}{E_{11}} + \frac{2m_{12}}{E_{11}} + \frac{1}{E_{22}} - \frac{1}{G_{12}}\right) Sin^2 2q\right] \qquad (44)$$

Fig.6 represents the laminate geometry. Fig.7, Fig.8 and Fig.9 represents the effect of angle of ply of laminate on mechanical properties of composite, coefficient of thermal expansion and coefficient of moisture respectively.

Table 15 represents the Mechanical properties of Carbon-epoxy laminate with angle of ply. Table 16 represents the comparison of properties of composite and steel . It is observed that the composite has five times more damping coefficient than steel and this is very useful in ship. If this composite material is incorporated in the engine foundation and to the wall of engine room then vibration and noise from the engine to the hull will be reduced.

Table 15 Mechanical properties of Carbon-epoxy laminate with angle of ply

θ^0	E_{xx} (GPa)	E_{yy} (GPa)	G_{xy} (GPa)	$\mu_{xy} = \mu_{12}$	$\mu_{yx} = \mu_{21}$
0^0	108.7	4.48	1.6	0.3533	0.0145
45^0	4.703	4.699	4.186	10.845	10.836
-45^0	-10.351	4.699	4.186	10.845	10.836
90^0	4.48	108.7	1.6	0.3533	8.582

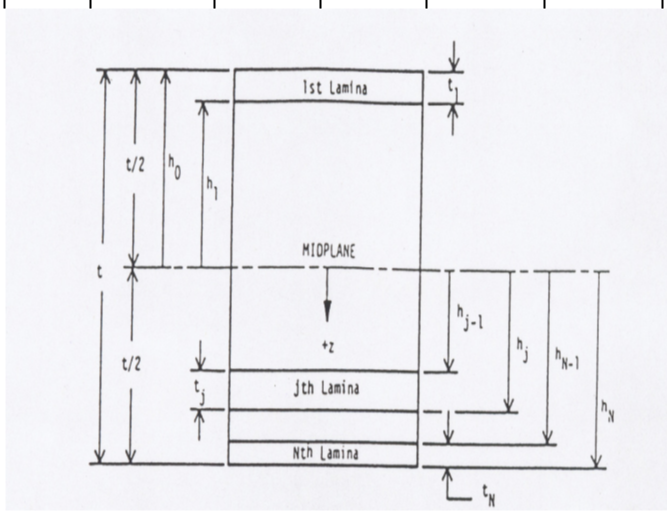

Fig.6 Laminate geometry

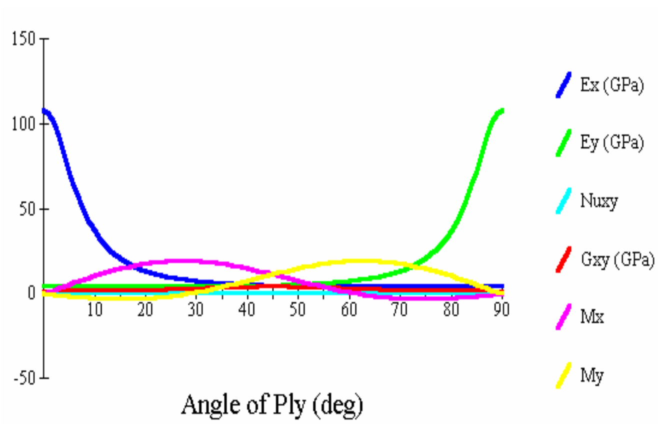

Fig.7 Effect of angle of ply of laminate on mechanical properties of composite

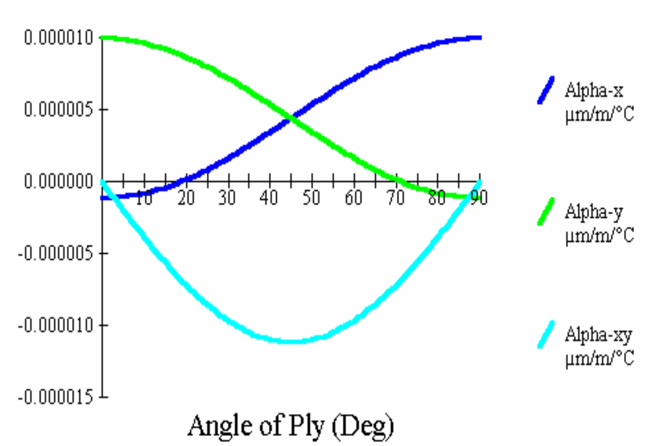

Fig.8 Effect of angle of ply of laminate on coefficient of thermal expansion

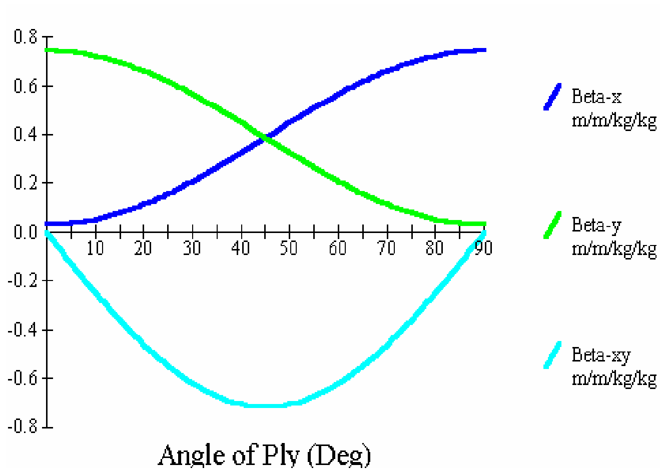

Fig.9 Effect of angle of ply of laminate on coefficient of moisture

Table 16 Comparison of properties of composite and steel

Sr. No.	Properties	Carbon - Epoxy Laminates	Carbon steel
01	Longitudinal Youngs Modulus, GPa	108.6	207
02	Transverse Youngs Modulus, GPa	4.48	79
03	Major Poisson’s Ratio	0.3533	0.20
04	Inplane Shear Modulus	1.6	--
05	Ply Thickness, mm	0.21875	--
06	Longitudinal Tensile Strength	1.55E+03	380 N/mm^2
07	Longitudinal Compressive Strength	4.7E+02	300 N/mm^2
08	Transverse Tensile Strength	31.0	---

09	Transverse Compressive Strength	31.0	---
10	Inplane Shear Strength	31.0	
11	Coefficient of Thermal Expansion / C^{o} Dir 1	-1.10E-06	12 E-06
12	Coefficient of Thermal Expansion Dir 2	1.0E-05	---
13	Coefficient of Moisture Expansion Dir 1	3.26E-02	---
14	Coefficient of Moisture Expansion Dir 2	7.468E-01	---
15	Damping Coefficient	0.005	0.001

RESULT AND DISCUSSION

The Fig. 10 represents the effect of LBP on second harmonics in cpm, it is observed that the second harmonics of horizontal vibrations are having the greater values than the second harmonics of vertical vibration. In both the cases the frequency of vibration decreases with the increase in LBP of ships upto 360m. The further increase in LBP=460m will have the reduction in second harmonics of horizontal hull vibration. This could be due to the rise in dwt capacity. The increase in dwt and LBP values also reduces the cavitation effect since propeller rpm is quite low. The third and forth mode vertical vibrations are accommodated within these second harmonics of hull vibrations, (Volcy 1976, Volcy 1984).

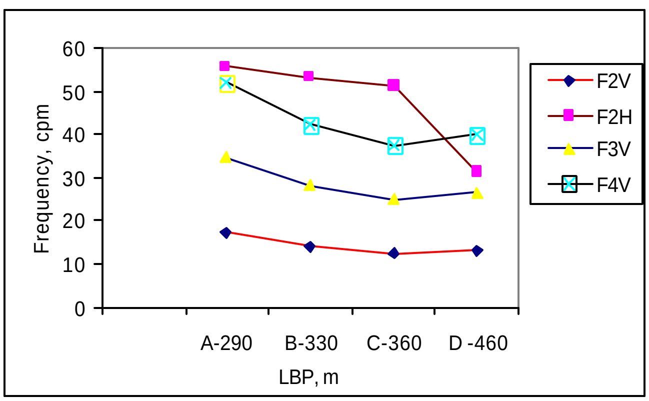

Fig.10 Effect of LBP on vertical and horizontal natural frequency of hull vibration

The Fig.11 represents the effect of LBP on highest second harmonics, cpm. It is obvious that the highest second harmonics of hull vibration is gradually decreased with increased in LBP till LBP =360m and further increase in LBP = 460m causes the sudden drop in the highest second harmonics of hull vibration. The result of fig. 3 – 4 pointed out that the optimized design dimensions of the containership obtained are more appropriate in reduction of hull vibrations. (Volcy 1978, Volcy 1986).

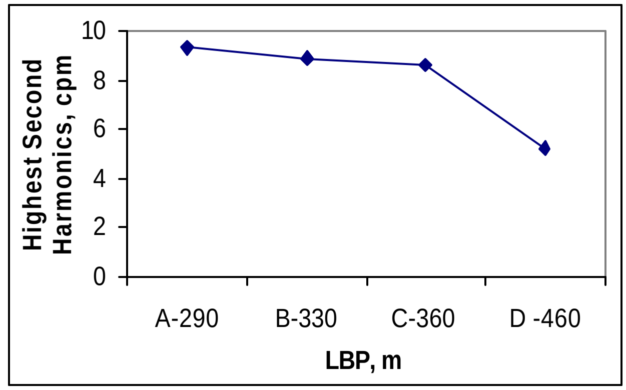

Fig. 11 Effect of LBP on highest second harmonics of hull Vibration

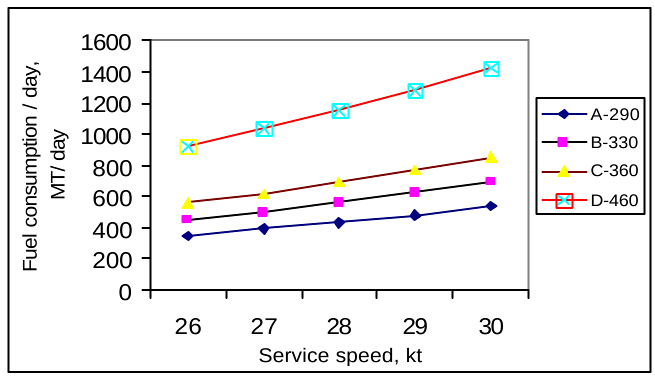

Fig. 12Effect of LBP on service speed and fuel consumption per day

The Fig.12 represents the effect of LBP on the service speed and fuel consumption. It is observed that, the increase in the service speed from 26kt–30kt for respective containership A-D demanded more fuel per day. This could be due the more power requirement of ship since the frictional as well as total resistances offered to the ship by the sea is increased.

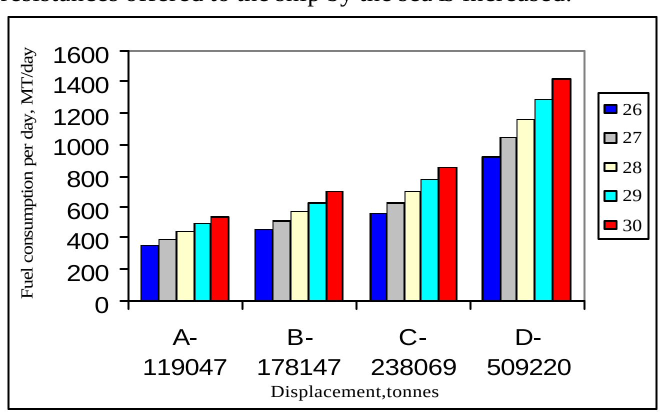

Fig. 13 Effect of displacement on service speed and fuel consumption per day

Fig.13 represent the effect of displacement on service speed and fuel consumption per day. It is observed that the fuel consumption per day increases with increase in the dwt of ship. It is also observed that the increase in the service speed of the respective dwt ship is also demands more fuel per day. The reason behind that could be again same as mentioned above.

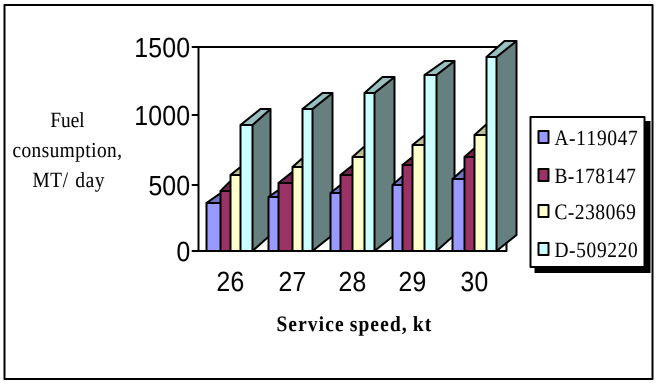

Fig. 14 Effect of service speed on the fuel consumption per day for containerships A-D

Fig. 14 represents the effect of service speed on the fuel consumption per day for containership A-D. The observation of the plot is similar to that of the observations of the fig 6 except 3D effect in the plot which may highlighted the gravity of the fuel consumption per day.

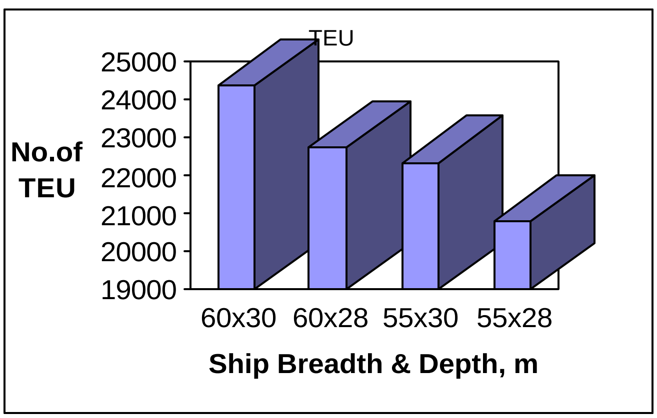

Fig.15 Effect of containership's two dimensions on TEU carrying capacity with LBP 460 m.

The Fig. 15 represent the effect of the ship dimensions (Breadth and Depth only) on TEU carrying capacity for LBP = 460m. It is obvious that the change in the dimensions of the ship directly reflected on the carrying capacity of TEU. For the dimensions 55m x 28m, the TEU carrying capacity is tremendously decreased, which will further affected the initial cost of ship.

CONCLUSIONS AND FUTURE SCOPE

The conclusion has been made that, the hull vibration due to the engine and propeller can be reduced by incorporating the more stiffened material for engine foundation and running the propeller with 6 no. of blades at low speed. The propeller pitch as well as blade area plays vital role in determination of diameter of propeller which further decides the second harmonics of vibration. Ref. Table 17, it is observed that an increase in the value of LBP and dead weight (Dwt) of ship decreases the 2-node horizontal and vertical vibration natural frequencies of hull which has been supported analytically and by FEM results.

Table 17 Comparison of 2-node horizontal and vertical vibration natural frequencies

LBP(m)	Dwt (t)	Present Work, Hz		
		Analytical		FEM
		Horizontal :	Vertical	
A-290	100000	0.92	0.28	-
B-330	150000	0.88	0.23	-
C-360	200000	0.85	0.20	0.117
D-460	463565	0.52	0.22	0.328

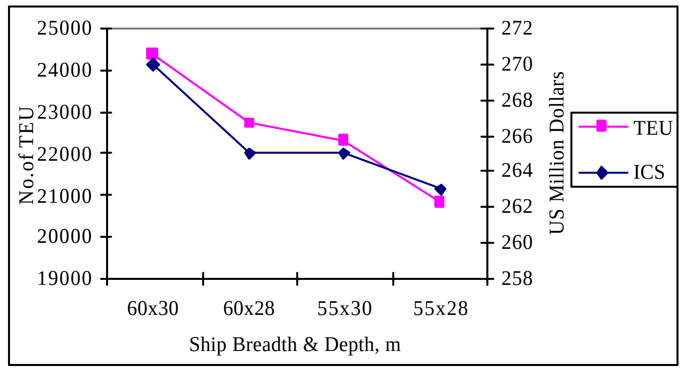

Fig.16 Effect of containership's two dimensions on TEU carrying capacity and initial cost of ship with LBP 460m.

Fig.16 represents the effect of containership's dimensions (Breadth and Depth) on TEU carrying capacity and initial cost of the ship with LBP 460m. It is observe that the initial cost of ship (ICS) drops gradually for ship's dimensions from 60x30 to 60x28, where as the TEU carrying capacity of these ships drops suddenly. Similarly for the ship dimensions 60x28 and 55x30, the initial cost of ship does not pay more attention on reduction of ICS but their TEU carrying capacity of these ships are notable. It is also observed that, variations in the ship's two dimensions from 55x30 to 55x28 not only reduces the initial cost of ship but also reduces its TEU carrying capacity, which is highly uneconomical. Finally, it is commented that the TEU carrying capacity of the containership should be high which will be beneficial to the ship owner, since this would reduces the ICCTEU, (Scheneekluth 2004).

Fig.17 represents the variations in the ICCTEU in US$ / TEU with respect to the ship dimensions and it is observed that for the optimized ship dimensions (460m x 60m x 30m), the ICCTEU cost is comparatively less and more economical to the ship owner since that ship has highest carrying capacity of TEU with marginal increase in initial cost of ship. The further scope for the present work is to analysed and control the cost of fuel consumed per day and marpol. The cost of fuel consumed per day can be control by introducing the alternative fuel for an engine where as the emission control of an engine would support to reduce the marpol. The further scope of the present work is also to reduce the light weight of ship by incorporating the composite materials for the super structures, wall and foundation of the engine room. The reduction in the light weight of the ship will not only reduces the fuel consumption per day but also further reduces the initial cost of the containership with enhancement in dead weight capacity of ship, (Carlton 2005, GL-Tech 2001, ABS 2006, Jones 1975)

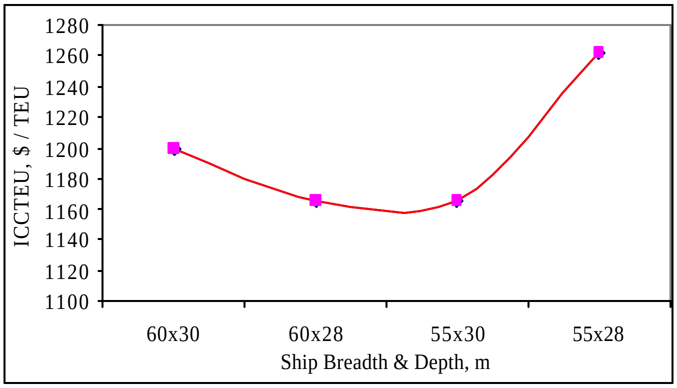

Fig. 17 Effect of container ship dimensions on ICCTEU cost in US $ per TEU for LBP 460m

Thus it is recommended to incorporate the said optimized dimensions of the container ship i.e. Containership-D, for which the second harmonics of hull vibrations is quite low and its TEU carrying capacity is 24343 TEU. As discussed earlier, KOREA'S STX Co. has already planned for the smash box containers ship with TEU capacity records to carry 22,000 TEU, where as the presented work comfortably overtaken and exceeds the carrying capacity of STX container ship by 10.60 % and able to carry 24343 TEU at 24-26knots with just one propeller.

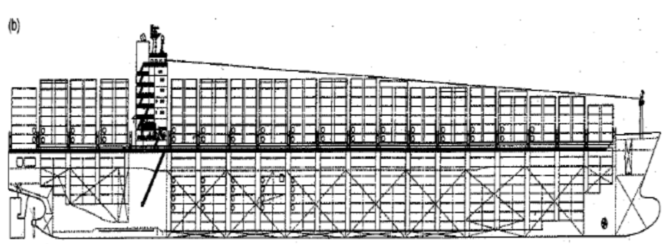

Fig.18 Propose model with optimized dimensions of newly designed containership - D (Not to the scale)

Fig.18 represents propose model with optimized dimensions of newly designed containership - D as per the requirement to reduce the initial cost of the ship (ICS) as well as ICCTEU.

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