*Int. J. Mech. Eng. & Rob. Res. 2015*

International Journal of Mechanical Engineering and Robotics Research

*ISSN 2278 – 0149 www.ijmerr.com*
*Vol. 4, No. 2, April 2015*


*Research Paper*

# DESIGN OF MARINE PROPULSION SHAFTING SYSTEM FOR 53000 DWT BULK CARRIER

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The basic operational requirement for a marine propulsion shafting system is to transmit the torque over the required range of speeds. In the present work we have designed a shafting system for 53000 DWT BULK CARRIER, which has to transmit a power of 9840 KW and 9960 KW@124 RPM. Considering the needs of propulsion shafting system, the diameter of intermediate shaft, propeller shaft, journal bearings, flange couplings are calculated taking into consideration, the various forces acting on the system and some empirical relations from the data book presented on the bibliography of this book. In accordance with the bearing loads and speed of the engine. The design is done from the first principles, making some assumptions and the results are checked with the Lloyd's register of shipping rules, which are the standard rules followed to maintained quality of the products for better performance, reliability and durability. Sectional views of the shafting are shown for better understandability.

Keywords: Propulsion shaft, Torque, Power, Bending moments, Lloyd's register of shipping rules

## PRINCIPLES OF SHIP PROPULSION

A ship moves through the water through propelling devices, such as paddle wheels or propellers. These devices impart velocity to a column of water and moves it in the opposite direction in which it is desired to move the ship. A force, called reactive force because it reacts to the force of the column of water, is developed against the velocity-imparting device. This force, also called thrust, is transmitted to the ship and causes the ship to move through the water.

The screw-type propeller is the propulsion device used in almost all naval ships. The thrust developed on the propeller is transmitted to the ship's structure by the main shaft through the thrust bearing (Figure). The main shaft extends from the main reduction gear shaft of the reduction gear to the propeller. It is

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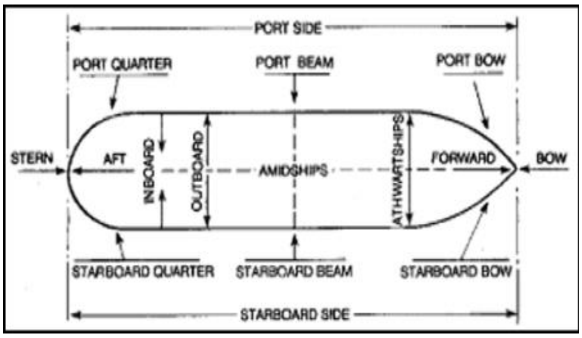
Figure 1: General Layout of Ship




supported and held in alignment by the spring bearings, the stern tube bearings, and the strut bearing. The thrust, acting on the propulsion shaft as a result of the pushing effect of the propeller, is transmitted to the ship's structure by the main thrust bearing. In most ships, the main thrust bearing is located at the forward end of the main shaft within the main reduction gear casing. In some very large ships, however, the main shaft thrust bearing is located farther aft in a machinery space or a shaft alley.

The main reduction gear connects the prime mover (engine) to the shaft. The function of the main reduction gear is to reduce the high rotational speeds of the engine and allow the propeller to operate at lower rotation speeds. In this way, both the engine and the propeller shaft rotate at their most efficient speeds.

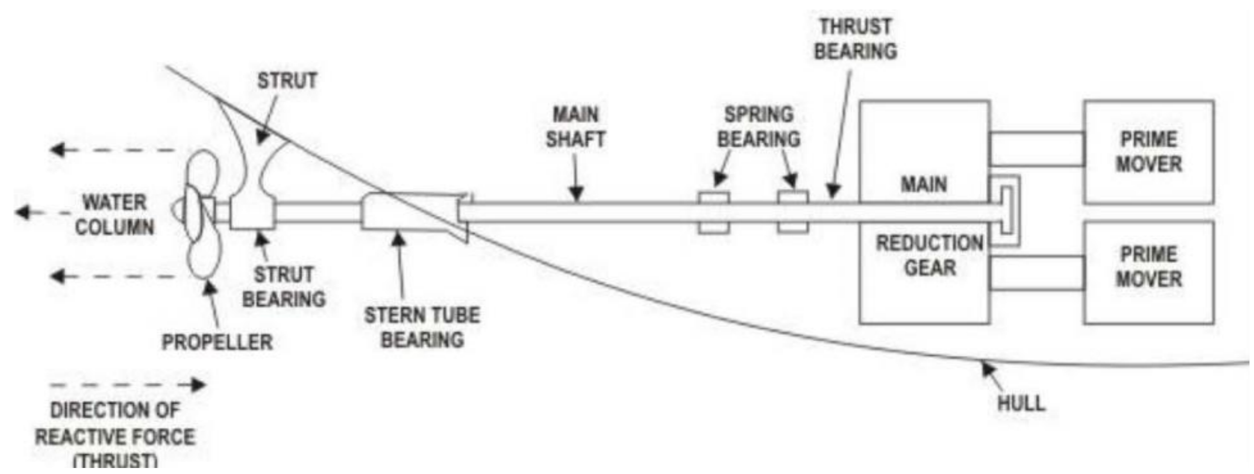
## TYPES OF PROPELLERS

### Controllable Pitch Propeller

The machinery must develop enough torque to turn the propeller at the revolutions appropriate to the power being developed or the machinery will lock up. This matching is not always possible with fixed blades and some ships are fitted with propellers in which the blades can be rotated about axes normal to the drive shaft. These are termed Controllable Pitch Propellers (CPPs).

The pitch of the blades is changed by gear fitted in the hub and controlled by linkages passing down the shaft Thus the GPP has a larger boss than usual which limits the blade area ratio to about 0.8 which affects cavitation performance. It is also mechanically fairly complex which limits the total power that can be transmitted. By reversing the pitch an astern thrust can be produced thus eliminating the need for a reversing gear box.

Figure 2: General Principle of Geared Ship Propulsion

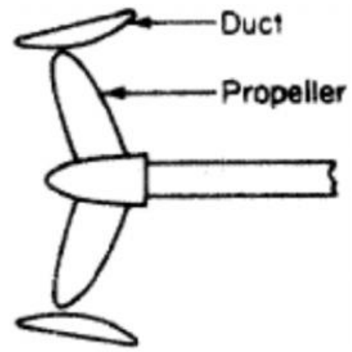


The advantages and disadvantages of Controllable-pitch propellers (CPP) are

Fast Stop Maneuvers are Possible

**Shrouded or Ducted Propellers:** The propeller is surrounded by a shroud or duct as depicted in Figure. The objects are to improve efficiency, avoid erosion of banks in confined waterways and shield noise generated on the blades.

Figure 3: Shrouded or Ducted Propellers



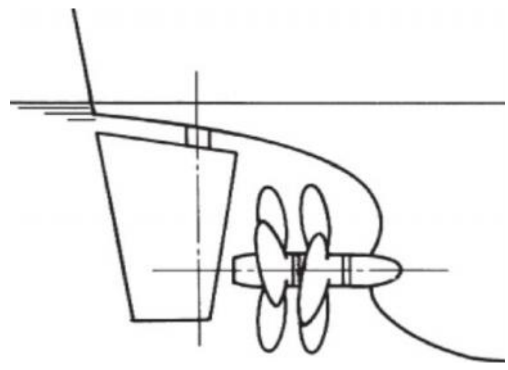

The duct can be designed so that it contributes to ahead thrust so offsetting the drag of the shroud and its supports. Most early applications were to ships with heavily loaded propellers like tugs. Its use is now being extended and it is considered suitable for large tankers.

**Pump Jets:** This is an advanced variant of the ducted propeller 10 for use in warships, particularly submarines, where noise reduction is important. A rotor with a large number of blades operates between sets of stator blades the whole being surrounded by a specially shaped duct.

The Main Engine Does Not Need to be Reversible

**Contra-Rotating Propellers:** Rotational exit losses amount to about 8-10% in typical cargo

Figure 4: Contra-Rotating Coaxial Propellers



ships. Coaxial contrarotating propellers can partially compensate these losses increasing efficiency by up to 6% (Isay, 1964). To avoid problems with cavitation, the after-propeller should have a smaller diameter than the forward propeller.

Contra-rotating propellers have the following advantages:

1. The propeller-induced heeling moment is compensated (this is negligible for larger ships).
2. More power can be transmitted for a given propeller radius.
3. The propeller efficiency is usually increased.

## DESIGN OF THE SHAFTING SYSTEM FROM THE FIRST PRINCIPLES

Shaft material = CK 45

Tensile strength = 600 N/mm$^2$

Yong's modules = 207 GPO

Density of material = 8740 kg/m$^3$

**Note:** The shaft is of forged steel with coupling flanges forged integrally to it and the material of coupling bolts is same as that of shaft.

Power to be transmitted, P = 9480 kw @124 rpm

With a 10% over load to account for quick reversals.

**Twisting Moment**

Torque, $T = (60\ p/2\Pi)$ x *over load*

$= (60 \times 9480 \times 10^3)/2\Pi \times 124$

$= 803.06 \times 10^6$ N-mm

Based on design of shaft subjected to pure torsion from the theory of pure torsion, we have

$T/r = T/J$

$J$ – Polar moment of inertia

$= d^4/32$ for a shaft of diameter $d$

Taking a factor of safety F.S = 4

Max operating shear stress

$$= T\ \max / F.S = \frac{255.62}{4}$$

$D = (16T/T)^{1/3}$

$$= \left(\frac{16 \times 803.472 \times 10^{6\ 1/3}}{63.095 \times \Pi}\right)$$

= 400.037 mm

## Design of Propeller Shaft (dp)

Since the propeller shaft is subjected to both bending moment due to propeller weight and twisting moment diameter of propeller shaft dp = 25% extra then the designed by pure torsion.

dp = 1.25 x 400.135

**dp = 500 mm**

## Design of Intermediate Shaft (d)

The intermediate shaft is subjected to bending moment along with twisting moment. (Proceeding with d = 4000 mm. we find the shaft fails under combined bending and twisting and for timer to be > 63.75 N/mm$^2$)

We find, optimize diameter of intermediate shaft.

**d: 400 mm**

## Design of Flange Coupling

For the diameter of the shaft, we get the number of bolts, nb = 10.

## Design of Intermediate Shaft Flange

Thickness of flange = 0.27d + 5

= (0.27 x 400) + 5

**ti = 113 mm**

Diameter of flange, df; = 2.2 di = 2.2 x 400

**df; = 880 mm**

Diameter of bolt circle, P.C.D = 1.6 x di; = 1.6 x 400

**P.C.D = 640 mm**

Mass of coupling

$= \Pi/4 \times (df^2 - d_j^2) \times ti \times p$

$= \Pi/4 \times (8802 - 4002) \times 113 \times 8740 \times 10^{-9}$

= 476.33 kg x 9.81

= 4672.82 N

## Design of Propeller Shaft Flanges

Thickness of flange = 0.25 $d$ + 5

= (0.27 x 500) $T$ = 140 mm

Mass of flange (excluding shaft) $\Pi/4 \times (dp^2 - d_j^2) \times tf \times p$

dp = 500 mm

# BENDING MOMENT CALCULATION

## For Propeller Shaft

$R_{aft} + R_{fwd}$ = 14.71 x 4980 + 6462.93 + 155653 = 235371.73 ...(1)

$-R_{fwd} \times 3040 + 6462.9334 \times 4313.33 - 155653 \times 666.67 - {}_0\int^{666.67} 14.71 \aleph dx + {}_0\int^{4313.33} 14.71 \aleph dx = 0$ ...(2)

**Figure 5: Layout of Ship Propulsion System**

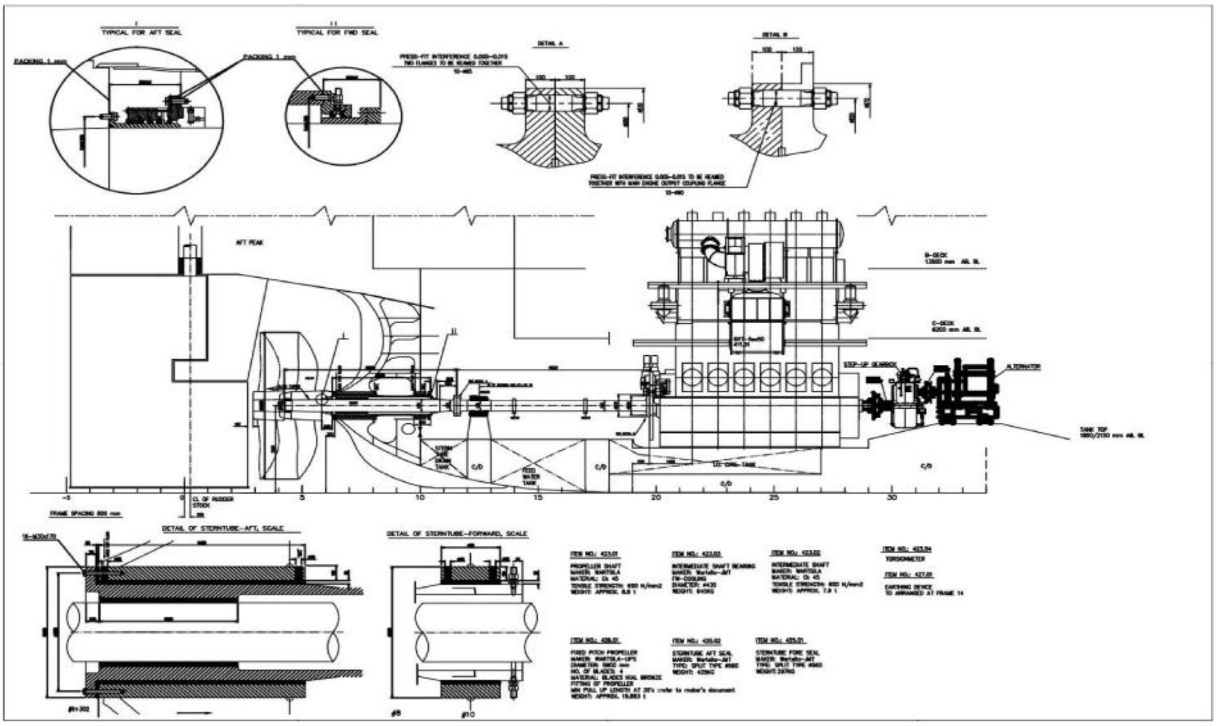


**Figure 6: Layout of Shaft from Engine Thrust Beraing to Propeller**

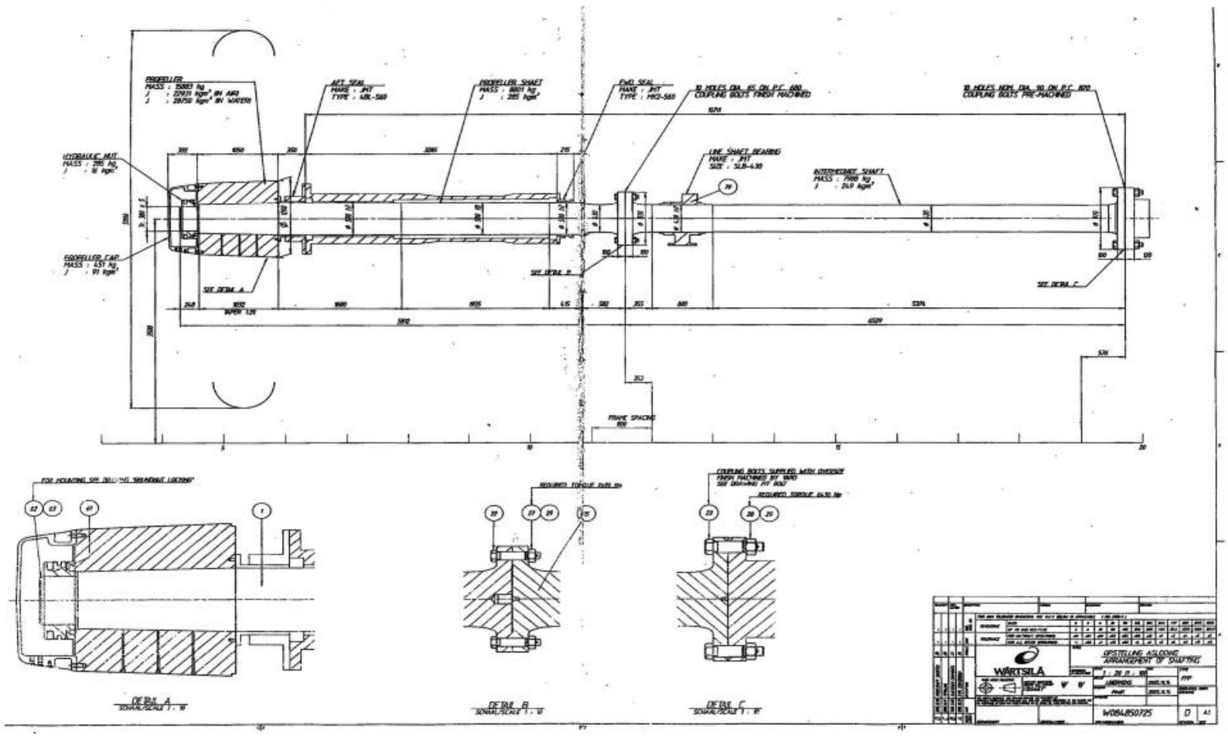
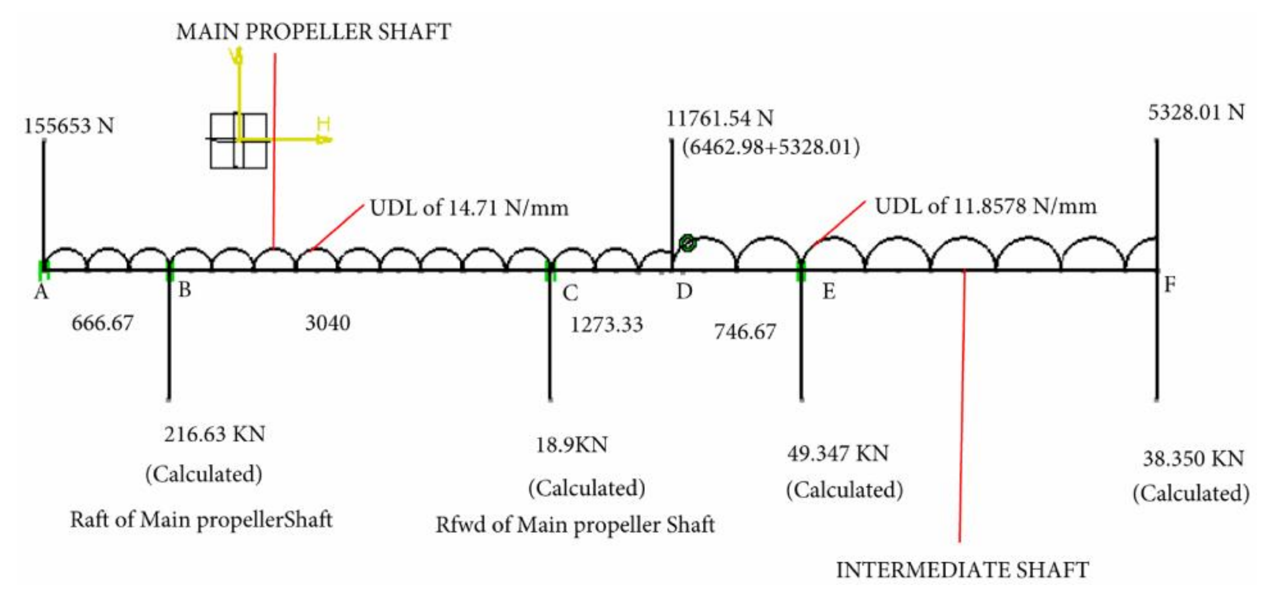


Figure 7: Loading Diagram of Propeller and Intermediate Shafts (From Propeller End to the Thrust Bearing of Main Engine)




From (1) and (2)

$R_{aft}$ = 216.63 kN

$R_{fwd}$ =18.9 kN

For intermediate shaft:

5328.510 + 5328.01 + (11.8 x 6529) = $R1$ + $R2$ ...(1)

$$0\int^{5782.33} 11.8578\,\aleph dx + (5328.01 \times 5782.33)$$

$$-R2(5782.33) - (5328.015 \times 746.67)$$

$$-0\int^{746.67} 11.8578\,\aleph dx = 0 \qquad ...(2)$$

From (1) and (2)

$R1$ = 49.347 kN

$R2$ = 38.350 kN

**Section AB**

$$-14.71 \times \aleph^2 / 2 - 155653\aleph = Bm$$

$@A\,\aleph = 0 = Bm = 0$

$B\,\aleph = 666.67 = Bm =$ –10, 70, 38, 107.1 N-mm

$Bm@B$ = –107.038 x 106 N-mm

**Section BC**

Eq: $-155653\,\aleph + 1\ 3972.\ 72(\aleph - 666.67)$

$$-\frac{14.71 \times \aleph^2}{2}$$

$X@C$ = 3706.67 mm

$Bm@C$ = –19.388 x $10^6$ N-mm

**Section CD**

Bm Eq: $-155653\,\aleph + 216639\,(\aleph - 666.67)$

$$-\frac{14.71 \times \aleph^2}{2} + 18972(1273.33)$$

$\aleph\,@\,D$ = 4980 mm

$Bm@D$ = 1.03423 x $10^6$ N-mm

**Section DE**

Bm Eq:

$$\frac{-155653\aleph + 216639(5060) - 14.71(4980)^2}{2}$$

$$\frac{-11.8578(746.67)^2 + 18972(2020) - 6462.93(746.67)}{2}$$

$x@E$ = 5726.67 mm

$Bm@E$ = 52.605 x $10^6$ N-mm

**Section EF**

Bm Eq:

$-155653\aleph + 18972.72(\aleph - 666.67) - 14.71\aleph^2/2 - 11761.54(\aleph - 4980) + 216399(\aleph - 3706.67) + 11.85.418(x - 4980)^2 + 49347.37 - 5726.67)$

$X@F = 11509$ mm

$Bm@F = 402.229 \times 10^6$ N-mm

### Stress Induced in the Intermediate Shaft

Maximum bending moment in intermediate shaft

$M$ Max = $424.30 \times 10^6$ N-mm

Torque to be transmitted = $803.473 \times 10^6$ N-mm

Value of thrust force acting on the intermediate shaft

$$F^T = \frac{326 \times H \times PC}{V(1-t)} \text{ in lb}$$

$H$ = Horse power of the Engine = 9480 kw = 12712.88 hp

$V$ = ship speed in knots = 14.3 knots

$T$ = 0.2 for single screw ships

$PC$ = 0.73 FOR " " "

$$F_t = \frac{326\ v12712.88 \times 0.73}{14.3 \times (1-0.2)} = 264459.615\ N$$

Taking 10% over load

$F_t$ = 290.905 kN

$$T_e = \sqrt{\left(k_m M \times \sim \frac{(fd)}{8}\right)^2 + (k_t T)^2} = \frac{\Pi}{16}{}^2 d^3$$

$$\sim = \frac{1}{1 - 0.0044(L/k)} \text{for } L/K < 115$$

$K$ = Radius of gyration,

$$\sim = \frac{1}{1 - 0.0044(6529/100)} = 0.1533$$

$$T_e = \sqrt{\left(k_m M \times \sim \frac{(fd)}{8}\right)^2 + (k_t T)^2} = \frac{\Pi}{16}{}^2 d^3$$

$$\sqrt{(1.5 \times 424.30 \times 10^6) + 0.1533\ (290.905 \times 10^3 \times 400/8) + (1 \times 803.437 \times 10^6)^2}$$

$$= \frac{\Pi \times {}^2 \times (420)^3}{16}$$

${}^2{}_{induced}$ = 81.677 N/mm² < 85.01 N/mm²

$M_e$ = ½ [km.m + ~($fd$/8) +

$$\sqrt{\left[k_m M + \sim (fd/8)^2 + (k_t x T)^2\right]}$$

$$= \frac{1/2\left[\left(1.5 \times 424.30 \times 10^6 + 0.1533(290.905 \times 10^3 \times 400) + 1026392095\right)\right]}{8}$$

$\pm b$ = 132.502 N/mm < 150 N/mm² (Design is safe)

### Stress Induced in the Propeller Shaft

The maximum bending moment is propeller shaft occur at the oft bearing

$M$ max = $1.0 \times 10^6$ N-mm

Torque to be transmitted $T = 803.473 \times 10^6$ N-mm

The value of thrust force acting on the propeller shaft is give by

$$F^T = \frac{3261 \times H \times PC}{V(1-t)}$$

$H$ = Horse power of Engine = 9480 kw

= 12712.88 hp

$V$ = ship speed in knots = 14.3 knots

$T$ = 0.2 for single screw ships

$PC$ = 0.73 for single screw ships

$$F_T = \frac{326 \times 12712.88 \times 0.73}{14.3 \times (1-0.2)}$$

= 264459.165 N

= 264.459 kN

Taking 10% over load $F_T = 290.905$

For shafts subjected to axial load in addition to combined torsion and bending loads we have

$$T_e = \sqrt{\left(k_m M \times \sim \frac{(fd)}{8}\right)^2 + (k_t T)^2} = \frac{\Pi}{16}{}^2 d^3$$

$$\sim = \frac{1}{1-0.0044} for\ (1/k) < 115$$

$t$ = Radius of gyration

$$\sqrt{\frac{I}{A}} = \frac{d}{4} = \frac{500}{4} = 125\ mm$$

$$\sim = \frac{1}{1-0.0044(4980/125)1.212}$$

With $km = 1.5$ $kt = 1.0$, $d = 500$ mm

$$T_e = \sqrt{\frac{\left[1.5\times1.0\times10^6 + 1.212(290.905\times10^3\times500)\right]^2 + \left(1\times8030065\times10^6\right)^2}{8}}$$

$T_e = 803817645.7$

² = 32.750 N/mm² < 63.75 N/mm² (safe)

$$M_e = 1/2[km.m + \sim(fd/8)]$$

$$+\sqrt{\left[k_m M + \sim(fd/8)^2 + (k_t x T)^2\right]}$$

= 1/2[1.5 x 1.0 x $10^6$ + 1.212(290.905 x $10^3$ x 500/8) + 803817645.7]

±$b$ = 33.709 N/mm² < 150 N/mm² (design is safe)

# DESIGN OF BEARINGS

## Design of the Intermediate Bearing

Bearing load $W_a$ = 645 Kg

= 632 IN

Diameter of journal (D) = 400 mm

Length of bearing (L) = 2D = 800 mm

Pressure on bearing = Load/Projected Area

= 6321/400 x 800

= 0.019 N/mm²

Lubricating oil type is SAE – 30

Taking operating temp. Of the bearing as 50 °C, the abs. Viscority of oil is 0.048 Kg/m.s

Speed of journal is $N$ = 124 rpm

Peripheral speed of journal

$$V = ®DN/60 = \frac{® \times 0.400 \times 124}{60}$$

= 2.59 m/sec

**c/d = 0.001**

**d/c = 10³**

The bearing characteristic number,

$$\frac{ZN}{P} = \frac{0.048\times124}{0.019} = 302.13$$

Critical Pressure $Pcr$ =

$$\frac{ZN}{4.75\times10^6}(d/c)^2\ (1/d+1)\ N/mm^2$$

$$= \frac{0.048\times124}{4.75\times10^6}\left(10^3\right)^2(800/400+800) = 0.835\ N/mm^2$$

= 0.835 N/mm² > Pressure acting (0.019)

Therefore the lubricating oil film is stable.

Coefficient of friction, « = (33/$10^8$) ($ZN/P$) ($d/c$) + $K$

$$= \frac{\left(33/10^8\right)(0.48\times124)\left(10^3\right)+0.02}{0.019}$$

« = 0.101

Heat generated $Qd = C.A\ (tb - ta)$

= $C(DL)\ (tb - ta)$

Taking heat transfer Cofficient ($C$) as 550 w/m² $C$ we get,

$Qd$ = 550 x 0.4 x 0.8 x 1/2(50 – 35)

= 1320 w

Since the heat dissipated is less than heat generated artificial cooling of the bearing is necessary

Amount of artificial cooling necessary

= $Qg - Qd$

= 1653 – 510 – 1320

= 333.510 w

Heat to carried away by oil, $Q0$ = 333.510 w

= $mcS(St)$

$mc$ – mass flow rate of cooling oil (Kg/s)

$S$ – Specific heat of Cooling oil = 1840 J/Kg $K$ and

$St$ is temp change in cooling oil = 50-35 = 15 °C

Therefore $mc = \dfrac{Q0}{S.St}$

$$= \frac{333.510}{1840 \times 15} = 0.012\ Kg/\sec$$

= 0.725 Kg/min

## DESIGN AS PER LYOYD'S REGISTER OF SHIPPING RULES

### Diameter of the Propeller Shaft

The minimum diameter of the propeller shaft is given by

$$dp = 100K3\sqrt{\frac{P}{R}\left(\frac{560}{\pm 4 + 160}\right)}\ mm$$

where

$R$ = Rated speed = 124 RPM

$P$ = Rated power = 9480 KW

$K$ = A factor for the shaft design features

= 1.22 for keyless propeller.

$\pm 4$ = Tensile strength = 600 N/mm$^2$

$$d = 100 \times 1.223\sqrt{\frac{9480}{124}\left(\frac{560}{600 + 160}\right)}$$

$$= 100 \times 1.223\sqrt{76.451 \times 0.7368}$$

= 100 x 1.22 x 3.83

= 467.66 mm

### Diameter of the Intermediate Shaft

The minimum diameter of the intermediate shaft is given by

$$d_j = F \times k3\sqrt{\frac{P}{R}\left(\frac{560}{\pm u + 160}\right)}\ mm$$

$K$ = 1 for shaft with integral coupling flanges coupling with 3, 7 on shrink fit coupling

$F$ = 100 for other oil engine installations.

$$d_j = 100 \times 1 \times 3F \times k3\sqrt{\frac{9840}{124}\left(\frac{560}{600 + 160}\right)}$$

$$= 100 \times 1 \times 3\sqrt{76.451 \times 0.7368}$$

= 100 x 1 x 3.833

= 383.336

### Flange Coupling

### Thickness of the Flange Coupling

- Thickness of propeller, shaft flange

  $Tp \geq 0.2xd_j$ = 0.2 x 467.67

  = 93.53 mm

- Thickness of intermediate shaft flange

  $T_i \geq 0.2 \times d_i$ = 76.6672 mm

### Diameter of the Coupling Bolt

The diameter of the coupling bolts is not less than

- Propeller shaft and intermediate shaft

$$d_{bp} = \sqrt{24010^6 P/nd \pm uR}\ mm$$

where

*D* = designed pitch circle diameter of the flinger = 800

*N* = number of bolts in the coupling = 12

- Intermediate shaft and thrust shaft

where *n* = 12

*D* = 640

$$db_j = \sqrt{240 \times 10^6 9480/12 \times 800 \times 124 \times 660}$$

= 63.1 mm

### Intermediate Bearing

$L = 2d_i = 2 \times 400 = 800$ mm

## CONCLUSION

The purpose of this paper is to provide detailed information about the various aspects related to the application of shaft generators.

According to the society rules. We get a diameter which is less than that of the designer value. Which is very suitable for the alignment and gives a better performance than that of the designer value so we can use the shaft.

By using these dimensions we can approach some useful values, those are:

- We can reduce the cost of the entire system.
- The total weight of the system is comparatively reduced.
- Lubrication also become easy.
- Speed of the engine is rapidly increases.
- Maintenance cost decreases.

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