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DIAGNOSTICS OF MARINE PROPELLER SHAFTS

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Abstract

This author provides a description of the construction and function of a ship's propeller shaft, and states that the methods and resources for diagnosing a marine propeller shaft are insufficient. It is underlined that the diagnostics of machines mounted in plain bearings successfully makes use of measurements of the shaft journal centre trajectory. An attempt has been made to transfer this kind of diagnostics to the field of marine propeller shafts. A physical model of a propeller shaft was built in a test stand ROTOR KIT OIL WHIRL/WHIP OPTION made by Bently Nevada. The trajectory of a shaft journal centre and its maximum radius-vector were examined. The need to develop this method of diagnosing propeller shafts has been confirmed.

Keywords: *vibration diagnostics, marine propeller shaft, trajectory of the shaft journal centre*

1. Introduction

The propeller shaft is an important element of the marine propulsion system. What characterizes the propeller shaft is that it is supported in the stern tube bearing, it transfers the torque from the engine to the propeller and the axial forces from the propeller to the thrust bearing, and it runs through the hull. Set at the end of the propeller shaft is a screw, or propeller. As a rule, the stern tube bearing is a slide bearing lubricated with a liquid, mostly oil. The propeller shaft has to be sealed to prevent water from mixing with the lubricating oil, or oil mixing with sea water, finally, oil and sea water from getting into the ship.

The operation of propeller shaft components leads to their wear. The shaft undergoes tribological wear at places it co-acts with seals and sleeves. Corrosion and abrasive wear affects the shaft on the surface of its contact with the propeller, so that it can even suddenly break at this place. Seals mostly undergo abrasive wear and aging. When a seal is damaged, it results in oil leaks. When water leaks into the oil, tribological wear of shaft journals and bearing sleeves is accelerated. Tribological wear results in damage of the propeller shaft and the propulsion system failure, which in bad weather may lead in extreme cases to the sinking of a ship. Other causes of propeller shaft failures include ship's hull deformation, and changes in the position of sleeves in relation to the journals of the propeller shaft.

Classification societies consider propeller shafts as very high-risk devices. To enhance ship's safety, propeller shafts are periodically surveyed. Condition monitoring of these shafts is even recommended, which yields a bonus consisting in every second complete survey being replaced by a simplified one.

2. Required and recommended methods of propeller shaft condition assessment

The propeller shaft is a smooth shaft with a flange or journal on one end, which is used for mounting a coupling connecting the propeller shaft with the drive. On the other end the shaft has a collar or conical profile facilitating the mounting of a propeller. As a rule, the propeller shaft is supported by two or three hydrodynamic slide bearings. One bearing block (additional bearing) is fixed to the foundation bed in the ship's double bottom, while the other two bearing blocks (main bearing of propeller shaft) – aft stern tube bearing and forward stern tube bearing - are placed at the ship's stern frame or post. As standard, the stern tube is provided with forward and aft stern tube seals of the lip ring type having three lip rings in the aft seal and two lip rings in the forward seal.

For slide bearing important is proportion between bearing sleeve length *l* and shaft journal diameter *d*.

In compliance with classification societies rules:

- for aft stern tube bearing l/d=1,5,
- for forward stern tube bearing l/d=0,5.

Date of the example propeller shaft:

- distance between the stern tube bearings – 9057mm,
- diameter of the stern bearing journal – 589mm,
- clearance in aft stern tube bearing – from 0,8 to 1,08mm,
- reaction in aft stern tube bearing in operating condition – 246,4kN,
- kinematic viscosity of lubricating oil at 100°C – 11,3 $\left[\frac{mm^2}{s}\right]$ (Marine Oil Gulfmar AC 307).

The value of the relative clearance in slide bearings is chosen depending on the material of the bearing sleeve, load and revolutions per minute. The literature [6] includes reports that the relative clearance in bearings with sleeves made of white metal should range from 0.4 to 1‰, while for sleeves made of plastic 1.5 – 10.0‰.

Overhauls make up the basic methodology of propeller shaft condition assessment. Overhauls, however, require that all elements of the propeller shaft assembly be accessible for the evaluation of their structure and geometry. To this end, non-destructive tests are carried out (mostly visual, penetrating, magnetic-powder and ultrasound) and measurements of geometrical dimensions. An unlimited access to the propeller shaft assembly components requires that the propulsion is stopped, the ship is docked, the propeller is removed, its seals and stern bearing are dismantled, and the shaft is taken out.

Besides, classification societies recommend certain assessment methods that go beyond technical diagnostics. According to the PRS [1] diagnostics consist in:

- measurements of sleeve temperatures at points regarded as the most loaded,
- sampling the oil lubricating the stern tube bearing for analysis.

On the basis of energy conservation law one can explain that the temperature gradient – a measure of change in the sleeve internal energy – is connected with the intensity of tribological processes taking place in a slide bearing: it indicates that energy is accumulated and that there is a danger of converting this energy into the work of destructive processes that result in bearing damage. The diagnostics using this method are called thermal diagnostics.

Based on convective diagnostics, analysis of an oil sample can deliver a range of information about:

- technical condition of the oil – the third element of a tribological node;
- condition of the other components of the propeller shaft: journal, sleeve and seal.

A disadvantage of thermal diagnostics is that they indicate the intensity of destructive processes as they happen at the moment of observation, even with some delay resulting from the inertia of the measurement path. Convective off-line diagnostics provide information with delay needed for taking samples, delivering them to a laboratory, analysis and sending the results back.

3. Developments in methods and resources of propeller shaft diagnostics

Condition monitoring of propeller shafts consists of diagnosing three critical units: propeller – shaft, seal – shaft journal, and bearing sleeve – shaft journal.

Diagnosing of seals comes down to detection and measurements of oil leaks [2]. The existing methods and resources used for diagnosing propeller shaft seals are insufficient. There are reports on attempts to detect water leaks into the oil, where detectors are placed in the sealing.

However, available publications on the subject do not include any methods for diagnosing the propeller – shaft assembly. To assess its condition, the assembly has to be dismantled for the examination of the structure and geometry of the shaft journal and the propeller.

For the inspection of 'land-based' machines where shafts are supported by plain bearings diagnosing is successfully performed based on the trajectory (position) of rotor shaft journal centre. The measurement principles are laid down in standard [3] and numerous publications, e.g. [4, 5]. Both fixed and portable systems for measurements and monitoring of shaft vibrations are available. The position of the shaft centre within the measurement plane depends on the position of supports, bearing capacity and the load acting on the rotor. The capacity of a plain bearing for a given rotation speed (rpm) depends on the technical condition of the bearing (journal and sleeve) and of the lubricant. The rotor load consists of the working load, its own weight, load due to rotor unbalance and load due to misalignment of the motor and power receiver shafts. The shaft centre trajectory is also affected by dynamic properties of the rotor and bearing, including the lubricant [4, 5, and 6]. We may draw conclusions on changes in machine loads and technical state from changes in the centre trajectory, its shape, dimensions and direction of displacement.

There are grounds to think that this type of diagnostics can be also used for examining propeller shafts. In this case condition monitoring would consist of measurements and analysis of the journal centre trajectory, position of the journal centre inside the clearance circle and the changes of clearance circle. We may consider conclusions that condition of stern tube bearings, its seals and propeller will have an important influence on centre trajectory of aft stern tube journal. While the seals influence can have double impact [7]:

- reaction force between shaft and its seals have an effect on trajectory, the bigger eccentricity the bigger the reaction force,
- at too big radius vector (too big eccentricity), the seals can loose the expected sealing effect it means tail shaft can reach non-operational or unserviceable state.

4. Examination of the propeller shaft journal centre trajectory

Transverse vibrations of a propeller shaft model were examined. A propulsion system model including a propeller shaft is shown in Fig. 1. The physical model was built in a test stand ROTOR KIT OIL WHIRL/WHIP OPTION from Bently Nevada. The propeller shaft, 10mm in diameter, connected through a flexible coupling with the shaft of an electric motor, was supported by two journal bearing blocks, one hydrodynamic (aft stern tube bearing) and the other self-lubricating. The journal diameter in the hydrodynamic bearing was 25mm. At four points distributed on the hydrodynamic bearing circumference a lubricating liquid Chevron GST Oil 32, was supplied from an autonomous lubrication system (without cooling radiator). Two eddy-current sensors were mounted in the sleeve, perpendicular to the bearing axis, according to standard [3]. At the shaft end, apart from bearings, was a disk with a mass of 800g (heavier than the shaft mass) simulating a propeller. As the sealing is an integral part of the propeller shaft assembly, action of the aft seal was simulated by an elastically supported rolling bearing. The rolling bearing was fixed with an inner ring on the shaft, while the outer ring was stretched with four springs placed in a frame, which in turn was mounted on a common foundation bed of the test stand. The rolling bearing axis overlaps the axis of journal bearings of the propeller shaft, and the spring tension is approximately the same as the tension of the sealing rings. Another function of the elastic support was to prevent the direct contact between the shaft and the sleeve. This is due to the fact that in the test stand used, the hydrodynamic bearing sleeve, on account of the working principle of eddy-current sensors, is not made of antifriction metal.

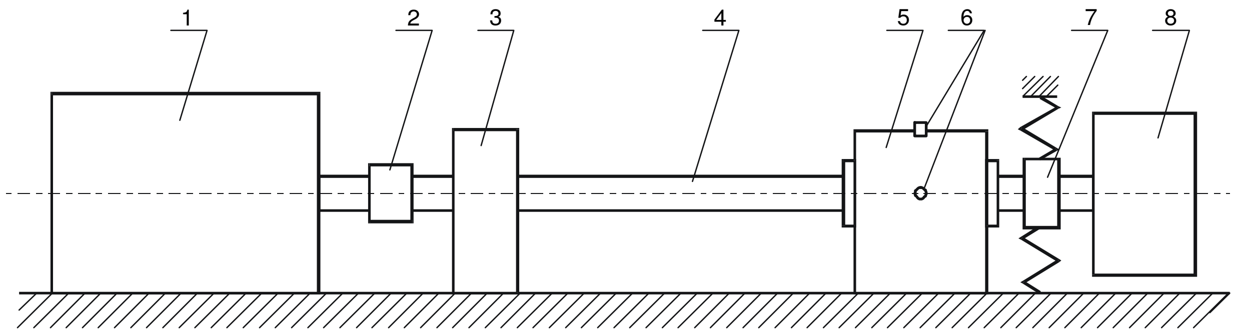

Fig. 1. Physical model of a propeller shaft: 1 – driving motor, 2 – coupling, 3 – plain bearing (self-lubricating), 4 – shaft, 5 – hydrodynamic plain bearing, 6 – eddy-current sensors, 7 – rolling bearing with elastic support (sealing), 8 – rotor disk (propeller)

The test parameters: absolute clearance of 0.35mm and the shaft journal diameter of 25mm gave a relative clearance of 14‰. The distance between bearing blocks was 340mm.

The Sommerfeld number was chosen to match rotational speed - revolution of actual shaft model. The results of calculation are presented in table no. 1.

Table 1. Determination of Sommerfeld number for actual tail shaft bearing and model tail shaft bearing

Quantity	Example of actual tail shaft	Model of actual tail shaft
b/d (bearing length / journal diameter)	1.5	25/25 = 1
Load of stern tube bearing	246.4kN	(0.800 + 0.338kg) x 9.81= 0.011kN
Relative clearance ‰	min = 0.8/589 = 1.36 max = 1.08/589 = 1.83	14.0
Square clearance	Min=1.8496 Max=3.3489	196
Bearing length x journal diameter *bd* $[m^2]$	0.883x0.589=0.520	0.025x0.025=0.000625
kinematic viscosity η at 100°C $\left[\frac{mm^2}{s}\right]$	11.3	5.2
Bdη	0.52x11.3=5.876	0.000625x5.2=0.00325
Sommerfeld number for min. clearance Sommerfeld number for max. clearance	$\frac{1}{\omega}$ 77.6 $\frac{1}{\omega}$140.5	$\frac{1}{\omega}$663.4

According to wide literature sources on this subject for example [6], slide bearings have the same position of journal in sleeve and the same friction coefficient when they are similar it means they have the same relative length *b/d* and the same angle of contact of bearing journal and sleeve and the same Sommerfeld number.

Bearing sleeves of the actual tail shaft and the model tail shaft are closed type and therefore, they have the same angle of contact of bearing journal and sleeve - for closed sleeves angle of contact is equal to 2Π (360°).

Sommerfeld number S_o is given by the following formula:

$$S_o = \frac{P\psi^2}{bd\eta\omega}$$

where:

P – load,

ψ – relative clearance = bearing clearance *s* / bearing diameter *d,*

b – sleeve length,

η – kinematics viscosity,

ω – journal angular velocity in relation to sleeve.

According to [6] for *b/d*>1 increase of *b/d* value is not creating, for the same Sommerfeld number significant changes in eccentricity. We may draw conclusions that the bearing in model shaft will hold the same eccentricity as in the actual tail shaft if Sommerfeld numbers hold the same values.

Permissible clearance changes in bearing (bearing wear) can cause changes in Sommerfeld number in range of 1/ω(77,6 – 140,5). Sommerfeld number in bearing in model will be equal to Sommerfeld number in actual bearing if the model shaft revolution is *k* times bigger than rotational speed of actual shaft. The substitution of data from table 1 *k* is vary from 663,4/77,6 to 663,4/140,5 (from 8,5 to 4,72). Because rotational speed of actual shafts is vary from 60 to 170 rev/min, it gives after calculation that shaft revolution during researches should be in range of 283 till 1445 rev/min:

- in range of 283 to 802 rev/min model bearing remains in the same way as actual bearing with max clearance,
- In range of 510 to 1445 rev/min model bearing remains in the same way as actual bearing with min clearance.

For the purpose of wider view, the tests were made in the range of 0 – 2500 rev/min. The displacement, i.e. trajectory of the journal centre was examined by measuring the maximum radius-vector. The trajectory was examined using a system of eddy-current sensors and a digital real-time oscilloscope TDS 210. The radius was measured by the eddy-current sensors combined with a WIBROPORT 41 system. The TDS 210 oscilloscope allows to filter the signal by using the in-menu 'coupling' function and setting DC or AC: DC passes both AC and DC components of the input signal; AC blocks the DC component of the input signal.

The following issues were examined:

- effect of sealing and rotation speed on the position of the trajectory inside the clearance circle, Figs. 2, 3, 4, 5;
- maximum radius-vector depending on the rotation speed, Fig. 6;
- maximum radius-vector $\mathbf{S_{max}}$ for a selected speed *n* depending on additional mass *m* of the disk, Table 2. The mass was added on the disk circumference at two opposite points φ = 0° and φ = 180° (position φ = 0° was chosen in random way).
- effect of the 'sealing' on shaft centre trajectory image for a speed close to the resonance speed of the propeller shaft, Fig.6;
- effect of the intensity of the lubricant flowing through the bearing for a selected rotation speed, Fig. 8.

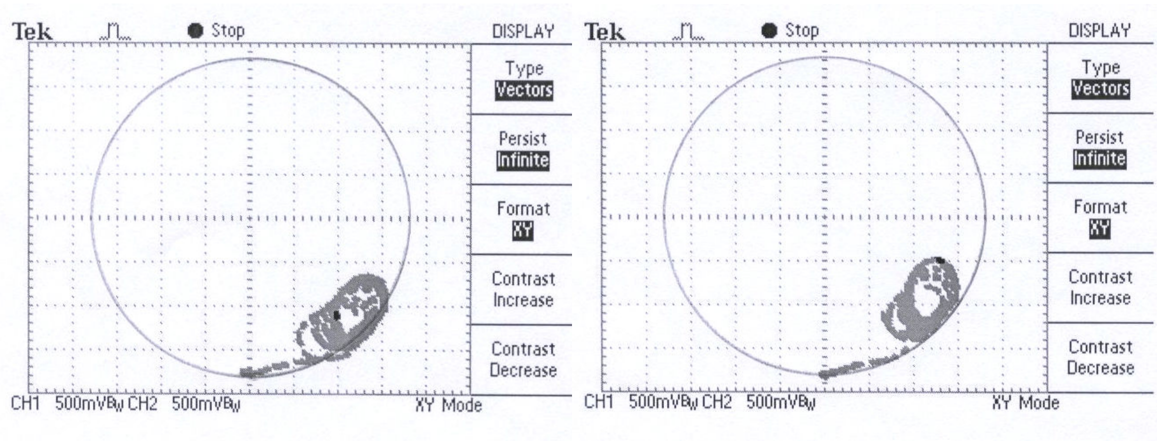

Fig.2. Trajectory of the journal centre for a shaft for $\dot{m}_{oil}$ = max *at transition from 0 to 265 rev/min without 'sealing'(left)and with 'sealing'(right)*

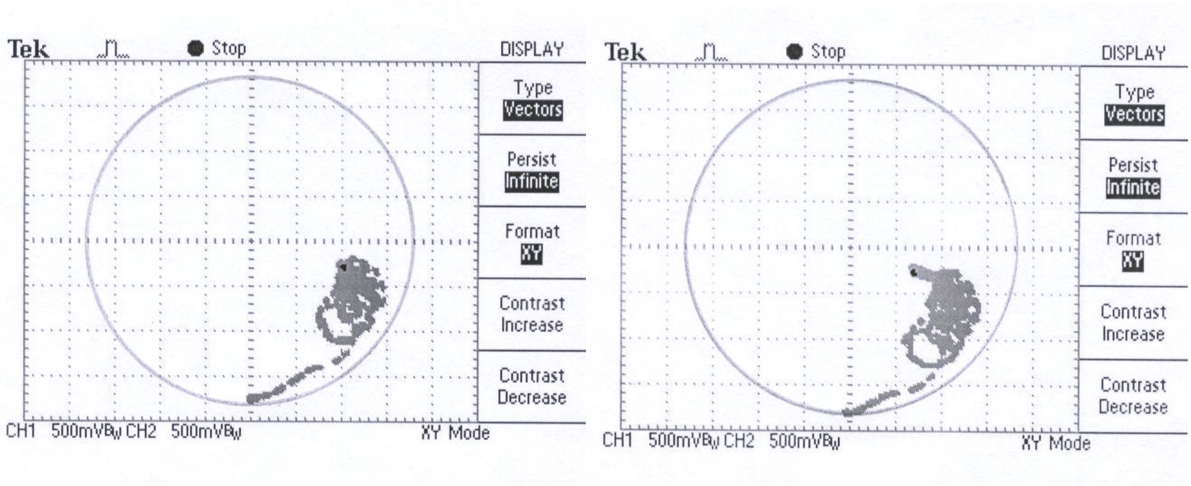

Fig. 3.Trajectory of the journal centre for a shaft with 'sealing' for $\dot{m}_{oil}$ = max *at transition from 0 to 720 rev/min (left) and at transition from 0 to 1300 rev/min (right)*

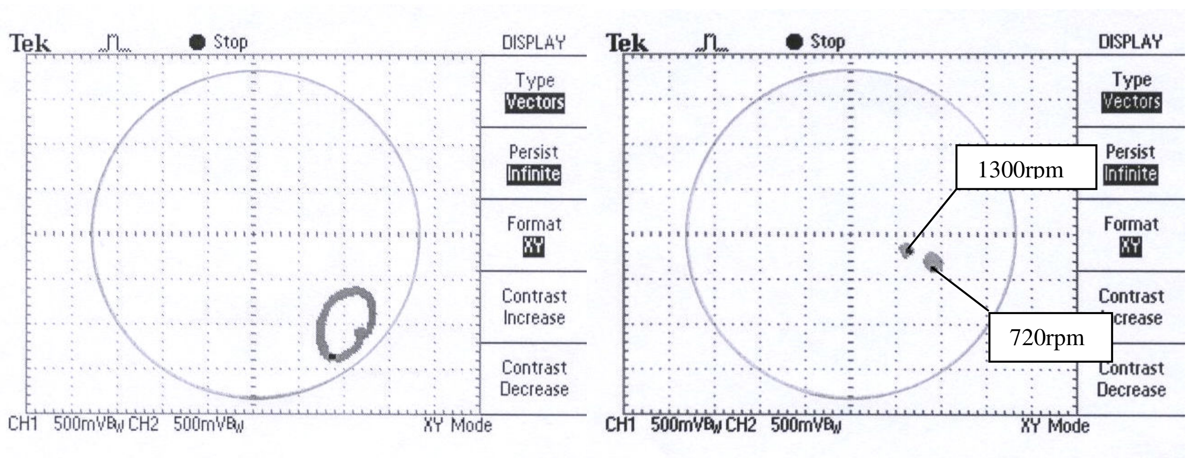

Fig.4. Trajectory of the journal centre for a shaft with 'sealing' for $\dot{m}_{oil}$ = max *at: 260 (left), 720, 1300 (right) rev/min*

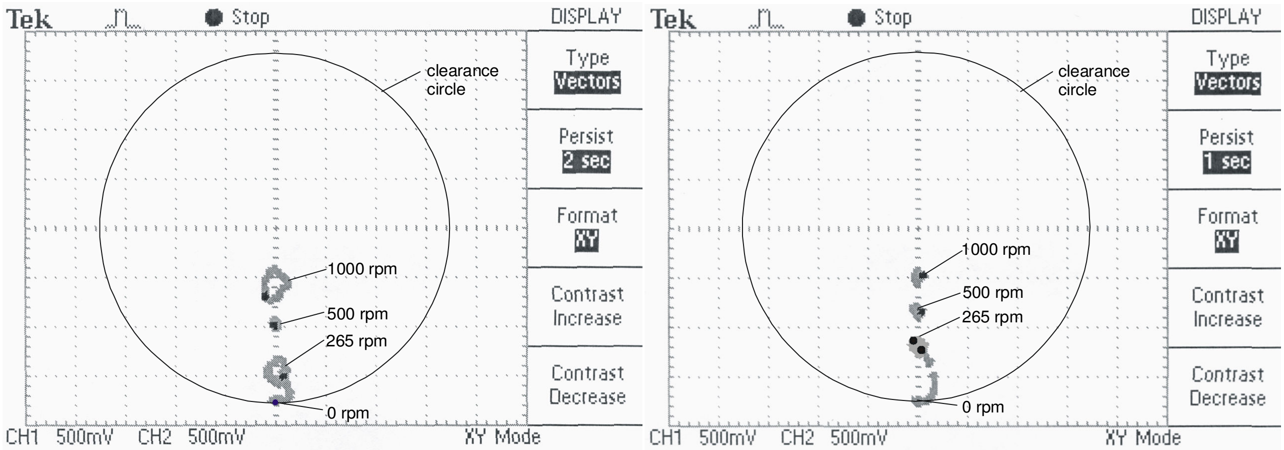

Fig.5. Trajectory of the journal centre for a shaft without 'sealing' for $\dot{m}_{\text{oil}} = \max$ *at: 0, 265, 500, 1000 rev/min (left) and trajectory of the journal centre for a shaft with 'sealing' for* $\dot{m}_{\text{oil}} = \max$ *at: 0, 265, 500, 1000 rev/min (right) (the DC component of the input signal in the horizontal direction is blocked)*

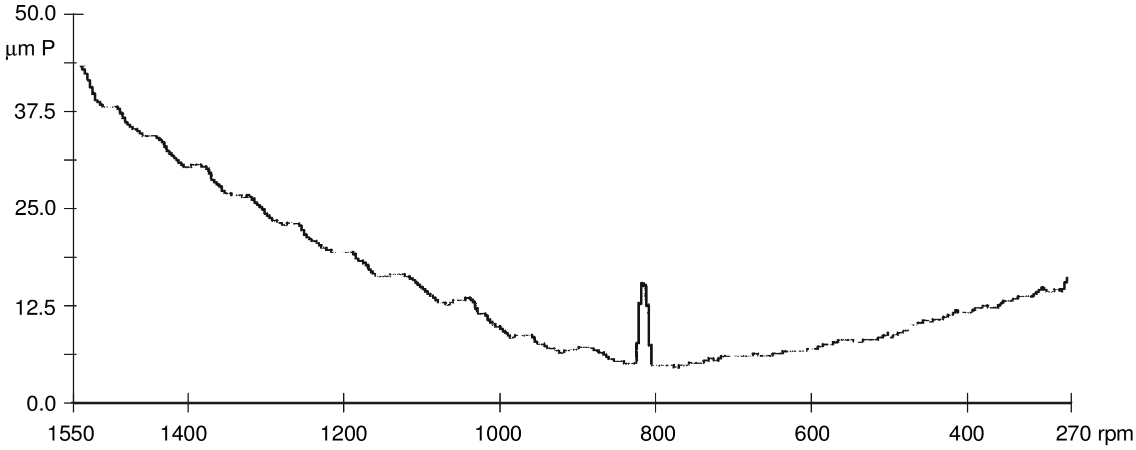

Fig. 6. $s_{\max} = f(n)$ *for n from 1550 to 270 rev/min*

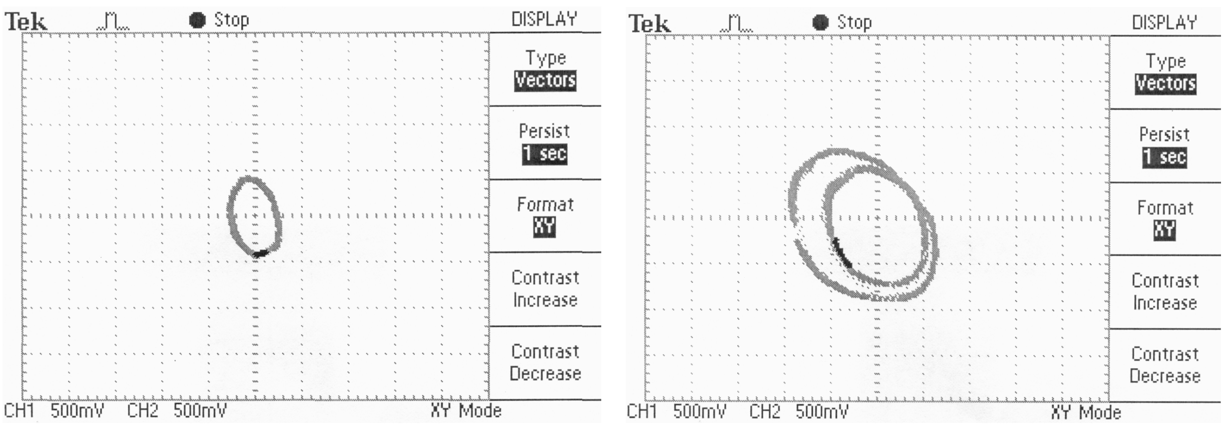

Fig.7. Trajectory of the journal centre for a shaft with 'sealing' (left) and without sealing (right) for $\dot{m}_{\text{oleju}} = \max$ *at n = 1610 rev/min (the DC component of the input signal in the horizontal direction is blocked)*

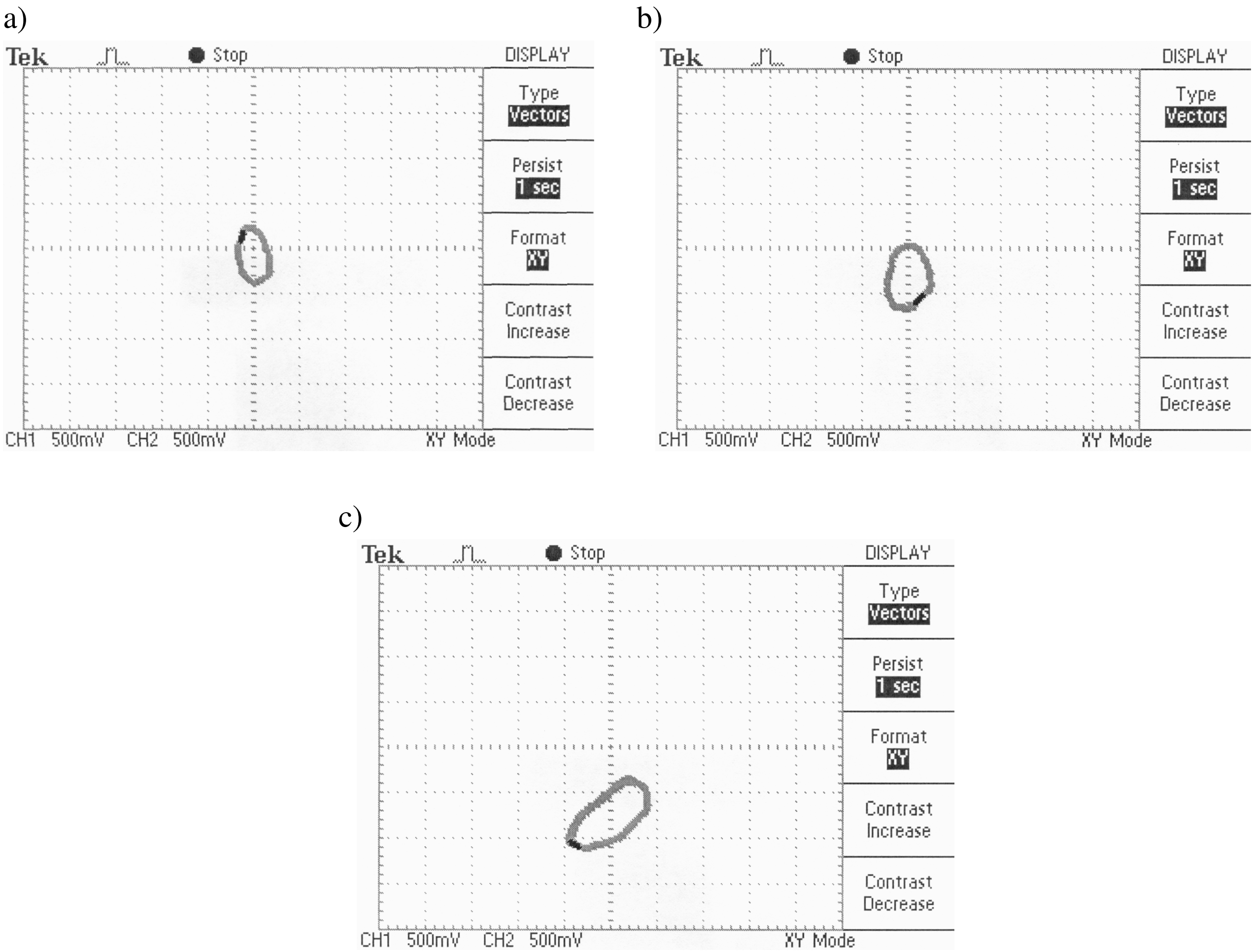

Fig.8. Trajectory of the journal centre for n = 1500 rev/min at various oil flow intensities: a) maximum, b) reduced by one revolution of the reduction valve, c) reduced by two revolutions of the reduction valve (the DC component of the input signal in the horizontal direction is blocked)

Tab. 2. Maximum radius-vector $\boldsymbol{S_{max}}$ *for a selected speed* ***n*** *in depending on additional mass* ***m*** *of the disk*

φ = 0 [°]			φ = 180 [°]		
n [obr/min]	m[g]	S_{max} [μm]	n [obr/min]	m [g]	S_{max} [μm]
1001	0	8,34	1001	0	8,34
1009	0,1	8,73	1003	0,4	7,50
1004	0,2	8,66	1006	1,0	5,50
996	0,4	9,51	1004	2,0	6,89
1005	2,0	15,0			

5. Analysis of the results

The following conclusions can be drawn from the journal centre displacements:

– for the rotation speed 0 rev/min, when the journal generatrix contacts the sleeve generatrix, the journal centre lies on the clearance circle line. As the rotation speed increases, initially the shaft journal rolls sliding on the sleeve, then lifts and remains in contact with oil film formed between the journal and the sleeve. The journal centre first travels on the clearance circle itself, then moves towards the circle inside. The higher the speed is, the thicker oil film forms and the closer the journal centre gets to the clearance circle centre, Fig. 2, 3, 4;

- similar effects are observed when the DC component of the signal in the horizontal direction is blocked, Fig. 5;
- for a preset rotation speed, the journal centre trajectory in a general case is a figure resembling an ellipse, with dimensions (e.g. maximum radius-vector) that get smaller as the bearing work is more stable (fixed load and better working conditions), see Figs. 4, 5;
- the position of the trajectory shows a sufficient difference between the minimum and maximum clearance (Fig. 4: 720 and 1300 rev/min., similarly in Fig.5: 500 and 1000 rev/min);
- additional elasticity, other than the one resulting from oil film elasticity, has a stabilizing effect on the trajectory of shaft journal, see Figs. 2, 5, 7. At the given revolution the average value of trajectory is not undergoing the significant changes, but the instantaneous value of trajectory is undergoing the significant changes, Fig. 7. During the start-up additional elasticity facilitates the formation of oil film and reduces friction, Figs. 2, 5;
- dimensions of the journal centre trajectory depend on the journal rotation speed, Fig. 6. Each rotor with one high mass has at least one significant frequency of free vibration. When the rotation speed is equal to the free vibration the bearing loses its stability, and the journal centre trajectory reaches the dimensions of the clearance circle. It was found during measurements that in the examined model of propeller shaft the free vibration frequency (with additional stabilizing elasticity and at maximum intensity of lubricant flow) corresponds to 1680 rev/min. One characteristic of plain bearings is that at 1/2 resonance speed the so called oil whirl appears, causing an essential increase in the rotor vibration amplitude, which translates into increased values of journal centre trajectory dimensions. In Figure 6 the effect of oil whirl is visible at a speed over 800 rev/min;
- amount of oil flowing through the bearing significantly affects the journal centre position and trajectory dimensions in stable working conditions. When the oil flow intensity is reduced, the trajectory, falling towards the clearance circle, increases its dimensions, Fig. 8.

It follows from the obtained measurements of the maximum radius-vector values that additional mass put on the rotor disk causes the radius value to change, Table 2. The value by which the radius-vector changes depends on additional mass as well as the place at which this extra mass is added relative to the residual unbalance of the rotor disk.

6. Conclusions

1. For ship's safety, a marine propeller shaft is a very important element of the propulsion system. The applied and recommended methods of its condition monitoring are insufficient. This author proposes diagnosing these shafts using relative vibrations measured in at least two planes perpendicular to the shaft axis. The measurement planes should be located in the plain stern and bow bearings of the shaft or close to them. In operational practice the journal centre trajectory can be visualized at the blocked DC component of the horizontal signal. In particular, this refers to propulsion systems operating at a constant rotation speed – systems with a controllable pitch propeller. At the constant rpm rate of the propeller shaft the vertical sensor axis should overlap the straight line connecting the point of thinnest oil film with the sleeve centre.
2. There is an assumption that existing relation between tightness and eccentricity and between eccentricity and reaction force (and sealing wear) in the sealing can be used to draw

conclusion on sealing condition. In this way the position and trajectory value of the shaft journal centre might provide data for conclusions concerning:

a) position of ship's propeller shaft,
b) wear of journals and bearing sleeves,
c) technical state of lubricating oil,
d) condition of shaft seals,
e) condition of the propeller (whether balanced or not).

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