CZESŁAW DYMARSKI, D.Sc.,M.E.
Technical University of Gdańsk
Faculty of Ocean Engineering
and Ship Technology
Chair of Ship and Offshore Units' Equipment

Laboratory investigations of the stern tube bearings and pitch control mechanism of the low-power CP propeller with screw-toothed gear

SUMMARY

Superiority of the CP propellers over the fixed pitch propellers is evident in the case of their application on the ships intended to operate at very changeable loading conditions and on the ships required to have higher manoeuvrability or even dynamic positioning features. Demand for low-power CPPs grows in Poland. To face this trend, two alternative, low-power CPPs of different pitch control mechanisms and systems were designed and manufactured by a team of the Faculty of Ocean Engineering and Ship Technology, Technical University of Gdańsk. A description and results of the laboratory tests of the stern tube bearing and seal assembly and of the pitch control mechanism with screw-toothed gear are presented in the paper.

INTRODUCTION

The controllable pitch propellers find wider and wider application in ships. They are fitted on very small vessels and supertankers as well, and manufactured in the range of diameters from 1 m to 11 m.

The growing interest for CPPs results from their several very important advantages in comparison with the fixed pitch propellers, namely :

⇒ Possible stepless change of propeller pitch within full operation range, which enables to utilize full power rating of the engine irrespective of actual navigation conditions.
⇒ Possible fast change of thrust direction without changing direction of revolution of the propeller shaft. This makes applying the non-reversible engines possible which is specially important for the ships propelled by steam and gas turbines as it makes their propulsion systems simpler and contributes to increasing their durability in the case of the ships with piston engines.
⇒ Possible maintaining the engine rotational speed constant at ship variable speed, which enables to utilize the main engine for driving the auxiliary devices which require keeping their rotational speed constant.
⇒ Lowering the shaftline torsional stresses during ship course change, relative to those appearing in the system equipped with the fixed pitch propeller.
⇒ Easiness of automation and remote control of the entire ship propulsion system equipped with CPP which makes personnel reduction and improvement of the ship's navigation safety possible.
⇒ Improving the ship manoeuvrability and safety as well as making execution of difficult manoeuvres easier.

Apart from the above mentioned advantages CPPs have also some disadvantages, among which the two following are deemed the most important :

- Highly complicated construction and in consequence its lower reliability.
- High initial and maintenance cost and demand for high qualified operators.

Summing up, at the recent state of technical development and production quality, a superiority of the CPPs over the fixed pitch propellers is evident in the case of their application on the ships intended to operate at very changeable loading conditions and on the ships required to have higher manoeuvrability or even dynamic positioning features.

In Poland however demand for CPPs has changed in the last several years. Demand for the medium-size CPPs drops along with dropping number of the medium-tonnage newbuildings, but the number of small private shipowners who own small ships usually still equipped with the fixed pitch propellers, increases. Demand for better propellers, i.e. first of all for low-power CPPs grows in result of growing repairs of the old ships and in consequence of growing demand for new, modern ships of the kind.

Four years ago, to face this trend, two alternative, low-power CPPs of different pitch control mechanisms and systems were designed and manufactured by a team of the Chair of Ship and Offshore Units' Equipment, Faculty of Ocean Engineering and Ship Technology, Technical University of Gdańsk. Recently both devices have been tested in the Chair's laboratory prior to their application on ships.

A description and results of the laboratory tests of the stern tube bearing and seal assembly and of the pitch control mechanism with screw-toothed gear are presented in the paper.

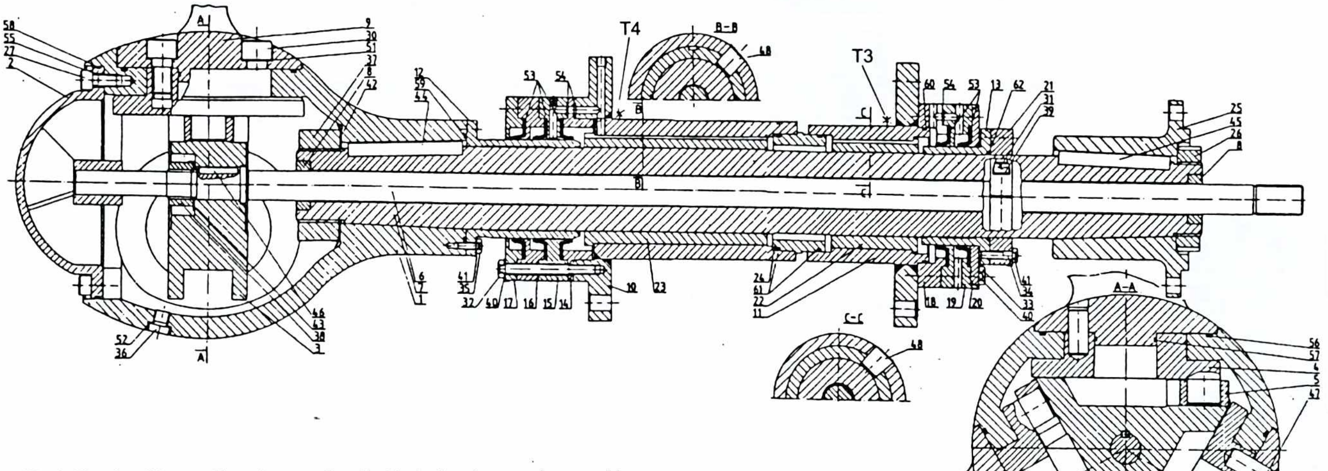
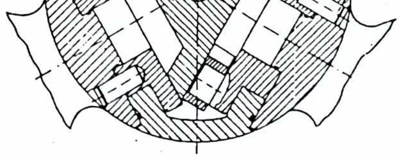


Fig.1. The ship CP propeller : the propeller (hub), shaft and stern tube assembly



DESCRIPTION OF THE TEST STAND

The tested propeller, shaft and stern tube assembly

The tested propeller, shaft and stern tube assembly (Fig.1) consists of the following sub-assemblies and more important elements :

- propeller hub together with blade pivoting mechanism (parts no: 1÷5, 9, 27, 30, 36, 38, 43, 47, 51, 55÷58)
- propeller shaft (7, 8, 37, 42, 44)
- reset rod (6, 38, 43, 46)
- stern tube aft sliding bearing (10, 23, 24, 48, 61)
- stern tube fore sliding bearing (11, 22)
- stern tube aft seal (12,14÷17, 32, 35, 40, 41, 53, 54)
- stern tube fore seal (13,18÷21, 31, 33, 34, 39÷41, 53, 54, 60, 62)
- flange of the coupling connecting the propeller and intermediate shafts (25, 26, 45).

In Fig.1 and 3 the symbol (*) indicates the points where the temperature of the stern tube body around both its bearings is measured.

The load-setting assembly of the propeller pitch control mechanism

The load-setting assembly of the propeller pitch control mechanism (Fig.2) installed directly on the propeller hub (in replacement of the cover 2, Fig.1) was provided to make generating the axial force of an arbitrary value in the reset rod possible.

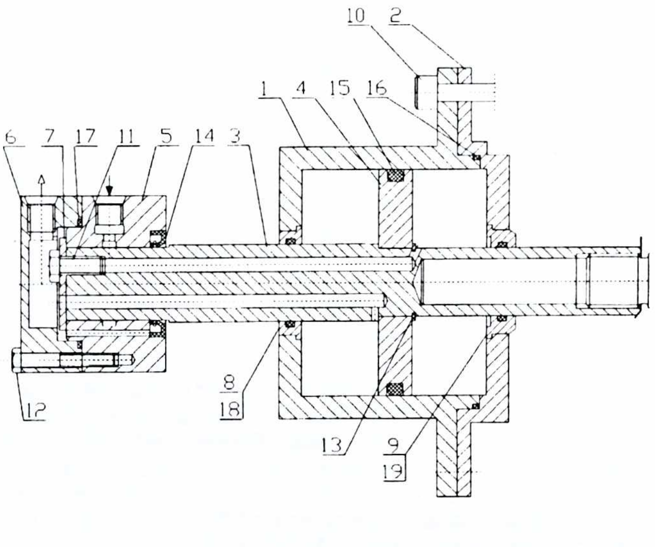


Fig.2. Hydraulic load-setting assembly of the propeller pitch control mechanism

The assembly consists of the cylinder 1 which, together with the face cover 2, is fastened by the screw bolts 10 directly to the propeller hub in replacement of its cover. The piston rod 3 is fixed at the reset rod end by means of the screw joint. The bronze guiding bushes 8 and 9 together with the „O" sealing rings 18 and 19 provide the guiding and sealing of the piston rod in the place where it penetrates the cylinder end and face cover. Inside the cylinder the piston 4 fixed by the spring ring 13 is located. The piston is sealed also by means of „O" ring of a greater cross-section area (15).

At the end of the piston rod the radial sliding seal is mounted which consists of the casing 5 and cover 6 connected to each other by means of the screw bolts 12, and sealed by the ring 17. The casing is secured against rotation and axial displacement. The disk 7 fixed to the piston rod face by the screw bolts 11 serves as a thrust bearing. The motionless casing is sealed in respect to the rotary piston rod by means of the lipseal 14.

The tested propeller pitch control mechanism with the sliding axial bearing

The propeller pitch control mechanism is presented in Fig.3.

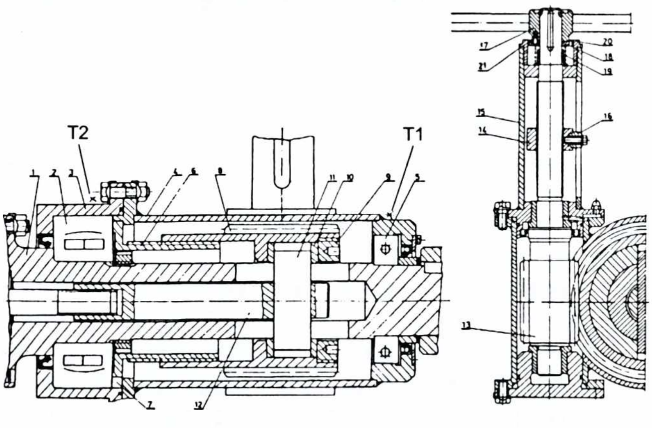


Fig.3. The propeller pitch control mechanism

The following sub-assemblies of the mechanism can be distinguished :

- the intermediate shaft 1 together with the flange coupling
- the rolling thrust-journal bearing 2, journal bearing 5 and casings 3 and 4
- the sliding axial bearing 9, 10, 11 of the reset rod 12

- the screw-toothed gear 6, 8, 13
- the mechanism driving wheel 17 together with the blocking coupling 18, 19, 20, 21
- the propeller pitch setting indicator 14, 15, 16.

The operation principle of the applied sliding axial bearing of the reset rod is schematically illustrated in Fig. 4 where also the basic geometric parameters and axial force F are marked.

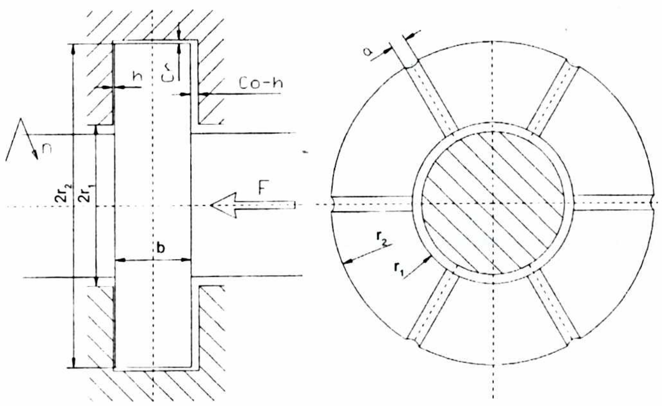


Fig.4. Scheme of the reset rod bearing
Cr - nominal radial clearance, Co - nominal axial clearance

The operation principles of the assembly and its construction are described in [2,3].

The tested propeller pitch control mechanism with the rolling axial bearing

The propeller pitch control mechanism with the rolling axial bearing of the reset rod was generally the same as that shown in Fig. 3 with the difference that the sliding bearing of it was replaced by two rolling cone bearings. In consequence small constructional changes of the centrally located gear wheel, only in its iterior part as shown in Fig.5, were introduced with maintaining all important elements of the mechanism unchanged.

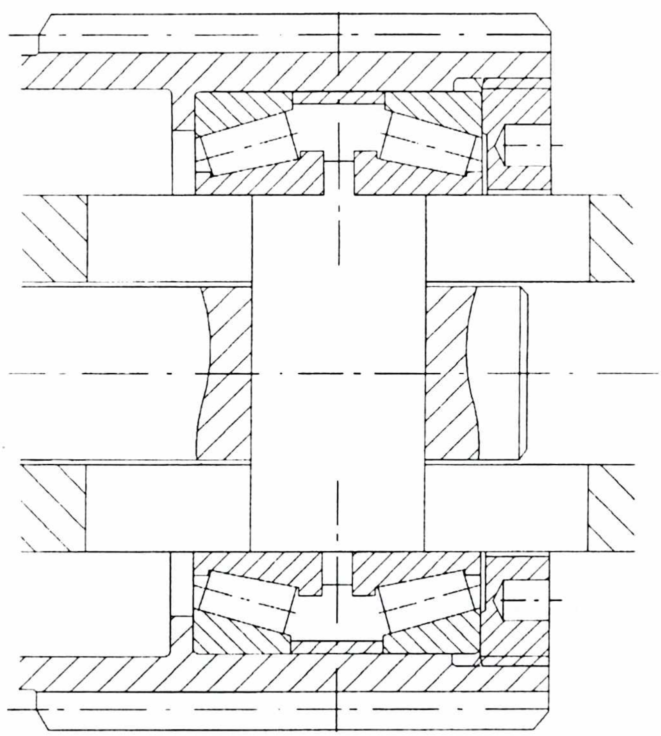


Fig.5. Arrangement of the rolling bearings of the reset rod

The propeller shaft driving assembly

The propeller shaft is driven by the DC electric motor of 3.5 kW output.The drive provides the stepless rotational speed control and rotation reversal. The motor is fed by using the Ward-Leonard system, a permanent laboratory installation.

The measuring system

The following magnitudes were measured during the tests :

★ the pressure of the oil delivered to the hydraulic load-setting assembly (measuring by a pressure gauge, reading out on a milivoltmeter)
★ the temperature of the metal parts of the bearing casings and pitch control mechanism (measuring by means of a thermocouple gauge, reading out on a microvoltmeter)
★ the shaft rotational speed (measuring by a photoelectric gauge, reading out on a milivoltmeter)
★ the force on the handle of the manual pitch change drive (reading out on a dynamometer)
★ the voltage and intensity of the electric current supplying the shaft driving motor (reading out on a voltmeter and ammeter of the control desk)
★ the propeller pitch value (reading out on a scale indicator).

The indicator was calibrated to eliminate the error of determination of the force in the reset rod by using the pressure measurements, which results mainly from difficulties in determining the seal rings friction within the load-setting cylinder. The calibration of the load-setting assembly was performed together with the propeller assembly shown in Fig.1. It consisted in mounting a special fixture on the fore, screwed end of the reset rod and exerting the axial force to it through the dynamometer, of the value sufficent to move the rod and thence to change the pitch. Three measurements were taken for each of the three pressure values of the oil delivered to the load-setting assembly : p = 0, 0.5 and 1.0 MPa.

It made possible to establish the following, approximate relationship for the determining of the rod force value required to reset the propeller pitch in the direction opposite to that of the load in the load-setting assembly :

$$F\ [kN] = 2 + 0.1\ p\ [MPa]$$

INVESTIGATIONS OF THE PROPELLER PITCH CONTROL MECHANISM WITH THE SLIDING AXIAL BEARING OF THE RESET ROD

The investigations consisted mainly in measuring the necessary torque value on the propeller pitch control wheel during simulation of the CPP operation at the speed n = 8.3 rps (500 rpm) and different axial load values of the reset rod. The resetting of the pitch was performed in both directions. The measured values are graphically presented in Fig.6.

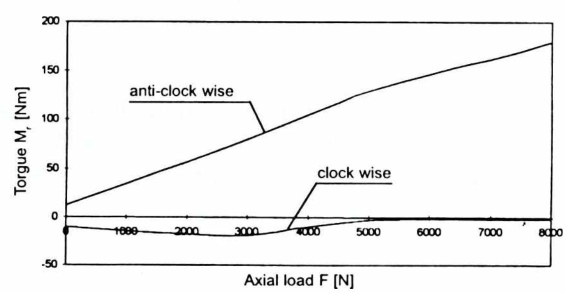


Fig.6. The torque M_r required to reset the propeller pitch in function of the axial load F applied to the reset rod

According to expectations the values of the torque necessary to reset the propeller pitch in each of the directions were clearly different and the difference was growing fast along with growing load applied to the mechanism. The measured torque values at the reset operation in the direction consistent with the load at first grew insignificantly along with the load and then, after reaching the value of F = 3.0 kN, they decreased almost to zero and thus appeared difficult to be measured. However it should be stated that the mechanism maintained its self-locking and no self-acting propeller pitch drift was observed. The measurements were carried out at small load values of the mechanism F < 9.0 kN only. The limitation was caused by too low output of the shaft driving electric motor and relatively large resistance to motion.

The measurements of the motor power input and its rotational speed in function of the rod loading at the constant, set to zero, propeller pitch were performed to better assess performance of the bearing in question. The measured values of the motor power input and friction power of the loaded axial bearing are graphically presented in Fig.7.

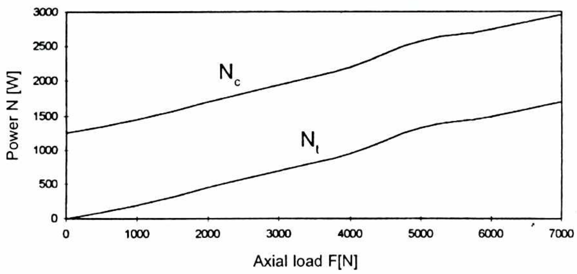


Fig.7. The input power of the motor, N_c, and of the loaded axial bearing, N_t, in function of the reset rod load F at the oil temperature inside the mechanism casing, $T_{ol} \approx 40°C$

It was possible to calculate the mean friction coefficient between sliding surfaces on the basis of the known values of the power lost in the bearing and of its loading. The calculated friction coefficient value $\mu = 0.126$ was obtained for the maximum load value shown in Fig.7 and the sliding disk dimensions $r_1 = 52$ mm and $r_2 = 36$ mm (acc. to Fig.4). This very high friction value shows that the bearing performance is unfavourable and the mixed friction appears in it. The too low power of the shaft driving motor and improper way of operation was probably the main reason of the observed phenomenon. It led to overloading the motor and drop in the shaft rotational speed when large loads were applied to the mechanism, and in consequence to worsening the conditons for generating the sufficient hydrodynamic lift between the sliding face surfaces till the mixed friction had appeared.

In the situation the pitch control mechanism was decided to be disassembled to survey elements of the bearing. During the survey the small scratches on both face surfaces of the steel disk were observed, which circumferentially spread off the oil grooves and visible on the disk surfaces, similarly orientated, yellow tints formed of bronze particles. Many particles of bronze were also found in the oil removed from the gear casing. It was the sign of an unsuitable work of the bearing.

Therefore it was decided to replace the sliding axial bearing by two typical, rolling cone bearings, taking into consideration rather unfavourable conditions of operation, maintenance and repairs of such device on small vessels. Moreover such rolling bearings are easily available and relatively cheap and can be replaced by new ones if necessary.

INVESTIGATIONS OF THE PROPELLER PITCH CONTROL MECHANISM WITH THE ROLLING AXIAL BEARINGS OF THE RESET ROD

The investigations, like the previous ones, consisted mainly in measuring values of the torque necessary for resetting the propeller pitch in both directions. Results of the measurements are given in Fig.8.

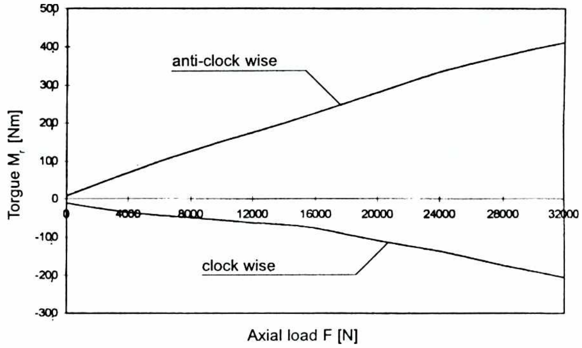


Fig.8. The torque M_r required to reset the propeller pitch in both directions, in function of the axial load F of the reset rod

It was possible, due to far lower resistance to motion of the applied rolling bearings, to increase considerably the axial load of the reset rod. However the measurements were performed for the loading up to F = 31.5 kN only. The value resulted from the rule limitation of the force which may be exerted on the handle of the load-setting assembly by its operator.

The curves presented in Fig.8 are the mean torque values calculated from at least three measured values at the relatively low rotational speed of about 1 revolution per 10 s. Rapid resetting of the pitch in the direction opposite to the loading resulted in increasing the torque at the load-setting assembly and increasing the motor power input, but the resetting in the direction compatible with the loading resulted in dropping values of both the magnitudes.

The test which consisted in measuring the reset rod friction in function of the load F was undertaken to better assess performance of the rod. The measurements were performed after one-hour operation of the device when the temperature at the measurement points shown in Fig.1 and 4 attained the following values : $T_1 = 49.5°C$, $T_2 = 52.0°C$, $T_3 = 45.0°C$, $T_4 = 43.0°C$. The results are presented in Fig.9.

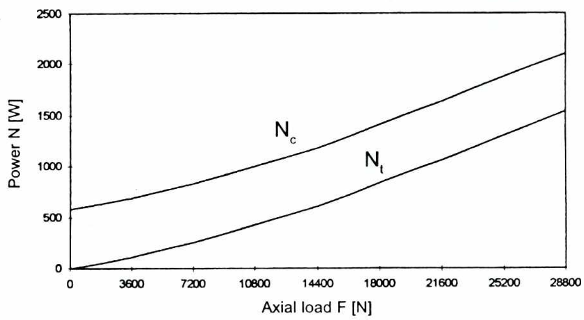


Fig.9. The input power of the motor, N_c, and the friction power of the reset rod axial bearing, N_t, in function of the reset rod load F at the oil temperature inside the mechanism casing, $T_{ol} \approx 52°C$

The calculated friction coefficient at the maximum load F value and the rolling element pitch diameter was $\mu = 0.025$ which showed correct bearing performance.

The next test was aimed at determining the influence of the propeller shaft rotation direction on the resetting torque value. Three measurements were performed at the load F values : 5.7, 9.35, and 13.0 kN. No perceptible and repeatable influence of changing the propeller shaft rotation direction on the resetting torque values was observed.

INVESTIGATIONS OF THE STERN TUBE BEARINGS

The stern tube bearing investigations were first of all aimed at determination of the optimum lubrication system for the tested propeller. Therefore several oil holes in the stern tube structure (Fig.1) were provided which made applying the following lubrication system possible :

- with solid lubricant
- with oil-filled stern tube
- with natural circulation of oil
- with forced circulation of oil.

The simple system with the oil-filled stern tube supplied by a single oil pipe from the gravitation tank located about one meter higher than the shaft, was first tested. The tests consisted in monitoring the system's performance and, at the same time, measuring the temperature of the casings of both bearings (see $*T_3$ and $*T_4$ points in Fig.1), as well as those of the thrust-journal bearing $*T_2$ and the pitch control mechanism $*T_1$ (Fig.3). The measurements commenced at the moment of putting the shaft in motion and were carried out under various loadings until steady thermal conditions were obtained, with measuring the driving motor power input once in a while. Measurement results of the temperatures and ambient temperature T_0 are presented in Fig.10.

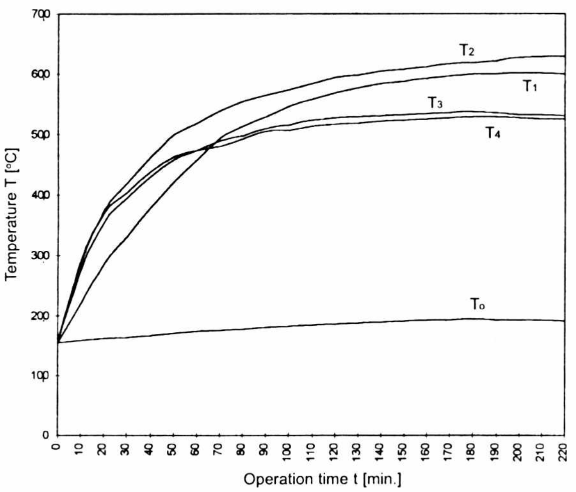


Fig.10. Courses of the measured temperatures T_1, T_2, T_3 and T_4 (acc. to Fig.1 and 3) and the ambient temperature T_0 during long operation time of the device until the steady thermal condition was obtained

During the measurements, the oil in the pipes which delivered it from the gravitation tanks either to the stern tube or the gear casing was all the time cool which showed that practically oil circulation was not present. Nevertheless the temperature values settled at a relatively favourable, not very high level. But much higher heat amount would be transferred through the shaft and propeller hub to the surrounding water in the real operation conditions of the system.

Therefore any further investigation of the lubricating systems with natural or forced oil circulation was regarded unjustified. Only the operational tests of the device with the bearings lubricated by solid lubricant was performed. However it was not possible to bring the test to its end due to excessively high resistance to motion of the device and too low power of the shaft driving motor which made carrying out the measurements at the nominal shaft speed not possible. Moreover very intensive heating of the elements of the stern tube bearings was observed.

ENERGY BALANCE OF THE TESTED DEVICE

The shape of the curves depicted in Fig.10 is close to parabolic. It results from the variable resistance to motion of the device and heat amount transferred to the environment in different thermal states of the device. It yields from the measurements of the motor power input taken during its idle operation time from the start of the device up to settlement of the thermal equilibrium that the power consumption of the device with the rolling bearings of the reset rod was almost three times greater at the beginning of the measurements in the ambient temperature $T_0 = 15.5°C$ than in the steady thermal conditions of the operation.

Values of the lost power due to resistance to motion of the particular tested elements measured at the temperature of oil in the gear casing, $T_p \approx 30°C$, were as follows :

- of the motor itself - 54 W
- of the propeller, shaft and stern tube assembly - 640 W
- of the pitch control mechanism :
  - full of oil - 337 W
  - half - oil filled - 300 W
  - with very small oil amount - 265 W

Working conditions of the device installed onboard a small vessel will obviously differ from those in the laboratory. The ambient temperature inside the ship can be much higher than that during the measurements, but on the other hand the heat amount transferred to the water through the shaft and propeller hub and the casing flanges of the stern tube bearings, connected to the ship hull, will certainly be much greater than that in the laboratory conditions. Therefore it is justified to conclude that the state of thermal equilibrium of the particular assemblies of the tested CP propeller would settle at a lower, or at least not higher, temperature level.

FINAL REMARKS

The obtained investigation results make it possible to better assess usefulness of the particular assemblies of the tested CP propeller design in view of their practical application on small sea-going ships.

➲ It can be stated that the type of bearing of the reset rod considerably influences the propeller pitch control mechanism operation. The axial bearing of the reset rod with the use of the typical rolling bearings provides higher quality of operation at lower resistance to motion irrespective of the mechanism's loading and ambient temperature. It also makes replacement of the worn or failed bearings easy which is advantageous due to standard features, low cost and availability of the applied bearings.

➲ The tested lubrication method of the bearings of the stern tube and pitch control mechanism, consisting in filling the casings of the mechanism and stern tube with oil delivered through the piping from the separate gravitation tanks, turned out quite sufficient. It ensured appropriate operation of the device at a very low resistance to motion and rather low working temperature in the settled thermal equilibrium conditions. Continuous tightness control of the stern tube sealing was possible by monitoring the oil level in the gravitation tank.

➲ Moreover the tested CP propeller design makes applying the lubrication system with natural or forced oil circulation possible. It requires only to connect the existing (but now blanked) holes to the tank through an additional pipe, or to the oil circulation pump in the case of forced circulation.

The tested CP propeller was designed and manufactured by the team of the Faculty of Ocean Engineering and Ship Technology, Technical University of Gdańsk. The applied propeller pitch control mechanism is of an original, patent protected design.

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Appraised by Józef Krępa, Assoc.Prof.,D.Sc.,M.E.