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 containerships 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
F2y -- Second mode of vertical vibration, cpm
F2h -- Second mode of horizontal vibration, cpm
F3v -- Third mode of vertical vibration, cpm
F4v -- Forth mode of vertical vibration, cpm
t0 -- Blade thickness extrapolated to zero radius
t1 -- Blade thickness at blade tip
t/R -- Blade section radius to propeller radius
Sx -- Compressive stress of blade due to thrust and torque, MN/m²
Sy -- Tensile stress of blade due to thrust and torque, MN/m²
Sx′ -- Compressive stress blade due to centrifugal force, MN/m²
Sy′ -- Tensile stress of blade due to centrifugal force, MN/m²
β -- Todd’s constant = 8 x 10⁴ --- 11 x 10⁴
λ -- Wavelength, mts
Δv -- 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 F1= 662 cpm and F2=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 F1= 753 cpm, F2 = 1685 cpm and F3= 2365 cpm fallen into the resonance with two lowest propeller excited frequencies of F4= 346 cpm and F5= 692 cpm with amplitude of 3–4 x 10⁻² 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 an 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, \( C_b = 0.75, C_d=0.96 \)

\[ \text{Displacement} \ W = \text{L} \times \text{B} \times \text{H} \times C_b \times \rho \]

\[ = 460 \times 60 \times 24 \times 0.75 \times 1.025 \]

\[ = 509220 \text{ tonnes} \quad (1) \]

But, Light weight of ship = \[ Lwt. \text{Part} - A + \text{Lwt. Part} – B \]

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 \times \text{density of steel }]

\[ = (4622 + 10\% 4622) \times 7.80 \]

\[ = 39656 \text{ MT} \]

Table1. Light weight Part – B

<table>
<thead>
<tr>
<th>Engine + Machinery</th>
<th>Boiler + E&amp;OF</th>
<th>Super structure + Engine room</th>
</tr>
</thead>
<tbody>
<tr>
<td>4500</td>
<td>1100</td>
<td>400</td>
</tr>
<tr>
<td></td>
<td></td>
<td>6000</td>
</tr>
</tbody>
</table>

Now,

Light weight Part-B = 6000 tonnes

Hence from eqs. (2) and (3),

Light weight of ship = \[ Lwt. \text{Part} - A + \text{Lwt. Part} – B \]

\[ = [39656 + 6000] \]

\[ = 45656 \text{ tonnes} \quad (4) \]

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

<table>
<thead>
<tr>
<th>Item</th>
<th>Volume m³</th>
<th>Density T/ m³</th>
<th>Weight MT</th>
<th>No. of tank</th>
</tr>
</thead>
<tbody>
<tr>
<td>Heavy fuel oil conus. Per day (1100 MT / day for Max. ( V_s = 28 \text{kt} )) Journey Max. 20 days</td>
<td>24444.44</td>
<td>0.90</td>
<td>22000</td>
<td>10</td>
</tr>
<tr>
<td>Marine diesel oil</td>
<td>2565.90</td>
<td>0.88</td>
<td>2258</td>
<td>1</td>
</tr>
<tr>
<td>Lubricating Oil</td>
<td>2566.03</td>
<td>0.742</td>
<td>1904</td>
<td>1</td>
</tr>
<tr>
<td>Fresh water</td>
<td>5130</td>
<td>1</td>
<td>5130</td>
<td>2</td>
</tr>
<tr>
<td>Ballast water</td>
<td>26709</td>
<td>1.025</td>
<td>27377</td>
<td>10</td>
</tr>
<tr>
<td><strong>Total</strong></td>
<td><strong>58669</strong></td>
<td><strong>24</strong></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

Here, Dead weight Part – A = 58669 MT

Table 3. Dead weight Part – B

<table>
<thead>
<tr>
<th>No. of Containers</th>
<th>Average Capacity, MT</th>
<th>Total weight, MT</th>
</tr>
</thead>
<tbody>
<tr>
<td>On Deck</td>
<td>10726 TEU</td>
<td>16.63</td>
</tr>
<tr>
<td>On Hold</td>
<td>13617 TEU</td>
<td>16.63</td>
</tr>
<tr>
<td><strong>Total</strong></td>
<td>24343 TEU</td>
<td>16.63</td>
</tr>
</tbody>
</table>
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 m³
Therefore, Dead weight, dwt = 58669 + 404825
= 463564 tonnes

But,
Dead weight, dwt = Displacement(W) – Light weight (Lwt)
= 509220 - 45656
= 463564 tonnes

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 m³

Now increase in base area of each TEU with clearance between two TEU along the length and breadth of ship,

\[ \text{TEU}_{BA} = 6.15 \times 2.51 \]
= 15.43 m²

**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,

\[ \frac{382}{((6.15 \times 4 ) + 1.5)} \]
= 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,

\[ 4 \times 21 \times 11 = 924 \text{ TEU} \]

and No. of TEUs in Zone - AB = 924 x 14 = 12936 TEU

**Zone – BC**

Available length = 13 m

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

3rd Bay → → 19 x 8 = 152 TEU
2nd Bay → → 17 x 8 = 136 TEU
1st Bay → → 15 x 8 = 120 TEU

Thus, no. of TEU’s in Zone BC = 408 TEU

**Zone DA’**

Available length = 15 m

Therefore only two bays are available with 7 Tiers along the breadth as follows,

2nd Bay → → 20 x 7 = 140 TEU
1st Bay → → 19 x 7 = 133 TEU

Thus, No. of TEU’s in Zone DA’ = 273 TEU

Therefore the total no. of TEU’s on HOLD (Ref. eq. 7, 8 and 9),

\[ \text{TEUH} = 12936 + 408 + 273 \]
= 13617 TEU

**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 382m from super structure,
The no. of TEU’s are as,

\[ (14 \times 4) \times (21) \times (8) = 9408 \text{ TEU} \]

**Zone – BC**

Available length = 13 m
Therefore only three bays are available with 8 Tiers along the breadth as follows,

3rd Bay  →  19 x 7 = 133 TEU
2nd Bay  →  17 x 6 = 102 TEU
1st Bay  →  15 x 5  =  75 TEU

Thus, no. of TEU’s in Zone BC = 310 TEU

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,

7th Bay  →  21 x 7 = 147 TEU
6th Bay  →  21 x 7 = 147 TEU
5th Bay  →  21 x 7 = 147 TEU
4th Bay  →  21 x 7 = 147 TEU
3rd Bay  →  21 x 7 = 147 TEU
2nd Bay  →  20 x 7 = 140 TEU
1st Bay  →  19 x 7 = 133 TEU

Thus,No. of TEU’s in Zone DA’ = 1008 TEU

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

TEUD = 9408 + 310 + 1008
      = 10726 TEU

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

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 / Cb)^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

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

<table>
<thead>
<tr>
<th>Container ship dimensions , m</th>
<th>L:B:D</th>
<th>Displacement, W , tonnes</th>
<th>Weight of Steel Hull Structure, MT</th>
<th>Total no. of TEU</th>
<th>ICS US Millions</th>
<th>ICCTEU US $ / TEU</th>
<th>TEU Capacity %</th>
<th>Recommandations</th>
</tr>
</thead>
</table>

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

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),
\[ R_f = fAV^n \]
\[ f = 0.417 + \frac{0.733}{L + 2.68} \]
\[ A = 2.59\sqrt{W_L} \]
\[ P_{ne} = R_fV \]
\[ P_e = P_{ne} + 10\% P_{ne} \]
\[ P_t = P_e \eta_{hull} \]
\[ P_d = P_t \eta_{prop} \]
\[ P_p = P_d \eta_{shaft} \]
\[ P_i = P_p \eta_{engine} \]

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**

\[ w_f = \frac{C_b}{2} - 0.05 \]
\[ w_f = \frac{V_s - V_a}{V_s} \]
\[ D_p = \frac{632.7(P_t)^{0.2}}{(RPM)^{0.6}} \]

The area of blade can be calculated as,
\[ A_b = 0.709 \frac{\pi}{4} D_p^2 \]

**Table 5 Frictional resistance and power requirement of ship**

<table>
<thead>
<tr>
<th></th>
<th></th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>Frictional Resistance ( R_f ), kN</td>
<td>2271.79</td>
<td>2961.16</td>
<td>3562.22</td>
</tr>
<tr>
<td>Total Resistance ( R_t ), kN</td>
<td>3786.32</td>
<td>4935.27</td>
<td>5937.03</td>
</tr>
<tr>
<td>Naked effective power ( P_{ne} ), kW</td>
<td>48696.34</td>
<td>63473.06</td>
<td>76356.88</td>
</tr>
<tr>
<td>Effective</td>
<td>53565.98</td>
<td>69820.36</td>
<td>83992.57</td>
</tr>
</tbody>
</table>

**Table 6 Wake fraction and propeller details**

<table>
<thead>
<tr>
<th>Container Ship - A (100000t)</th>
<th>Container Ship - B (150000t)</th>
<th>Container Ship - C (200000t)</th>
<th>Container Ship - D (463565t)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Wake fraction, ( w_f )</td>
<td>0.299</td>
<td>0.311</td>
<td>0.318</td>
</tr>
<tr>
<td>Velocity of advance, ( V_a )</td>
<td>17.5</td>
<td>17.22</td>
<td>17.04</td>
</tr>
<tr>
<td>Theoretical speed, ( kt )</td>
<td>26.76</td>
<td>28.25</td>
<td>29.75</td>
</tr>
<tr>
<td>Propeller</td>
<td>9.00</td>
<td>9.50</td>
<td>10.00</td>
</tr>
</tbody>
</table>

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 harmonics 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 harmonics 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 harmonics 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 synchronization of rpm of propeller with highest value of second harmonics 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>
<thead>
<tr>
<th></th>
<th>Diameter, m</th>
<th>Thrust on propeller, $T_p$ kN</th>
<th>Thrust on propeller blade $T_b$ kN/m²</th>
<th>Propeller pitch, $P_p$ m</th>
<th>No. of blades &amp; Area of blade, m²</th>
<th>Area of each blade, m²</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td>6067.12</td>
<td>8040.15</td>
<td>9775.71</td>
<td>6.0 44.16</td>
<td>7.36</td>
</tr>
<tr>
<td></td>
<td></td>
<td>8281.67</td>
<td>6067.12</td>
<td>8040.15</td>
<td>9775.71</td>
<td>7.36</td>
</tr>
</tbody>
</table>

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,

$$F_{2v} = \beta \sqrt{BD^3 \Delta L^3}$$

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,

$$F_{2h} = \beta \sqrt{DB^3 \Delta L^3}$$

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 $\lambda_2 = L = 460$ m and
- Frequency $F_{2v} = 13.23$ cpm

II) Third vertical mode,

- Wavelength $\lambda_3 = 2/3 \, L = 306.66$ m and
- Frequency $F_{3v} = 2 \times F_{2v} = 26.46$ cpm

III) Fourth vertical mode,

- Wavelength $\lambda_4 = 1/2 \, L = 230$ m and
- Frequency $F_{4v} = 3 \times F_{2v} = 39.69$ 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

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

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:

\[ P_{bo} = 76714.61 \text{kW}, \quad N_o = 102 \text{ rpm} = 1.7 \text{s}^{-1} \]

Maximum rated torque, \[ T_o = P_{bo} / 2\pi N_o = 7182.06 \text{kN-m} \]

For continuous service rating:

\[ P_{b1} = 0.85 P_{bo} = 65207 \text{kW} \]
\[ N_1 = 0.95 N_o = 96.9 \text{rpm} = 1.615 \text{s}^{-1} \]

For trial condition:

\[ P_{b\text{trial}} = kN^3 = P_{b1} / 1.2 = 54339.50 \text{kW} \]

Here,

\[ N_{\text{trial}} = 108.7 \text{ rpm} \]

Therefore,

\[ k = P_{b\text{trial}} / N_{\text{trial}}^3 = 12900.24 \text{kWs}^3 \]

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, \( N = 108.7 \text{ rpm} \) 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,

\[ P_b = 1.4 P_{b\text{trial}} = 1.4 \times 12900.24 \text{ (N)}^3 \text{kW} = 18060.33 \text{ N}^3 \text{kW} \]

and Torque, \( T = 1.10 T_o = 1.10 \times 7182.06 \text{kN-m} \)

So that,

\[ T = P_b / 2\pi N = 7900.26 \]
\[ = 18060.33 \text{ N}^3 / 2\pi N = 7900.26 \]

Thus,

\[ N = 1.65 \text{ rpm} = 99.47 \text{ rpm} \]

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,

\[ F_c = \frac{W^{2/3}V^3}{\text{FuelCons./day}} \]

\[ \text{FuelConsumption/day} = \frac{W^{2/3}V^3}{F_c} \]

\[ \text{FuelConsumption/day} = \frac{(119047.61)^{2/3}}{120000} (25)^3 \]

\[ = 312.657 \text{ tonnes per day} \]

(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**

<table>
<thead>
<tr>
<th>Service speed, kt ( \rightarrow )</th>
<th>( V_5 = )</th>
<th>( V_\circ = )</th>
<th>( V_2 = )</th>
<th>( V_3 = )</th>
<th>( V_4 = )</th>
</tr>
</thead>
<tbody>
<tr>
<td>Frictional Resistance, kN</td>
<td>3826.</td>
<td>4099.</td>
<td>4380.</td>
<td>4669.</td>
<td>4968.05</td>
</tr>
<tr>
<td>Total Resistance, kN</td>
<td>6376.</td>
<td>6831.</td>
<td>7300.</td>
<td>7783.</td>
<td>8280.09</td>
</tr>
<tr>
<td>Naked effective Power, ( P_{ne} ), kW</td>
<td>85295</td>
<td>94892</td>
<td>10515</td>
<td>11611</td>
<td>127789</td>
</tr>
<tr>
<td>Fuel Consumptions, MT / day</td>
<td>557.9</td>
<td>8</td>
<td>624.8</td>
<td>7</td>
<td>696.9</td>
</tr>
</tbody>
</table>

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

<table>
<thead>
<tr>
<th>Service speed, kt ( \rightarrow )</th>
<th>( V_5 = )</th>
<th>( V_\circ = )</th>
<th>( V_2 = )</th>
<th>( V_3 = )</th>
<th>( V_4 = )</th>
</tr>
</thead>
<tbody>
<tr>
<td>Frictional Resistance, kN</td>
<td>1974.</td>
<td>2114.</td>
<td>2259.</td>
<td>2408.</td>
<td>2561.4</td>
</tr>
<tr>
<td>Total Resistance, kN</td>
<td>3290.</td>
<td>3524.</td>
<td>3765.</td>
<td>4013.</td>
<td>4269.0</td>
</tr>
<tr>
<td>Naked effective Power, ( P_{ne} ), kW</td>
<td>44008</td>
<td>48950</td>
<td>54232</td>
<td>59878</td>
<td>65886.</td>
</tr>
<tr>
<td>Fuel Consumptions, MT / day</td>
<td>925.7</td>
<td>1036.</td>
<td>1156.</td>
<td>1284.</td>
<td>1422.0</td>
</tr>
</tbody>
</table>

**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,

$$S_c = \frac{C_g P_d}{Z n D^3 (c / D) (t_o / D)^2}$$

(19)

$$S_T = S_c (0.666 + C_1 \frac{t}{c})$$

(20)

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

$$S'_c = C_2 \rho n^2 D^2 \left( \frac{C_1 \tan \epsilon}{2 (t_o / D)} - 1 \right)$$

(21)

$$S'_T = C_2 \rho n^2 D^2 \left( \frac{C_1 \tan \epsilon}{3 (t_o / D)} + \frac{C_4 \tan \epsilon}{c_{max} / D} + 1 \right)$$

(22)

Where $C_0, C_1, C_2, C_3, C_4$... Coefficients for Taylor’s method

$\epsilon$ - rake angle = 10°

$D = D_p$ – Propeller diameter, m

Selecting blade section geometry of B-series propellers with skew angle 15° and rake angle 10°.

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

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

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

<table>
<thead>
<tr>
<th>r/R</th>
<th>S_c MN/m²</th>
<th>S_t MN/m²</th>
<th>S'_c MN/m²</th>
<th>S'_t MN/m²</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.2</td>
<td>216.12</td>
<td>166.51</td>
<td>25.61</td>
<td>33.05</td>
</tr>
<tr>
<td>0.3</td>
<td>190.85</td>
<td>142.74</td>
<td>25.61</td>
<td>33.05</td>
</tr>
<tr>
<td>0.4</td>
<td>175.21</td>
<td>128.21</td>
<td>25.61</td>
<td>33.05</td>
</tr>
<tr>
<td>0.5</td>
<td>166.91</td>
<td>120.10</td>
<td>25.61</td>
<td>33.05</td>
</tr>
<tr>
<td>0.6</td>
<td>164.24</td>
<td>116.58</td>
<td>25.61</td>
<td>33.05</td>
</tr>
<tr>
<td>0.7</td>
<td>167.53</td>
<td>117.55</td>
<td>25.61</td>
<td>33.05</td>
</tr>
<tr>
<td>0.8</td>
<td>182.33</td>
<td>126.71</td>
<td>25.61</td>
<td>33.05</td>
</tr>
<tr>
<td>0.9</td>
<td>227.05</td>
<td>156.61</td>
<td>25.61</td>
<td>33.05</td>
</tr>
<tr>
<td>1.0</td>
<td>--</td>
<td>--</td>
<td>25.61</td>
<td>33.05</td>
</tr>
</tbody>
</table>

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

<table>
<thead>
<tr>
<th>r/R</th>
<th>t/D</th>
<th>t</th>
<th>c/D</th>
<th>t/c</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.2</td>
<td>0.030</td>
<td>0.33</td>
<td>0.2146</td>
<td>0.1332</td>
</tr>
<tr>
<td>0.3</td>
<td>0.033</td>
<td>0.33</td>
<td>0.2430</td>
<td>0.1045</td>
</tr>
<tr>
<td>0.4</td>
<td>0.022</td>
<td>0.22</td>
<td>0.2647</td>
<td>0.0838</td>
</tr>
<tr>
<td>0.5</td>
<td>0.022</td>
<td>0.22</td>
<td>0.2779</td>
<td>0.0684</td>
</tr>
<tr>
<td>0.6</td>
<td>0.022</td>
<td>0.22</td>
<td>0.2824</td>
<td>0.0559</td>
</tr>
<tr>
<td>0.7</td>
<td>0.011</td>
<td>0.11</td>
<td>0.2769</td>
<td>0.0455</td>
</tr>
<tr>
<td>0.8</td>
<td>0.011</td>
<td>0.11</td>
<td>0.2544</td>
<td>0.0369</td>
</tr>
<tr>
<td>0.9</td>
<td>0.011</td>
<td>0.11</td>
<td>0.2043</td>
<td>0.0303</td>
</tr>
<tr>
<td>1.0</td>
<td>0.000</td>
<td>0.00</td>
<td>0.000</td>
<td>--</td>
</tr>
</tbody>
</table>

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

<table>
<thead>
<tr>
<th>r/R</th>
<th>S_c MN/m²</th>
<th>S_t MN/m²</th>
<th>S'_c MN/m²</th>
<th>S'_t MN/m²</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.2</td>
<td>264.73</td>
<td>203.96</td>
<td>30.99</td>
<td>40.00</td>
</tr>
<tr>
<td>0.3</td>
<td>233.81</td>
<td>180.13</td>
<td>30.99</td>
<td>40.00</td>
</tr>
<tr>
<td>0.4</td>
<td>214.65</td>
<td>165.37</td>
<td>30.99</td>
<td>40.00</td>
</tr>
<tr>
<td>0.5</td>
<td>204.45</td>
<td>157.52</td>
<td>30.99</td>
<td>40.00</td>
</tr>
<tr>
<td>0.6</td>
<td>201.20</td>
<td>155.01</td>
<td>30.99</td>
<td>40.00</td>
</tr>
<tr>
<td>0.7</td>
<td>205.27</td>
<td>158.14</td>
<td>30.99</td>
<td>40.00</td>
</tr>
<tr>
<td>0.8</td>
<td>223.33</td>
<td>172.06</td>
<td>30.99</td>
<td>40.00</td>
</tr>
<tr>
<td>0.9</td>
<td>278.21</td>
<td>214.34</td>
<td>30.99</td>
<td>40.00</td>
</tr>
<tr>
<td>1.0</td>
<td>--</td>
<td>--</td>
<td>30.99</td>
<td>40.00</td>
</tr>
</tbody>
</table>
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,

$$f_f = \left[ \frac{0.305t_o}{(R - r_o)^{0.5}} \left( \frac{C_o T_m g E}{C_m T_o \rho_m} \right)^{0.65} \right] \times 0.65$$

$$= 15.36 \text{ Hz}$$

$$f_t = \left[ \frac{0.92t_o}{(R - r_o C_o)^{0.5}} \left( \frac{C_o (g G)^{0.5}}{C_m (\rho_m)^{0.5}} \right) \right] \times 0.65$$

$$= 150.08 \text{ Hz}$$

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](image)

![Fig 3 Torsional vibration of containership of LBP = 460m](image)

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/mm², 0.29 and 7850 kg/m³. 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

<table>
<thead>
<tr>
<th>Container</th>
<th>Container</th>
<th>Container</th>
<th>Container</th>
</tr>
</thead>
<tbody>
<tr>
<td>KB, m</td>
<td>6.935</td>
<td>7.422</td>
<td>7.921</td>
</tr>
<tr>
<td>BM₁₄</td>
<td>10.209</td>
<td>14.45</td>
<td>17.59</td>
</tr>
<tr>
<td>KM₁₄</td>
<td>17.14</td>
<td>21.87</td>
<td>25.51</td>
</tr>
<tr>
<td>BM₂₄</td>
<td>443.41</td>
<td>536.25</td>
<td>599.27</td>
</tr>
<tr>
<td>KM₂₄</td>
<td>450.34</td>
<td>543.67</td>
<td>607.19</td>
</tr>
<tr>
<td>Width of Influence,</td>
<td>397.11</td>
<td>475.48</td>
<td>533.19</td>
</tr>
<tr>
<td>Depth of Influence,</td>
<td>91.95</td>
<td>94.94</td>
<td>99.11</td>
</tr>
<tr>
<td>Water plane area, WPA, m²</td>
<td>9641.54</td>
<td>13763.48</td>
<td>17330.08</td>
</tr>
<tr>
<td>-------------------------</td>
<td>--------</td>
<td>---------</td>
<td>--------</td>
</tr>
<tr>
<td>Tonnes per cm immersio n, TPC</td>
<td>98.82</td>
<td>141.07</td>
<td>177.63</td>
</tr>
<tr>
<td>Moment to change trim one cm, MCTC, tm/cm</td>
<td>1715.98</td>
<td>2841.29</td>
<td>3949.45</td>
</tr>
</tbody>
</table>

**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 \([0/90/45/-45]_{2s}\); 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.

**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/cm³

Poisson’s Ratio for Fiber, \(\mu_f = 0.3\)

Poisson’s Ratio, \(\mu_m = 0.4\)

Young’s Modulus for Epoxy Matrix, \(E_m = 2.41\) GPa

Matrix density, \(\rho_m = 1.1\) gm/cm³

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

$$
\frac{1}{E_{xx}} = \frac{\cos^2 \theta}{E_{11}} + \frac{\sin^2 \theta}{E_{22}} + \frac{1}{4 \left( G_{12} \right)} \sin^2 2\theta
$$

$$
\frac{1}{E_{yy}} = \frac{\sin^2 \theta}{E_{11}} + \frac{\cos^2 \theta}{E_{22}} + \frac{1}{4 \left( G_{12} \right)} \sin^2 2\theta
$$

$$
\frac{1}{G_{xy}} = \frac{1}{G_{11}} + \frac{2 \mu_{12}}{E_{11}} + \frac{1}{E_{22}} \left( \frac{1}{E_{11}} + \frac{2 \mu_{12}}{E_{11}} + \frac{1}{G_{12}} \right) \cos^2 2\theta
$$

$$
\mu_{xy} = E_{11} \left[ \frac{\mu_{12}}{E_{11}} - \frac{1}{4 \left( \frac{1}{E_{11}} + \frac{2 \mu_{12}}{E_{11}} + \frac{1}{E_{22}} - \frac{1}{G_{12}} \right)} \sin^2 2\theta \right]
$$

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

<table>
<thead>
<tr>
<th>θ°</th>
<th>$E_{xx}$ (GPa)</th>
<th>$E_{yy}$ (GPa)</th>
<th>$G_{xy}$ (GPa)</th>
<th>$\mu_{xy} = \mu_{12}$</th>
<th>$\mu_{yx} = \mu_{21}$</th>
</tr>
</thead>
<tbody>
<tr>
<td>0°</td>
<td>108.7</td>
<td>4.48</td>
<td>1.6</td>
<td>0.3533</td>
<td>0.0145</td>
</tr>
<tr>
<td>45°</td>
<td>4.703</td>
<td>4.699</td>
<td>4.186</td>
<td>10.845</td>
<td>10.836</td>
</tr>
<tr>
<td>90°</td>
<td>4.48</td>
<td>108.7</td>
<td>1.6</td>
<td>0.3533</td>
<td>8.582</td>
</tr>
</tbody>
</table>

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

<table>
<thead>
<tr>
<th>Sr. No.</th>
<th>Properties</th>
<th>Carbon - Epoxy Laminates</th>
<th>Carbon steel</th>
</tr>
</thead>
<tbody>
<tr>
<td>01</td>
<td>Longitudinal Youngs Modulus, GPa</td>
<td>108.6</td>
<td>207</td>
</tr>
<tr>
<td>02</td>
<td>Transverse Youngs Modulus, GPa</td>
<td>4.48</td>
<td>79</td>
</tr>
<tr>
<td>03</td>
<td>Major Poisson’s Ratio</td>
<td>0.3533</td>
<td>0.20</td>
</tr>
<tr>
<td>04</td>
<td>Inplane Poisson’s Ratio</td>
<td>1.6</td>
<td>--</td>
</tr>
<tr>
<td>05</td>
<td>Ply Thickness, mm</td>
<td>0.21875</td>
<td>--</td>
</tr>
<tr>
<td>06</td>
<td>Longitudinal Tensile Strength</td>
<td>1.55E+03</td>
<td>380 N/mm²</td>
</tr>
<tr>
<td>07</td>
<td>Longitudinal Compressive Strength</td>
<td>4.7E+02</td>
<td>300 N/mm²</td>
</tr>
<tr>
<td>08</td>
<td>Transverse Tensile Strength</td>
<td>31.0</td>
<td>--</td>
</tr>
</tbody>
</table>
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 1986).

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).

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.

The Fig.13 represents the effect of displacement on service speed and fuel consumption per day.
Fig. 13 represents 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.

![Figure 13: Effect of displacement on service speed and fuel consumption per day.](image)

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

<table>
<thead>
<tr>
<th>LBP (m)</th>
<th>Dwt (t)</th>
<th>Present Work, Hz</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td>Analytical</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Horizontal : Vertical</td>
</tr>
<tr>
<td>A-290</td>
<td>100000</td>
<td>0.92</td>
</tr>
<tr>
<td>B-330</td>
<td>150000</td>
<td>0.88</td>
</tr>
<tr>
<td>C-360</td>
<td>200000</td>
<td>0.85</td>
</tr>
<tr>
<td>D-460</td>
<td>463565</td>
<td>0.52</td>
</tr>
</tbody>
</table>

Fig. 14 represents the effect of service speed on the fuel consumption per day for container ships A-D.

![Figure 14: Effect of service speed on the fuel consumption per day.](image)

**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.

![Figure 16: Effect of containership’s two dimensions on TEU carrying capacity and initial cost of ship with LBP 460 m.](image)
consumed per day and marpol. The cost of fuel consumed per day can be controlled 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 container ship 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 containership 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.

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REFERENCES