Assessment of destabilizing factor for automatic control systems in propulsion systems of mechatronic and maritime transport objects

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Abstract. It is known that many of today's ships and vessels have a shaft generator as a part of their power plants. Modern automatic control systems used in the world's fleet do not enable their shaft generators to operate in parallel with the main diesel generators for long-term sustenance of the total load of the ship network. On the other hand, according to our calculations and experiments, a shaft generator operated in parallel with the main power plant helps save at least 10% of fuel while making the power system of the ship more efficient, reliable, and eco-friendly.
The fouling and corrosion of the propeller as well as the weather conditions of navigation affect its modulus of resistance. It changes the free component of the transient process of shaft generator stress frequency changes in transient processes. While the shaft generator and the diesel generator of the ship power plant are paralleled, there emerges an angle between their EMF. This results in equalizing currents generated between them. The altering torque in the drive–shaft line—propeller system causes torsional fluctuations of the ship shaft line. To compensate for the effect of destabilizing factors and torsional fluctuations of the shaft line on the dynamic characteristics of the transient process that alters the RPM of the main engine, sliding mode controls can be used. To synthesize such a control, one has to evaluate the effect of destabilizing factors.

1. Introduction

When the dynamic characteristics of the ship propeller, the generator, the main engine (ME), or the rotation frequency regulator wear are changed, the free component of the main shaft rotational speed transient process is changed as well. If the ship is equipped with shaft generators (SG), such changes result in the fluctuations of the voltage they generate [1-2].

The fouling and corrosion of the propeller as well as the weather conditions of navigation affect its modulus of resistance. It changes the free component of the transient process of shaft generator stress frequency changes in transient processes. While the shaft generator and the diesel generator of the ship power plant are paralleled, there emerges an angle between their EMF. This results in equalizing currents generated between them [3].

To compensate for the effect of destabilizing factors and torsional fluctuations of the shaft line on the dynamic characteristics of the transient process that alters the RPM of the main engine, sliding

mode controls can be used. To synthesize such control, one has to evaluate the effect of destabilizing factors [2].

2. Torsional fluctuations of the propulsive system shaft line

The altering torque in the drive–shaft line—propeller system (Figure 1) causes torsional fluctuations of the ship shaft line [4]. This is due to the uneven generation of mechanical energy in the piston engine and the uneven field of rates at which the water flows onto the propeller. These fluctuations peak at resonant RPM numbers when the variable torsional stresses are maxed.
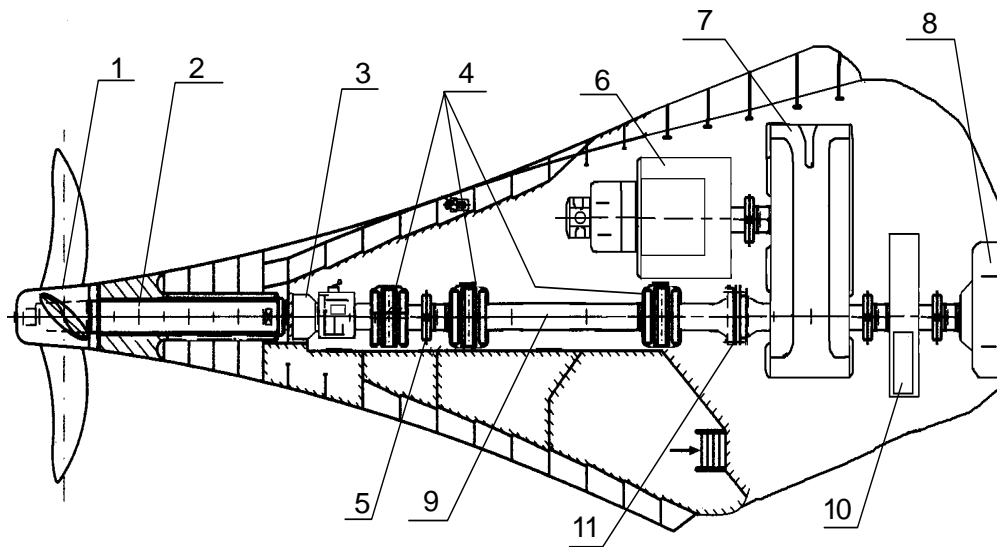

Figure 1. Main engine, propeller and shaft line diagram: 1 – variable pitchpropeller (VPP); 2 – shaft line; 3 – propeller pitch shifting mechanism(PPSM); 4 – sliding plain bearings; 5 – flange connection; 6 – shaft generator; 7 – reducer; 8 – main engine (ME); 9 – intermediate shaft; 10 – flywheel; 11 – elastic coupling.

Let us calculate the stiffness of the shaft line in torsion for Project 503 seiner trawler ship that fishes in the Black Sea; for that let us use the calculation diagram in Figure 2.
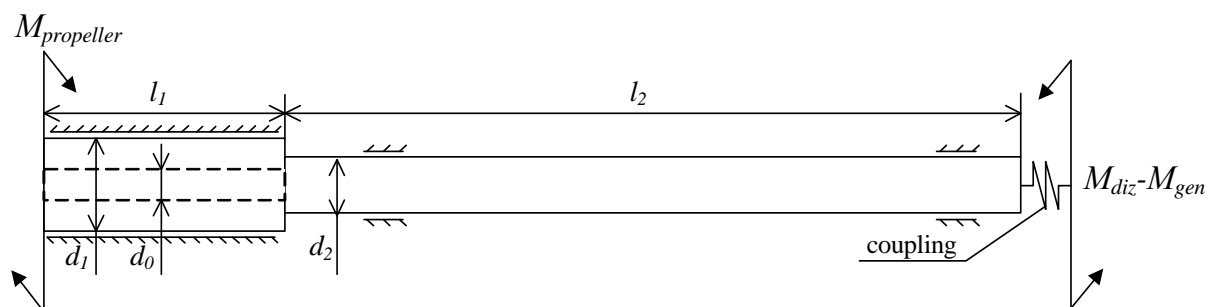

Figure 2. Shaft line calculation diagram

Figure 2 contains the dimensions for such calculations: $l_1=2\ m$, $d_1=0.183\ m$, the axial hole diameter $d_0=0.040\ m$, the propulsion shaft; $l_2=6\ m$, $d_2=0.154\ m$, the intermediate shaft. To the right, the

difference in the torques of the main engine $M_{диз}$ and the generator $M_{ген}$ is applied to the shaft. To the left, there is the propeller torque $M_{винт}$.

Polar moments of inertia [2] is:

$$J_{p1} = \frac{\pi}{32}\left(d_1^4 - d_0^4\right) = \frac{3.14}{32}\left(18.3^4 - 4^4\right) = 10980\left[cm^4\right],$$

$$J_{p2} = \frac{\pi}{32}\left(d_2^4\right) = \frac{3.14}{32}15.4^4 = 5519\left[cm^4\right].$$

Material shear modulus is: $G = 8 \cdot 10^5 \, kg/cm^2$. Shaft flexibility modulus is:

$$\begin{cases} e_1 = \dfrac{l_1 \cdot 10}{G \cdot J_{p1} \cdot 9.81} = \dfrac{200 \cdot 10}{8 \cdot 10^5 \cdot 10980 \cdot 9.81} = 2.32 \cdot 10^{-8}\left[\dfrac{rad}{Nm}\right], \\[4mm] e_2 = \dfrac{l_2 \cdot 10}{G \cdot J_{p2} \cdot 9.81} = \dfrac{600 \cdot 10}{8 \cdot 10^5 \cdot 5519 \cdot 9.81} = 13.8 \cdot 10^{-8}\left[\dfrac{rad}{Nm}\right]. \end{cases}$$

The gearbox shafts are considerably shorter than the intermediate shaft or the propulsion shaft, which is why the gearbox flexibility is ignorable [3-4].

The coupling in Figure 2 is in general a non-linear component in the torsional system [1, 5-6]. This nonlinearity is due to deformation constraints that distort the proportionality of the torque and the shaft torsion angle. The following formula can be used to find the coupling flexibility coefficient for low fluctuations:

$$e_3 = \frac{12 \cdot l}{Embh^3} = \frac{12 \cdot 10 \cdot 10}{2 \cdot 10^5 \cdot 8 \cdot 0,5 \cdot 1,5^3 \cdot 9,81} = 4530 \cdot 10^{-8}\left[\frac{rad}{Nm}\right],$$

where $E = 2 \cdot 10^5 \, kg/cm^2$ is the spring material elasticity modulus; $l = 10 \, cm$ is the length of each spring in the coupling; $m = 8$ is the number of springs in the coupling; $h = 1.5 \, cm$; $b = 0.5 \, cm$ are the width and the thickness of the spring section.

Shaft stiffness factor in torsion [5, 7, 8] is:

$$C = \frac{1}{e_1 + e_2 + e_3} = \frac{1}{2.32 \cdot 10^{-8} + 13.8 \cdot 10^{-8} + 4530 \cdot 10^{-8}} = 2.2 \cdot 10^{-4} \cdot 10^8\left[\frac{rad}{Nm}\right].$$

Based on Figure 1, the shaft line twisting moment is:

$$M_{sk} = C\varphi = C\left(-M_{propeller} \cdot e_1 + \left(M_{diz} - M_{gen} - M_{propeller}\right) \cdot \left(e_2 + e_3\right)\right) =$$

$$= 2.2 \cdot 10^{-4} \cdot \left(-M_{propeller} \cdot 2.32 + \left(M_{diz} - M_{gen} - M_{propeller}\right)4543.6\right),$$

where the shaft line twisting angle is:

$$\varphi = \left(-M_{propeller} \cdot e_1 + \left(M_{diz} - M_{gen} - M_{propeller}\right) \cdot \left(e_2 + e_3\right)\right) =$$

$$= \left(-M_{propeller} \cdot 2.32 + \left(_{diz} - M_{gen} - M_{propeller}\right)4543.6\right) \cdot 10^{-8},$$

According to the calculations, the nominal shaft line twisting moment is no more than *10%* of the main engine moment.

Let us proceed to the deviations in the twisting moment [9-12]:

$$\Delta M_{sk} = 0.999 \cdot \left(\Delta M_{diz} - \Delta M_{gen}\right) - 1.0005 \cdot \Delta M_{propeller}.$$

If we factor in the gear ratios:

$$\Delta M_{sk} = \left(0.999 \cdot \left(\Delta M_{diz}H_1 - \Delta M_{gen}H_1\right) - 1.0005 \cdot \Delta M_{propeller}\right)\frac{1}{H_1} =$$
$$= 1 \cdot \left(\Delta M_{diz} - \Delta M_{gen}\right) - 0.584 \cdot \Delta M_{propeller}, \tag{1}$$

where: ΔM_{gen} is the deviation in the moment of the generator driven by the main engine shaft, ΔM_{diz} is the deviation in the main engine moment.

The equation (1) is further to be used in the main power plant dynamics equations.

When studying the mechanical fluctuations of the power plant [1, 12], one has to take into account the internal resistance forces. Those occur as a result of the parts of the fluctuation-torsional system affecting each other in case of the relative rotation of their sections. The viscous friction force depends on the internal resistance that is calculated by referring to the system energy dissipation coefficient *b*. For calculations, *b=0.005*. Generalized resistance force is determined as follows:

$$Q = -b \cdot \dot{\varphi}.$$

Let us proceed to the deviations in the twisting angle equations:

$$\Delta\varphi = \left(-\Delta M_{propeller} \cdot 2.32 + \left(\Delta M_{diz} - \Delta M_{gen} - \Delta M_{propeller}\right)4543.6\right) \cdot 10^{-8}$$

and the equation of viscous friction determined by internal resistances is:

$$\Delta Q = \frac{\left(\left(\Delta M_{diz} - \Delta M_{gen}\right) \cdot 22718 - \Delta M_{propeller} \cdot 22730\right)}{\Delta t} \cdot 10^{-11}.$$

If we factor in the gear ratios:

$$\Delta Q = \frac{\left(\left(\Delta M_{diz}H_1 - \Delta M_{gen}H_1\right)22718 - \Delta M_{propeller} \cdot 22730\right)}{\Delta t \cdot H_1} \cdot 10^{-11} =$$
$$= \frac{\left(\Delta M_{diz} - \Delta M_{gen}\right) \cdot 2,3 - \Delta M_{propeller} \cdot 1,3}{\Delta t} \cdot 10^{-7}. \tag{2}$$

The equation (2) is further to be used in the main power plant dynamics equations.

3. Conclusion

The altering torque in the drive–shaft line—propeller system causes torsional fluctuations of the ship shaft line. The fouling and corrosion of the propeller as well as the weather conditions of navigation affect its modulus of resistance. It changes the free component of the transient process of shaft generator, stress frequency changes in transient processes. While the shaft generator and the diesel generator of the ship power plant are paralleled, there emerges an angle between their EMF. This results in equalizing currents generated between them. This is due to the uneven generation of mechanical energy in the piston engine and the uneven field of rates at which the water flows onto the propeller. These fluctuations peak at resonant RPM numbers when the variable torsional stresses are maxed.

According to our calculations, the nominal shaft line twisting moment for Project 503 seiner trawler is no more than *10%* of the main engine moment.

Acknowledgments

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