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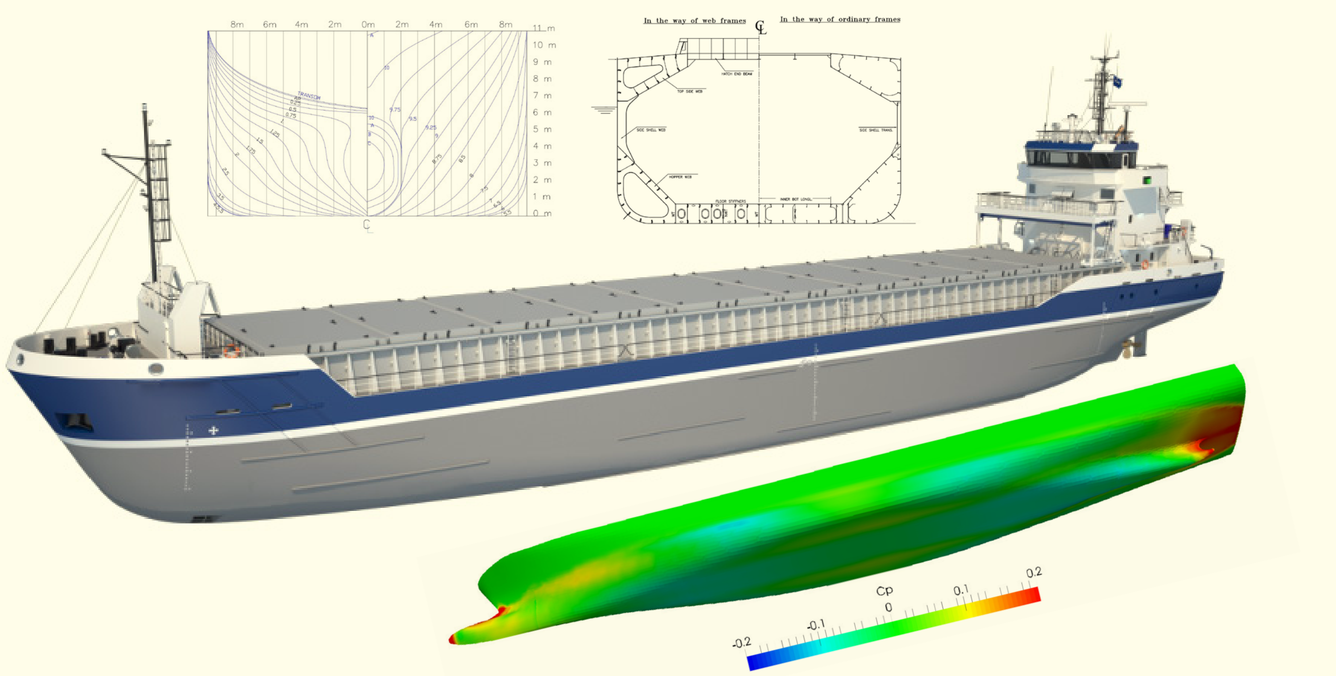
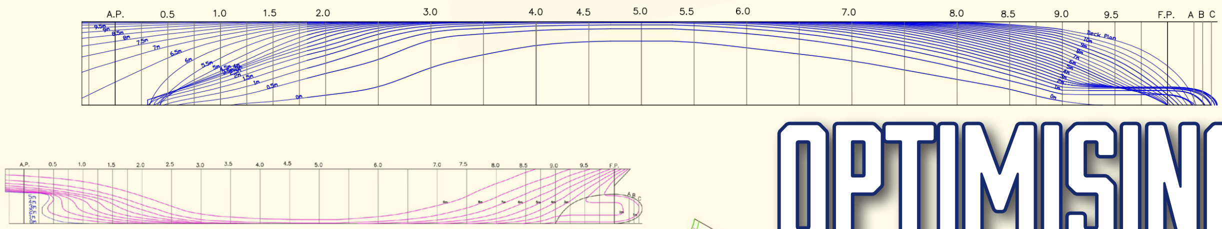
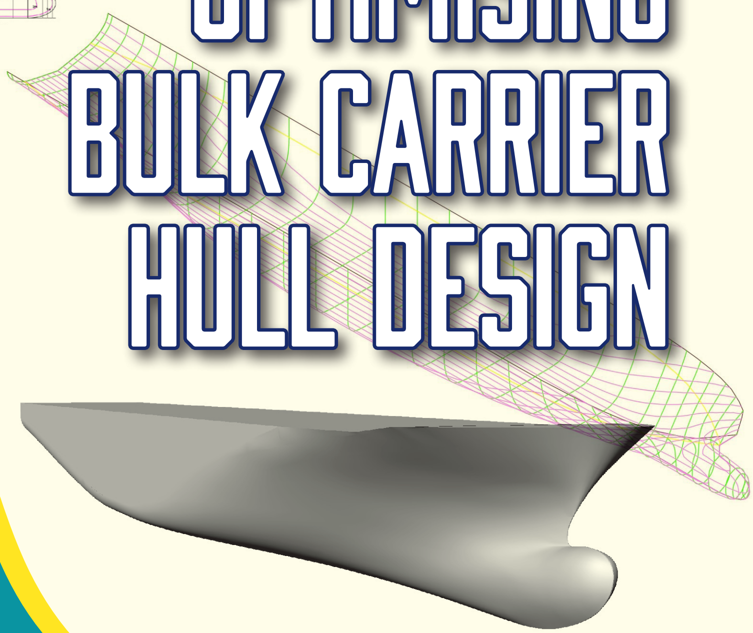




OPTIMISING BULK CARRIER HULL DESIGN



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EDITORIAL

If I had learned education, I would not have had time to learn anything else.

- Cornelius Vanderbilt

The Virus Vacation seems to be coming to a close. Indian schools are opening up to education-as-usual track. The Budget bus passing by, and the bean counters are crunching the numbers on the GDP, employed/unemployed and of course, the infected/injected/immune/departed. Missing in this wake are the education infographics. India's 320 million school children lost almost 69 weeks (can throw a few more while we settle down), which could possibly be the highest hiatus (maybe Uganda with >83 is on top), if compared globally. Some peripheral areas of the peninsula (also in the islands), the connectivity had been poor causing much lower dissemination efficiencies.

And Maritime Education and Training hiccupped with the higher education programmes. We might be staring at stigmatised 'Corona Cohorts' or 'Pandemic Pass-outs', becoming the less preferred among the aspirants. This again is another moment for the Nation to gain wisdom...

Let us just step back and imagine that we never had the internet... the computers... the communication... Would we have been better?

Would we have innovated methods and kept the learning levels ticking?

May be we would have invented *Gurukuls...* isolated igloos for home-honing of the children...

May be the learning would have been more relaxed... more relevant... more conducive... more fitting... more fulfilling than the present education format, which our society has turned into an employment ePass gateway.

Moving back into the teach-with-technology times... let us hope that this inflection point would lead the generations to a learning without the education hindering it.

In this issue...

Optimisation is the catchword for all real-life problems that science and technology try to address. Optimising the design being an integral part of ship construction, has economic and efficiency merits. Dr. Kesavadev *et al*., present a study of a bulk carrier's mid-section design. The valuable takeaway is the Genetic Algorithm (GA) process and the establishments of the constraints. GA approaches, built on the premise of 'Natural selection of the Fittest' have found good favour for many engineering and other applications. This should be of interest to the Naval Architecture students pursuing research.

From optimisation, we move into a slow variable speed drive of propulsion systems. Dr. Veda, in his inimitable style, expounds reliability assessments for a DP System. The system dynamics and the analyses are described falling back on the Power Generation and Management System discussed in earlier parts. An interesting takeaway to those working on DP vessels is the tabulation highlighting the modes of DP applications. The inputs needing advertence are: vessel resistance parameters obtained through hydrodynamic models; the shaft speed calculation from propeller-engine torque balance and the discussion on IGBT redundancies. This Part E is another absorbing read in the series.

Under Lube Matters, Sanjiv Wazir continues to give valuable insight into viscosity. The Heritage Hourgalss is full on both sections (or say, a double delight) in this instance... We have an eminent marine engineer/scribe, Narasiah talking about ancient maritime trade. This will be a worthwhile distraction from the technical tenor. And Sadaf Khan takes us back to the Mughal time boat building.

This March must bring the much needed relief to normalcy. With that hope, here is the March issue.

Dr Rajoo Balaji
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In This Issue

ARTICLES

COLUMNS

OPTIMISATION OF BOTTOM STRUCTURE OF SHIPS USING GENETIC ALGORITHMS

Kesavadev V. K., Vidyasagar D.S.P., Avinash Godey, Jayaram S

Abstract: Structural design of ship's hull is a critical activity that has to be carried out by naval architects or ship designers to ensure that a ship can withstand different loading conditions it undergoes throughout its lifespan. Reduction of structural weight without compromising with the strength is an important task in ship design because an increase in structural weight leads to reduced pay load and more fuel consumption which causes increased fuel cost and environmental pollution due to increased emissions. Classification society rules may be followed to ensure structural strength. At the same time, it is possible to optimise the scantlings using various optimisation techniques subject to rule requirements. Optimisation of scantlings of bottom structural members in the midship region of a bulk carrier using genetic algorithms, subject to LR rules and regulations of classification of ships is discussed in this article.

Keywords: Midship section design, Optimisation, Genetic algorithm.

1. INTRODUCTION

Over the past few decades design of hull structures has become more sophisticated because of latest developments in computational methods, higher standards set by classification societies and other statutory bodies, fast developments in the field of structural design software and enormous growth of computer hardware. These have made complex calculations and simulations dealing with huge amount of data possible in relatively short period of time.

Cost of hull steel in case of conventional ship types usually ranges between 20 to 25 percent of total cost of the ship

Structural strength of the ship is one of the most important factors which should be ensured by designers because any sort of structural failure may result in major financial losses, human fatalities and marine pollution. Cost of hull steel in case of conventional ship types usually ranges between 20 to 25 percent of total cost of the ship. Therefore, reducing the hull weight without compromising with the strength of ship's structure is of paramount importance. In general, hull structure is divided into three regions namely forward, midship and aft.

Midship region consists of 40 percent of a ship's length. Design of structures in these three regions of the ship are addressed separately in classification society rules because of the difference in the type of loads, probable failure modes, structural configuration and hull shapes. In rule-based design, structural scantlings are obtained from formulae given in the rule books. However, currently rules insist on FEM analysis and other advanced methods to ensure the adequacy of scantlings obtained from rule book formulae. It is possible to optimise the midship scantlings in the preliminary design stage using optimisation techniques ensuring that the rule requirements are satisfied.

2. GENETIC ALGORITHMS

Genetic algorithms are adaptive heuristic search algorithms based on the evolutionary ideas of natural selection and genetics. These are commonly used to generate optimal or sub optimal solutions of optimisation and search problems by depending on biologically inspired operators such as mutation, crossover and selection. In a genetic algorithm, a population of candidate solutions in the form of chromosomes which are encoded form of solutions are used. These solutions are also called as **individuals** or **phenotypes**. A standard representation of each candidate solution is as an array of bits. Arrays of other types and structures may be used to represent solutions in certain problems.

A set of individuals or solutions with specific size is called a population and number of solutions in a population is termed as population size which depends on the nature of the optimisation problem to be solved. Initial population, which may be termed as the first generation, consists of randomly generated individuals. Evolution is initiated from first generation itself by applying selection, crossover and mutation. To carry out selection, fitness of individuals in each generation are calculated using fitness function.

Usually, fitness function is same as the objective function. However, fitness function may be defined as sum of the objective function and a suitable penalty function to deal with infeasible solutions included in the population. Genetic search from both feasible and infeasible regions of the solution space can be made possible by including certain number of infeasible solutions with feasible ones in each generation.

Most of the selection methods, follow a stochastic approach to select more fit solutions from a population. These selected solutions are modified by applying genetic operations such as crossover and mutation to obtain next generation or a new population. This new population is used in the next iteration of the genetic algorithm. The algorithm terminates when it completes pre-decided number of generations or reaches satisfactory fitness level. Crossover and mutation which may be termed as the major genetic operations carried out in genetic algorithms are briefly discussed below.

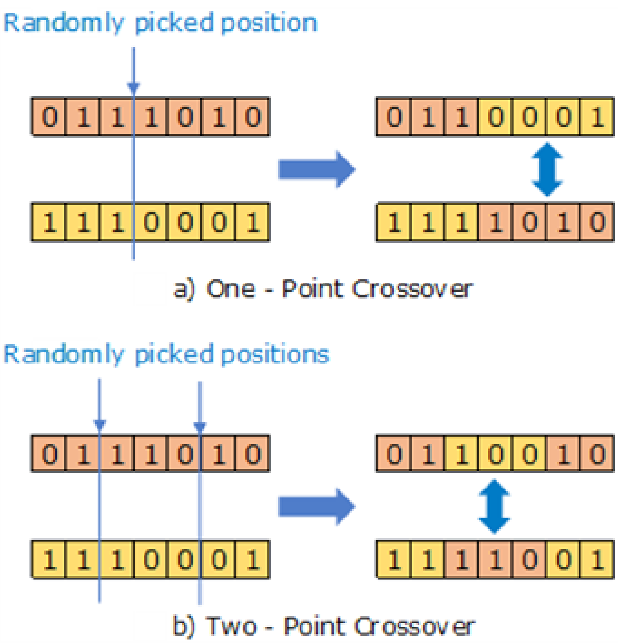

Figure 1 Crossover

Crossover: It is the main genetic operation in Genetic Algorithms. Crossover operates on two chromosomes at a time and generate offspring by combining the features of those chromosomes. Commonly used methods of crossovers are one-point crossover and two-point crossover as shown in **Figure 1**. The crossover rate p_c is defined as the ratio of number of offspring produced in each generation to the population size. This ratio controls the expected number of chromosomes to undergo crossover operation.

Mutation: Mutation is a background operation to produce spontaneous random changes in various chromosomes.

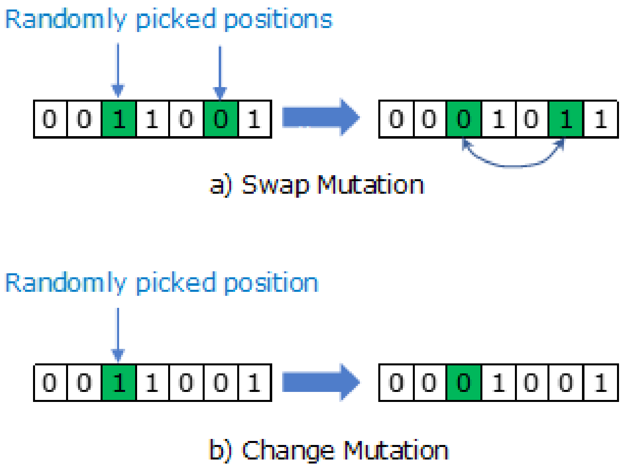

Figure 2 Mutations

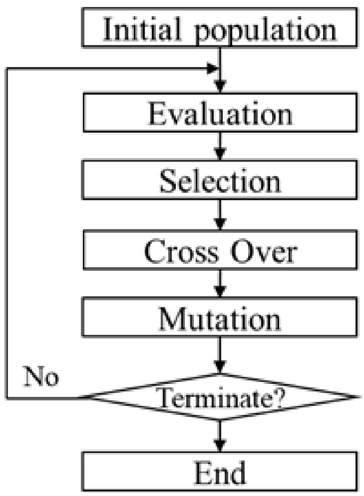

Figure 3 Simple Genetic Algorithm

Commonly used mutation methods are change mutation and swap mutation which are as shown in **Figure 2.** The mutation rate p_m is defined as the percentage of total number of genes in the population considered for mutation. The mutation rate controls the rate at which new genes are introduced into the population.

There are different variants of Genetic Algorithms such as Simple Genetic Algorithm, Niched Pareto Genetic Algorithm, Non-dominated Sorted Genetic Algorithm and so on. In this paper Simple Genetic Algorithm, as shown in **Figure 3**, is considered for the optimisation of bottom structure of ships.

3. DEFINITION OF THE OPTIMISATION PROBLEM

In rule-based design of ship structures, scantlings of structural components are calculated using the formulae given in the classification society rules. In this process, loads acting on the ship are also calculated using the rule book formulae. After obtaining the scantlings of the structural components, its adequacy is checked for different failure modes such as yielding, buckling, fatigue and so on. In most of these structural adequacy checks, local strength analysis is carried out.

In rule-based design of ship structures, scantlings of structural components are calculated using the formulae given in the classification society rules

However, global strength of the ship is also checked by calculating bending stress developed at the mid ship section and by carrying out ultimate strength analysis of hull girder following the calculation procedures given in the classification society rules. In addition to these calculations, class rules insist for strength adequacy check using Finite Element Method analysis for different loading conditions after making three hold finite element models.

Optimisation of bottom structure of the midship region can be carried out in the early design stage by formulating the rule-based design as an optimisation problem. In the optimisation problem, there are design variables, objective functions and constraints. Optimisation of bottom structure in the midship region is defined as a single objective constrained optimisation problem. In order to make the problem an unconstrained optimisation one, a penalty function is introduced and defined using the constraints. Initial conditions, Design variables, Objective Function, Fitness Function and constraints are as given below.

Initial conditions: Structural configuration and scantlings of all parts in the midship region except bottom structures are finalised prior along with scantlings of floors and bottom girders. Bottom and inner bottom longitudinal stiffeners are assumed as T-sections.

Design variables: Following are the design variables considered.

- Number of bottom or inner bottom longitudinal stiffeners (N_s)
- Thickness of bottom plating (tb_j, here '*j*' indicates strake number)
- Thickness of Inner bottom plating (tib_j, here '*j*' indicates strake number)
- Flange width of bottom longitudinal stiffeners ($wfbl_j$, here '*j*' indicates stiffener number)
- Flange width of inner bottom longitudinal stiffeners ($wfibl_j$, here '*j*' indicates stiffener number)
- Flange thickness of bottom longitudinal stiffeners ($tfbl_j$, here '*j*' indicates stiffener number)
- Flange thickness of inner bottom longitudinal stiffeners ($tfibl_j$, here '*j*' indicates stiffener number)
- Web depth of bottom longitudinal stiffeners ($dwbl_j$, here '*j*' indicates stiffener number)
- Web depth of inner bottom longitudinal stiffeners ($dwibl_j$, here '*j*' indicates stiffener number)
- Web thickness of bottom longitudinal stiffeners ($twbl_j$, here '*j*' indicates stiffener number)
- Web thickness of inner bottom longitudinal stiffeners ($twibl_j$, here '*j*' indicates stiffener number)

Constraints: Requirements as per LR rules and regulations of classification of ships (2018) applicable to bottom structures in the midship region are considered as constraints in this optimisation problem.

$C_{1j} = tb_j - tbr \geq 0$ *here j = 1 to l wherein l indicates total number of bottom strakes.*

$C_{2j} = tib_j - tibr \geq 0$ *here j = 1 to m wherein m indicates total number of inner bottom strakes.*

$C_{3j} = zb_j - zb \geq 0$ *here j = 1 to n wherein n indicates total number of bottom longitudinal stiffeners.*

$C_{4j} = zib_j - zib \geq 0$ *here j = 1 to n wherein n indicates total number of inner bottom longitudinal stiffeners.*

$C_5 = \sigma - \frac{175}{k} \geq 0$ *here 'σ' is the maximum value of bending stress at the midship section and 'k' is the material factor. In order to calculate bending stress, bending moment is obtained from formulae given in the rule book.*

In the above expressions tbr, tibr, zb and zib indicate minimum values of bottom plate thickness, inner bottom plate thickness, section modulus of bottom longitudinal stiffeners and section modulus of inner bottom longitudinal *stiffeners respectively, as per LR rules and regulations of classification of ships (2018).*

Objective function: Steel weight of bottom structural members for unit length is considered as the objective function ***'f (x)'*** which has to be minimised to achieve an optimum solution.

$$f(x) = \sum_{j=1}^{k} Wbs_j + \sum_{j=1}^{l} Wibs_j + \sum_{j=1}^{m} Wbl_j + \sum_{j=1}^{n} Wibl_j \quad (1)$$

Here,

Wbs_j *=Weight of* j^{th} *strake of bottom plate.*

$Wibs_j$ *=Weight of* j^{th} *strake of inner bottom plate.*

Wbl_j *=Weight of* j^{th} *bottom longitudinal.*

$Wibl_j$ *=Weight of* j^{th} *inner bottom longitudinal.*

*Here '**k**' and '**l**' indicate number of bottom and inner bottom strakes.*

*'**m**' and '**n**' indicate number of bottom and inner bottom longitudinal stiffeners.*

Fitness Function: Fitness function, ***F (x)*** is defined as sum of objective function and penalty function in the optimisation of bottom structure in the midship region.

$$F(x) = f(x) + p(x) \quad (2)$$

*Here **f(x)** is the objective function whereas **p (x)** is the penalty function. Expression for objective function is as given in (1). Penalty function is as given below.*

$$p(x) = a.\sum_{j=1}^{k} P_{1j} + b.\sum_{j=1}^{l} P_{2j} + c.\sum_{j=1}^{m} P_{3j} + d.\sum_{j=1}^{n} P_{4j} + e.P_5 \quad (3)$$

*Here **a**, **b**, **c**, **d** and **e** are multipliers for constraints related to scantlings of bottom plating, inner bottom plating, bottom longitudinal stiffeners, inner bottom longitudinal stiffeners and maximum bending stress at midship respectively.*

$$P_{ij} = \begin{cases} C_{ij} & if\ C_{ij} < 0 \\ 0 & if\ C_{ij} \geq 0 \end{cases} \quad \text{Here } i = 1 \text{ to } 4$$

$$P_5 = \begin{cases} C_5 & if\ C_5 < 0 \\ 0 & if\ C_5 \geq 0 \end{cases}$$

4. IMPLEMENTATION OF SIMPLE GENETIC ALGORITHM

In order to materialise optimisation using simple genetic algorithm, it is required to encode the solutions. In the problem of optimisation of bottom structure of ships, a solution can be represented with a set of design variables mentioned in the above section and each solution can be encoded as a binary chromosome. To carry out selection process, the fitness function is to be calculated. For this purpose, objective function and penalty function are calculated after decoding the chromosome and finally fitness function is obtained.

Tournament selection with suitable tour size, one-point crossover and change mutation are the genetic operations to be used. Suitable population size and generations may be considered to achieve an optimal solution.

5. EXAMPLE

In order to demonstrate applicability of the optimisation approach discussed in this paper, optimisation of bottom structure of a bulk carrier of 82000 tonnes deadweight is carried out using Excel VBA codes. Bottom structural arrangement of the ship is shown in **Figure 4**. Range of design variables are given in **Table 1**. GA parameters and operations considered for carrying out the optimisation are given in **Table 2**. Optimisation results are given in **Figure 5**.

Scantling values corresponding to four solutions obtained during the optimisation process are given in **Table 3**. Values of objective function, penalty function, fitness and feasibility of the four solutions are given in **Table 4**. Out of the four solutions given, Solution-1 and 2 are not feasible. Solution-3 and 4 are feasible ones and Solution 4 may be considered as the optimum.

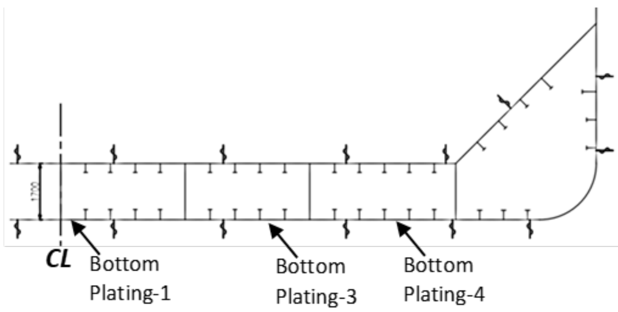

Figure 4 Bottom structural arrangement of an 82000 tonnes dwt bulk carrier

Table 1: Design variables with their ranges

Design Variable	Range
Floor Spacing (***S***)	[2000 mm, 2511 mm]
Number of Bottom / Inner Bottom Longitudinal stiffeners (N_s)	[6, 27]
Web Depth of Inner Bottom Longitudinal stiffeners (d_{w-ibl})	[200 mm, 327 mm]
Web Thickness of Inner Bottom Longitudinal stiffeners (t_{w-ibl})	[8 mm, 15 mm]
Flange Width of Inner Bottom Longitudinal stiffeners (w_{f-ibl})	[100 mm – 163 mm]
Flange Thickness of Inner Bottom Longitudinal stiffeners (t_{f-ibl})	[17 mm, 24 mm]
Web Depth of Bottom Longitudinal stiffeners (d_{w-bl})	[200 mm, 327 mm]
Web Thickness of Bottom Longitudinal stiffeners (t_{w-bl})	[8 mm, 15 mm]
Flange Width of Bottom Longitudinal stiffeners (w_{f-bl})	[80 mm, 207 mm]
Flange Thickness of Bottom Longitudinal stiffeners (t_{f-bl})	[15 mm, 30 mm]
Thickness of Inner Bottom Plating (tib_1)	[16 mm, 30 mm]
Thickness of Bottom Plating-1 (tb_1)	[15 mm, 22 mm]
Thickness of Bottom Plating-2 (tb_2)	[15 mm, 22 mm]
Thickness of Bottom Plating-3 (tb_3)	[15 mm, 22 mm]
Thickness of Bottom Plating-4 (tb_4)	[15 mm, 22 mm]
Thickness of Bottom Plating-5 (tb_5)	[15 mm, 22 mm]

Table 2: GA parameters and operations

Population size	100
Generations	100
Selection method	Tournament selection with tour size of 2
Crossover	One – point crossover with cross over rate of 90 %
Mutation	Change mutation with mutation rate of 1 %

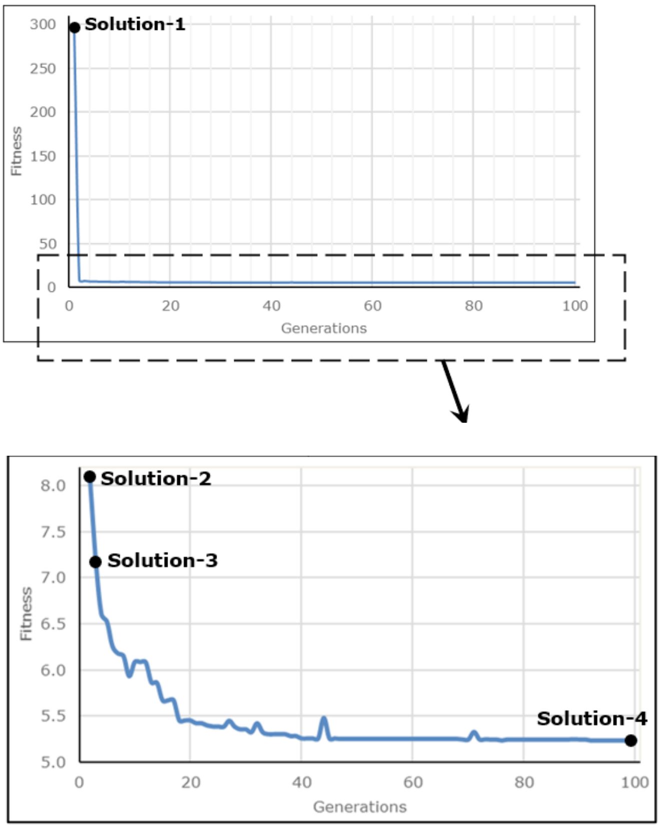

Figure 5. Optimisation Results (Fitness Vs. Generations)

6. CONCLUSIONS

Obtaining an optimal ship structure with reduced structural weight without compromising the strength is a significant task in design of ships. A process for obtaining optimum design of midship section by combining rule book concepts and genetic algorithms is presented in this paper. In order to demonstrate the applicability of the method, some simple codes were written in Excel VBA and an example problem was solved.

Table 3: Solutions obtained during the optimisation process

Scantlings ↓	Scantling values in "mm"			
	Solution-1	Solution-2	Solution-3	Solution-4
Floor Spacing	2025	2099	2268	2268
Longitudinal Spacing (mm)	1250	417	469	469
Inner Bottom Longitudinal Web depth (mm)	230	242	242	200
Inner Bottom Longitudinal Web thickness (mm)	15	12	12	8
Inner Bottom Longitudinal Flage width (mm)	109	109	109	100
Inner Bottom Longitudinal Flage thickness (mm)	20	24	24	17
Bottom Longitudinal Web depth (mm)	318	292	292	200
Bottom Longitudinal Web thickness (mm)	8	8	8	8
Bottom Longitudinal Flage width (mm)	97	151	151	80
Bottom Longitudinal Flage thickness (mm)	18	22	22	15
Thickness of Inner Bottom Plating (mm)	23	22	22	20
Thickness of Bottom Palting-1 (mm)	16	19	19	19
Thickness of Bottom Palting-2 (mm)	22	18	18	17
Thickness of Bottom Palting-3 (mm)	20	20	19	19
Thickness of Bottom Palting-4 (mm)	21	16	19	19
Thickness of Bottom Palting-5 (mm)	22	21	22	17

Table 4: Feasibility and fitness of the solutions

	Solution-1	Solution-2	Solution-3	Solution-4
Value of objective function	5.09	7.56	7.18	***5.23***
Value of Penalty function	288.92	0.48	0	***0***
Fitness	294.01	8.04	7.18	***5.23***
Feasibility	Not Feasible	Not Feasible	Feasible	***Feasible***

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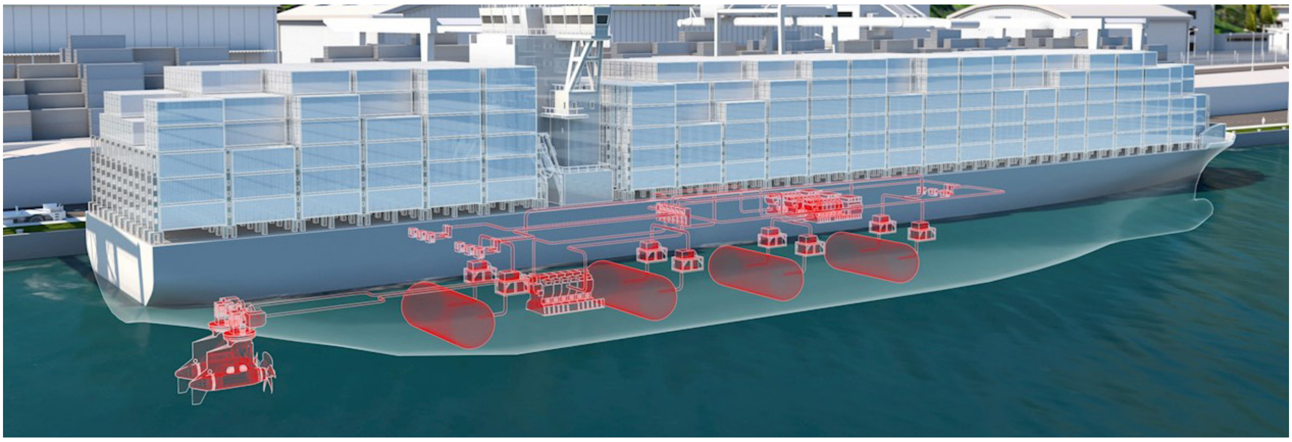
RAMS-CENTERED SYSTEM ENGINEERING AND OPERATIONS OF MODERN MULTI-MEGAWATT CAPACITY MARINE POWER SYSTEMS – PART E



N.Vedachalam
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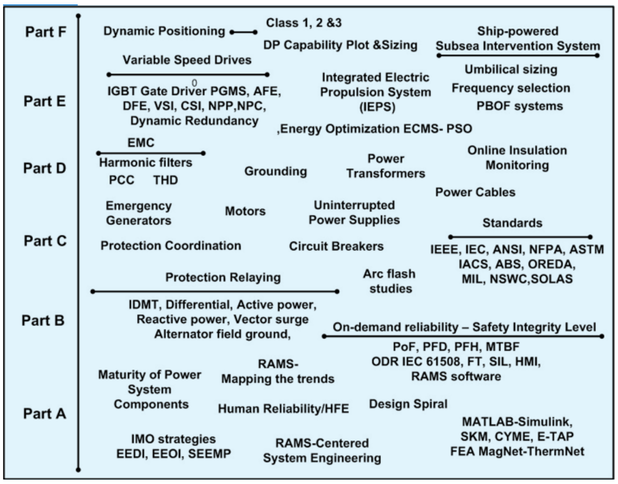
Abstract

This article in six parts (this is the fifth, Part E) discusses the importance, trends and integrated approach to RAMS-centred system engineering, key design and operational considerations for low- and medium-voltage marine power systems, including alternator protection, effective protection coordination, integrity requirements of relaying, emergency diesel generators and uninterrupted power supplies, significance of grounding, condition monitoring of power transformer, cables, motors, harmonics filters and the methodologies for realising fault-tolerant voltage source inverter based variable speed drives and dynamic positioning systems.



Index terms: Alternator, Cables, Grounding, Harmonics, Medium Voltage, Motors, Protection Coordination, VSD

Part A (first part), Part B (second part), Part C (third part) and Part D (fourth part) of the series were published in the November 2021, December 2021, January 2022 and February 2022 issues respectively.

Dynamic Positioning Systems

Reliability is the key requirement for Dynamic Positioning (DP) systems used in seismic survey, offshore supply vessels and platforms engaged in complex offshore developments and processes including Mobile Offshore Drilling Units (MODU), diving, pipelay, umbilical lay, Remotely Operated Vehicle (ROV) support, floating production, lifting and accommodation.

DP is based on measuring the attitude changes of the vessel/platform and applying linear and angular forces to counteract the external forces and to maintain the vessel/platform in a fixed position exclusively by means of active thrusters, without the aid of mooring lines and/or anchors.

The developments in high-precision attitude sensors, environmental sensors, commercialisation of the global navigation satellite systems, state-of-the-art acoustic base line positioning systems, computer-based control systems and algorithms, combined with RAMS-centered design practices have resulted in reliable DP systems with position-keeping accuracies up to 2m.

The architecture of a typical vessel DP system including Power Generation and Management Section (PGMS), Computer Control and Sensor Section (CCSS) and Thruster Systems (TS) are shown in **Figure 1**.

The guidelines published by the International Marine Contractors Association (IMCA) for operational planning and execution of offshore marine vessel projects and routine offshore support activities comprise of Critical Activity Mode (CAM), Task Appropriate Mode (TAM) and Activity Specific Operating Guidelines (ASOG).

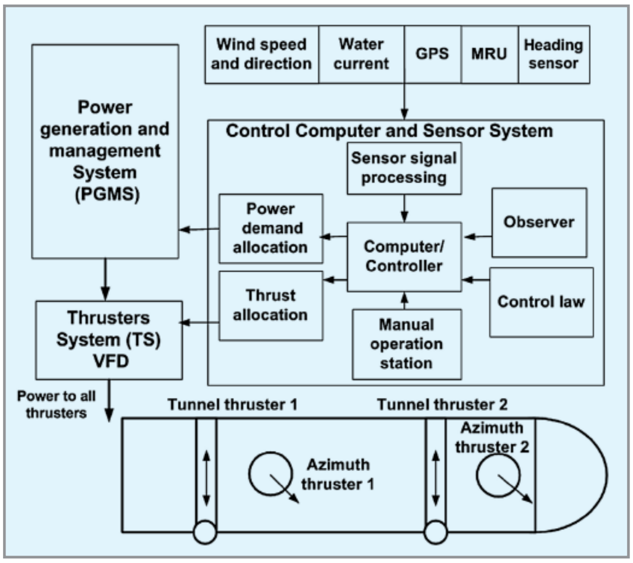


Figure 1. Architecture of a typical vessel DP system

The CAM is the configuration that the vessel's DP system should be set up and operated so as to deliver the intent of the vessel's DP class notation, with the objective that no single failure should result in exceeding the Worst-Case Failure (WCF).

The TAM is a risk-based mode in which the configuration of the vessel's DP system may be set up and operated in, accepting that a single failure could result in exceeding the WCF and could result in blackout or loss of position.

The design of a vessel or platform DP system involves optimising the hull, propulsion, thruster and power configuration. The DP capability plots (IMCA M140 Rev 1) define, by theoretical calculation, the vessel's capability to maintain position in various environmental conditions (including wind, sea state, water current) appropriate to the area in which the DP vessel is to operate, and external forces such as pipe tension, and in various thruster/power configurations, including all thrusters running, loss of most effective thruster, and following the WCF.

Simulation tools based on accurate vessel modeling and time-domain simulations helps to evaluate the vessel statistics of motion in all degrees of freedom (DoF), thruster loads, power utilisation and fuel consumption.

The results of simulations are presented in Polar diagram with the centre of rotation of he vessel at the centre of the plot. The power envelope is shown in **Figure 2**, where the angle in the envelope is the wind direction relative to the vessel heading. The wind rotates 360° and current and waves follow with the relative angles.

The design of a vessel or platform DP system involves optimising the hull, propulsion, thruster and power configuration

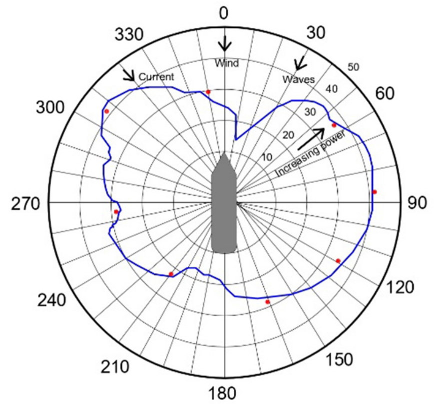


Figure 2. Power envelope for a DP system with estimated and actual performances in red dots

Modern tools have the capability to simulate conditions for DP operations such as black-out prevention, load limitation and sharing, auto-start and auto-stop of the power generators

The total power consumption from the experimental results is shown in red dots, while the simulated power consumption is shown by the blue envelope, which defines the acceptance criteria, that is evaluated during the sea trials. Modern tools have the capability to simulate conditions for DP operations such as black-out prevention, load limitation and sharing, auto-start and auto-stop of the power generators.

During full scaled trials, DP capability (heading deviation and position deviation) pertaining to head wind, bow quartering wind, beam wind, stern quartering wind ,etc., are carried out for various values of water currents and wave influences.

During regular operations, DP footprint plots are taken whenever opportunities arise, such as during standby periods, weather downtime, or on arrival at the field. Plots should be taken for the thruster configurations that are used in the DP capability plots. The details on the various modes of DP operations are summarised in **Table 1**.

According to IMCA database with 71 DP-related incidents reported from 54 vessels during 2014, 54% of the failures were due to technical failures in the DP system and the rest were due to human error, procedural and environment-related (**Figure 3**) [1].

Taking into consideration the increasing number of DP vessels and the demand on risk reduction with increased HSE regulations, IMO and International Association of Classification Societies (IACS) insist fault-tolerant DP systems using redundant architectures to prevent vessel Loss of Position (LoP) during critical operations.

Table 1. Various modes of DP applications

Mode	Details
Target follow	Enables the DP vessel to follow a moving target e.g. to follow an ROV along a pipe line
Heavy Lift	Takes account of the effects of the load transfer on the mass of the vessel and the additional lateral force, normally by reducing gain and relaxing the DP controller.
External Force Compensation	Where the measured external force acting on the vessel, which is separate from environment, is included in DP calculation and treated as a force feed forward. This mode is used to account for pipe tensions.
Fire Monitor Compensation	Used to compensate for the varying forces exerted on a vessel from the fire monitors
Weathervane	Enables the DP vessel to rotate with the wind, current, and waves around a fixed or moving point called the terminal point. Neither the heading nor the position of the DP vessel is fixed. The heading of the vessel is controlled to point towards the terminal point. The position of the vessel is controlled to follow a circle, called the set point circle, around the terminal point. This mode is appropriate for connected shuttle tanker/Floating Production Storage and Offloading (FPSO) operations

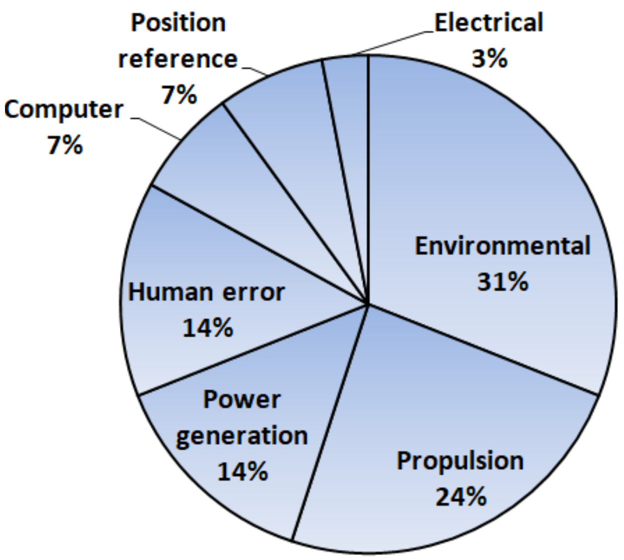


Figure 3. Contributors to DP failure

According to IMO guide-lines Msc.1/Circ 1580, DP vessels are classified under Class 1, 2 and 3, and their requirements in terms of system operational availability are shown in **Table 2**.

Reliability assessment for DP system

Based on the component/subsystem failure rates summarised in **Table 3**, reliability modeling is done for IMO-defined DP1, DP2 and DP3 architectures. The redundancy definition for the sensors and systems recommended by IMO for the three DP classes are summarised in **Table 4**.

The GRIF-FTA software is used for modeling the failure tress (FT) and exponential failure law is chosen as the degradation pattern of the components during the simulations [3]. In the FT, the gates are represented based on IEC 60125 conventions. icons with rectangles boxes representing events below, but folded for easier representation in the FT diagram.

Table 2. IMO Classification of DP and its fault tolerance [2]

Class	Feature
1	A LOP may occur in the event of a single fault
2	A LOP may not occur in the event of a single fault in any active component or system such as generator, thruster etc. But LoP may occur in the event of a single failure in static equipment such as power cables, power bus bars etc
3	A LOP may not occur in the event of single fault of any or static component including complete loss of compartment due to fire or flood. The redundant systems are separated by bulkheads and decks that are fire insulated as per A-60 class divisions.

Table 3. Failure-in-time (FIT) data used for the analysis

Subsystem	FIT	Source
Engine	17000	IEEE [4]
Alternator	11000	
Circuit breaker and relays	1400	
PGMS	27530	OREDA [5]
Control Computer System	17370	
Wind speed and direction sensor	51505	NIOT [6]
Motion Reference Unit	11273	
Gyroscope	33000	
Water current log	63800	
GPS hardware	10200	Novatel [7]
Thruster electric motor	12520	OREDA
Planetary gear for azimuth thruster and fixed pitch propeller	10200	NSWC [8]
MV Variable Frequency Drive (MV-VFD) with active front end and N + 2 IGBT redundancy	8800	Vedachalam et al [9]

Table 4. Redundancy requirements defined by IMO [2]

Subsystem/ Equipment	DP Classification		
	1	2	3
Power generators	Non-redundant	Redundant	Redundant
No of PMS units	1	2	3
Thrusters	Non-redundant	Redundant	Redundant
Control computer	1	2	3
Position reference	1	3	3
Wind sensor	2	3	3
MRU	1	3	3
Heading sensor	2	3	3
UPS	1	2	3

The analysis considers a DP vessel configuration with 2 azimuth and 2 tunnel thrusters. The azimuth thrusters are used for maintaining the vessel in surge and heading DOFs, and the tunnel thrusters are used for maintaining the vessel in sway DOF.

It is considered that the capacity of each azimuth and tunnel thruster when operating at its fullest capacity could maintain vessel position within the envelope (without LOP) under the defined sea state. In this manner, the azimuth and tunnel thrusters are redundant by capacity.

Power Generation and Management System

The representative architecture considered for the PGMS with fully-redundant diesel generators (DG) based on DP2 redundancy requirements is shown in **Figure 4**. The generator sets and the Power Management System (PMS) connected to one bus (port or starboard side) shall be able to cater to the vessel design power.

A redundant bus with similar capacity and architecture shall be coupled to the active bus through a bus coupler arrangement. The PMS receives the total vessel power demand needs from the CCSS and allocates the active and reactive power allocation to the individual DG based on the droop characteristics and healthiness.

The PMS receives the total vessel power demand needs from the CCSS and allocates the active and reactive power allocation to the individual DG based on the droop characteristics and healthiness

The FTA for the PGMS is described in the third part of the article in which mains Probability of Failure (PoF) is computed for a combination of essential and redundant generator configurations. The results (in terms of demand frequency) were used as inputs for computing the Proof Test Interval (PTI) for safety integrity management of Emergency Diesel Generators (EDG) and Uninterrupted Power Supply (UPS) systems.

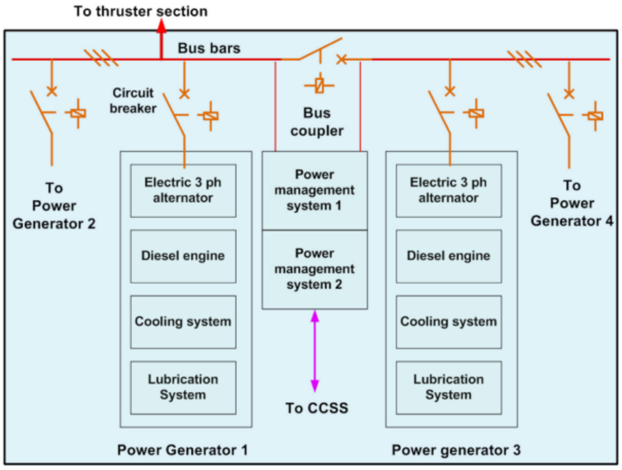


Figure 4. Architecture of the fully-redundant PGMS

Based on **Figure 3** (in the second published part) the PoF of 7% (for a configuration with 3 essential and 3 redundant generators) complying with DP2 requirements is considered for this study.

Computer Control and Sensor Section

Based on the IMO DP2 redundancy definition, **Figure 4** represents the architecture considered for the CCSS. The architecture features 2 computer/controller systems; 3 sets of environmental, attitude, absolute position sensors with signal processing circuits; and 3 sources of power supply fed by 1 power generator along with 2 independent line-interactive UPS sources.

The CCSS controller functions as a master and the PGMS and TS section control systems operate in the slave mode. However, the PGMS and TS controllers update the CCSS master with their respective section health information.

The computer/controller is programmed with the vessel Propulsion Plant Model (PPM) comprising of engine, propeller and maneuvering models in 3 and 6 DOF depending on the precision requirements. The maneuvering model based on ship translation dynamics calculates the ship speed by integrating the force balance between the vessel resistance and the thrust forces based on Newton's second law.

The vessel resistance parameters are usually obtained from the experimentally-validated hydrodynamic models. The propeller model based on shaft rotation dynamics calculates the shaft speed by integrating the torque balance between the engine and the propeller. The outputs from the PPM provide input to the engine model for computing the power generation requirements.

In addition to the PPM, the CCSS computer also receives the environmental disturbances from the environmental sensor suite including wind speed, wind direction and water speed; real time position from the Global Positioning System (GPS) and/or acoustic base line systems; and the vessel attitude from the Motion Reference Unit (MRU).

The propeller model based on shaft rotation dynamics calculates the shaft speed by integrating the torque balance between the engine and the propeller

With the aid of the PPM, environmental, attitude and position inputs, the control law/algorithm in the computer generates a control signal for the thrusters in all the 3- DOF proportional to the 3-D position and heading deviation error vector referenced to the vessel position relative to the desired set point (the proportional term), to the velocity deviation vector (differential term) and to the accumulated deviation vector (the integral term).

The controller also incorporates functions for system status handling, mode transition from auto to manual and model adaptation. Modern controllers include non-linear output feedback control, non-linear passive weather optimal, fault tolerant, back stepping and robust and adaptive fuzzy types and they helps to achieve wider technical solutions.

Thus, the CCSS controller provides the thrust allocation command to TS and power demand allocation to the PGMS through the data communication bus.

The FT corresponding to the sensor part of the CCSS is shown in **Figure 5**. The KOutofN5 gate represents the architecture with 3 redundant sensors, by which the PoF of wind speed and direction data at the same time can be reduced from 36% to 4.7% in a 1-year period.

Thus, with triple redundancy for all environmental, attitude and absolute position sensors, the PoF of the sensor gallery at the same time in a 1-year period could be reduced from 87% to 13.8%. This corresponds to an increase of MTTF from 0.48 years to 6.7 years. The FTA for the sensor section is shown in **Figure 6**.

The FTA for the CCSS section is shown in **Figure 7**. The power failure event which is common to all the sensors and control computers are considered as top-level events. The redundant power supply helps in significantly reducing the PoF of the power supply. The redundant sensor system, control computer and power supplies have a PoF of 13.8, 1.99 and 3 % in a 1-year period.

Thruster section

In the propulsion section, each thruster train comprises of the MV-VFD, 3 phase squirrel cage induction electric motor, speed reduction gear and fixed-pitch propellers. All

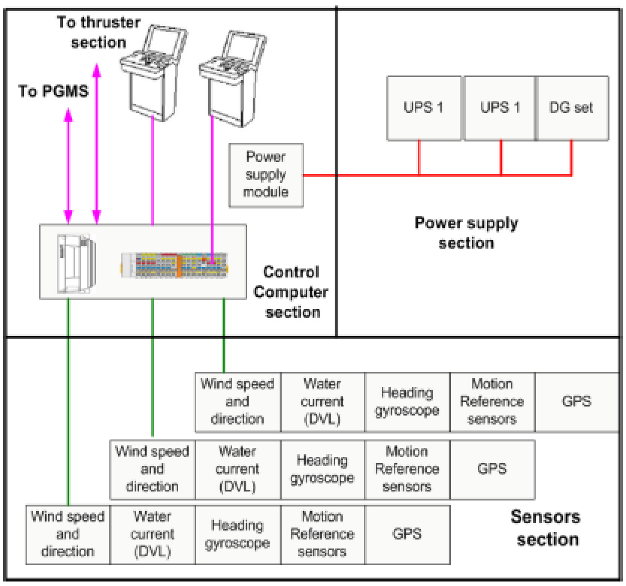


Figure 5. Architecture of the CCCS

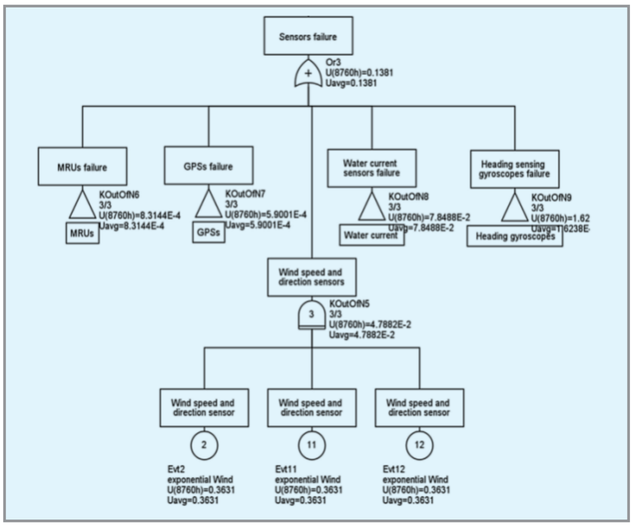


Figure 6. FTA results for the sensors section

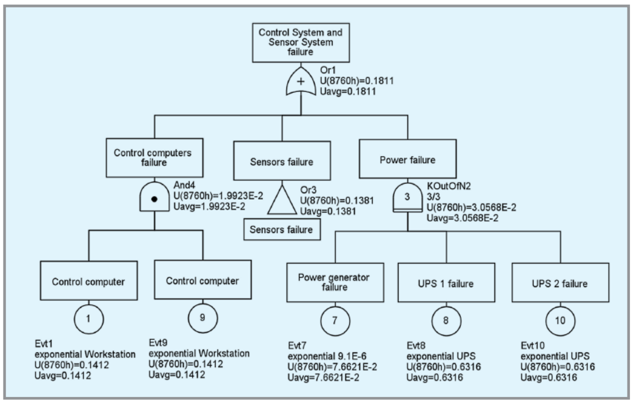


Figure 7. FTA results for the CCCS section

the thrusters including 2 azimuth and 2 tunnel thrusters are operated by independent trains.

The MV-VFD, on receiving the thrust allocation command from the CCSS, dispatches the respective power to the electric motor. Power regulation by varying the output voltage and frequency is done in the power electronics-based inverters in the machine bridge and active front end (AFE) sections of the VFD.

The Insulated Gate Bipolar Transistors (IGBT) are active components in the VFD, which carry out the power regulation based on Pulse Width Modulation (PWM) techniques. The computations of the firing angles for the IGBT are done by the power electronic controller unit. The cooling section provides adequate cooling to the IGBT by circulating the deionised cooling water through the heat sinks.

The AFE is used for enabling bidirectional power transfer between the motor and the PGMS power generators during sudden vessel deceleration. The VFD also incorporates soft starting, fault stress limiting and overload protection features improving the reliability of the electric motor, mechanical power transmission gears and the propeller [10].

IGBT dynamic redundancy

A 3-level NPC (Neutral Point Clamped) inverter has 12 IGBT, with 2 IGBT in each half phase. A VSD with AFE configuration has 24 IGBTs in both machine bridge and AFE. In the control sequence, when any one of the GD reports a fault, the master controller stops issuing the PWM trigger command (locks the pulses) to all the other IGBT GD.

The GD Complex Programmable Logic Device (CPLD) based control system ensures a GD fault detection time of <10μS, which is the condition required for protecting the series-connected IGBT under faulty conditions [10]. The advantage of providing redundant IGBT is computed using FTA with the respective failure data (**Table 3**) and the results are represented in **Figure 8**.

It is found that N+2 and N+3 redundancies in the IGBT level reduce the PoF to 15% from 68% and 54 % for AFE and DFE configurations, respectively. Thus, by N+2 and N+3 IGBT redundancy in all the phase limbs, the MTBF of a typical MV-MW-3level-NPC-VSD could be increased to 3.6 and 6.3 years, respectively. Increasing the redundancy more than N+2 does not offer any significant improvement in the VSD reliability.

Dynamic Redundancy (DR) of the series-connected IGBT is essential for uninterrupted operation of the VSD, as failure/unavailability of the thrusters during critical vessel maneuvers could be catastrophic.

The DR which is presently operational in advanced VSD is implemented uses a string of suitably rated reverse-biased ultra-fast transient protection Zener diodes (TPZD) connected between the collector and gate terminals of all the press-pack IGBT (**Figure 9**) of the machine side and AFE inverters.

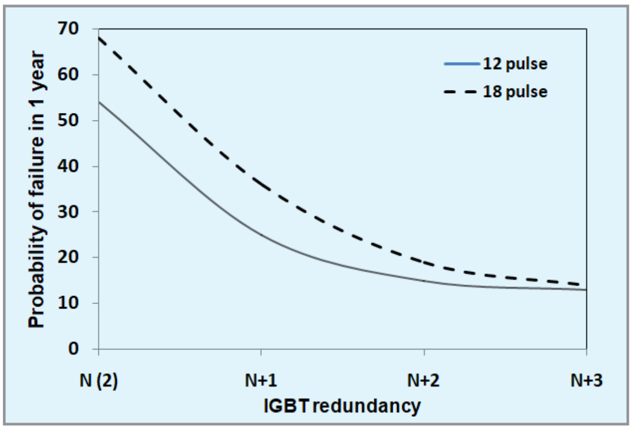


Figure 8. Reduction in PoF of VSD with IGBT redundancy

Their instantaneous response of the TPZD to overvoltage (above knee point voltage) with turn-on duration typically in the order of tens of picoseconds makes them suitable for implementing DR.

As indicated in **Figure 9**, when a trigger command is applied to all the 3 IGBTs (forming part of one-half phase of a 12-pulse inverter) and when IGBT T2 fails to get triggered, the entire DC bus voltage occurs across T2, with the circuit closed through the motor winding and other IGBTs.

As the resultant voltage across C and E is greater than the knee point voltage of the TPZD string, the string breaks down resulting in a sudden spike in voltage (with a steep slope) being applied to the gate terminal of the untriggered IGBT2, which switches on the IGBT at a higher VCE. This "hard-switching" results in increased IGBT junction temperature.

The hard switching process drives the IGBT out of the Safe Operating Area (SOA) (**Figure 10**). Thus, the switching of IGBT2 under overvoltage is continued until

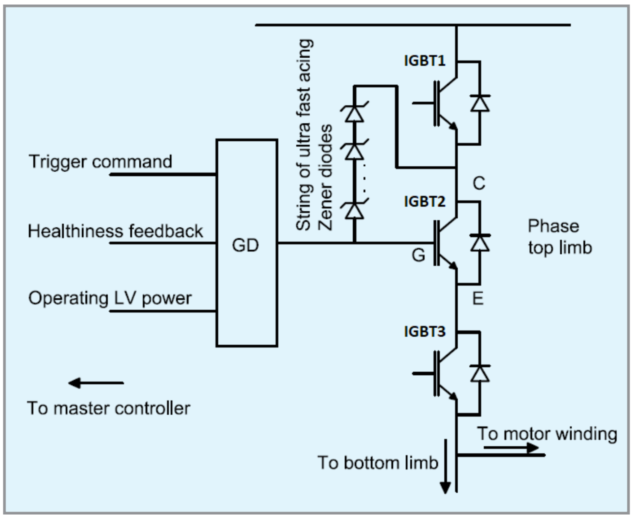


Figure 9. IGBT dynamic redundancy (DR) in VSD inverter

it is damaged due to excessive junction temperature (typically in <1s).

The temperature rises in the 1200V rated IGBT junction modeled using MATLAB Simpower/Sim electronics systems and Simscape thermal model based on two-cell Cauer network operating at 1 kHz. Simulation results indicate the junction temperature crosses the maximum allowed limit of 120 °C in 730ms (**Figure 11**) and reaches 400 °C in 3s.

For PWM switching frequencies of 50kHz and 100 kHz, the time for IGBT to get short circuited is 212ms and 56ms, respectively, which is advantageous for implementing DR in high-speed switching inverters.

For higher power marine propulsion drives multi-level inverters are used. They include diode-clamped, flying capacitors and cascaded H-bridge types. Multilevel cascaded H-bridge inverters (**Figure 12**) are used to eliminate the bulky transformers required in conventional multi-phase inverters, clamping diodes required in diode clamped inverters and flying capacitors required in flying capacitor inverters.

Multi-level inverters produce common-mode voltage reducing the motor winding stresses, produce lesser ship mains distortion and operate at wide range of switching frequencies. The FTA results for the thruster section with N+2 redundancy in the IGBT in the AFE and machine bridge inverters is shown in **Figure 13**.

DP system reliability

Based on the failure rates computed for the PGMS, CCSS and TS, the PoF of a DP2 system (**Figure 14**) in a period of 1 year is 37.3% and the corresponding MTBF is 2.1 years. Based on the similar methodology, FTA is performed for the DP1 and DP3 configurations. The MTBF of DP1, 2 and 3 configurations are 0.27, 2.12 and 2.51 years, respectively.

Conclusions

The reliability of the propulsion, protection and life support power systems in an Integrated Electric Power System needs careful evaluation during the design and operational phases, as the ramifications of non-operation or mal-function could be catastrophic.

Hence, safety-centered system engineering with a reasonable reliability, safety and efficiency trade-off is essential for multi-megawatt vessel power systems.

The first part of the article discussed the maturity of marine power system components, importance of RAMS-centered system engineering and developments in the RAMS computation tools.

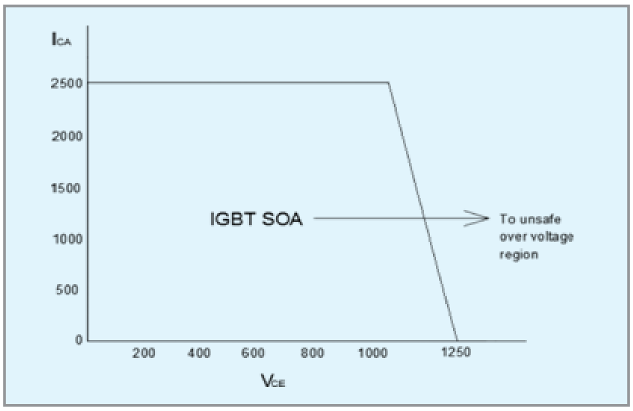


Figure 10. Reduction in PoF of VSD with IGBT redundancy

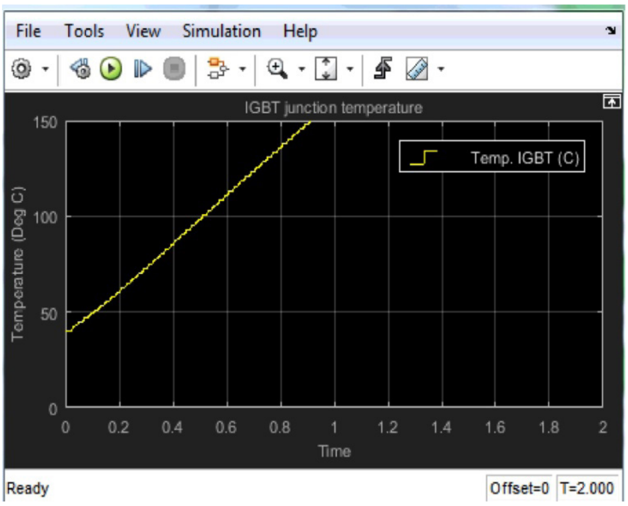


Figure 11. Junction temperature of the untriggered IGBT2

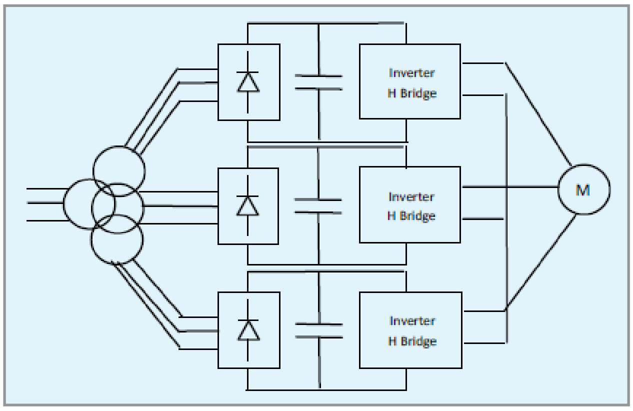


Figure 12. VSI 5-level H-bridge VSD for high power applications

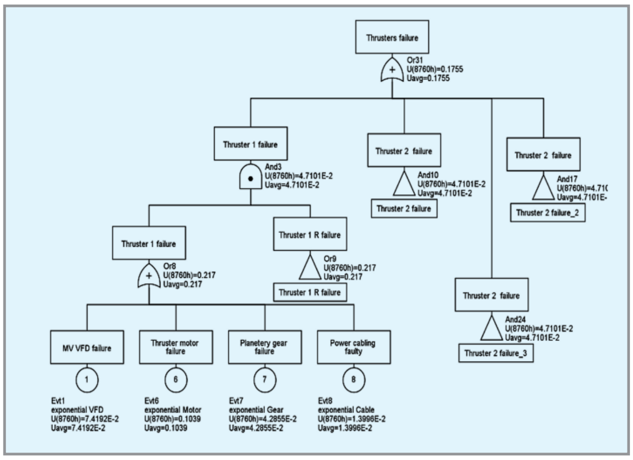


Figure 13. FTA results for the thrusters section

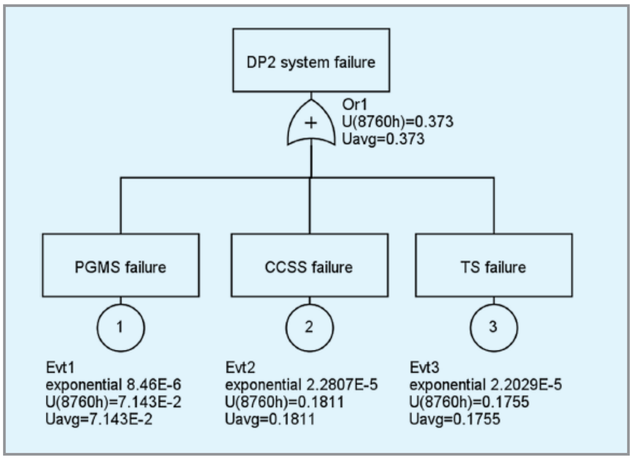


Figure 14. FTA results for the DP2 system

The second part discussed on the integrated approach to RAMS estimation, power system protection coordination based on current-time characteristics and methods for determining SIL-compliant circuit breaking based on numerical tools with field-failure data as inputs.

The third part of the article discussed the methodologies for determining the proof test intervals for emergency diesel generator sets and uninterrupted power supplies. Various possible induction motor failure modes, winding and rotor fault detection methods and bearing failures are detailed. The importance of off-line and on-line condition assessment methods for motors and transformers were discussed. The developments in the active magnetic bearings and its reliability are presented.

The fourth part the article described the advantages of ungrounded neutral, recent developments in online insulation monitoring techniques and the need for higher insulation voltage to withstand ground faults, the stringent quality needs for marine grade power cables, their integrity monitoring and the ampacity de-rating for multi-layered configurations with the help of finite element analysis tools, maturing variable speed propulsion drives and inverter topologies, role of variable speed drives in vessel power optimisation and regulations for electromagnetic compatibility.

This part, taking into consideration the importance of reliability of the propulsion systems during vessel critical maneuvers and operations, describes methodologies for reducing the failure rate of the active-front-end high power density variable speed propulsion drives by incorporating dynamic redundancy in the inverter power electronics. Based on dynamic redundancy, the reliability assessments for various classes of dynamic positioning systems are computed.

The importance of RAMS in the strategic marine sector including autonomous ships, subsea boosting systems, renewable power grids, offshore power transmission and remotely operated vehicles shall be discussed in the (sixth/final) part. It is evident that RAMS studies based on field-failure data shall be of immense help in identifying a tradeoff between the capital investments, operating expenditure, redundancy requirements, system modularity and maintainability, as well as meet the stringent regulatory requirements.

ACKNOWLEDGEMENTS

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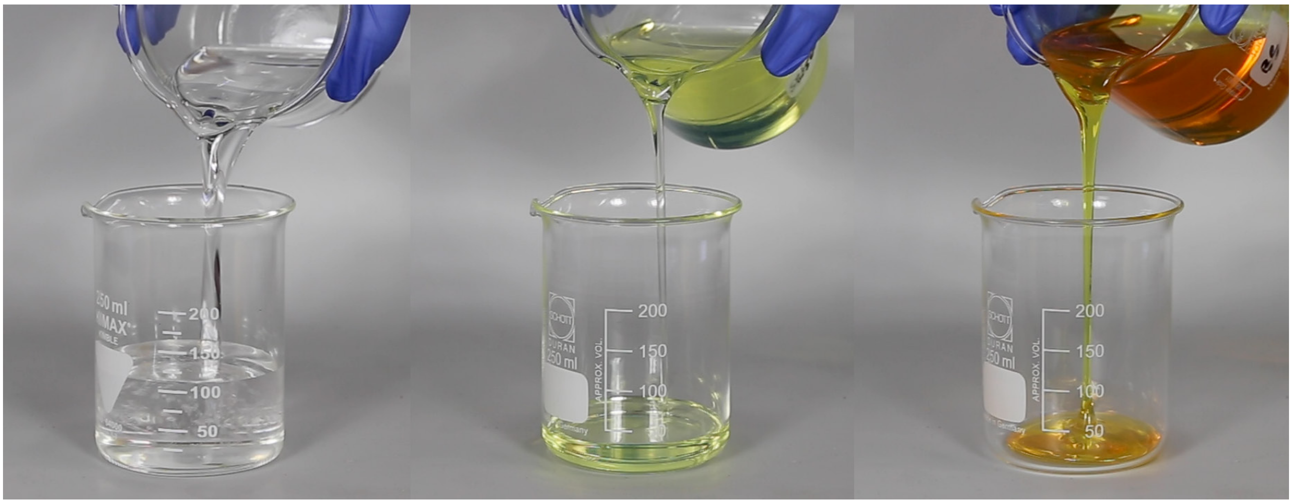
LUBE MATTERS # 9
VISCOSITY



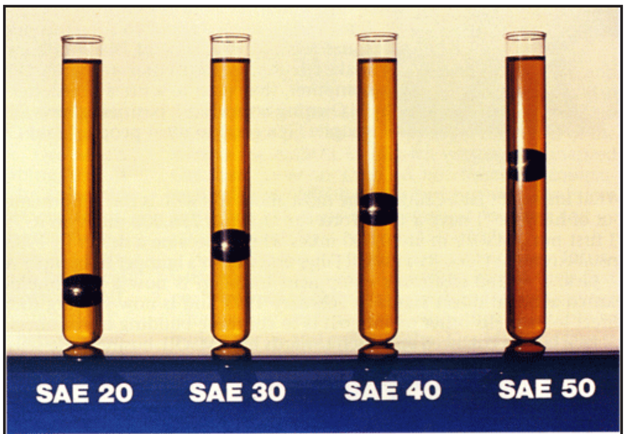
Sanjiv Wazir

Introduction

The single most important property of a lubricating oil is its viscosity. If oil viscosity is too low, the film thickness may not be enough to prevent asperity contact, friction, and wear. If the viscosity is too high, the oil film thickness will be thick, causing the machine to work harder, generating heat, and use more energy. Choosing the right viscosity for a lubricant is often the first decision that needs to be made when selecting a lubricant for a particular usage.

What is Viscosity

Viscosity is a fundamental property of all fluids. When a fluid flows, it has an internal resistance to flow caused by friction within the fluid. **Viscosity is the measure of the fluid's resistance to flow and shear**. It is the result of intermolecular forces between particles within a fluid. These intermolecular forces resist the shearing motion of the fluid, and the viscosity of the fluid is directly proportional to the strength of these forces.



Steel balls of equal weight dropped into test tubes filled with motor oils fall at different rates. Their rate of fall depends on the viscosity of the oil. The ball travelling through the light SAE 20 oil has travelled farthest, while the ball in the heavy SAE 50 has travelled least.

(SAE image source: TheLubricantStore.com)

Impact of Base Oil Composition on Viscosity

Viscosity Index

The flow properties of a lubricant are strongly affected by the composition of the base oil used in its formulation. Viscosity of all mineral oils decrease with increase in temperature. Viscosity Index (VI – a dimensionless number) reflects the relationship between base oil viscosity and temperature. Comparing hydrocarbons of similar molecular weight, paraffins tend to raise base oil VI, while aromatics and naphthenes strongly decrease VI. In addition to removing undesirable molecular species, refining also generally improves the viscosity index (VI) of the fluid. In general, higher VI products are considered more desirable because they maintain higher viscosity and lubricating film building properties at high temperatures (100-150°C) while exhibiting lower viscosity, better fluidity and pumpability at low temperatures (< 0°C) when treated with pour point depressants. The effect of temperatures on

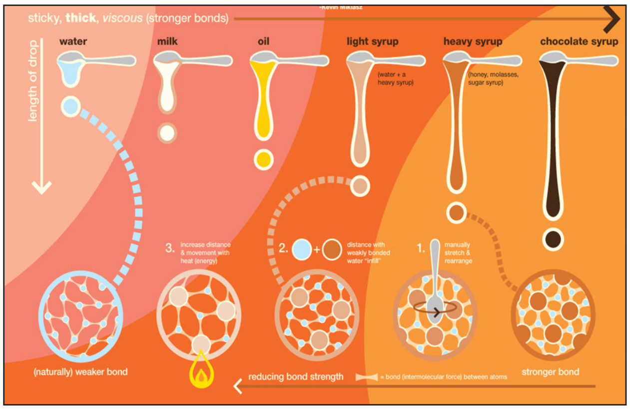


Figure 1. Viscosity behaviour of some household liquids [1]

the viscosity of different VI base oils that have a viscosity of 4 cSt at 100C is shown in **Figure 2**. **[2]**

Viscosity-Pressure Coefficient

Most organic fluids, including base oils, exhibit a reversible viscosity increase with the application of high pressure due to the restrictions on molecular mobility imposed by the forces being exerted. This phenomenon is especially important for lubricants, since in most mechanical applications, films of fluid are compressed between sliding or rolling surfaces under very high loads. The degree to which a fluid "thickens" under high pressure is expressed by the Viscosity-Pressure Coefficient (known as α). Under high pressure conditions, such as experienced under hydrodynamic and elasto-hydrodynamic lubrication regimes, thicker lubricant films are produced by fluids exhibiting higher α value. **Figure 3** illustrates the magnitude of viscosity increase that can be expected for typical base oils, over pressure ranges experienced in certain mechanical components.

Kinematic Viscosity (KV)

The most familiar measure of lubricant viscosity is its KV. It is a measure of the fluids resistance to flow and shear under gravity. KV is determined by using a capillary viscometer in which a fixed volume of fluid is passed through an orifice at a controlled temperature under gravity (**Figure 4**).

It is usually reported in centistokes (cSt).

Dynamic Viscosity

(also known as Absolute Viscosity)

Absolute Viscosity is the ratio of sheer strain experienced by a fluid film moving under a sheer stress. It is usually reported in centipoise (cP).

A common device used to measure dynamic viscosity is the Brookfield Viscometer. It employs a rotating paddle that experiences torque as it rotates against fluid friction (**Figure 5**).

Viscosity Classification

Various technical bodies have created classifications that are used by lubricant manufacturers, OEMs, and users to designate the KV of the lubricant.

SAE J300 (Engine oil viscosity)

In 1911, the Society of Automotive Engineers (SAE) introduced the first classification for automotive engine oils called Specification 26. These were revised over the years and in 1962, Specification 26 became the first SAE

Base Oil VI	Temperature			
	0°C	40°C	100°C	150°C
75	192	21	4	1.87
100	155	19.5	4	1.91
125	117	17.7	4	1.97
150	92	16.3	4	2.03

Figure 2. VI – The Effect of temperature on viscosity [2]

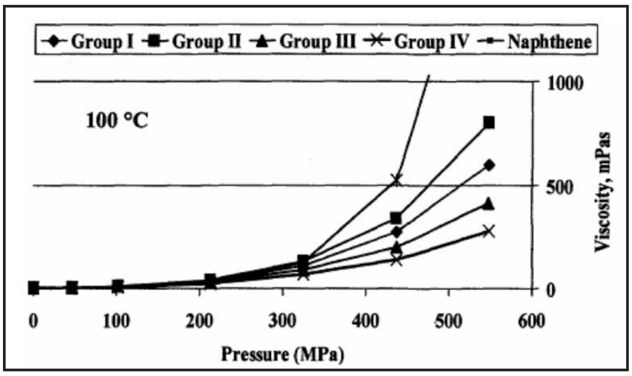


Figure 3. Viscosity-Pressure relationship for different base oils at 100°C (Nominally 4cSt at 100°C, at 1 bar pressure) (2)

(Base Oil composition and Base Oil groups I-V mentioned above will be discussed later in this series, on Base Oils).

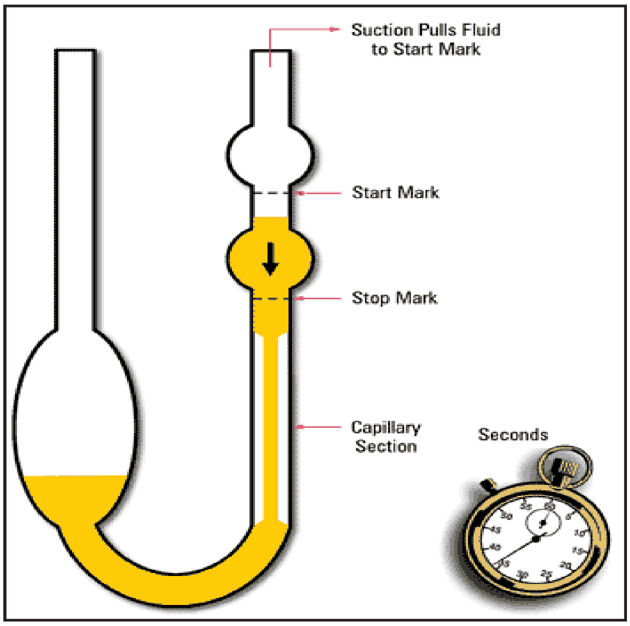


Figure 4. Determination of Kinematic Viscosity

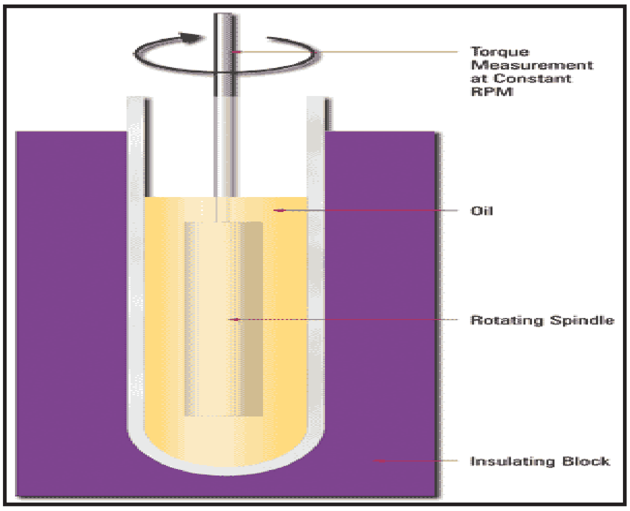


Figure 5. Determination of Dynamic Viscosity

SAE Viscosity Grade	Low-Temperature Cranking Viscosity (CCS)	Low-Temperature Pumping Viscosity (MRV)	Low-Shear-Rate Kinematic Viscosity (KV100) at 100 °C		High-Shear-Rate Viscosity (HTHS) at 150 °C
	ASTM D5293	ASTM D4684	ASTM D445		Various methods
	mPa.s (max)	mPa.s (max)	mm²/s (min)	mm²/s (max)	mPa.s (min)
0W	6200 @ -35 °C	60000 @ -40 °C	3.8	-	-
5W	6600 @ -30 °C	60000 @ -35 °C	3.8	-	-
10W	7000 @ -25 °C	60000 @ -30 °C	4.1	-	-
15W	7000 @ -20 °C	60000 @ -25 °C	5.6	-	-
20W	9500 @ -15 °C	60000 @ -20 °C	5.6	-	-
25W	13000 @ -10 °C	60000 @ -15 °C	9.3	-	-
8	-	-	4.0	< 6.1	1.7
12	-	-	5.0	< 7.1	2.0
16	-	-	6.1	< 8.2	2.3
20	-	-	6.9	< 9.3	2.6
30	-	-	9.3	< 12.5	2.9
40	-	-	12.5	< 16.3	3.5 (0W, 5W, 10W)
40	-	-	12.5	< 16.3	3.7 (other grades)
50	-	-	16.3	< 21.9	3.7
60	-	-	21.9	< 26.1	3.7

Figure 6. SAE J300 Standard for KV of Engine Oils

J300 standard. The SAE J300 standard is widely used for designating the KV of engine oils. The grades denoted with the letter "W" were intended for use in winter. The current standard is SAE300_201501 issued in 2015.

Since the viscosity range of each SAE grade is quite wide, often "equivalent" oils may be of different viscosity. For instance, for marine cylinder oils the OEM requirement for viscosity is SAE 50 (16.3 cSt at 100°C to 21.9 cSt at 100°C). Most LUKOIL cylinder oils are blended to be 20 cSt at 100°C (**Figure 6**). Other makers may have different target viscosity within the SAE range. This may result in difference in operating characteristics.

The original viscosity grades were all mono grades, e.g., a typical engine oil was a SAE 30. This was an operating temperature viscosity. Basically, the higher the monograde oil SAE viscosity, the thicker the oil would be at operating temperature. This was great for operating temperature but the oil would also be thicker at cold temperature which made it harder to pump and also slower to provide lubrication where it was needed **[5].** Often the user had to change to a "W" grade in cold conditions.

Multigrade oils were developed to resolve this issue. Nowadays Viscosity Index Improver (VII) additives are added to the oil to increase the VI of base oil so that a single lube can be used in cold and hot conditions (**Figure 7**).

A multigrade oil is one that meets both, a "W" low temperature viscosity requirement and a 100°C "operating temperature" requirement. For example, a 15W-40 engine oil (a grade commonly used on board ships for high-speed engines such as emergency generator engine) meets the requirements for a SAE 15W oil at cold temperatures and of a SAE 40 oil at high temperatures.

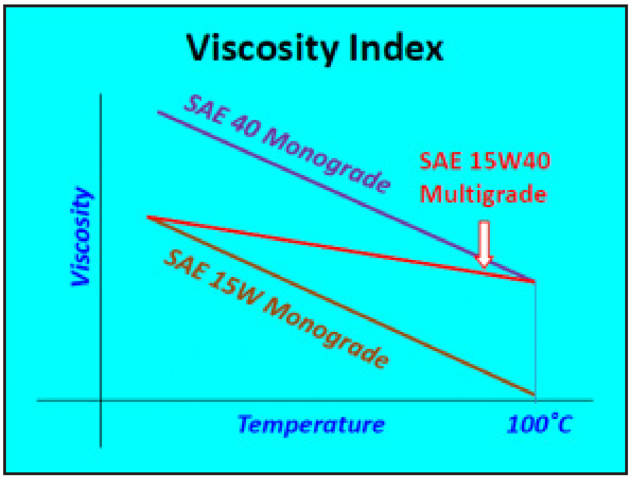


Figure 7. VI Improvers additives are used to make multigrade oils

Because of relatively uniform operating conditions prevailing in the engine room, marine engine oils typically are mono-grade oils of the following KV grades.

2-S X-Head Engine System oil: SAE 30 (between 9.3 cSt and 12.5 cSt at 100°C)

2-S X-Head Engine Cylinder oil: SAE 50 (between 16.3 cSt and 21.9 cSt at 100°C)

ISO Viscosity Grade	Midpoint Kinematic Viscosity mm²/s at 40°C (104°F)	Kinematic Viscosity Limit mm²/s at 40°C (104°F) Minimum	Kinematic Viscosity Limit mm²/s at 40°C (104°F) Maximum
ISO VG 2	2.2	1.98	2.42
ISO VG 3	3.2	2.88	3.52
ISO VG 5	4.6	4.14	5.06
ISO VG 7	6.8	6.12	7.46
ISO VG 10	10	9.00	11.0
ISO VG 15	15	13.5	16.5
ISO VG 22	22	19.8	24.2
ISO VG 32	32	29.8	35.2
ISO VG 46	46	41.4	50.6
ISO VG 68	68	61.2	74.8
ISO VG 100	100	90.0	110
ISO VG 150	150	135	165
ISO VG 220	220	198	242
ISO VG 320	320	288	352
ISO VG 460	460	414	506
ISO VG 680	680	612	748
ISO VG 1000	1000	900	1100
ISO VG 1500	1500	1350	1650
ISO VG 2200	2200	1980	2420
ISO VG 3200	3200	2880	3520

Figure 8. ISO Viscosity Grades

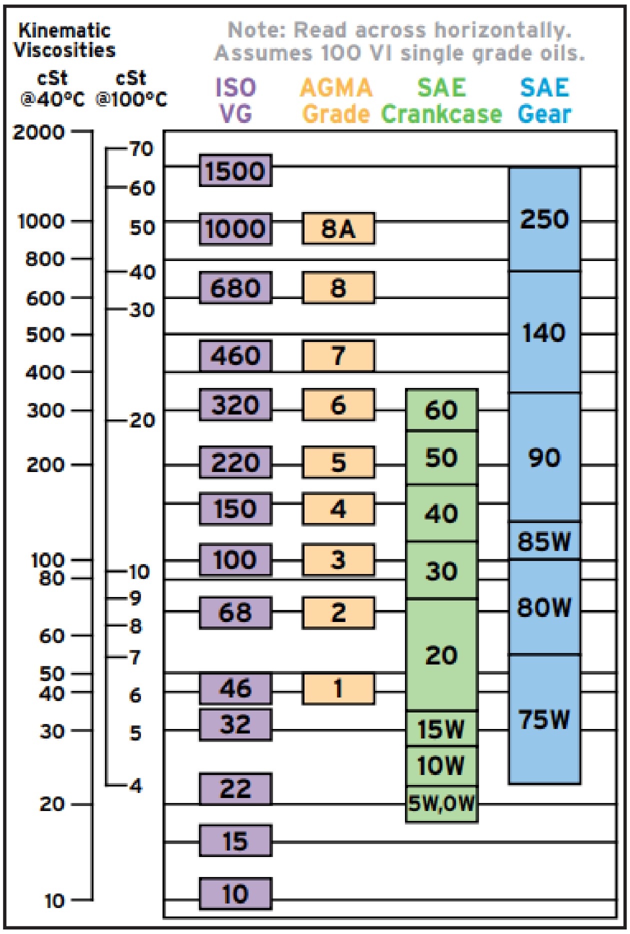


Figure 9. Various KV classifications [4]

4-S Trunk Piston Engine Oil: SAE 40 (between 12.5 cSt and 16.3 cSt at 100°C)/ SAE 30

Automotive engine oils are now predominantly "Multigrade" oils (e.g., 5W-20, 0W-16, etc).

ISO Viscosity Grades (Industrial oil viscosity)

The most widely used viscosity classification system for industrial lubricants is the ISO 3448 Viscosity Grades (ISO VG). KV of these oils are measured at 40°C, and each succeeding grade is about 50% more viscous than the previous grade. The mid-point of each grade viscosity range is the ISO VG of that grade (**Figure 8**).

Amongst marine ancillary grades, the following ISO VG grades are most used:

Hydraulic Fluids: ISO VG 15/32/46/68/100/150. (Note marine hydraulic grades are usually VI Improved oils).

Gear Oils: ISO VG 68/100/150/220/320/460/680

Turbine & Compressor Oils: ISO VG 32/46/68/100

Other Viscosity classifications

SAE also has a classification for Automotive axle & manual transmission Lubricants (SAE J306) that are widely used in the transportation industry. An 80W90 gear oil often used in Lifeboat gear boxes is based on this grading. AGMA (American Gear Manufacturers Association) viscosity classification is another grading system commonly used for gear oils (**Figure 9**).

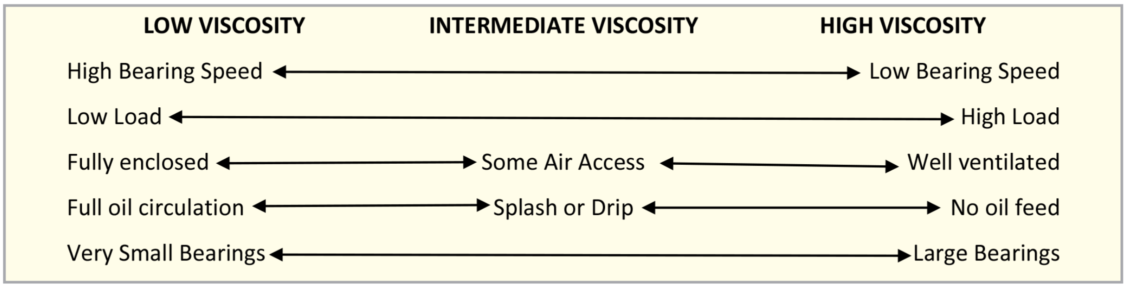


Figure 10. Factors affecting choice of viscosity [3]

Choosing the right oil viscosity

Although viscosity is the single most important property of an oil, in practice the choice is not very critical (**Figure 10**). Often the viscosity already giving satisfactory results in similar equipment is chosen. Only where choice of the right viscosity is critical, specialists use fluid mechanics fundamentals (Reynolds, Dowson-Higginson equations) to arrive at a suitable viscosity & choose the standard closes to the requirement.

Conclusion

Starting with the right oil viscosity and monitoring and trending changes in viscosity of the in-use oil is key to trouble-free operation of all machinery.

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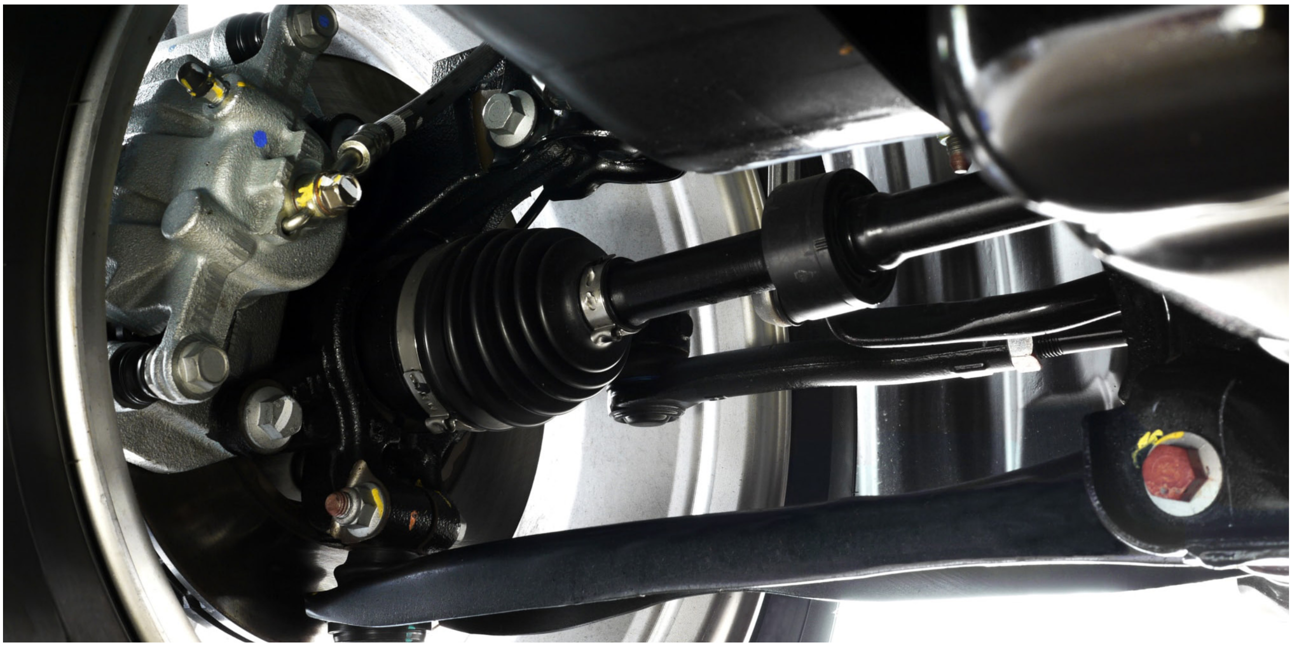
STRESSES IN KEYED PROPELLER – PROPELLER SHAFT ASSEMBLIES (PART 1)



Agaram Ramanujan

1 INTRODUCTION

During the fifties of the last century the average size of ships were in the region of 6000 to 7000 tons' displacement and there were a few ships over 10000 tons' displacement. Most of these ships had conventional shafting systems, sea water lubricated. In this era the mode of tightening large bolts was by brute force using heavy sledge hammers. The limitation of tightening was observed by a rule that before opening a nut a suitable mark is made on the nut flat side and a corresponding mark made on the part, and when tightening if both marks coincide at the maximum tightening effort, it is considered fully tight. This used to be the procedure for tightening cylinder head covers and all running gear bearing bolts of the main engine. The process of tightening was by slogging on a short ring spanner by a heavy sledge hammer.

In the case of the propeller shaft, for withdrawal purpose in dry dock the same procedure was used. In this case the hammer was heavier weighing about 200 kg slung from a support at the stern frame by a rope and made to swing from a higher position so that it struck on the spanner head with a strong impact causing the nut to ease off.

When assembling the same method was used to push up the boss on the taper till it reaches the mark by slogging on the spanner to start and move the nut.

To free the boss from the shaft taper, steel wedges were inserted between the stern frame post and the propeller boss and the wedges were slogged in the same manner till the propeller jumped off the taper. In some adamant and difficult situations application of heat by large gas torches was resorted to and had to be done very carefully to prevent cracking of the propeller boss.

As ship sizes started increasing the shaft diameters and propeller sizes also increased accordingly and the primitive method of tightening or slackening had to give way to the use of hydraulic force which was controlled, shock free and hence safe as well as easy to use. Special hydraulic jacks were developed to cater to these needs.

2 MODEL PROPELLER SHAFT ASSEMBLY

All keyed shafts have a taper of 1:14 as per class rules. The taper ratio defines the taper angle and is the Tangent of the taper angle. The material of the shaft is medium carbon forged steel having a carbon content of 0.6 percent, manganese 1.5 percent, sulphur 0.4 pe5rcent (max), phosphorous 0.4 percent(max), silicon 0.45 percent (max). The residual alloying elements are copper 0.3 percent (max), chromium 0.3 percent (max), molybdenum 0.15 percent (max), and nickel 0.4 percent (max).

The diameter of the shaft is 400 mm. The taper length at the propeller assembly end has a length of 400mm. The shaft end has a threaded end with a mean diameter of 200mm and provided with standard acme threads having a trapezoidal section. The trapezoidal section aids in providing shear strength to the threads, which is necessary

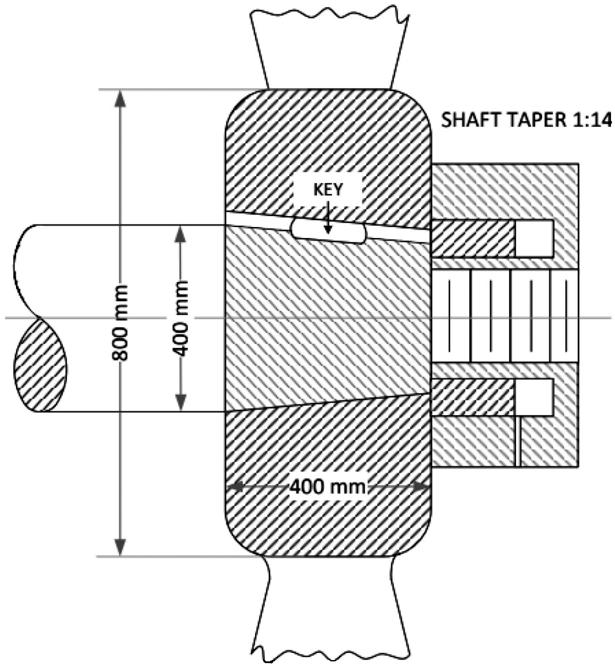


Figure 1.

to resist the high hydraulic force which occurs during the assembly of the propeller on the shaft (**Figure 1**).

3 PROPELLER DISMANTLING

(a) A light mark(line) made on taper of shaft in line with forward edge of the boss made. Reference for limit of push up during assembly.

(b) Strong back of propeller withdrawal tool attached with propeller nut removed and refitted in after turning it to enable the inbuilt jack to force on the strong-back (**Figure 2**).

(c) By pressurising the inbuilt jack in the pilgrim nut using a powerful electric pump with adequate oil pressure to force the propeller off the taper.

The strong back and studs are ship supply tools and so held on board. The electric pump is shipyard supply.

(d) The intermediate shaft coupling bolts at both ends and the plummer block holding down bolts dismantled and the intermediate shaft with the plummer block lifted up and suspended, to provide space for the propeller shaft to be withdrawn in.

(e) When the propeller is forced out it is suspended by two heavy chain blocks with strong slings suspended from the ships stern bollards and lowered on the floor of the dry dock ,whilst the propeller shaft is withdrawn inside the machinery space and made to rest with proper supports in the position of the intermediate shaft.

4 PROPELLER ASSEMBLY

The method of forcing the propeller on the shaft taper is shown in **Figure 3**. The propeller nut is provided with

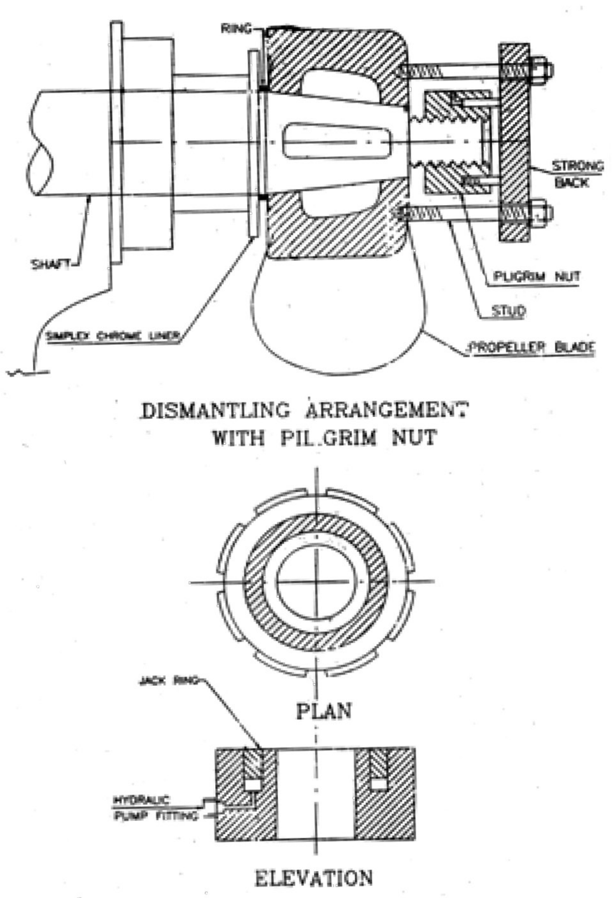


Figure 2.

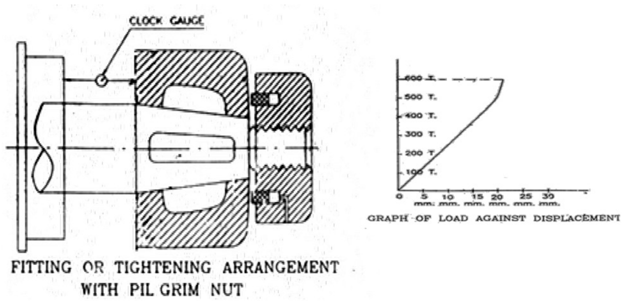


Figure 3.

an inbuilt ring jack and uses hydraulic oil pressurised by a powerful electric pump to force the propeller on the taper till it reaches the original mark made before dismantling.

The forcing is done in stages to prevent shock loading of the propeller since it is made of phosphor bronze which cannot stand any shock loading. The displacement of the propeller along the taper is noted on the dial gauge placed between the propeller boss and the seal box fitted on the stern frame. For record purposes a graph of propeller displacement against The hydraulic pressure or forcing load is drawn and is preserved for future reference. This record is referred when the propeller is to be dismantled.

The propeller is mounted on the shaft taper and tightened initially by the spanner using a sledge hammer till it does not move any more.

The pilgrim nut is now hydraulically pressurised by the pump in stages of 1mm displacement with the corresponding pressure increasing from zero to 8000 kg/cm^2. The boss front has now reached the original mark made on shaft taper. The pump pressure is increased to verify if the propeller can move more but there is no movement and the final pressure recorded is 8100 kg/cm^2 and the corresponding displacement is 8mm The inbuilt jack has a mean diameter of 25cms with the annular ring space of 3cms.

5 FORCING AND RESULTANT LOCKED IN STRESSES CALCULATION

* The final tightening force = (8100* 3.142*25*3)/1000*9.81=18725 KN
* The mean diameter of the taper for shaft and propeller is:

  {400 + [400— (2*400/14)]} = 371.43 mm.

* The forcing displacement of 8mm along the longitudinal length of the taper creates an increase in mean diameter and is given by

  (0.8/14) * 2 = 0.1143mm

* The mean diameter increases to 371.43 +0.1143 = 371.5443 mm.

The force fit causes a shrink in the shaft taper and a swell in the propeller taper. The swell and shrink are inversely proportional to their corresponding bulk modulus.

The forcing is done in stages of approximately 1 mm with a rest between every stage of displacement so that the shrink and swell settle uniformly and without shock at every stage.

Let D be the final interface mean diameter between the propeller boss taper and shaft taper.

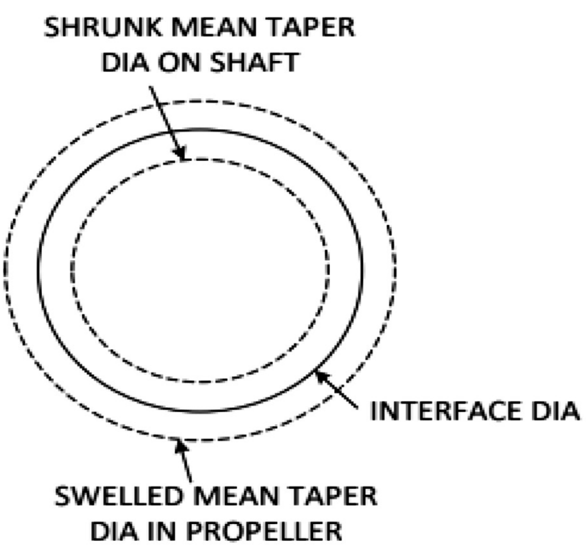


Figure 4.

* Diametrical swell of propeller taper is (371.5443 – D.) mm.
* Diametrical shrink of shaft taper is (D---371. 43) mm.
* the interface radial pressure or stresses are equal and opposite

The equation for this radial stress is given by:

Swell strain of propeller boss* bulk modulus of bronze = shrink strain of shaft taper* bulk modulus of steel.

The corresponding equation is given below.

$$\Pi/4\ (371.5443^2 - D^2)/\ (\Pi/4*371.43^2)*112*10^9 = 0\Pi/4\ (D^2 - 371.43^2)/\ (\Pi/4*371.43^2)*165*10^9 \quad (1)$$

Where 112*10^9 N/M^2 is the bulk modulus of the bronze propelled And 165* 10^9 N/M^2 is the bulk modulus of the steel shaft.

The above equation (1) reduces to

$$(371.5443^2 - D^2)/\ (D^2 - 371.43^2) = 165/\ 112 \quad (2)$$

From above equation we get D= 371.4762mm

The radial stress is 112*10^9* (371.54432 ----371.47622)/ 371.432 (3)

(Which gives the stress as 41N/mm^2)

The surface area of the propeller taper is:

400* 3.142* 371.43 – 400*10 ==462813.224 mm^2

Where 400*10 is the area of the keyway in the propeller

The normal reaction on shaft taper is:

(41 * 462813.224)/1000 = **18975 KN**.

- The shaft delivers a power of 10,000 KW at service RPM 110.
- **The Torque at SCR IS 868 KNM**
- **At MCR RPM of 120 and 11000 KW the torque is 875 KNM**
- The friction slip torque between propeller and shaft assuming there is no key is given by

  18975*0.3 * 0.1857 = 1057 KNM

Where 0.3 is the coefficient of friction between bronze propeller and steel shaft at the taper fit.

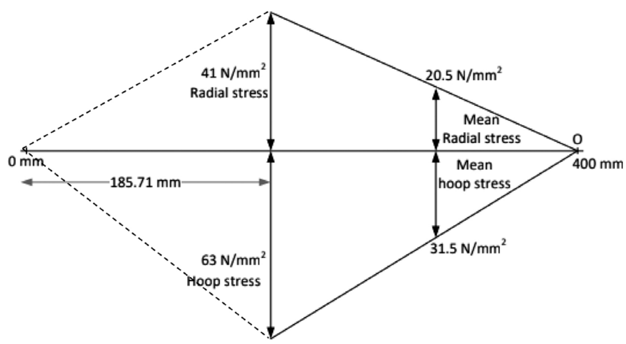
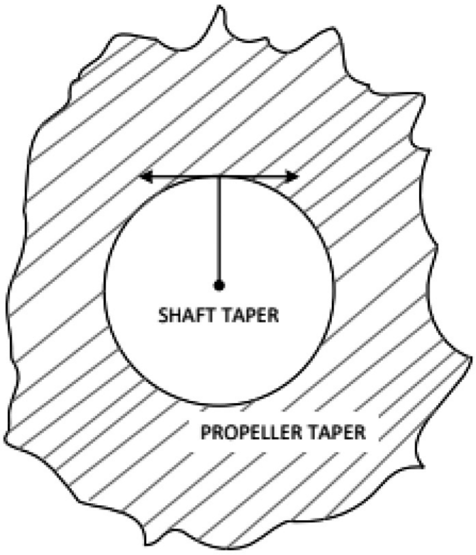


Figure 5.



AHEAD AND ASTERN FRICTION SLIP LOCK

Figure 6.

This friction torque is 20 percent greater than the maximum torque and can be named the friction lock. So it can be seen that the shaft key and key way is redundant and in reality the key sides and top do not contact the propeller taper.

6 CALCULATION OF HOOP STRESS INDUCED IN THE PROPELLER BOSS

To find the hop stress in the propeller boss caused by the force fit, we use LAME'S equations for finding the hoop stress in thick cylinders, since the propeller boss is like a thick walled cylinder with the shaft taper.

The two equations are given as under:

Radial pressure or stress $P = a/r^2 - -b$ (1)

Hoop stress $f = a/r^2 + b$ (2)

Where an and b are constants

Applying these equations to the end conditions in the propeller boss we can find the constants values and the hop stress.

So we have $41 = a/371.476^2$ (3)

$0 = a/800^2$ (4)

Where 800 mm is the boss diameter and the radial pressure on the boss is zero neglecting the hydrostatic pressure of the sea water which is negligible,

The constant an and b obtained from above equations are

a = 7213014 and b= 52

the hoop stress at the interface of propeller taper is 63N/mm^2 and it is tensile.

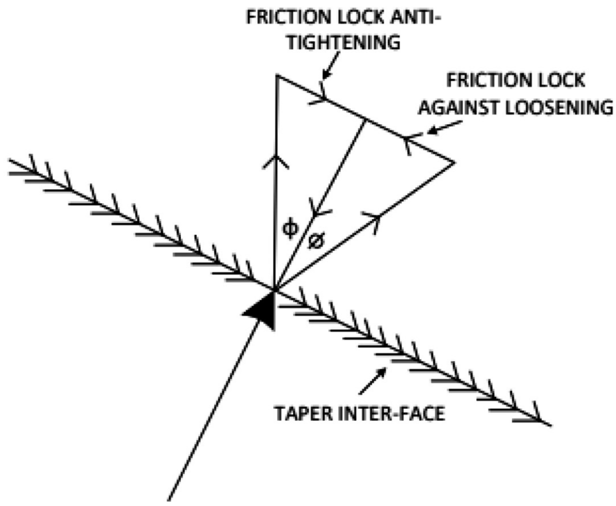


Figure 7.

The hoop stress at the outer diameter of the boss is zero. Hence the hoop stress and the radial stress in the propeller boss reduces from 62 N/mm^2 to 0 and 41N/mm^2 to zero respectively in a straight line.

*The required dismantling oil pressure in the inbuilt jack is given by:

Friction slip force/Cos taper angle = oil pressure * jack area

Which is

(18975*0.3)/ 0.9974= P* 3.142*25*3

P = 2420 kg/cm^2.

By giving a quick pressure pulse of 3000kg/ cm^2 the propeller jumps off he taper easily.

*The torsional stress at the mean taper diameter of shaft at SCR is given by:

(868*1000000*16)/3.142*371.43^3 = 86.26 N/mm^2

7 TABLE OF STRESSES

STRESS	MAGNITUDE
maximum radial stress	41 N/mm^2
mean radial stress	20.5 N/mm^2
maximum hoop stress	63 N/mm^2
mean hoop stress	31.5 N/mm^2
friction slip torque	1057 kink
maximum forcing pressure	8100 kg/cm^2
dismantling pressure	3000 kg/cm^2
Torsional stress of shaft taper at SCR	86.26 N/mm^2
Torsional stress of shaft taper at MCR	86.95 N/mm^2
Radial stress on shaft taper(max)	41 N/mm^2 (compressive)
Hoop stress on shaft taper(max)	63 N/mm^2 (compressive)

8 CONCLUSION

From the above study we may conclude that the key and key way is redundant. It may be retained as a stand by anti-slip device, but it provides a source of fatigue at the shaft taper because of the abrupt change in shape caused by the keyway. The edges of the keyway are locations for fatigue cracks and when shaft sizes proportionately rose with advent of large vessels like the super-tankers and bulk-carriers in the late fifties of the last century, there were a number of cases of fatigue cracking of shafts at the taper. Classification societies have therefore made it mandatory to include magna-flux crack detection test for shaft tapers in the scope of survey of propeller shafts. From then on, detailed study into their causes gave way to the development of keyless shafts.

(The development of keyless shafts will be dealt in part2)

References

*Observed shipyard practice.

*Strength of materials by Paradise and Church.

About the Author

Agaram Ramanujan is an octogenarian marine engineer, passed out from DMET in 1955. He has served in various capacities as surveyor, examiner of engineers and Engineering superintendent of shipping companies. Since 1989 he has taught in various training institutes including I.M.A.R.E Mumbai. He holds an extra first class certificate of competency, issued by UK MOT.

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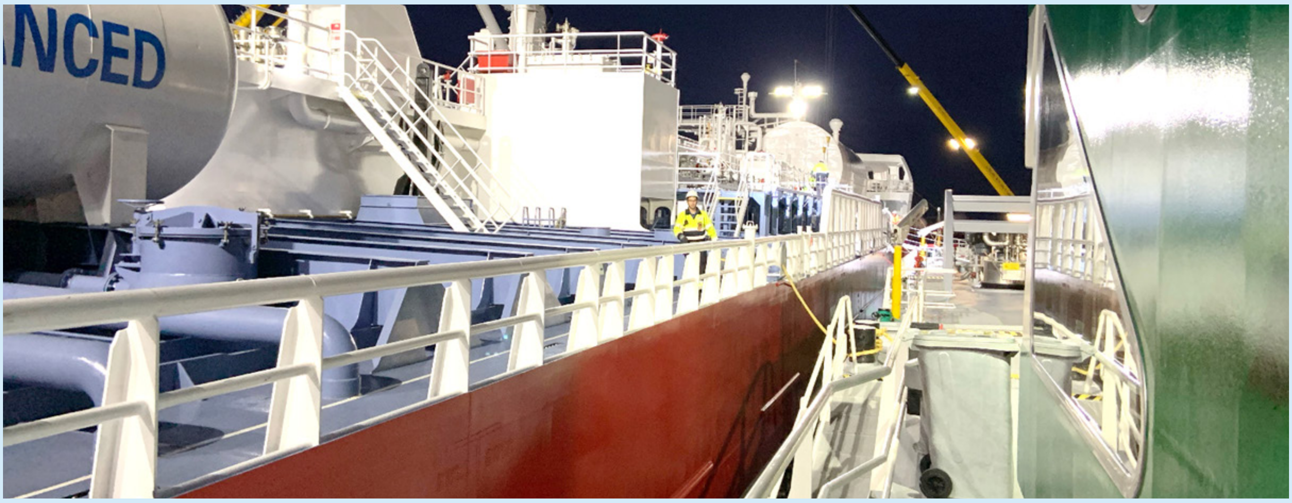
ALTERNATE FUELS



Anuj Kumar Chouhan and Rahul Singh
Marine Engineering Students
Indian Maritime University, Mumbai Campus.

Abstract: The world is facing a huge problem of high fuel prices, air pollution and a lot of climatic changes. Alternate Fuels play an essential role in the present scenario in Internal Combustion Engines as the mineral fuels are depleting. This paper presents a discussion on the use of alternative fuels in internal combustion ignition engines. This article highlights various alternative fuels utilised in India and all over the world.

This review paper covers potential alternative fuels for automotive engine application for both spark ignition (SI) and compression ignition (CI) engines. It also includes applications of alternative fuels in advanced combustion research applications. The representative alternative fuels for SI engines include compressed natural gas (CNG), hydrogen (H2) liquefied petroleum gas (LPG), and alcohol fuels (methanol and ethanol); while for CI engines, they include biodiesel, di-methyl ether (DME), and jet propellent-8 (JP-8).

Naphtha is introduced as an alternative fuel for advanced combustion in premixed charge compression ignition. The production, storage, and the supply chain of each alternative fuel are briefly summarised and are followed by discussions on the main research motivations for such alternative fuels. Literature surveys are presented that highlight the relative advantages and disadvantages of these alternative fuels for application to engine combustion.

Keywords: *Alcohol fuels, Ammonia, Carbon neutral negative fuel, Hydrogen, Natural gas- LPG and CNG).*

I. Introduction

Alternative fuels research has been on-going for well over many years at a number of institutions. Driven by oil price and consumption, engine emissions and climate change, along with the lack of sustainable fossil fuels, transportation sector has generated an interest in alternative, renewable sources of fuel for internal combustion engines. The focus has ranged from feed stock optimisation to engine-out emissions, performance and durability.

Biofuels for transportation sector, including alcohols (ethanol, methanol...etc.), biodiesel, and other liquid and gaseous fuels such as methane and hydrogen, have the potential to displace a considerable amount of petroleum-based fuels around the world.

The second-generation biofuels are still in the development stage. Combining higher energy yields, lower requirements for fertilizer and land, and the absence of competition with food, second generation biofuels, when available at prices equivalent to petroleum derived products, offer a truly sustainable alternative for transportation fuels.

There are main four issues related to alternative fuels: production, transportation, storage, handling and usage. This article presents a review of recent literature related to the alternative fuels usage and the impact of these fuels on fuel injection systems, and fuel atomisation and sprays for both spark-ignition and compression-ignition engines.

Global Alternative fuel consumption, Energy densities of fuel and Properties of different fuels are shown in **Figure 1**, **Figure 2** and **Figure 3** respectively.

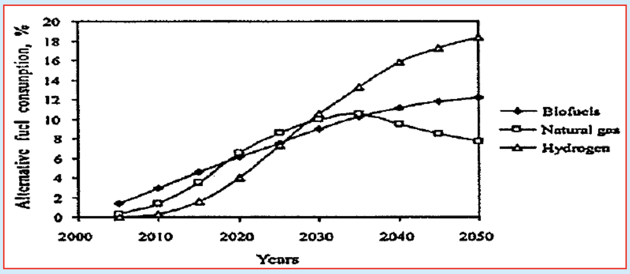


Figure 1. Alternative fuel consumption by world

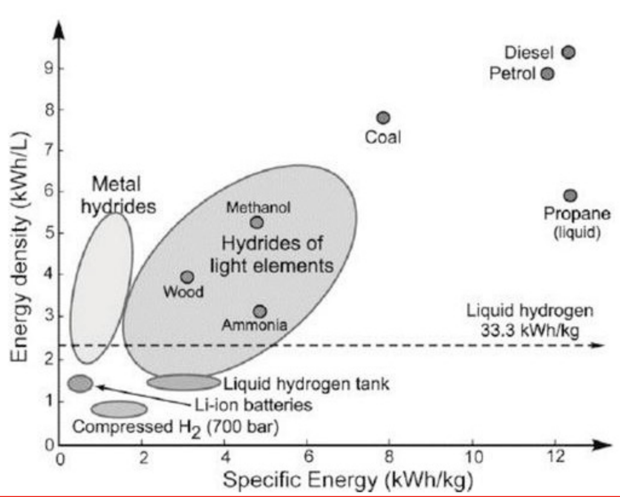


Figure 2. Energy densities of fuel

Fuels	Resource	Expended energy [MJ/MJ fuel]	Greenhouse emissions [g CO_2/MJ]
Gasoline	Crude oil	0.18	13.8
Diesel	Crude oil	0.20	15.4
Natural gas	EU-mix NG	0.17	13.0
	Imported NG 7000 km	0.29	22.6
	Imported NG 4000 km	0.21	16.1
	LNG*	0.28	19.9
	Shale gas	0.10	7.8
	Synthetic from wind electricity	1.05	3.3
Ethanol	Sugar*	1.20	28.4
	Wheat*	1.31	55.6
	Other*	1.66	41.4
Hydrogen	Natural Gas*	1.10	118
	Coal*	1.45	237
	Biomass*	1.05	14.6
	Electricity*	3.11	190

Figure 4. Greenhouse emissions of different fuels

Properties	Acetylene	Hydrogen	CNG	Ethanol	Gasoline	Diesel
Formula	C_2H_2	H_2	CH_4	C_2H_5OH	C_4–C_{12}	C_8–C_{20}
Density (1 atm, 20°C (kg/m³))	1.092	0.08	0.65	809.9	720–780	820–860
Auto ignition temperature (°C)	305	572	540	363	257	254
Stoichiometric ratio (kg/kg)	13.2	34.3	17.2	9	14.7	14.5
Motor octane number	45–50	130	105	89.7	95–97	–
Flammability limits in air (%Vol.)	2.5–81	4–74.5	5.3–15	3–19	1.4–7.6	0.6–5.5
Adiabatic flame temperature (K)	2500	2400	2320	2193	2300	2200
Min. quenching diameter (mm)	0.85	0.9	3.53	2.97	2.97	–
Min. ignition energy (MJ)	0.019	0.02	0.29	0.23	0.23	–
Maximum flame speed (m/s)	1.5	3.5	0.42	0.61	0.5	0.3
Lower heating value (kJ/kg)	48.225	120.000	49.990	26.700	43.000	42.500

Figure 3. Properties of different fuels

II. Alcohol fuels

Methanol and ethanol fuel are primary sources of energy; they are convenient fuels for storing and transporting energy. These alcohols can be used in internal combustion engines as alternative fuels. Butane has another advantage: it is the only alcohol-based motor fuel that can be transported readily by existing petroleum-product pipeline networks, instead of only by tanker trucks and railroad cars. Greenhouse emissions of different fuels are shown in **Figure 4**.

III. Bio-diesels

Bio-diesel is made from animal fats or vegetable oils, renewable resources that come from plants such as atrophy, soybean, sunflowers, corn, olive, peanut, palm, coconut, safflower, canola, sesame, cottonseed, etc. Once these fats or oils are filtered from their hydrocarbons and then combined with alcohol like methanol, diesel is brought to life from this chemical reaction.

These raw materials can either be mixed with pure diesel to make various proportions or used alone. Despite one's mixture preference, bio-diesel will release a smaller number of pollutants (carbon monoxide particulates and hydrocarbons) than conventional diesel, because bio-diesel burns both cleanly and more efficiently.

Property	Value	Test Method
Density @ 15°C, kg/L	0.8801	ASTM D-1298
Copper Strip Corrosion	1A	ASTM D-130
Sulfur, %wt	0.0010	ASTM D-4294
Flash Point, °C	> 110	ASTM D-93
Kin. Viscosity @ 40°C, cSt	4.70	ASTM D-445
CFPP, °C	–6	IP-309
Cloud Point, °C	–2.0	ASTM D-2500
Pour Point, °C	–3	ASTM D-97
Cetane Number	61	DIN 51773
Water, mg/kg	243	ASTM D-1744
Carbon Residue, %wt	0.22	ISO 10370
Net Heating Value, kcal/kg	8906	ASTM D-2015

-BIODIESEL PROPERTIES

Figure 5. Bio-diesel properties

Even with regular diesel's reduced quantity of sulphur from the LSD (ultra-low sulphur diesel) invention, bio-diesel exceeds those levels because it is sulphur-free. Properties of Bio-diesel are shown in **Figure 5**.

IV. Carbon-neutral and negative fuels

Carbon neutral fuel is a synthetic fuel such as methane, gasoline, diesel fuel or produced from renewable or nuclear energy used to hydrogenate waste carbon dioxide recycled from power plant flue exhaust gas or derived from carbolic acid in seawater. Such fuels are potentially carbon neutral because they do not result in a net increase in atmospheric greenhouse gases.

To the extent that carbon neutral fuels displace fossil fuels, or if they are produced from waste carbon or seawater carboxylic acid, and their combustion is subject to carbon capture at the flue or exhaust pipe, they result in negative carbon dioxide emission and net carbon dioxide removal from the atmosphere, and thus constitute a form of greenhouse gas remediation.

Such carbon neutral and negative fuels can be produced by the electrolysis of water to make hydrogen used in the Sabbatical reaction to produce methane which may then be stored to be burned later in power plants as synthetic natural gas, transported by pipeline, truck, or tanker ship, or be used in gas to liquids processes such as the Fischer–Tropsch process to make traditional transportation or heating fuels.

V. Hydrogen

Hydrogen is an emission less fuel. The by-product of hydrogen burning is water, although some mono-nitrogen oxides NOx are produced when hydrogen is burned with air. Nitrogen Oxide values are shown in **Figure 6**.

Hydrogen in the gas phase is about 14 times lighter than the air. Moreover, it is the cleanest fuel in the world. On the other hand, because of its high ignition limit (4–75%), low ignition energy, needs special design to use as pure hydrogen in internal combustion engines.

It is proved that an addition of 20% hydrogen, improves the combustion, emissions and performance.

CR	H_2 (%)	λ = 1.0	λ = 1.15
9.6	0	2000	3620
	5	2100	3825
	10	1710	4185
	20	1535	4225
12.5	0	2040	4410
	5	1940	4200
	10	2260	4520
	20	2210	4695
15	0	2045	4465
	5	2570	4700
	10	2660	4565
	20	3030	4350

*(CR= compressed ratio)
NOx (nitrogen oxide) value (ppm) for λ=1.0 & λ=1.15
Figure 6. NOx (nitrogen oxide) value

Figure 7 shows the skeletal arrangement of introducing the fuel. **Figure 8** is the graph showing minimum ignition energy of hydrogen in air.

VI. NATURAL GAS - LNG & CNG

Natural gas is a mixture of hydrocarbons- mainly methane (CH4) and is produced either from gas wells or in conjunction with crude oil Production. Due to its low energy density for use as a vehicular fuel, it is compressed to a pressure of 200-250 bars to facilitate storage in cylinders mounted in vehicle and so it is called compressed natural gas (CNG).

Low noise, low exhaust emissions, less maintenance, not prone to adulteration, driver's comfort, etc. are some of the attractive features of CNG as an automotive fuel. It can be stored on a vehicle either in a compressed gaseous state (CNG) or in a liquefied state (LNG). **Figure 9** shows the composition of CNG and LPG. **Figure 10** shows the comparative emission from CNG and Diesel.

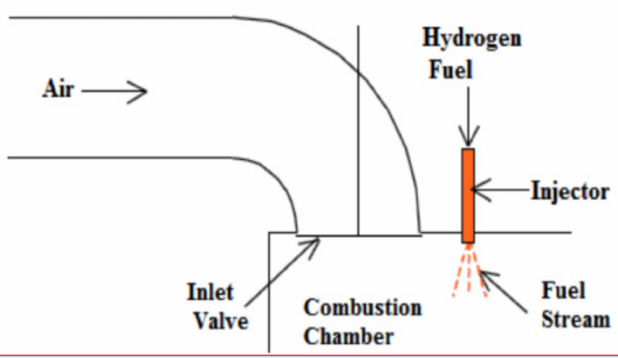


Figure 7. Hydrogen induction mechanism

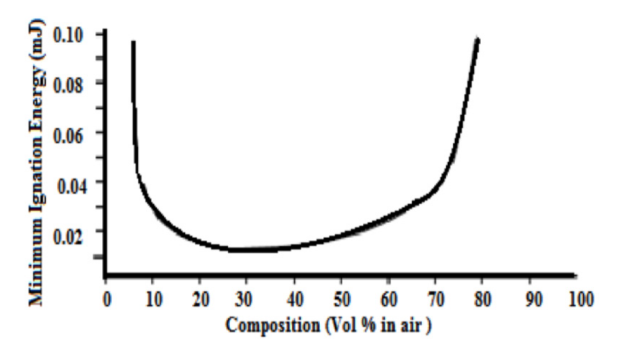


Figure 8. Minimum ignition energy of hydrogen in air

Composition of LPG & CNG

% composition	CNG	LPG
Methane	93.20	-
Ethane	4.27	0.2
Propane	01.38	57.3
Butane	0.58	41.1
Pentane	00.07	1.4

Figure 9. Composition of CNG and LPG

Comparative Emissions from CNG & Diesel

Fuel	Emissions in g/km				PM emissions relative to CNG
	CO	NMVOC	NOx	PM	
Low Sulphur diesel (500 ppm)	1.32	0.50	14.72	0.22	340% higher
ULSD (50 ppm)	1.41	0.52	14.32	0.16	220% higher
CNG	0.66	2.75	9.87	0.05	-

Source: Tom Beer et al 2000, Lifecycle emission analysis of alternate fuels CSIRO report to Australian Greenhouse Office March, mimeo

Figure 10. Comparative emission from CNG & Diesel

VII. P-Series

P-series fuels are a family of renewable, non-petroleum, liquid fuels that can substitute for gasoline. The blend of methyl tetrahydrofuran (MTHF), ethanol, and hydrocarbon constitute the P-series fuel.

They are a blend of 25 or so domestically produced ingredients. About 35% of P-Series comes from liquid by-products, known as "C5+" or "pentanes-plus", which are left over when natural gas is processed for transport and marketing.

Ethanol, fermented from corn, comprises about 45%, and the remaining 22% is MTHF, an ether derived from lignocellulose biomass which can be paper sludge, wastepaper, food waste, yard and wood waste, agricultural waste, and so on. **Figure 11** shows the non-petroleum energy content of the P-series fuels.

Positive: P-Series fuels can be used alone or mixed with gasoline in any ratio by simply adding it to the tank. Negative: Manufacturers are not making flexible fuel vehicles.

P-Series fuels were officially designated as an alternative fuel by the U.S. Department of Energy (DOE) in 1999. The Since P-Series is not derived from petroleum, the DOE concluded that P-Series fuels would effectively help replace petroleum imports. DOE also found P-Series to have environmental benefits because of the reductions in hydrocarbon and CO emissions, toxics, and greenhouse gases.

Much like gasoline, P-Series fuels range from 89-93 octane (mid-grade to premium) and can be formulated specifically for winter or summer use. Refuelling with P-Series is as quick and familiar as with gasoline. But P-Series is not gasoline and cannot be used in a regular gasoline engine.

-Verified Non-petroleum Energy Content of the P-series Fuels

Constituent	Regular	Premium	Cold weather (percent)
Pentanes plus	36.2	33.3	19.1
MTHF	37.7	22.1	32.3
ethanol	26.1	44.6	37.5
normal butane	0.0	0.0	11.2
Non-petroleum (excluding pentanes plus, butane).	63.8	66.7	69.8

Figure 11. Verified Non-Petroleum Energy Content of the P-Series Fuels

The basic capability for utilising P-Series in vehicles has already been incorporated into methanol/ethanol flexible-fuel vehicles (FFV's). FFV's are designed to operate on alcohol, on gasoline, or on any mixture of the two. Nearly three million FFV's have been manufactured since 1996.

P-Series is a relatively new alternative fuel. It is a blend of natural gas liquids, ethanol, and biomass extracts. It has significant emission benefits over gasoline, the greenhouse emissions are about 50% lower than that of gasoline. The use of P-Series would also reduce fossil fuel energy use by 49%-57%, as well petroleum use by 80%.

P-Series comes almost completely from domestic renewable sources. Already major companies such as Dodge and Ford have begun making models that support P-Series as an alternative fuel. P-Series was first added to the list of alternative fuels in 1999. This is actually its major weakness.

Since it is very recent, not much is known on this. Also, the usage of P-Series is also not as wide to have good information. Information based on experience is still awaited.

As engine fuels, the most popular alternative fuels are bioethanol, biodiesel, and hydrogen. Recently, in addition to these, there are intensive researches on methyl-, and ethyl alcohols, natural gas, liquefied petroleum gas, P-series, electricity, and solar fuels. Alternative fuels for diesel engines are becoming increasingly important due to diminishing petroleum reserves and the environmental consequences of exhaust gases from petroleum-fuelled engines.

These fuels are inexpensive fuels generated by municipal and agricultural wastes. The National Renewal Energy Laboratory (NREL) showed that P-series would be 96% derived from domestic resources and reduce petroleum use by 80% as compared to gasoline. Use of P-series fuels also greatly reduces toxic emissions. P-fuels are economically competitive with gasoline.

Pros and Cons of P-Series fuels

One of the advantages of P-series is that they are very easy to use. There is no need for any special fuel management because gasoline and P-series can be freely intermixed in any proportion with fuel that is already in the vehicle's fuel tank. So, even if P-series is not available at a particular location, simply fill up with gasoline.

Using P-Series fuels has several benefits. It decreases the amount of petroleum used to power vehicles. It makes use of waste that would otherwise have to be placed in a landfill, incinerated, or transported to some other location. P-Series fuels are easy to use. Fuelling an FFV with P-Series fuel is identical to fuelling a vehicle with gasoline.

If consumers begin buying these fuels, however, they could be a good substitute for gasoline.

But P-Series fuels cannot be used in vehicles designed to burn gasoline only. FFVs designed to burn methanol or ethanol can burn it, but ordinary vehicles cannot. P-Series fuels are slightly more efficient than gasoline, but in practice, mileage for vehicles using P-Series fuels is about 10 percent less per gallon than those using gasoline.

The feedstock used to make MTHF is chemically digested by the process of making it; as a result, the raw material is completely consumed and no emissions enter the air. Burning P-Series fuels in vehicles releases many fewer emissions than burning fossil fuels.

VIII. Naphtha

Naphtha is a term used to refer to a group of volatile, flammable mixtures of liquid hydrocarbons, that are used mainly as solvents, diluents, or raw materials for gasoline conversion. It is a lightweight petrochemical feedstock that is separated from crude oil in the fractional distillation process along with kerosene and jet fuel.[3]

There are many specific types of naphtha that vary in the amounts and types of hydrocarbons contained in their unique blend. Refineries can produce various forms of naphtha, and each has specific guidelines in how it should be handled and stored. Generally speaking, the flammability and volatility of naphtha should be taken into consideration as they are significant safety hazards. **Figure 12** show the composition of Naphtha. **Figure 13** and **Figure 14** show the properties of Naphtha.

Uses and Safety

Naphtha is commonly used as a solvent. It is used in hydrocarbon cracking, laundry soaps, and cleaning fluids.

Component	Units	Composition
Butane	% mole	0.11
Iso-pentane	% mole	11.69
N-pentane	% mole	13.3
Cyclo-pentane (CP)	% mole	1.95
2,2-Di-methyle-butane (2,2 DMB)	% mole	0.49
2,3-Di-methyle-butane (2,3 DMB)	% mole	1.66
2-Methyl-pentane (2 MP)	% mole	10.4
3-Methyl-pentane (3 MP)	% mole	9.37
N-hexane	% mole	30.72
Methyl-cyclo-pentane (MCP)	% mole	8.69
Cyclo-hexane (CH)	% mole	5.84
Benzene	% mole	3.18
Heptanes	% mole	2.6
Copper	Ppb*	20
Lead	Ppb	10
Arsenic	Ppb	1
Fluorides	Ppb	0.1
Mercury	Ppb	< 1
HCl	Ppm**	0.5
Sulphur	Ppm	0.5
Nitrogen	Ppm	0.5

* Part per billion.
** Part per million.

Figure 12. Component of Naphtha

Physical properties				
Specific gravity (g/cm^3)				0.655
Initial boiling point (°C)				60.7
Final boiling point (°C)				120.3
Chemical composition (wt%)				
Carbon no.	n-Paraffin	i-Paraffin	Naphthene	Aromatics
C_4	2.16	0.12	0	0
C_5	27.34	21.38	3.58	0
C_6	10.19	12.29	3.84	1.58
C_7	3.29	3.82	4.34	1.57
C_8	1.04	1.2	0.92	0.55
C_9	0.26	0.5	0	0.03
Sum	44.28	39.31	12.68	3.73

Properties and composition of naphtha

Figure 13. Properties and composition of Naphtha

Composition (V%)	Reforming topped light naphtha	Hydrocracking light naphtha	Boiling point (°C)	Octane number RON	Octane number MON	Octane number BRON
n-butane	9.36	2.8	−0.50	93.6	93.1	–
2-methyl butane	15.4	26.5	28	92.3	90.3	–
n-pentane	15.7	12.0	36	61.7	61.9	–
2,2-dimethyl butane	9.0	12.0	50	91.8	93.4	–
2-methyl pentane	12.8	5.4	60	73.4	73.5	–
3-methyl pentane	7.2	22.5	63	74.5	74.3	–
n-hexane	13.2	11.5	69	24.8	26.0	–
3-methyl hexane	3.36	–	90.1	42.4	46.4	–
n-heptane	2.88	–	98.4	0	0	–
2,2-dimethyl pentane	–	10.8	79.6	93	96	89
Methyl-cyclopentane	6.2	6.5	71.8	91	80	107
Cyclohexane	8.6	0.9	80.8	83	77	110
1,2-Dimethyl-cyclopentane	2.8	–	99.4	95	–	95
Methyl-cyclohexane	1.4	–	100.8	104	71	104
Toluene	2.9	–	110.5	115	104	124

MON: motor octane number; RON: research octane number; BRON: blend research octane number.

Figure 14. Properties and composition of light Naphtha from hydro-cracking and reforming process

Naphtha is also used to make varnishes, and sometimes is used as a fuel for camp stoves and as a solvent (diluent) for paint.

Highly volatile and flammable, naphtha finds use in many human industries as a solvent, as a fuel, and for industrial purposes. Humans discovered it before the first century A.D. The term naphtha refers to a broad category of chemicals, each one is a potentially dangerous solution of hydrocarbons.

Coal tar, shale and petroleum make up three distinct forms of naphtha, each formed under different conditions and used for different purposes for their chemical properties. In modern production, naphtha often comes from crude oil distillation.

Naphtha has several other names including Enerade ED-6202, high-flash aromatic naphtha, light aromatic solvent naphtha and petroleum naphtha, although these terms can apply to a specific form of naphtha with an intended use.

Naphtha Safety Concerns

Naphtha chemicals can be harmful to humans in various ways. If a human's skin or eye comes into contact with naphtha, the area can become irritated and begin to swell and feel painful. Flush skin and eyes immediately after contact. Ingesting the substance causes nausea, lung damage, respiratory failure and death in some cases.

In the case of ingestion, do not induce vomiting and seek medical attention immediately. As naphtha produces a strong, chemical odour, long-term exposure to it can cause respiratory and mental issues. Some scientists list it as a carcinogen. A toxic chemical, naphtha should not be drained into natural ecosystems.

As most naphtha compounds give off intense, chemical aromas, they are often found in mothballs. Naphtha is flammable and can cause unexpected and dangerous fires.

Naphtha as a Fuel

Humans use naphtha to fuel products because it contains a large amount of chemical energy and is volatile. It can create 3.14 mega joules of energy per litre. Many camping goods stores and hardware stores sell it to power stoves, lanterns, heating units, blow torches and cigarette lighters, thanks to its ability to burn relatively cleanly. It also finds use as an additive to other fuels.

Why don't we use Naphtha as fuel for spark ignition engines?

It is difficult to adopt this fuel for the Otto engines, as the cycle dictates the fast burning which is very difficult to be achieved using the Naphtha.

The S. I. engines need almost homogeneous fuel air mixture, to be introduced at the suction stroke. Therefore, a special arrangement and modification are needed to carry out this preparation process. This might include preheating, atomisation of the fuel, usage of hot compressed air etc.

Naphtha has very low octane number (RON of 70 or less). Modern SI engines cannot be run on such low octane fuels because of knock.

According to a research study on using Naphtha in SI engine, it was found that adding 7% (by vol.) of methanol to Naphtha enhances the engine performance and reduces emissions (than using gasoline). Also, engine noise was also lower.

But according to other studies it was seen that the engine specifications mattered, for example the compression ratio.

Adding methanol increases the fuel octane number and will allow you to run the engine on higher load. Noise reduction might also be related to better knock resistance.

As for emissions - there might be slight reduction in HC and CO but an increase in knock. As for the compression ratio, naphtha is in the gasoline's boiling range and mixture preparation should be no different from gasoline. The burning velocity would also be similar.

As for emissions - there might be slight reduction in HC and CO but an increase in knock could be there. The last word here should be NOx not the knock characteristic.

Other studies did not measure the NOx but surmised that it will be lower due to the high heat required to evaporate the methanol.

However, do you think it is better to use this blend instead of gasoline given this blend has (almost) the same properties of gasoline?

So we think naphtha is cheaper than gasoline and we do not need to other processes to convert naphtha into gasoline. This means lower CO2 emission and low power consumption.

Yes. Naphtha is less processed than gasoline and hence easier to make. But it depends on the availability and ease of manufacture of methanol. You also have to remember that methanol has a very low volumetric energy content (km/litre will be less) and has material compatibility issues.

It is also toxic and this raises handling issues for market deployment.

But some researchers disagree with it. Addition 7% methanol to naphtha improve the fuel quality, combustion process and will decrease the fuel consumption. Regarding methanol toxicity, we think the advantages gained outweigh this disadvantage.

IX. JP-8

JP-8, or JP8 (for "Jet Propellant 8") is a jet fuel, specified and used widely by the US military. A kerosene-based fuel, JP-8 is projected to remain in use at least until 2025. **Figure 15** shows the properties of JP-8. **Figure 16** shows the emission measurement when JP-8 was used.

In order to make this type of fuel compatible with direct injection compression engines, the Fuels and Lubricants

Fueltype	JP-8	Method
Density(gr/ml,15°C)	0.7950	ASTMD1298
Viscosity(cSt,)20°C)	3.87	ASTMD445
Freezingpoint(°C)	)48.5	ASTMD2386
Flashpoint(°C)	41	ASTMD93
Conductivity(pS/m)	375	ASTMD2624
Sulfur(wt.%)	0.23	ASTMD4294
Nitrogen(ppm)	14	
Aromatics(vol.%)	15.3	ASTMD1319
Olefins(vol.%)	0.3	ASTMD1319
Water(ppm)	23	ASTM1744-83
Copperstripcor.	16¼	ASTMD130
Lubricity		CECF-06-A-96
Initialmeasurement(lm)	754	
Repeatedmeasurement(lm)	758	
Distillation(°C)	145	ASTMD86
IBP	174	
10%	181	
20%	200	
50%	233	
90%	250	
FBP	251	

Figure 15 Properties of JP-8

	0.1HP	1.25HP	2.5HP	3.75HP	5HP
NO emissions of JP8 base fuel (ppm)					
Mean value	180	326	352	360	423
Std deviation	2.4	6.7	7.8	3.5	7
NO_x emissions of JP8 base fuel (ppm)					
Mean value	209	351	377	378	433
Std deviation	2.0	2.9	9.4	5.5	5.8
PM emissions of JP8 base fuel (mg/m^3)					
Mean value	4.2	3.2	6.6	6.2	13.5
Std deviation	0.9	0.8	1.7	1.0	2.8

Figure 16. Emission measurement from the stationary petter engine, when JP-8 was used (base fuel measurement)

Laboratory of the National Technical University of Athens, used a stationary Diesel engine fuelled with fuel blends containing two different types of biodiesel, at proportions up to 50%.

In this paper, fuel consumption and exhaust emission measurements from a single cylinder, stationary, Diesel engine are described. The two types of biodiesel appeared to have equal performance, and irrespective of the raw material used for their production, their addition to the JP-8 aviation fuel improved the particulate matter emissions.

The research evaluates the effect of using JP-8 fuel in a heavy-duty diesel engine on fuel injection, combustion, performance, and emissions, and subsequently utilises the obtained insight to propose changes to the engine calibration to mitigate the impact of the trade-offs.

Experiments were carried out on a Detroit Diesel Corporation (DDC) S60 engine outfitted with exhaust gas recirculation (EGR). The results indicate that torque and fuel economy of diesel fuel can be matched, without smoke or NOx penalty, by increasing the duration of injection to compensate for the lower fuel density.

The lower Cetane number of JP-8 caused an increased ignition delay and increased premixed combustion, and their cumulative effect led to relatively unchanged combustion phasing.

Under almost all conditions, JP-8 led to lower NOx and particulate matter (PM) emissions and shifted the NOx-PM trade-off favourably. **Figure 17** shows the mixing mode for the base fuels. Comparison between automotive diesel and JP-8 is shown in **Figure 18**.

Can JP8 Be Used On Cars that Use Gasoline?

When used in highly supercharged diesel engines with the corresponding low compression ratio of about only 14:1 or below, JP-8 cause's troubles during cold start and idling due to low compression temperatures and subsequent ignition can delay. Modern common-rail diesel engines can experience wear problems in high-pressure fuel pumps and injectors.

Workers have complained of smelling and tasting JP-8 for hours after exposure. As JP-8 is less volatile than standard diesel fuel, it remains on the contaminated surfaces for longer time, increasing the risk of exposure.

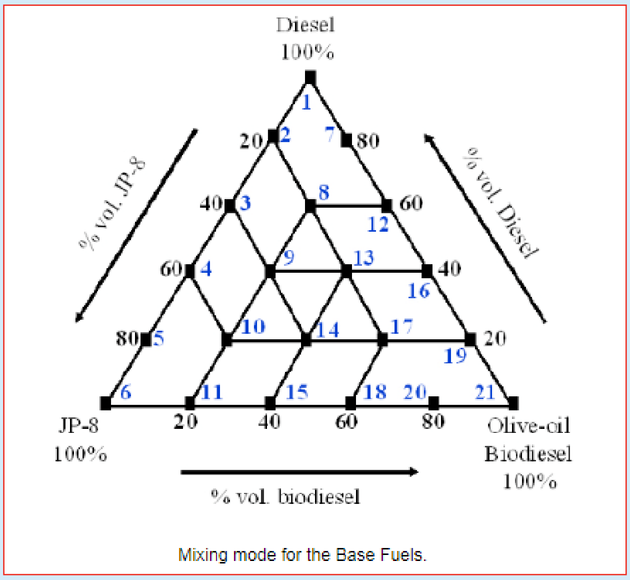


Figure 17. Mixing mode for the Base Fuels

Property	Automotive Diesel	JP-8	Test Method
Density @ 15°C, kg/L	0.8334	0.8001	ASTM D-1298
Distillation, °C			ASTM D-86
10% Rec. Temperature	218	151	
50% Rec. Temperature	283	200	
90% Rec. Temperature	348	238	
Sulfur Content, % wt	0.033	0.2532	ASTM D-4294
Copper Strip Corrosion	1A	1A	ASTM D-130
Flash Point	65	44.5 (D-56)	ASTM D-93
Kin. Viscosity @ 40°C, cSt	.92 (40°C)	4.05 (-20 °C)	ASTM D-445
Cetane Index	57	53	ASTM D-4737
Cetane Number	55	51	FuelTech IQT
CFPP	-7	n/a	IP-309
Freezing Point	n/a	-48	ASTM D-2386
WSD, µm	455	720	CEC F-06-A-96
Conductivity (pS/m)	n/a	420	ASTM D-2624

Figure 18. Comparison between automotive diesel and JP-8

Comparison between combustion, performance and emission characteristics of JP-8 and ultra-low sulphur diesel fuel in a single cylinder diesel engine

JP-8 is an aviation turbine engine fuel recently introduced for use in military ground vehicle applications and generators which are mostly powered by diesel engines. Many of these engines are designed and developed for commercial use and need to be adapted for military applications.

This requires more understanding of the auto- ignition and combustion characteristics of JP-8 under different engine operating conditions. This paper presents the results of a comparative analysis of an engine operation using JP-8 and ultra-low sulphur diesel fuel (ULSD).

Experiments were conducted on 0.42 litter single cylinder, high speed direct injection (HSDI) diesel engine equipped with a common rail injection system.

The results indicate that the distillation properties of fuel have an effect on its vaporisation rate. JP-8 evaporated faster and had shorter ignition delay as compared to ULSD. The fuel economy with JP-8 was better than ULSD. The gaseous emission components such as unburned Hydrocarbons (HC), Carbon monoxide (CO) reduced with JP-8 suggesting improved combustion quality. A slight reduction in NOx was recorded with JP-8, whereas Nano particle emissions increased compared to ULSD.

X. CONCLUSION

P-series fuels are inexpensive fuels generated by municipal and agricultural wastes. Use of P-series fuels also greatly reduces toxic emissions. P-fuels are economically competitive with gasoline.

There is no need for any special fuel management in P-series fuel because gasoline and P-series can be freely intermixed in any proportion with fuel.

If consumers begin buying these fuels, however, they could be a good substitute for gasoline. P-Series fuels are slightly more efficient than gasoline, but in practice, mileage for vehicles using P-Series fuels is about 10 percent less per gallon than those using gasoline.

Burning P-Series fuels in vehicles releases many fewer emissions than burning fossil fuels.

Naphtha chemicals can be harmful to humans in various ways. If a human's skin or eye comes into contact with naphtha, the area can become irritated and begin to swell and feel painful. Flush skin and eyes immediately after contact. Ingesting the substance causes nausea, lung damage, respiratory failure and death in some cases.

It is difficult to adopt naphtha for the Otto engines, as the cycle dictates the fast burning, which is very difficult to be achieved using the Naphtha.

The S. I. engines need almost homogeneous fuel air mixture, to be introduced at the suction stroke. Therefore, a special arrangement and modifications are needed to carry out this preparation process. This might include preheating, atomisation of the fuel, usage of hot compressed air etc.

Naphtha has very low octane number (RON of 70 or less). Modern SI engines cannot be run on such low octane fuels because of knock.

Adding methanol increases the fuel octane number and will allow you to run the engine on higher load if the engine is knock-limited- noise reduction might also be related to better knock resistance.

In order to make this JP-8 fuel compatible with direct injection compression engines, for this they used a stationary Diesel engine fuelled with fuel blends containing two different types of biodiesel, at proportions up to 50%.

Fuel consumption and exhaust emission measurements from a single cylinder, stationary, Diesel engine are described. The two types of biodiesel appeared to have equal performance, and irrespective of the raw material used for their production, their addition to the JP-8 aviation fuel improved the particulate matter emissions.

The research evaluates the effect of using JP-8 fuel in a heavy-duty diesel engine on fuel injection, combustion, performance, and emissions, and subsequently utilises the obtained insight to propose changes to the engine calibration to mitigate the impact of the trade-offs.

When JP-8 used in highly supercharged diesel engines with the corresponding low compression ratio of about only 14:1 or below, JP-8 cause's troubles during cold start and idling due to low compression temperatures and subsequent ignition can delay. Modern common-rail diesel engines can experience wear problems in high-pressure fuel pumps and injectors.

JP-8 evaporated faster and had shorter ignition delay as compared to ultra-low sulphur diesel fuel(ULSD). The fuel economy with JP-8 was better than ULSD. The gaseous emission components such as unburned Hydrocarbons (HC), Carbon monoxide (CO) reduced with JP-8 suggesting improved combustion quality. A slight reduction in NOx was recorded with JP-8, whereas Nano particle emissions increased compared to ULSD.

[This paper was included in the proceedings of the National e-Conference on Energy, Environment and Sustainable Shipping organised by IMU Chennai campus & IME(I) in December 2020. The article is published after several corrections.]

Heritage Hourglass

ANCIENT INDIAN MARITIME TRADE

K. R. A. Narasiah

Numismatic evidence of Trade is available to show that Roman-Tamil trade was going on for centuries without break, though now and then there was slackness. Roman coins are found from 1st century BCE to 5th century CE at various places of coastal area to prove this as symbols of trade. Maximum number of Roman Coins found is however dated to Roman Emperors Augustus and Tiberius. Chinese coins have also been found to show that trade was in directions, east and west.

Romans have complained that their treasury is drained by the trade with India. Truly this was the period of first globalisation and Rome that had enough wealth and the rich Romans had to show their wealth. They spent their money in acquiring pearls, spices and other exotic items from India that soon they found their wealth depleting.

It is said that in 22 BCE Roman Emperor Tiberius complained to his senate that the rage for jewels and pearls from India by the ladies of Rome drained the wealth of Rome. It was estimated that in 70 CE India drained Roman Gold to a tune of million pounds a year. To have such kind of trade existing, there must have been strong shore organisations supporting the trade, known as merchant guilds.

Indian articles like glass beads and gem stones manufactured in Arikkamedu have been found in Thailand also and reports say that copper coins of Cholas and Pallavas have been found in Thailand. All this prove that there was a sea-shore connection for trade and that was through the merchant guilds.

Unfortunately, equally spending kings of the peninsular India were draining a lot of their wealth in getting wine from abroad. Testimony to trade of wine from West for the kings of this region is sherds of pottery, which carried the wine; they are found in excavations right in the interior as well which makes it an important point to note about the spread of the trade inland through the river waters, indicating a strong inland water connectivity and recognition of hinterland fort ports.

The foreign traders with their ware did travel inwards through the rivers. Important evidence provided by the Amphorae jars is the trade of commodities. Although the goods brought in them had been more valuable than these, the bits of such jars remain long after the contents were consumed. All these jars and pots were used only during the flourishing sea trade period of the peninsula. As this trade declined, there was also decline of the presence of jars.

An important point seen is that intermodal transportation was well known then. Cargo came to west coast and from thence to east coast by land and later connected to sea. After Emperor Nero's passing away in Rome, there was a lull in sea trade and later it is found other metallic coins have been in circulation like silver and copper.

In fact, the Roman trail from West Coast to East Coast ports confirms this. History has it that Karur was at various times under Chera, Chola or Pandyas. It was a well-known trading town on the banks of river Amaravathi. Epigraphical, literary and numismatic evidences show that it was an emporium of trade. Ptolemy mentions it as early as 2nd century. When Indian sea trade was well established, the ruler of the day did not interfere with the merchant guilds that managed the trade. These guilds, known by various names such as *Anjuvannam, AinnuRRuvar, Manigramam, Padinenvishayam* etc., had well established trade practices and had their own methods of collecting levies for cargo imported and exported along with prescribed fees to have protected harbours and clearing methods.

No matter where they functioned from, they established certain welfare measures for the people at large of the locality such as water tanks and places of worship and generally acted totally independent of the government. This ensured that their functioning was smooth in spite of changing governments.

The Ruler however ensured safe transit of ships and provided various supporting facilities, in addition to collecting custom duties for imported articles. The classic case is that of Rajendra Chola who with one of the best known navies of the world, ensured that traders were well protected and had easy passage ensured.

According to well-known historians Noboru Karashima and Y. subbarayulu, *Padinenvishayam* was an organisation of the high order, which controlled other guilds such as *Manigramam, Senamugam* etc., and these names have been found in various countries with whom the merchant of the Chola period carried on trade. *Padinenvishayam* itself means eighteen countries. In a gloss on Tamil grammar, the treatise *Nannul*, names the eighteen countries. It may therefore be assumed that the guilds operated in organised network between various countries for good logistics support.

The craftsmen who went out in the ships to follow their profession in foreign countries and continue with their profession were supported by the merchant guilds. Smiths migrated to far off eastern countries to establish the trade there and Noboru Karashima of Taisho University states in a paper that the inscriptions to prove the chronological order.

The first is from Thailand. In the museum there, kept is a stone with an inscription *Perumpatan Kal* in Brahmi script meaning the touch stone of the Chief Goldsmith. This, according to experts may belong to third or fourth century CE. Several beads and similar objects have been found and therefore it must have been a centre of trade for such material.

During the period between fifth and eighth centuries, Karashima[1] says, a good number of Sanskrit inscriptions have been found in Southeast Asia. He quotes the text found in a stone now preserved in Nakhon Si Thammarat museum, that was first published by E. Hulzsh in 1913 and later by K. A. N. Sastri, says that a tank was constructed by a benefactor and put under the protection of a merchant guild – *Manigramam.*

K. A. N. Sastri read the tank's name as *Sri Avaninaranam*. Karashima says that as the name suggests this could be

connected to Pallavas. There are along with it a sculpture of Vishnu and other artifacts in Pallava style and their time could be ninth/tenth centuries. Karashima has observed two more Tamil inscriptions now kept in a Buddhist temple and says one of them mentions the name of a donor Dhanma Senapathi, who made grant to Brhamanas. The name could be that of a merchant. He mentions that in Pagan (Myanmar) there is an inscription belonging to 13th century with first half showing a *Mantra* of Vaishnavites and the later-half saying that a hall was constructed with some facilities by *Irayiran Kulasekhara Nambi.* The temple is itself constructed by *Nanadesis.*

But the most amazing inscription is the one that was introduced by Late T. N. Subramaniam. This was from Quanzhou, a medieval port of south China. The text reveals that one Champanda Perumal also known as Thava-Chakravarthigal, having got a grant of land from the then King Khan, built a temple there and called it *ThiruKhaneeswaram* after the Khan.

All the above evidences clearly show that system of merchant guilds were strong and controlled with its own set of rules. An inscription found in the Vishnu temple in north Chennai, states that to make *Mylapore* a protected harbour certain levy was laid on goods imported or exported. Similarly, we find a levy called *Pattina Pakudi* (Pattinam means a port town) was levied for maintenance of the harbour.

Thus we find that apart from a strong merchant fleet and sailing community the ancient South Indians had supporting organisation called by various names but controlling the trade in a most efficient manner. This the organisation did on its own irrespective of who ruled the country.

About the Author

K.R.A. Narasiah is a Marine Engineer with a flair for research into Maritime History and heritage. He is an eminent and a popular scribe. He writes regularly to The Hindu and The Times of India.

Email: narasiah267@gmail.com

1. Ancient and medieval Commercial Activities in the Indian Ocean: Testimony of Inscriptions and ceramic sherds. Edited by Noboru Karashima.

COLUMN

Heritage Hourglass

EXPLORING BOAT BUILDING IN MEDIEVAL INDIA UNDER THE MUGHAL RULE

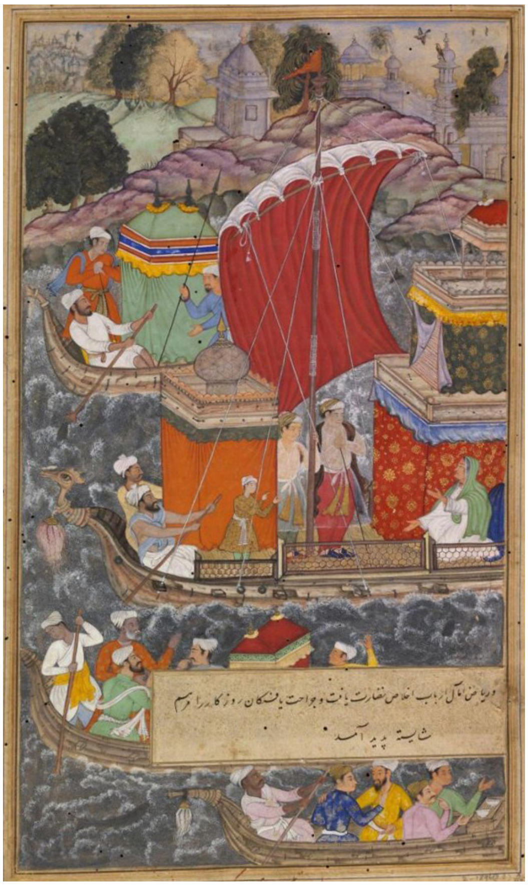
Ships near Chittagong, Source: Chughtai Museum, https://bit.ly/3pzcSPC

Sadaf Khan

Maritime activities like sea-borne trade and commerce, marine navigation along with maintenance of flotilla have been one of India's key characteristics since ancient times. With a long coastline of about 7,516.6 km, evidence of shipbuilding along the Indian Coast has been found since the Indus Valley Civilisation. During the Mughal rule an Imperial flotilla was maintained and set up in Bengal which was one of the centres of Indian Ship building. Abul Fazl's Ain-i-Akbari (16th Century) presents exhaustive information on Mughal ship building and the organisation of the Mughal flotilla.

Most of the Mughal rulers took great interest in shipbuilding beginning with Akbar who built his first ship at Lahore in 1593; followed by Jahangir, Shah Jahan, and Aurangzeb, all of whom invested in ship building primarily to send voyages to Mecca. When Prince Khurram (later

During the Mughal rule an Imperial flotilla was maintained and set up in Bengal which was one of the centres of Indian Ship building



Akbar's mother travels by boat to Agra, Source: Victoria & Albert Museum, https://bit.ly/3dvAhf5

Shah Jahan) was the governor of Gujarat, he built and managed his own ships, famous among them being '*Shahi'* and '*Ganjhawar*'. Women too participated in maritime commerce; Princess Jahan Aara invested in shipbuilding and Noor Jahan owned ships from which she collected revenue.

It was the abundant supply of timber amongst other factors that helped further this shipbuilding industry during the Mughal period

Mughal Ship-building

Ship building was carried across the Mughal Empire, the records of which are maintained in Abul Fazl's Ain-i-Akbari. He notes that on the Indian sea-coast, in the east, west, and south, large ships were built, which became a source of comfort to the seafarers, as ports obtained prosperity. It was the abundant supply of timber amongst other factors that helped further this shipbuilding industry during the Mughal period. It is in this context that Surat became one of the centres of shipbuilding; the work being carried out at the Swally port (Mookerji 1957).

On the East Coast, Bengal was the primary site of boat and ship building. It was known for building large ships some of which measured hundred yards long and twenty yards wide. The boats were used for port to port riverine trade and the collection of merchandise. With the availability of some of the best timber from Chittagong, the Mughals were able to maintain a *Nowwara* (flotilla) specifically in this region. Shah Jahan has been credited for developing the Bengal *Nowwara* which consisted of 70,000 personnel, besides soldiers. The Mughals also built small cannons that could fire cannon-balls of 9 seers by weight which were mounted on these fighting boats.

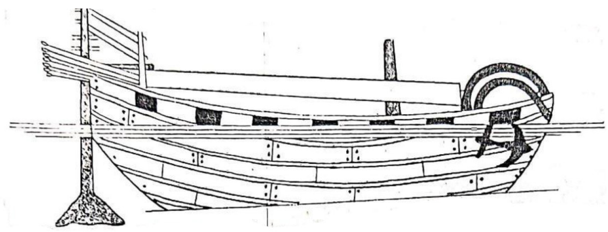
Besides Bengal, the province of Sind was also a big centre of Indian shipping. Shipbuilding was carried out at Lahore due to availability of timber from the Himalayan region. As Lahore was part of the mainland, the port nearest to Lahore was Lahari Bandar on the western bank of river Indus which was a few miles away from the city of Thatta. Thatta was far inland and it took about three days to reach its port - Lahari Bandar (Paliwal 1999). Larger vessels and ships were anchored at Lahari Bandar, once the cargoes were unloaded the merchandise was brought to Thatta in small boats up to the river and then animals were used for transportation by land. Abul-Fazl writes that in Thatta, Pakistan alone, there were 40,000 vessels ready for hire. He also gives details on the vast quantity of iron nails which was used in the building of two ships at Lahore by Akbar in 1593 and 1596. Lahari Bandar in those days was an important seaport on the Indus. Thus Lahari Bandar was a great centre of riverine trade which provided a good harbour for ships.

During Aurangzeb's time (1658-1707), a marked development was seen in shipping and maritime commerce throughout India. Thomas Bowrey, an English traveller to India during A.D. 1669-79, left a very valuable account of kingdoms around the Bay of Bengal, in which he gives descriptions and representations of ships and boats, which are "among the best of the kind for this period" (Mookerji 1957). There are also records of Aurangzeb retaining four great ships at all times which were always in pay to carry pilgrims to Mecca free of cost from Surat.

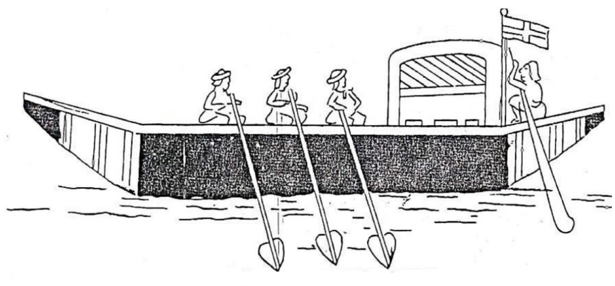
Mughal Boats & Types

During the Mughal period, the waters and areas where the vessels were used can be broadly divided into three classes; the first class of vessels were 'Junka' and 'Jahaz' which were only used in the ocean. The second class of vessels were 'ghurab', 'taory', 'sambuk', 'shibar', 'manchua', 'balloon', and other kinds of boats which were used on both the ocean and along the coast. The third class of vessels known as 'parao', 'purgo', catamaran, 'patella', 'jelia' etc. was used for internal navigation i.e in the rivers.

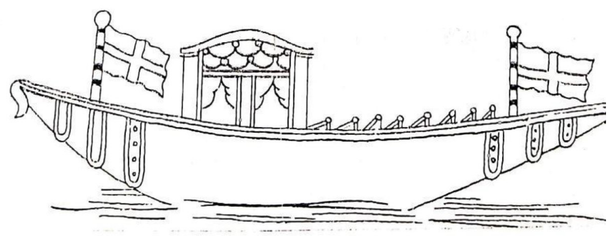
Apart from this, the 'Massoola' boats were used in loading and unloading ships or vessels, they were slightly built, had no timbers in them or 'thafts' to hold their sides together. Their planks were very broad and thin, sewn



PATELLA



OLOAKO



BUDGAROO

Mughal Boat Forms, Source: Mukherjee, R.K. "Indian Shipping." Longmans, Green & Co. Ltd, 1957 pp.169.

Boats on the Ganges, Source: Maritime History Society, https://bit.ly/3ECx8pw

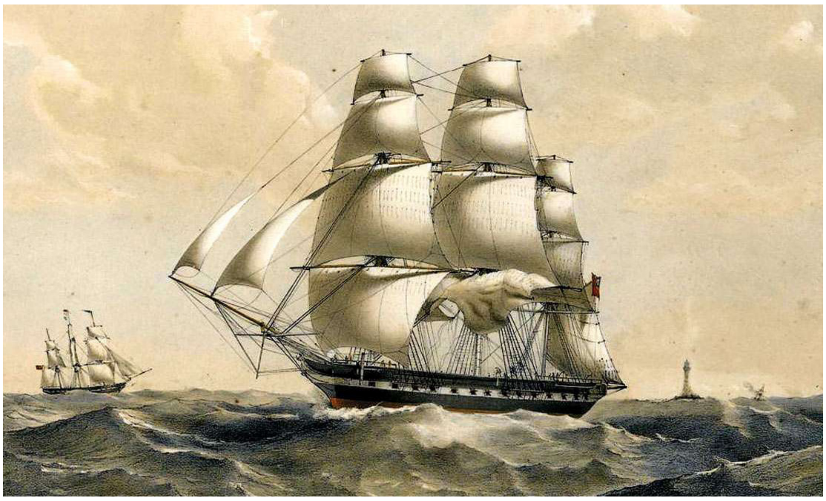
together with coir; they were flat-bottomed and were deemed fit for the Coromandel Coast. There was another kind of boat called the catamaran, made of four, five, or six large pieces of buoyant timber upon which they could load three to four tons of weight (Mookerji 1957, 167).

In Bengal, great flat-bottomed vessels known as 'Patellas' of considerable strength sailed the waters. Various kinds of boats were in use on rivers like the 'Oloako' boats rowed either with four or six oars, which plied for a fare. A 'Budgaroo', was a pleasure boat, used by the upper classes.

The Emperor owned pleasure boats which were built with convenient apartments. There were also boats built with floating markets and flower gardens. A 'Bajra' was a kind of large boat, the centre of which formed a little room. The 'Purgoos' which were seen for the most part between 'Hugli', 'Pipli', and 'Balesore' were used for loading and un- loading ships. 'Booras' were light boats, rowed either with twenty or thirty oars (Mookerji 1957, 167). They carried salt, pepper, and other goods from Hugli downwards, and engaged in trade to Dacca; they also served as tow-boats for the ships bound up or down the river.

Mughal Flotilla

According to Abul Fazl, this department was named as the office of '*Meer Behry',* and its function was to see to the supply of ships and boats for navigation and supervise their building. These vessels were built of various sizes; they were used for transportation of elephants and for



the conveyance of merchandise. The duty of this office was to supply efficient mariners who knew the nature of tides, the depth of channels, the coasts to be avoided and the character of the winds. The Mughal flotilla was also supposed to watch the rivers for which resolute men were appointed who managed ferries, regulated the tonnage and provided travellers with boats. They imposed duties and maintained the imperial *Nowwara* (flotilla). Under Akbar the *Nowwara* consisted of 3000 vessels which during the later period were reduced to 768 armed cruisers and boats. The office of '*Meer Behry*' was headquartered at Dacca and it guarded the coast of Bengal from foreign pirate invasions. The revenue for the support of this naval establishment was derived from *Mheer Baree, a tax* on the building of boats which depended on the size of the vessels. *Mheer Baree* was also levied upon the boats arriving or leaving the headquarters whose crew were non-residents of the district.

Officers and men of the following titles and descriptions were stationed in every ship:

(1) The Nakhoda was commander of the vessel, who directed the course of the ship.

(2) The Maullim (the mate), knew the soundings, the situation of the stars, and guided the ship safely to her destination.

(3) The Tundeilf was the chief of the khelasses or sailors.

(4) The Nakhoda-khesheb, his duty was to provide fuel for the people and assist in loading and unloading the ship.

(5) The Sirheng supervised the docking and launching of the ship.

(6) The Bhandaree was in charge of the ship's store.

(7) The Keranee, or ship's clerk, kept the accounts and also served water to the people.

(8) The Sukangeer, or the helmsman, were sometimes twenty per ship.

(9) The Punjeree, whose duty was to look out from the top of the mast and give notice when he saw land or a ship, or discovered a storm rising, or any other object worth notifying.

(10) The Goomteey or those particular khelasses were those who threw the water out of the ship.

(11) The Gunners differed in number according to the size of the ship

(12) The Kherwah, or common seamen, were employed to set and furl the sails and to stop leaks, and in case of the anchor sticking fast in the ground they had to go to the bottom of the water to set it free (Mookerji 1957, 148).

Conclusion

Mughals possessed ships of every kind, big or small for both civil and military use. These were made in good numbers thus showcasing that shipbuilding was a flourishing occupation during Mughal times. The Mughals built various types of boats and ships which could be categorised as river ships, sea-going vessels, navigation ships, war ships, Haj pilgrimage ships, and pleasure boats for their personal usage. The indigenous designs of Mughal ships complimented the environment and the context of the water bodies in which they sailed thus showcasing a high level of command and control on their empire and knowledge of their waterscape. The Mughals, though they didn't sail into the high seas they understood the importance of maritime trade and hence also shipbuilding.

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About the Author

Sadaf Khan has a MA in Archaeology from University of Mumbai. Sadaf is an Archivist with experience in the research analysis and appraisal of historical documents and has worked with CSMVS as a Research Intern in Education Sector. She also has a MA in History from the University of Mumbai. The trajectory of her endeavours in History and Archaeology and her research are highlighted in "Exploring boat building in India under the rule of Mughals". With a blended background in academic and real-life experience in History and Archaeology, she is working at Maritime History Society as a Staff Assistant (Academics & Collections).

Blessings from a Bhishma

Dear Sir,

I have just now received my copy of MER and I must say how delighted I felt on reading it. Being the founder Editor of MER, it gives me great satisfaction to see that our journal has made such tremendous progress in terms of quality of articles and its presentation. Please accept my heartfelt congratulations and good wishes.

Thanking you,

Hari Taneja
F0001

It is very humbling to see these kind words from men of stature. Our team will try to keep up the good work and make things better.

- Hon.Ed.

Kudos from our Cartoonist

The quality and enriching nature of the articles in MER over the past Decade was never a matter of debate. But I would like to convey my first impression that under the able Steering and Propulsive thrust of our beloved Hon Editor, the Look, Feel and Quality of the presentations in the journal have touched unmatched levels. Kudos to his efforts. It would be a great disservice to the esteemed authors if we dare to pick any one individual article for special mention. Each and every article is of the topmost quality. Congrats and may the Voyage continue.

Ramesh Subramanian
F4498

A Note of Appreciation

Dear Dr. Rajoo Balaji and Ms. Rashmi,

I have been reading the Marine Engineers Review (MER) journal over the last few years. The informative and insightful technical articles published have greatly enhanced my application knowledge and they significantly contribute to the professional development of marine engineers. The meticulous editing done by your team and the editor's foreword in every issue is commendable. I am proud to be a regular reader and recently a contributor to this prestigious journal. Wish you and your team, all the very best.

Dr. N. Vedachalam
Scientist G - Deep Ocean Mission / Technology,
National Institute of Ocean Technology, Ministry of Earth Sciences

Corrigendum:

MER February 2022 Issue

Press Releases: Pages 42 & 44.

Referring to the Press Releases, one photograph tile was repeated for two different events. This inadvertent error is regretted.

PUNE BRANCH

TALK ON "UNDERSTANDING MARINE INSURANCE"

IME(I), Pune Branch arranged a talk on **"Understanding Marine Insurance"** on **Friday, 28 January 2022** through an online platform. The talk was delivered by Dr. Brijendra K. Saxena, Founder Principal (Retd.), Tolani Maritime Institute, Induri, Pune and also a Past President of IME(I).

Mr. Sanjeev Ogale, Chairman, IME(I), Pune Branch welcomed the attendees and introduced the speaker. The event was attended by senior industry professionals as well as students undergoing preparatory courses. More than 220 attended the lively interactive session.

The topics covered included an introduction of marine insurance and how it is different from other branches of insurance; basic terms and principles; areas of coverage of Hull & Machinery, cargo and liability insurances (P&I) etc. He specifically explained the new developments in the English law due to the amendments to the Marine Insurance Act 1906 because of their new Insurance Act 2015 and requested the industry that such changes should also be brought on in the Marine Insurance Act 1963 of India. The talk was followed by a small quiz and a lively Q&A session.

Mr. Anant Sahasrabudhe thanked the speaker and the participants and acknowledged the support of the Head Office of IME(I) in hosting the event.

MOU Renewal Signed between IME(I) and GESCO for Skill Enhancement Courses

Obituary

Ravi Bhardwaj was a brilliant man with a very practical side. He graduated from DMET in 1972 with many awards and accolades. After graduation, he joined Scindia Steam Navigation Co. Over the next few years, he took his licenses in India and UK and got promoted to Chief Engineer at a very young age.

A very happy person, Ravi always brought cheer to any gathering with his great sense of humour. He was truly fun to be with. From the DMET 1972-year book: "An extremely lovable and open-hearted person we doubt whether we will ever forget that spontaneous guffaw of his."

Ravi K. Bhardwaj (M582)

(20. 05. 1950 - 03. 02. 2022)

A straight shooter, he tried a short stint ashore but did not enjoy the office politics, so he went back to sea and continued sailing as Chief Engineer.

After his sailing phase, he with his family settled down in Ghaziabad where they had built their home. He was very proud of his garden, and regularly shared photos of many new gardening projects. He is survived by his wife Sujata, two daughters, Pieusha and Ruchika, and four grandchildren.

A wonderful son, brother, husband, father and grandfather, he was also a dear friend to his college batch mates. He will be sorely missed.

IN THE WAKE

Rajoo Balaji

MER Matters

This March brings only little ripples In the wake... and the ripples relate to our MER...

The Journal Management System (JMS) of MER is warming up but slowly. I request all Members to register and flip through the flipbook version and advise for improvements.

In same earnestness, it is requested that contributions may be attempted through the JMS portal. This would bring in the necessary discipline for the review-advise-corrections-revise process.

And do advise us for improvements.

About March

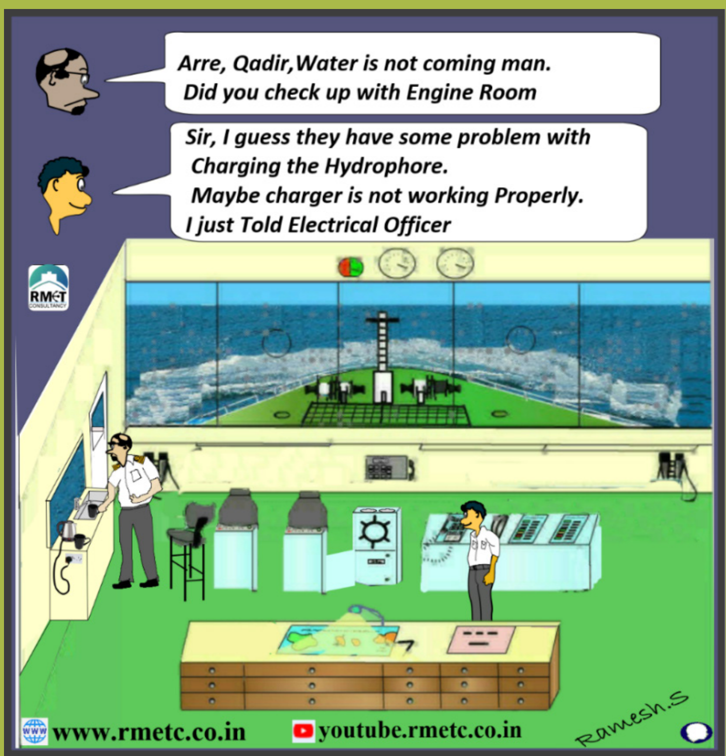
My darts found the following dates for this month... I am adding a few feathers to the darts...

Dart#1: **March 1: Zero Discrimination Day**. Feathers: Hats, Hijabs, Half-folded dhotis...let us love them all...

Dart#2: **March 4: National Safety day**: Safety First... and keep it last also as we complete any process.

Dart#3: **March 8: International Women's Day**: From Cheerleaders to Chief Engineers, may the tribe increase and be respected.

THE END VIEW

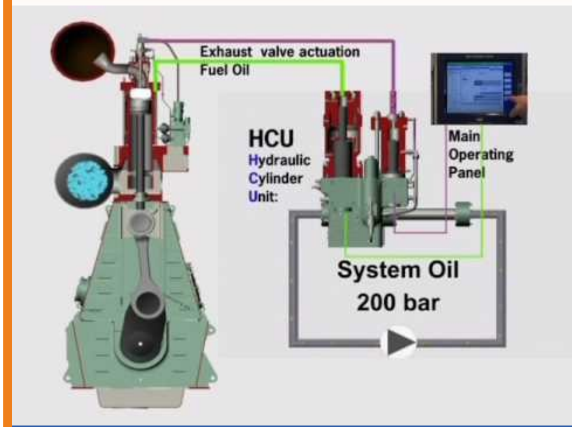


Idea, Words & Drawing: **Ramesh Subramanian**

MASSA Maritime Academy, Chennai

The Institute of Marine Engineers (India)

Electronic Engine Familiarisation Course (ME-Type Engine) Delivered online with Cloud access to ME Engine Simulator



This 3 days course is designed for all Ship's Engineer Officers and Electro Technical Officers responsible for the operation of ME Engine. This course consists of technical lessons and practical instructions on the design, principles, operating procedures and maintenance activities for the safe, efficient and optimal performance of the engine system.

Course Aims and Objectives:

The course aims to provide practical understanding of the principles, design, operation and maintenance of the ME Engine System, enabling participants to safely and efficiently operate the engine and perform fault-finding in the control system.

Coverage / Program Focus:

This course deals with the following training areas:

- **Introduction to ME Engine**
- **Hydraulic Power Supply (HPS)**
- **Hydraulic Cylinder Unit (HCU)**
- **Engine Control System (ECS)**
- **Main Operating Panel (MOP)**
- **Standard Operation**

Entry Requirement / Target Group:

Entry is open to all Ship's Engineers and Electro Technical Officers with basic knowledge of diesel engines.

DATE & TIMING : 15^{th} to 17^{th} **Mar'22**; 19^{th} to 21^{st} **Apr'22**; 17^{th} to 19^{th} **May'22**; 14^{th} to 16^{th} **June'22**; 19^{th} to 21^{st} **July'22**; 16^{th} to 18^{th} **Aug'22**; 20^{th} to 22^{nd} **Sep'22** **8:00 am - 4:00 pm IST**

VENUE : Web Platform / Zoom**. APPLICATION LINK:** https://forms.gle/e4As7kCucR5xoJBm9

REGISTRATION & PAYMENT : Rs. 15,000/- /- per participant – inclusive of taxes.
For IME(I) Members 13,500/- per participant - inclusive of taxes.
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(Under Category - Value added Courses) *10% discount available for IME(I) members*

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