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EMPIRICAL DESIGN FOR SMALL CONTAINER SHIPS

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ABSTRACT: *The paper presents a summary of the multidisciplinary/optimization method for the preliminary design of container feeder vessel. The current scenario in the ship building industry highly focuses on container ship design and construction proving the inherent demand in maritime industry. The design accomplishes the outer circle of the design spiral giving stress in areas of Hull Form Design, Resistance & Propulsion. Empirical relations, model test results, data from built ships, class rules and latest market demands stood as the criteria for the design. Optimization of the design as per the owners requirement, class rules and the trade route selected are the major challenges met with. Strength, reliability, structural safety and stability have been incorporated in compromising standards.*

1. Introduction

The paper presents the conceptual design of a Container Feeder Vessel based on empirical formula. The objective of the paper is to design a container feeder vessel with a carrying capacity of 625 TEU with a service speed of 15.75 knots. Container ships are designed for low block coefficients characterized by wide hatch openings, fixed cell guides and permanent ballast. Stability, Optimization of resistance & propulsion, torsional strength affects the design and construction. Space allotted for each container is called a cell. The cell themselves are made up of corner guide angles attached to the ship structures. Hatch covers are usually lift off pontoon type, clogged water tight by manual or hydraulic means. Disadvantages of this type is that all the containers must be of uniform length and width and with uniform fittings for lifting, stacking and locking; and that it can't be used for any other type of cargo or even another type of container without extensive conversion. The idea of unitized container cargo ensures safety and speedy transport. The paper concentrates mainly on the initial estimates of main dimensions, Hull form design from first principles, resistance and propulsion. The ship is internally divided by transverse bulkheads. Collision bulkhead and aft peak bulkhead enclose the engine room & cargo holds into watertight compartments. Water tight transverse bulkheads are placed in cargo holds as per classification society rules. Subdivisions in double bottom is based on tank capacities required for diesel oil, lub oil etc.

Superstructure & Deck houses are divided inside by decks, transverse and longitudinal bulkheads. Container ships have fine slender hull form with very little or no parallel middle body. Stern is made transom for increasing the main deck area, for easiness in construction and to reduce resistance by postponing the separation of flow at the stern. Stem is fitted with nabla type bulb. Ship is to be propelled by a low speed two stroke diesel engine fitted to a fixed pitch propeller.

2. Initial Estimates of Main Dimensions

Container ships are designed as linear dimension ship. The principle dimensions are multiples of the containers being stowed. Draft and breadth restrictions in the areas where she ply also impose limitations in design [3].

Table 1. Container Specifications

Length	6.096 m
Width	2.438 m
Height	2.438 m
Maximum load capacity	20.3 t
Weight of empty container	2.25 t
Total capacity	625 TEU

Parent ship analysis refers to carry 60% of the total containers under deck. Therefore the number of containers to be carried under deck (N_{CU}),

N_{CU} = 60% of 625 = 375

Parent ship analysis will give an appropriate idea of the main dimensions of the ship. The standard dimensions of a twenty feet container are marked as reference for breadth wise stowage.

As a result, the arrangement of containers in the midship region inside the hold is selected as 7 x 5 and the dimensions are fixed accordingly. Maximizing the container stowage in the midship, taking a form factor of 1.25, Number of files will come up to 14. The Length is estimated as per the container stowage as this is a Linear dimension ship.

From the number of files calculated, the length between perpendiculars can be calculated providing sufficient clearances in the longitudinal directions and considering the space requirement for the cargo hold, engine room, fore peak and aft peak tanks. The Draft corresponding to the required deadweight is 7.4 m. The preliminary dimensions and coefficients are shown in table 2

Table 2. Preliminary Main Dimensions

LBP	129 m
B	26.3 m
D	13.12 m
T	7.4 m
C_B	0.66
C_W	0.78
C_M	0.97
C_P	0.68

3.Development of Preliminary Lines

Preliminary lines are developed from first principles (TOWNSENDS Method [4]). Standard values of LCB, Parallel middle body, Angle of Entrance, Area of Water plane, Area of Stations and Sectional Area curve are the major criteria in determining the form of lines. Referring to charts of Townsends, following parameters are determined.

Table 3. Townsends Parameters

AB/ LBP	0.487
Position of LCB from AP	62.823 m
Half angle of entrance	13.2°
Length of entrance, Lent	0 .45LBP
Length of entrance, Lrun	0.45LBP
Length of entrance, Lpar	0.1LBP
Cp forward region, CPF	0.66
Cp aft region, CPA	0.72

Corresponding to the CPA and CPF values, areas as fractions of midship sectional areas are read out from the charts

Area midship section, $A_M = B \times T \times C_M = 188.78\ m^2$

The sectional area curve is plotted with reference to the ratio of area of stations with respect to the midship area. The sectional area curve is thus marked as the reference to the final fairing process and optimization of the volume displacement. The Hull is faired manually keeping all the parameters and form coefficients at the derived values.

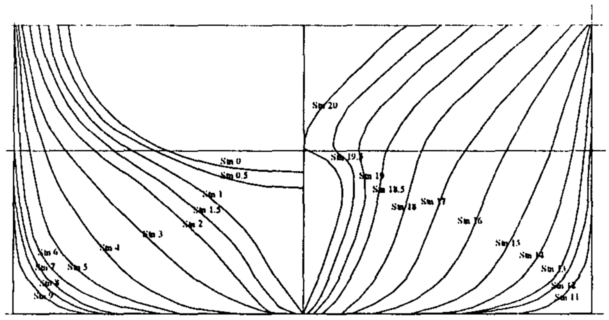

Fig. 1. Designed Body Plan

The fairing process took into consideration, the stowage plan to satisfy the required number of containers to be stowed and hence developing a hull form compromising both in terms of capacity to be carried as well as the seaworthiness of the ship.

4. Resistance & Powering

4.1 Resistance Estimation

The preliminary resistance calculations are performed using Guldhammer & Harvard method and validated using NavCAD Software. Guldhammer and Harvard method [8] divides the total resistance coefficient in to non-dimensionalized Residuary and Frictional Resistance coefficients. Residuary resistance coefficient has been plotted against Froude number for different values of prismatic coefficient. This data is available for different values of $L/\nabla^{1/3}$. Frictional Resistance Coefficient is calculated on the basis of the International Towing Tank Conference (ITTC) 1957. Appropriate corrections are given for deviations from the standard values.

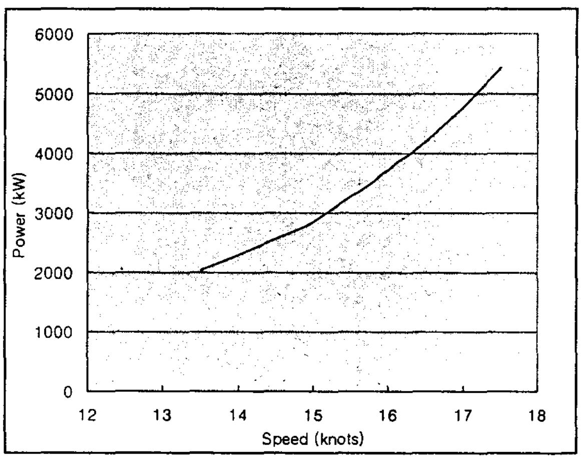

Fig. 2. Speed Power Curve

4.2 Powering & Propulsion

Powering [8] begins with initial approximation of the diameter of the propeller and the Thrust corresponding to the calculated resistance. The entire powering calculations and propeller design is done using T-J P-J charts. The required power and rpm of the engine are determined and an initial engine selection is done with sufficient deration using T-J charts. Again the thrust and diameter are found out using the selected engine power using P-J charts. Finally the calculated thrust should exceed the required thrust and the diameter should not exceed the initial approximation for optimum design. Propulsion system to be employed is of single screw fixed pitch type as it is mostly used in cargo ships. This system is much simple, and consequently less costly to purchase and operate. The ship has to operate normally under the service speeds and any kind of emergencies as in warships is hardly dealt with. Wageningen B series [Oosterveld, 1975] was selected for the design of propeller.

The design starts with the calculation of thrust deduction factor, wake fraction and hull efficiency using empirical formulas.

Wake Fraction

$W = 0.5\ C_B - 0.05 = 0.28$ (Taylors formula)

Thrust Deduction Factor

$t = 0.5\ C_P - 0.12 =$ 0.22 (Hecksher formula)

Hull efficiency

$$\eta_H = \frac{(1-t)}{(1-w)} = 1.126 \qquad (1)$$

Required Thrust

$$T = \frac{1.15\ R_T}{(1-t)} = 614.72 \text{ kN} \qquad (2)$$

Velocity of Advance

$$V_A = V(1-w) = 5.785 \text{ m/s}^2 \qquad (3)$$

Propeller Diameter

D = (2/3) T = 4.93 m

Assume the value of diameter to be 4.93 m initially, value of Td is calculated read the values of K_Q and J from T-J charts.

$$T_d = \frac{1}{DV_A}\sqrt{T/\rho} = 0.859 \qquad (4)$$

Minimum Blade Area Ratio

$$(A_E/Ao)_{\min} = \left[\frac{(1.3+0.3Z)T}{(P_{atm}+\rho gh - PV)D^2}\right] + K \qquad (5)$$

K = 0.2 for single screw propellers

Z = number of blades

h = height of LWL above shaft central line

$(A_E/A_O)_{min}$ is calculated for three and four bladed propellers and K_Q and J values are found out form T-J charts at the intersection of constant T_d and maximum efficiency (η_D) line for T_d constant.

P_{atm} = 101.366 kN/m2

PV = 1.704 kN/m2

h = 4.8 m

D = 4.93 m

K = 0.2 for single screw propellers

ρ = 1.025 t/m^3

g = acceleration due to gravity (9.81 m/s^2)

Number of blades is selected based on the open water efficiencies of three and four bladed propellers corresponding to minimum blade area ratio. The values of K_Q and J with respect to expanded blade area ratio (A_E/Ao) is found from T -J charts and then interpolated for the optimum value. Comparing open water efficiencies (ηo) of the two propellers the four blade propeller has more efficiency than three bladed propeller. Hence four bladed propeller is selected. Accounting a Shaft efficiency (η_S) of 97 % and interpolating the value of velocity of advance (J), rps (n) and K_Q,

P_D = $2\pi n^3 \rho D^5 K_Q$ = 6258.265 kW

P_B = $P_D / \eta S$ = 6451.82 kW

N = 121.18 rpm

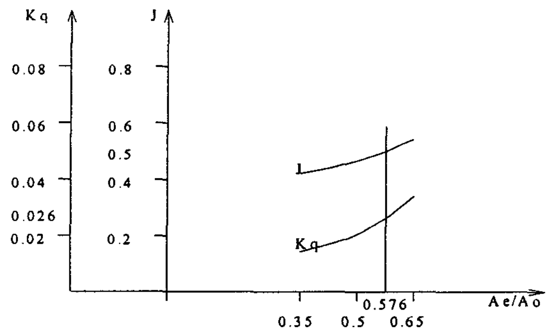

Fig: 3. A_E/A_O vs. KQ and J graph for 3 bladed propeller

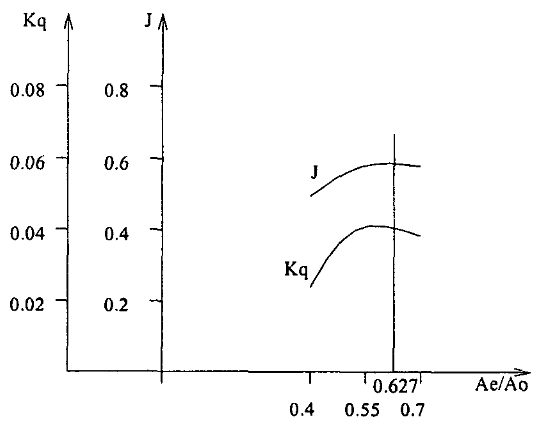

Fig: 4. A_E / A_O vs. K_Q and J graph for 4 bladed propeller

The engine is always run at 10% lesser than the maximum power. ∴ Giving 10 % increase in required main engine power,

P_B	=	7097 kW
N	=	133.3 rpm

Based on this power, Engine is selected.[5]

4.3 Engine Selection

Company	=	MAN B & W	
Model	=	10 S 35 MC	
Power	=	7,400 KW	
Taking 90% of MCR, P_B	=	6660 kW	
$\therefore P_D$	=	$P_B \times \eta_S$	(6)
	=	6460.2 KW	

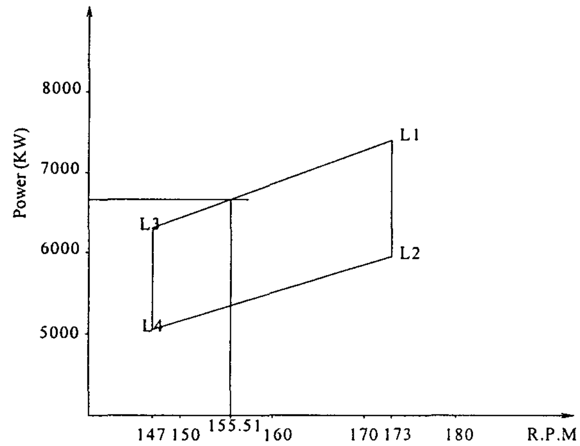

Fig 5. Engine Power v/s RPM Interpolation

Rpm (N) at derated power is taken from the engine catalogue by interpolation with respect to power as shown in figure 5.

Hence corresponding to P_B = 6660.2 kW,

Rpm = *155.51*

Now it is required to optimize the diameter of the propeller, so as to match the engine selected and completing the iteration. This is done using P-J charts. Here J values are read for constant Pn lines, intersecting at maximum efficiency line for Pn constant. The J values are increased by 6 % to get J', to account for the fouling and roughness of seawater. The K_T and P/D values are read off at the point of intersection of J' and Pn line. Then the following quantities are found using the formulas given below.

$$P_n = \frac{n}{V_A^2}\sqrt{\frac{P_D}{2\pi\rho V_A}} = 1.0199 \qquad (7)$$

$$D = V_A / Jn \qquad (8)$$

$$T = \rho n^2 D^4 K_T \qquad (9)$$

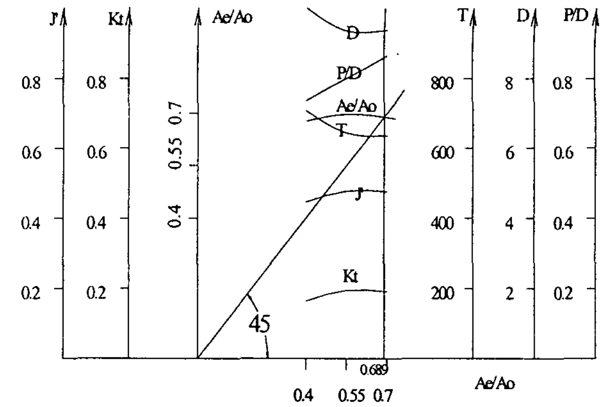

Fig 6. Propeller Characteristics Curve

K_T, J, P/D, T, D, A_E/A_0 min values are tabulated from the above formulas. A graph is drawn with A_E/A_0 on X-axis and other parameters on Y-axis (Fig 6). From the graph the optimum values are obtained by drawing a 45° line to X-axis. The values of T and D are with in the limits as stated. Hence the selected preliminary propeller design is satisfactory.

4.4 Propeller Geometry

The design geometry consists of four parts [9].

- An expanded blade outline
- Side Elevation
- Pitch Diagram
- Transverse View
- Propeller geometry made by using Wageningen B Series Charts.

The design and drawing of the propeller has been done in AutoCAD.

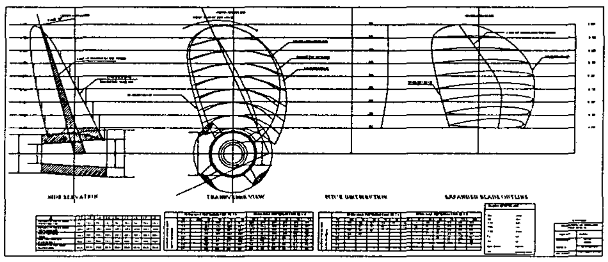

Fig 7. Propeller Geometry

5. Conclusion

It is required to check the propeller performance i.e. whether the propeller is able to propel the ship at the required speed. For this the propeller power is calculated for various speeds. For the same speeds the engine power is also calculated and the values are plotted against velocities. The curves intersect at some point, which will give maximum attainable velocity using this propeller. The performance characteristics are checked for trial and service conditions. For service condition 15 % and for Trial condition 7% increase in resistance is taken in to account. Thrust and the coefficient Td are found for different speeds. From T-J charts, values of K_Q and J are taken, utilizing these values rpm, P_D (propeller) and ηo are found.

$$rps = V_A / (JD)$$

$$P_D\,(propeller) = 2\,\pi n^3 \rho D^5 K_Q \qquad (10)$$

$$\eta o = TV_A / P_D$$

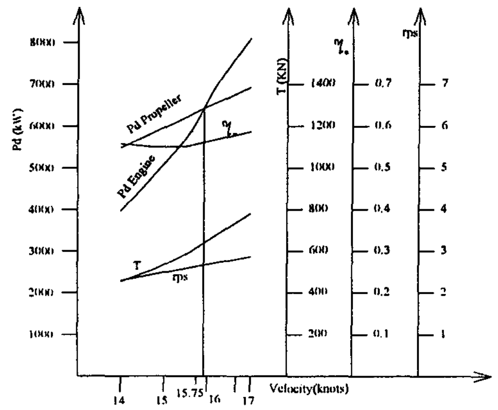

Fig 8. Performance Curves - Service Condition

A graph is drawn with speed on X axis and PD(propeller), P_D(engine), Thrust and ηo on Y axis as shown in figure 8 and 9 for service and trial conditions respectively. Hence, performance check is satisfied in both service and trial conditions.

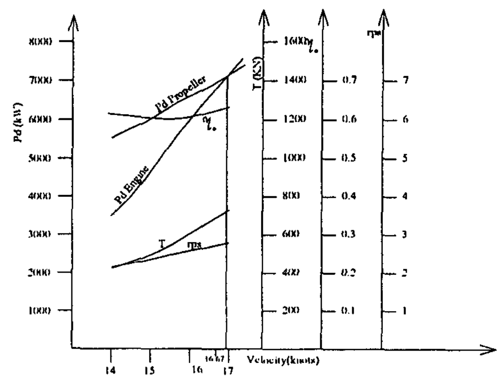

Fig 9. Performance Curves - Trial Condition

Although the optimization of ship lines is performed through the use of model tests, empirical methods are applied in the preliminary design stage. This empirical design strategy is a reliable and quick reference method in the initial stage of ship design with sufficient amount of accuracy and easiness. It will give a quick insight for the customers to assess the main dimensions of the vessel for the initial cost estimation. Owing to its simplicity and precision, this empirical formulation can be an efficient and proper tool in the preliminary stage of ship design.

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