BUILDING THE PROGRAM OF CALCULATION FOR THE THICKNESS AND REASONABLE LANDSCAPE APPLYING THE SHAFT LINERS

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ABSTRACT

This paper presents the theoretical basis of the process of building programming to calculate the reasonable thickness and thickness of the shaft liners. Through this calculation program, it is possible to determine and select the shaft liners for the shaft system. The content can be calculated as thickness; shaft torque and reasonable residual. The program results were calculated for the vessel with the load of 950 tons. This result completely complies with the design requirements of the shaft liners.

KEYWORDS

shaft liners, reasonable landscape, thickness, calculation.

1. INTRODUCTION

The shaft lubricated with natural water, the pinwheel shaft is the detail that works directly with foreign vessels, this is an environment prone to corrosive axes, especially with alloy steel which will be electrochemical corrosion [1].

At the same time, the shaft liner support is usually made of soft materials such as barbed wire, rubber, and textile that work well with copper material under natural lubrication with natural water [2]. Therefore, people need to have shaft protection measures. At the axial position working with the bearing shaft having a large load of friction, it is necessary to protect the shaft and create a good pair of friction between the neck and the shaft, called shaft liners [4][5].

At frictionless shaft parts protected by shaft liners with a smaller thickness or covered with copper tubes or protective layers made of materials such as epoxy resin, polysilipolam in, glass fiber cloth protects the shaft from loss. worn by the effects of seawater and convenient for repair. These shells must ensure that seawater does not enter the shaft, especially in transitions (the end of the coat and the wrap).[9][10]

2. THEORETICAL BASIS

2.1 Torque and stress due to twisting

If the profile is hard and there is no relative displacement: [11]

\[
\frac{d\phi}{d\zeta} = \frac{M_{x(z)}}{\sum GJ_{\rho}} = \text{const}
\]

Inside:

\[
M_{\text{total}} = \sum GJ_{\rho} M_{x(z)} = [GJ]_{\text{true}}
\]

\[
M_{\text{total}} = [GJ]_{\text{true}}
\]

\[
M_{\text{total}} = [GJ]_{\text{true}}
\]

\[
\frac{\pi d_{1}^{4}}{32} ; \quad \frac{\pi (d_{2}^{4} - d_{1}^{4})}{32}
\]

\[
J_{\text{pol}} = \frac{\pi d_{2}^{4}}{32}.
\]

\[
J_{\text{pol}} = \frac{\pi (d_{2}^{4} - d_{1}^{4})}{32}
\]

Therefore, the torque component of the liners is

\[
M_{\text{liners}} = \left( \frac{GJ_{\rho}}{GJ_{\rho}} \right)_{\text{true}} [GJ]_{\text{true}}
\]

\[
M_{\text{liners}} = \left( \frac{GJ_{\rho}}{GJ_{\rho}} \right)_{\text{true}} [GJ]_{\text{true}}
\]

Stress due to twisting:

The torque that causes the stress on the cross-section is determined by the formula.

\[
\tau_{z} = \frac{M_{\text{liners}}}{J_{\text{pol}}} \rho.
\]

In which: \( M_{\text{liners}} \) is the torque on the shaft liners. \( W_{30} \) is a torsion torque at the mounting surface.
\[ J_{s,i} = \frac{\pi}{32} (d_a^4 - d_i^4) \]  
(5)

\[ \rho \text{ is the radius of stress point. } \rho = d_i/2 \]

\[ \tau_x = \frac{16 M_{ao} a \rho d_i}{\pi (d_a^4 - d_i^4)} \]  
(6)

2.2 Bending moment and bending stress

a) \[ J_{s,oo} = \frac{\pi}{64} (d_a^4 - d_i^4) \]

b) \[ J_{s,oo} = \frac{\pi}{64} (d_a^4 - d_i^4) \]

At the assembly section \([13]\), we have

\[ \frac{M_{y(x)}}{\sum EJ_x} = \frac{1}{\rho} = \frac{M_{y(x)}}{EJ_x} \]  
(7)

Inside:

\[ M_{y(x)} = \text{the bending moment on the surface.} \]
\[ M_{ao} = \text{a mound tissue on the shaft.} \]
\[ E \text{ is elastic modulus.} \]
\[ J_x \text{ is the moment of inertia of the cross-section.} \]

\[ J_{s,oo} = \frac{\pi}{64} (d_a^4 - d_i^4) \]

So, the moment of bending on the shaft.

\[ M_{s,oo} = \frac{\sum EJ_x}{\sum EJ_x} M_{y(x)} \]  
(8)

*Bending stress:

The bending moment causes the stress on the cross-section, determined by the formula:

\[ \sigma_x = \frac{M_{y(x)}}{J_x} y \]  
(9)

Inside:

\[ M_{s,oo} = \text{the bending moment on the shaft liners.} \]
\[ J_x \text{ is the moment of inertia of the section.} \]

\[ J_{s,oo} = \frac{\pi}{64} (d_a^4 - d_i^4) \]

y is the magnitude of the point to calculate stress.

\[ y = d_i/2. \rightarrow \sigma_x = \frac{32 M_{s,oo} \sigma d_i}{\pi (d_a^4 - d_i^4)} \]  
(10)

2.3 Cutting force at section

Cutting force at section is following \([14][15]\):

\[ \frac{dM_u}{dz} = F_Q(0) \]

\[ Q_{Q(.)} = \sum EJ_x \text{. Q}_{z-} \]  
(11)

2.4 Redundancy ensures anti-slip

2.4.1 Anti-slip due to twisting

Considering from the free end of the shaft liners it is clear that the component \( \frac{dM_{u}}{dz} \) is due to the torque of the friction force on the assembly surface \( \tau_{ms} \) generated on the \( dz \) segment \([17]\).

\[ \frac{dM_u}{dz} = m_{\tau_{ms}} = 2 \pi r_{\tau_{ms}}. \tau_{ms} \]  
(12)

Inside:

\( P \) is the pressure on the assembled surface
\( f \) is the coefficient of friction.
\( z \) is the torque calculation.

When \( z = 0 \), then there is no more slipping. At this time the torque component due to frictional torque is zero to ensure the condition

\[ \frac{d \phi}{dz} = \frac{M_{s,oo}}{GJ_{y(x)}} \]  
(13)

Ta có:

\[ M_{s,oo} = \sum GJ_{y(x)} M_s \]

Tir công thức (2.12) → 2.πr_{\tau_{ms}}. f. L_o = \left( \frac{GJ_{y(x)}}{2 \pi r_{\tau_{ms}}} \right) M_s

So, the condition of overlapping slip is:

\[ P \geq \left( \frac{GJ_{y(x)}}{2 \pi r_{\tau_{ms}}} \right) M_s \]  
(14)
We can see that the required pressure or the required density is directly proportional to $M$, and the ratio $\frac{G_1}{\sum G_1}$ inversely proportional to $r_n$.

Through the ideas on the $z_0$ segment, it should not be in the work zone of the liners, so it should be $z_4$ in the chamfered section or with the convenience of stress-reducing trenches (when beveled or with the hard-wearing stress-reducing slit, it should be reduced). The friction generated by sliding is not larger and the stress on this part is not large.

\[ \sum \rho GJ d_2 r d \rho GJ d_1 r = u_0 (a) \]

**Figure 4:** The friction torque on the shaft liners[18]

The friction torque on the shaft liners

*Note: here $M$, considers it the same as the torque generated by the motor.

2.4.2 Anti-slip due to bending

If you ignore the weight of the liners itself and the pressure (not due to the thickness) because the shaft on the outer liners is only due to friction on the shaft liners[19].

\[ \rho d \rho \]

**Figure 5:** Balance torque on dz segment [20]

Consider balance on dz

\[ M_{(z)} + dM_{tot} = \frac{dM_z}{dz} + M_{(z)} \]

\[ dM_{tot} = \frac{dM_z}{dz} \]

\[ Q_{moz} dz = M_{moz} = \int_0^z \tau_{muz} r d \phi r d \rho \sin \phi |_{2} = 4. \tau_{muz} r^2 dz \]

\[ Q_{moy} = 4. \tau_{muy} r^2 \]

\[ \tau_{muz} = Q_{moz} / 4 r^2 \]

But according to the pressure generated during the installation process there is a redundancy

\[ \tau_{muz} = P. f \]

Thus, to ensure the anti-slip condition when bending, the contact stress due to close contact of pressure P must ensure:

\[ P.f > Q_{moz} / 4. \tau_{muz} r^2 \]

\[ P > Q_{moz} / 4. \tau_{muz} r^2 \]

2.5 Stress arises due to thickness

In a joint with pressure due to pressure on the assembly surface, the legal stresses appear in the center ($\sigma_1$) and the stress in the next direction ($\sigma_2$). [19]

\[ \sigma_1 = \frac{r_1^2}{r_2^2 - r_1^2} \left( 1 - \frac{r_2^2}{r_1^2} \right) P \]

**Figure 6:** Tensile stress due to the thickness on the liners[20]

*Tensile stress in the direction of the center determines the following formula [20]:

\[ \sigma_1 = \frac{r_1^2}{r_2^2 - r_1^2} \left( 1 - \frac{r_2^2}{r_1^2} \right) P \]

*Tensile stress in the tangent direction determined by the formula [13]:

\[ \sigma_2 = \frac{r_1^2}{r_2^2 - r_1^2} \left( 1 + \frac{r_2^2}{r_1^2} \right) P \]

with $r = r_1 = d_1/2$ $r_2 = d_2/2$  $\sigma_1 = -P$

\[ \sigma_2 = P \frac{d_1^2 + d_2^2}{d_2^2 - d_1^2} \]

3. DEVELOP A BLOCK DIAGRAM TO CALCULATE THE REASONABLE RESIDUALS OF PINWHEELS

3.1 Sequence and basis for calculating sections in the block diagram

3.1.1 Determination of maximum residue according to heating and assembly conditions

The coupling strength is the main factor affecting the pressure and stress generated on the shaft liners. If the excess is too small, the pressure and friction coefficient on the assembly surface is too small. This makes the joint not guarantee anti-slip conditions and affects the quality of the joint [21]. However, if the thickness is too large, the stress due to the assembly can cause stress on the larger parts to be allowed (greater than the yield limit causing the material to be deformed, pressure and ghosting close down reduces the ability of the splice load to decrease) or cannot be assembled [22].
The characteristics of the shaft liners are not responsible for the transmission of torque but the stress on the liners is mainly due to the deformed shaft causing deformed liners. So with shaft liners:

The minimum thickness must ensure no slip and limited by manufacturing errors. Maximum permeability is limited by durable conditions, assembly conditions and economy. First of all, we determine the maximum assembly thickness based on the heating conditions and assembly according to the formula:

$$\delta_{thad} = \Delta t \cdot \alpha \cdot d - \delta_0 \text{ (mm)}$$

(23)

$\Delta t$ is the maximum temperature difference of the shaft liners material, with this temperature difference the firing temperature does not cause a detailed deformation.

$\alpha$ is the coefficient of expansion due to heat (mm / mm0c).

d is the nominal diameter of the joint (by shaft diameter) (mm).

$\delta_0$ is the gap required for easy mounting, usually taken by the smallest gap of loose coupling H7 / g6, (mm).

3.1.2 Pressure calculation ensures anti-slip conditions due to twisting and bending

Pressure calculation ensures anti-slip conditions due to twisting:

a) From the capacity that the rotation of the shaft calculates the torque generated on the liners according to the formula (23).

b) The shaft length is usually taken about 4 times the shaft diameter.

c) Moment of inertia is calculated according to the size of the current loop.

d) Replacing the above values into the formula (14) we have pressure to ensure anti-slip conditions due to twisting.

Pressure calculation ensures bending conditions due to bending:

a) Calculating the cutting force at the position of the pillow No. 1, where there is a piece of paint, so the entire weight of the pinwheel with the weight of the shaft itself acting on the pillow creates the largest cutting force

$$Q_{lins_{max}} = Q_{shaft} \cdot q_{lo}$$

3.1.3 Calculation of actual pressure tests arising

a) Determine pressure according to the thickness and residual based on formula (22).

b) Compare the allowable pressure from the anti-slip condition due to twisting and bending due to bending to choose the pressure to satisfy both conditions [13].

c) Compare actual pressure generated with allowable pressure to check slip resistance condition.

d) If the pressure satisfies the anti-slip condition, then proceed to the next test.

e) If the anti-slip condition is not satisfied, the thickness of this loop is not satisfied, then we let the program run again to increase the thickness of the liners. After that, the calculation process will continue until achieving the anti-slip condition, then finish this step [17].

3.1.4 Calculation of stress tests arising under durable conditions

a) Calculating stress generated by bending moment of liners according to formula (10).

b) Calculating stress generated by the torque of the liners according to formula (6).

c) Calculating the French stress by centrifugal direction arising due to the redundancy according to the formula (17).

d) Calculating the stress of the tangential method arising due to the redundancy according to the formula (17).

e) Calculate the equivalent stress according to durable theory 4

$$\sigma_{eq} = \sqrt{\sigma_1^2 + \sigma_2^2 + \sigma_3^2 - \sigma_1 \sigma_2 - \sigma_2 \sigma_3 - \sigma_1 \sigma_3 + 3 \sigma_4 \sigma_5} \leq \sigma_{thad}$$

(24)

Inside

$$\sigma_1 = \frac{\sigma_y + \sigma_y + \sigma_y}{2} - \frac{1}{2}\sqrt{(\sigma_y + \sigma_y - \sigma_y)^2 + 4 \sigma_y^2}$$

(25)

$$\sigma_2 = 0$$

$$\sigma_3 = \frac{\sigma_y + \sigma_y + \sigma_y}{2} + \frac{1}{2}\sqrt{(\sigma_y + \sigma_y - \sigma_y)^2 + 4 \sigma_y^2}$$

(26)

Comparison of equivalent stress according to the theory of durability 4 with stress limit stress. If satisfied, the thickness and thickness of this loop have ensured both the anti-slip condition and the strength of the liners. However, this value is only the smallest possible thickness, in the process of exploiting the thickness, it will be smaller and may lead to dissatisfaction. Therefore, the need for liners thickness continues to increase. The increasing will now be equal to the amount of wear and tear due to both the operation and repair of the ship later.

After increasing the thickness to such a number of mutations, it may lead to a pressure that will no longer satisfy the required strength at the required thickness and thickness. The thickness of the coat must still ensure the residue of wear and repair, then the program will automatically return to the initial selection to reduce the residual to a certain amount and then continue to calculate again.

The program will automatically recalculate the loops until the thickness and thickness are calculated to satisfy both durable conditions, anti-slip conditions, wear and tear, then stop and notify the results to the screen picture.

3.1.5 Select the appropriate thickness and required thickness

In the event that the runners need to check the proper thickness and thickness of the available shaft and shaft set, after running the part on the program, it will be required for the ship parameters and available available. The process will calculate the reasonable thickness of the shaft to check the validity of the current coupling.

3.2 Block diagram of the program

![Figure 7: Block diagram of the calculation program](Cargo ship HT-03 load 950 tons)
Some parameters of the ship:
Main machine: 8NVD36-1U
Rated power: Ne = 408 cv
Rated rotation: n = 500 rpm
Shaft diameter: d1 = 116 mm
Length of axes: L0 = 700 cm.
L0 = 850 cm.
L1 = 315 cm.
L2 = 65 cm.

Shaft material is steel 35 with:
Permissible yield limit: $[\sigma_{ch}] = 32$ KN / cm2
Compression compression module $E = 2.105$ KG / cm2
Breakdown coefficient $\mu = 0.25$
Specific weight: $\gamma = 7.86$.
Fast weight: GB 214 kg
Material, CuSnZn 10-2, has:
- Tensile strength $\sigma_0 = 20-25$ KG / mm2.
- Flow limit $\sigma_f = 18$ KG / mm2.
- Stretching relatively long $\delta = 2-10 \%$
- Hardness HB = 80 - 90.
- Impact strength $a = 1-1.5$ KG / cm2
- Chemical composition Sn = 9 - 11%, Zn = 2 - 4%, the rest is Cu.

### 3.3 Calculation results by empirical formula

The thickness of the shaft liners at the working position with the bearing is determined by the following formula:

$$t_1 = 0.03d_\sigma + 7.5 \text{ (mm)}$$

Where: $t_1$ is the minimum thickness of the shaft liners (mm)
$d_\sigma$ is the pinwheel diameter (mm)
Using the formula above we can calculate $t_1 = 1.098$ mm.

Shaft liners thickness at sections does not work with bearings determined by the following formula:

$$t_2 = 0.75t_{11} \text{ (mm)}$$

Using the formula above we calculate $t_2 = 8.235$ (mm).

The thickness of the liners must be replaced when used is:
$$t = 0.02d_\sigma + 5 \text{ (mm)}$$

Using the formula above we calculate $t = 7.32$ cm.

The elongation of the pair of shaft and pinwheel coatings is:

$$\delta = 0.001d_\sigma \text{ (mm)}$$

Using the formula above we calculate $\delta = 0.116$ (mm).

### 3.4 Application of the program of calculating math programming language in Turbo-pascal programming language to calculate the axis system of ships on:

a) To calculate a specific ship, the program steps run in the following sequence:

b) Enter the input data as required, such as capacity, shaft diameter, rotation of the shaft etc.

c) Calculate maximum residue according to heating conditions

d) For increasing thickness from the smallest thickness

e) Calculate pressure to ensure anti-slip conditions for coat

f) Calculate the actual pressure generated with the thickness and upper thickness of the liners.

g) If the actual pressure arises less than the pressure to ensure the anti-slip condition, return to the upper part for the thickness to increase and then calculate it until the slip condition is met, then the loop sequence for pressure is finished.

![Figure 8: Load diagram of the ship system HT-03 with 950 ton load](image)

In the above process and at the same time for the program to calculate stresses arising in the liners, the end of the above loop sequence will check whether the stress generated has exceeded the permissible stress of the liners material. If stresses do not guarantee stable conditions, the program will automatically reduce the drop and run the pressure loop to satisfy.

![Figure 10: Graph the relationship between pressure and thickness](image)

After the pressure and stress are satisfied, the program will announce the calculation result and ask the question whether the runners want to change the assembly thickness so that the thickness can be calculated as desired. If you want to change, the person running the program can enter the value of the entry value and the program will give a reasonable value of the corresponding liners size. Calculation results will be displayed on the screen by the program and the program will automatically save this result to a file in text format so that users can easily print and use. Therefore, a reasonable tonnage of 950 ton HT-03 is: a) Shaft liners thickness at working position with bearing: = 10.2 mm b) Shaft liners thickness at sections not working with 7.7 mm bearing.

c) The required coat thickness must be replaced when used: 6.8 mm.

d) The elongation of the pair of shielding liners and pinwheel pair: 0.11 (mm)

### 4. CONCLUSIONS

After completing the article, the author draws some conclusions as follows: The article contributes to perfecting the design theory and calculating the fast-pivot coat of ship shaft system. Give block diagram to be able to program in many different programming languages, set up the program to calculate the thickness and reasonable redundancy of the pinwheel liners. Calculation results by experience and calculation according to the calculation program in the topic also have errors due to the program, taking into account the difference of load diagram of the axis system, main engine capacity and characteristics type of material for shaft.
and shaft. The results obtained can be applied to test when designing the propeller, testing any propeller. The method proposed in this thesis can be a good reference for professionals when designing, repairing, testing ship propellers in particular and calculating and testing the marine propulsion system in general.

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