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Empirical research on the sliding radial seals applicable to CP propellers

SUMMARY

Laboratory investigations of the influence of bearing sleeve transverse stiffness on operation effectiveness of the sliding radial seals applicable to ship CP (controllable pitch) propellers, are presented. The tests were carried out on the full-scale seal which cooperated at first with the typical, i.e. thick-walled, bearing sleeve and then the thin-walled one. Their results confirmed that development of a new generation of such seals, but lighter and more effective, is possible.

INTRODUCTION

Ship CP propeller mechanism design development is basically connected with the problem of oil delivery under pressure to the propeller shaft interior. In the contemporary high-power designs, e.g that shown in Fig. 1 (see page 12), the hydraulic cylinder 6, used to change propeller pitch, is fitted inside the propeller hub body 3, but the delivery system with the oil pump assembly 25 is located outside the servo shaft 22.

Working oil is delivered to the seal 18, slidewise-fitted on the shaft but not rotating, which makes the oil flow through radial holes in the shaft into the relevant co-axial pipings 13 and 14, by which it flows into one of two chambers of the oil cylinder 6.

The location of the oil cylinder in the front or rear part of the hub determines its large length and weight, but it enables application of the piston of the diameter large enough to maintain the required working pressure not higher than 4 MPa [1, 2].

The propeller pitch control mechanism can be much simpler and gabarites of the propeller hub smaller if the oil cylinder is placed at the propeller blade axis, between the blade carriers, as shown in Fig.1. However a small diameter cylinder is then required, and in result, higher oil pressure up to 10÷12 MPa, and even to 14 MPa in the case of high loaded CPPs.

Oil delivery under the pressure less than 4 MPa to the small-diameter propeller shaft interior has been effected with good results by using the simple radial seal shown in Fig. 2. Operation effectiveness of such seal has substantially dropped in the case of its installation in large-diameter shafts, particularly if a higher oil pressure has been applied. It has been caused by the seal elastic deformation, shown in Fig. 3., under the pressure of the oil contained in the gap between the shaft and sleeve.

The disadvantage is removed from a novel design of the unloaded radial seal developed by LIPS and being now applied also by other CPP manufacturers. In the not-unloaded seal the oil pressure acts only on the internal sleeve surfaces, but in the novel unloaded seal it acts, due to the applied casing, also on the outer sleeve surfaces delimited by rubber rings. In result the resultant radial loading of the seal and seal deformations are substantially reduced.

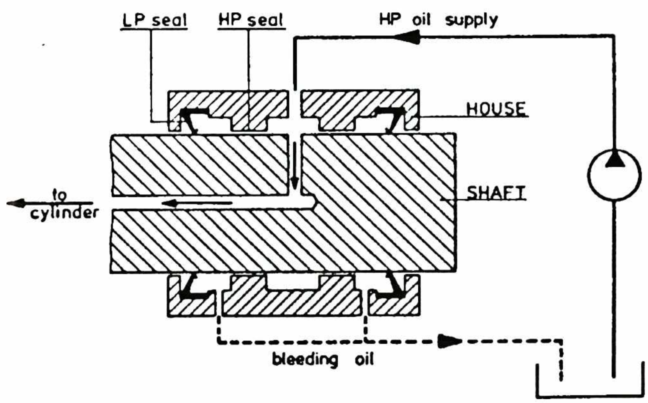


Fig. 2. The radial seal used in the CPP low-pressure installations

It seemed that the novel unloaded radial seal could solve the trouble and make applying a higher oil pressure possible. However its operational effectiveness and quality appeared unsufficient and many CPP manufacturers came back to the low-pressure installations. This was the reason for initiating research works on the seals in question, conducted in the Technical University of Gdańsk and ordered by ZAMECH Engineering Works, Elbląg, in the 1970s.

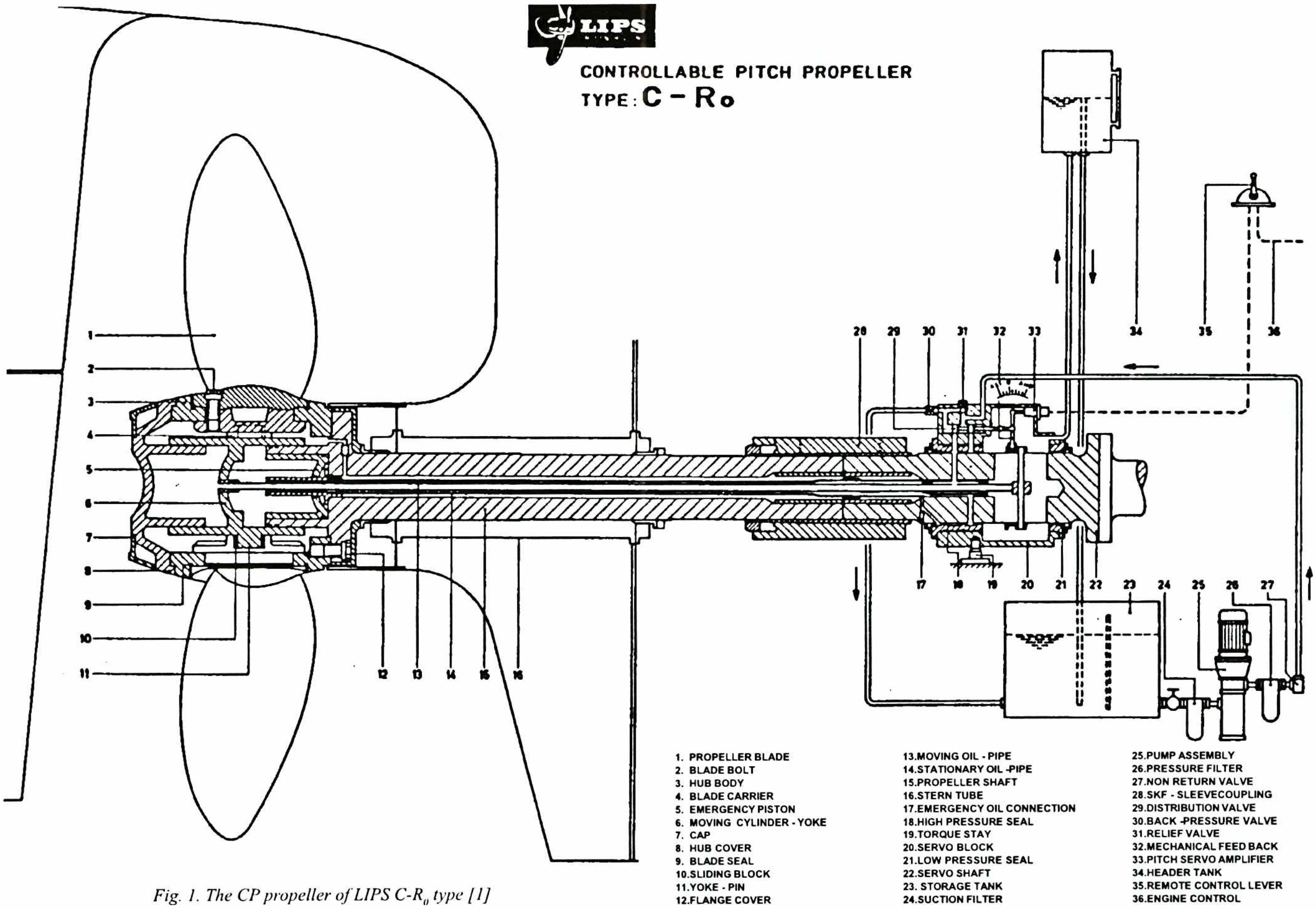


Fig. 1. The CP propeller of LIPS C-R_0 type [1]

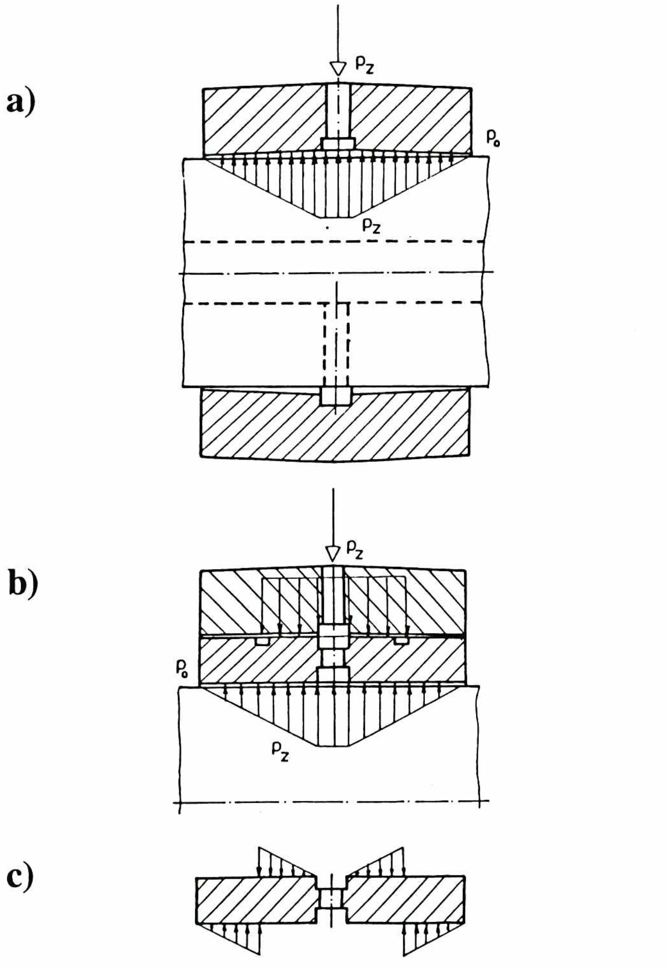


Fig. 3. Unloding principle applicable to radial seals
a) not-unloaded seal, b) unloaded seal, c) resultant pressure distribution over sealing sleeve

The research was carried out with the use of a full-scale seal installed on a laboratory stand specially arranged for the purpose. The tests revealed that wrong work of many seals was caused because of the disadvantageous dynamic phenomena which appeared in the oil film in the gap between the rotating shaft and bearing sleeve. It was also observed that the operation quality and effectiveness of the seal could much change if some its parameters, especially the design ones, are changed. For instance, the same seal can operate, depending on the position of sealing rings , in a stable and relatively effective way or in an unstable way with the oil leakage and power consumption even a dozen or so times greater than those observed in the former case. This is caused by different transverse loading of the sealing sleeve and thus its elastic deformation and bearing interspace shape which very much influences hydrodynamic phenomena in the oil film.

The bearing interspace shape depends also on temperature distribution over seal elements, especially that over the sealing sleeve, which is influenced by such parameters as: shaft speed, shaft-sealing sleeve design clearance, oil viscosity, and rate of the oil flow through the bearing interspace.

Results of the research in question made it possible, apart from providing some fundamental answers, to determine a seal unloading value which could ensure the stable seal operation under assumed conditions. Moreover the results were successfully used for the verification of a computer program for determining design parameters of such seal [5, 6].

It was found, however, in the course of the further investigations connected with attempts to improve the theoretical model and apply such seals to installations other than CPPs , that it is possible to increase their operational effectiveness and simultaneously lower their gabarites and power consumption. To confirm this, the empirical investigation on the influence of wide changes of sealing sleeve transverse stiffness on seal operation were undertaken by the research team of the Faculty of Ocean Engineering and Ship Technology in 1994.

The investigations were to be carried out in the natural scale due to the specific nature of the phenomena which accompany the seal operation. Measurements of the characteristic operation magnitudes of the seal in changeable conditions were performed on two sealing sleeves with different wall thickness (different transverse stiffness) only.

RESEARCH STAND DESCRIPTION

The investigations were carried out in the Faculty laboratory with the use of the stand described in detail in [3].

The stand makes it possible to test the seals under load of different levels and up to 16 MPa oil pressure, fitted on the shaft of 600 mm diameter, that rotates with 3.5 rev/s under transverse vibrations of 0÷0.5 mm amplitude and the frequencies equal to the first four multiplications of shaft speed.

The stand consists of :

- test bed
- hydraulic supply unit
- steering and operation control unit

zontal ends of the rockers 9 cooperate with the links 25, which, beared in their lower part on the excentric pins of the vibration generator shaft 21, operate like connecting-rods, forcing the rockers to rock and thus generating horizontal transverse vibration of the shaft.

The way, in which different levels of seal unloading can be achieved, is illustrated in Fig. 5 showing a fragment of the seal only. The sliding sleeve and casing are mutually sealed by means of eight, symmetrically placed, rubber rings of „O" type. Different unloading levels, characterized by an appropriate pressure distribution over the outer surface of the sliding sleeve, were obtained by connecting the relevant slots and oil supply pipes with the use of the distribution valve.

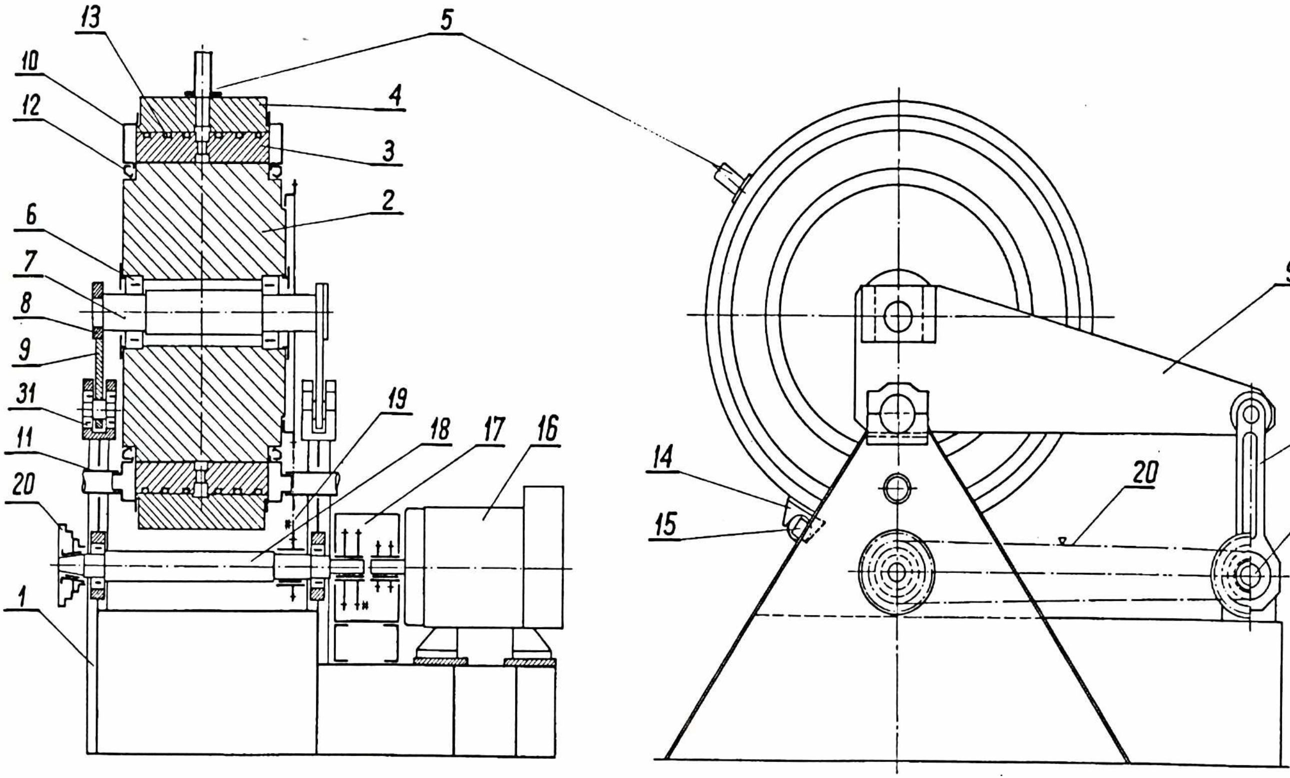


Fig. 4. Test bed of radial seal research stand

The test bed, shown schematically in Fig. 4, is fitted in the common frame 1. The sleeve 3 together with the casing 4 which is supplied with oil under pressure through the pipe 5, is slidewise-beared on the journal 2 which imitates a propulsion shaft segment. The side covers 10 fixed on both faces of the casing 4 are provided to collect the oil which flows out of the gap between the shaft 2 and the sliding sleeve 3 and to discharge it through two pipes 11 to the oil tank.

The shaft 2, beared on the non-rotating axle 7, is driven through two chain gears 17 and 19 by the DC electric motor 16 with the Ward-Leonard system. The casing 4 is secured against rotation and logitudinal displacement by means of the cantilever 14 and the transverse beam 15.

The propulsion is transmitted from the shaft 18, located in line with the electric motor axis, to the vibration generator shaft 21 by means of the belt gear 20 with four wheels of different diameters. Transmission ratios of the chain gear 19 and belt gear 20 are selected in such a way as to be able to obtain the rotational speed ratios of the vibration generator shaft 21 relative to the shaft 2 equal to 1:1; 2:1; 3:1; and 4:1, which correspond to four fundamental vibration frequencies of the propeller shaft.

The axle 7 is firmly fixed in the upper arms of the rockers 9 which are self-aligning supported on the frame 1. The hori-

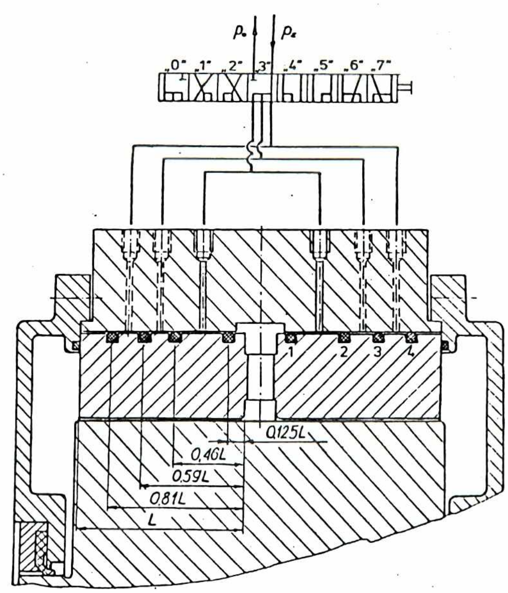


Fig. 5. Realization scheme of eight different unloading levels in the tested seal

The hydraulic supply unit shown in Fig. 6 is of two circulation paths : main and auxiliary. In the main circulation path the oil is sucked by the variable output radial pump 3 from the tank 1 through the filter 2 and pressed through the pipe equipped with the flowmeter 7 into the tested seal. From here it is then discharged through the triple-way cocks 9 either to the main tank 1 or to the measurment tank 8. The safety valve 5 and overflow valve 4 for pump pressure regulation are paralelly connected to the pressure pipe of the pump.

The auxiliary circulation path, provided to cool the oil, is equipped with the gear pump 11 and water cooler 12. Its cooling capacity is regulated by changing flow rate of cooling water.

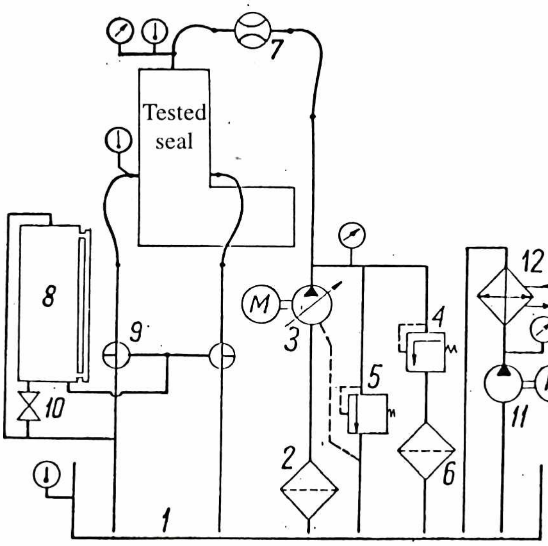


Fig. 6. Hydraulic supply unit scheme

The steering and control unit of the stand provides electric supply and control of the stand motors, and contains:

● the Ward-Leonard system with the DC electric motor steering console

● the steering unit with the star-delta switch for two other asynchronous motors.

MEASURED MAGNITUDES

The following magnitudes can be measured with the use of the research stand in question:

- oil pressure
- temperature
- sleeve displacement against shaft
- oil flow rate
- shaft rotational speed.

Number of possible measurement points as well as measurement and recording method is presented in detail in [3, 4].

Only a part of the measurement points, shown in Fig. 7 and 8, were used during the reported investigations.

The following magnitudes were measured:

● the oil pressure at four circle-denoted points, namely: one point at the sealing sleeve input and three points within the bearing interspace, located at I, II, III planes. The measurements were carried out with the use of manometers

● the oil temperature at the seal input and side cover output, measured by means of EMT-01 device with TP01A gauges which were NiCr- and NiAl- wired thermocouples

● the oil flow rate through the bearing interspace measured with the use of the measurement tank because of a low value of the flow

● the shaft rotational speed mesaured by means of a photo-electric system.

SCOPE AND COURSE OF THE INVESTIGATIONS

The investigations were aimed at the determination of the influence of sealing sleeve transverse stiffness change on seal operation quality (stability) and effectiveness.

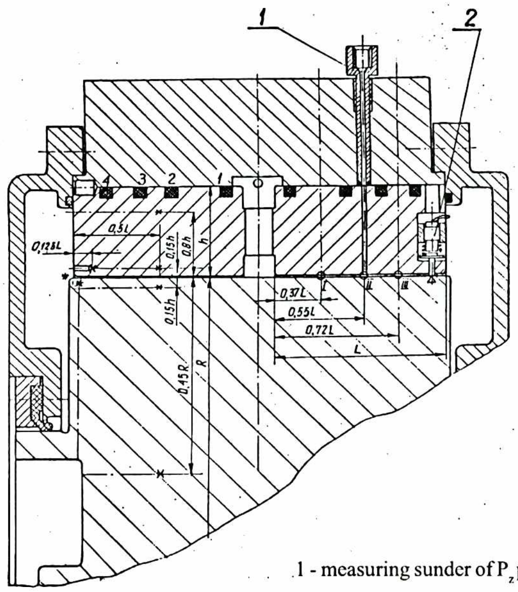
The investigations consisted in measuring of the oil pressure distribution within the bearing interspace and of the rate of oil flow through it, as well as the power of friction in two sealing sleeves of the following characteristic dimensions:

a)	b)
D = 600 mm = 2R	D = 600 mm = 2R
L = 120 mm	L = 120 mm
h = 60 mm	h = 17.5 mm

The measurements were comparative and carried out possibly for the same values of the seal operation characteristic quantities, namely:

- unloading levels
- the seal supply oil pressure p_z
- the seal supply oil temperature t_z
- the shaft rotational speed n

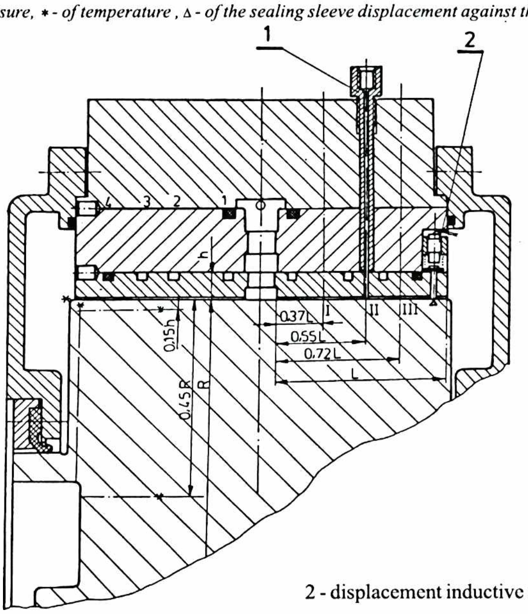
a)



1 - measuring sunder of P_z pressure

Fig. 7. Axial cross-section through the investigated seal with the sleeve a
Measurement point denotation:
o - of pressure, ∗ - of temperature , ∆ - of the sealing sleeve displacement against the shaft

b)



2 - displacement inductive sensor

Fig. 8. Axial cross-section through the investigated seal with the sleeve b
Measurement point denotation:
o - of pressure, ∗ - of temperature , ∆ - of the sealing sleeve displacement against the shaft

Beneath are given results of the measurements performed for the most characteristic load level „3" (see Fig. 5), in which the sleeve outer surface under seal supply oil pressure was the largest out of those available in the seal. In this case the resultant transverse load causes the largest elastic deformations in the direction of the possible seal-to-shaft clamping, and in result, of the decreasing of the bearing interspace height. Possible hazards of full seal-to-shaft lockup and seal seizure should have been revealed just during this test.

Moreover, the character of the elastic deformations, which cause decreasing height of the bearing interspace in the oil flow direction, is that favourable which the investigations should be focused at, because it ensures the advantageous, stable operation of the seal at almost co-axial mutual position of the sleeve and shaft.

The tests performed on both sleeves fully confirmed the expectations. The only exception was the starting operation phase of the seal with the thick-walled sleeve at the low oil supply pressure p_z = 1 MPa when an unstable work appeared with the characteristic, slowly declining rotary motion of the sleeve centre relative to the shaft centre, with the rotational speed twice smaller than that of the shaft. The phenomenon is described in detail in [4]. An oil supply pressure increase caused the sudden decay of the motion and stable work of the seal. If the oil supply pressure dropped again to 1 MPa value, with the shaft kept rotating, the seal operation maintained stable. In consequence, the measurements were continued without any need of diplacement recording. The use of the shaft transverse vibration generator which made environmental nuisance, was also given up as its influence on the seal operation was negligible.

Results of the oil pressure measurements in the bearing interspace are presented in Fig. 9 and 10. It can be observed that the thick-walled sleeve stiffness was large enough to cause almost uniform pressure dropping over the length of the interspace. It means that the interspace height was changed in a very moderate and almost linear way, an oil temperature increase within the bearing interspace accounted for.

A completely different situation was found in the case of the thin-walled sleeve where the pressure drop over the interspace length appeared highly changeable. It evidenced that the bearing interspace axial profile was very complicated with convergent and divergent segments in the oil flow direction, however with the convergent resultant profile.

It seems that, in the latter case, the design shaft-sleeve radial clearance much less influences the seal operation quality, and especially its effectiveness. However, a larger influence of the supply oil temperature and shaft rotational speed on the oil pressure within the bearing interspace was observed in its regions of high pressure gradients. This can be justified as both the quantities influence the temperature distribution over the sleeve and thus its thermal deformations and the final bearing interspace shape and, in consequence, the oil pressure distribution over its length. Results of the oil flow rate measurements within the bearing interspace of both sleeves are presented in Fig. 11. The effectiveness of the seal with the thin-walled sleeve *b* is several times higher than that of the seal with the thick-walled sleeve *a* . It is characteristic that the flow rate of the leaking oil becomes steady at the high supply pressure value p_z > 10 MPa , however at quite different flow rate levels in each case.

Smaller differences between the values at smaller values of the pressure p_z can be explained by smaller elastic deformations of the seals whose internal diameters are practically identical and therefore their interspace heights and flow conditions become more and more similar to each other as the pressure p_z is lower and lower.

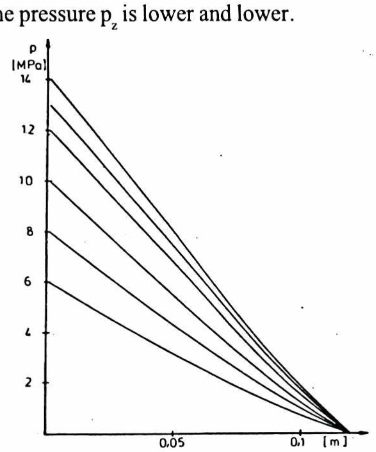


Fig. 9. Oil pressure distribution within the bearing interspace of the seal with the thick-walled sleeve a
Applied oil: Hydrol 30; shaft speed: n = 1.5 rev/s ; supply oil temperature: t_z *= 35 °C*

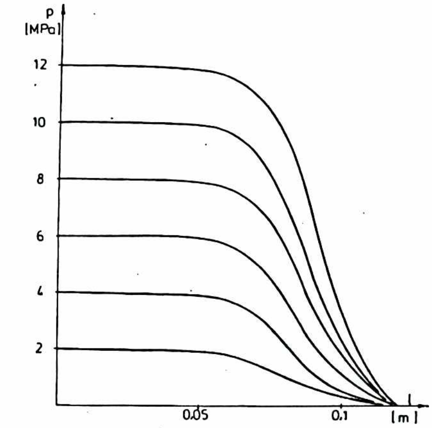


Fig. 10. Oil pressure distribution within the bearing interspace of the seal with the thin-walled sleeve b
Applied oil: Hydrol 30; shaft speed: n = 1.5 rev/s ; supply oil temperature: t_z *= 35 °C*

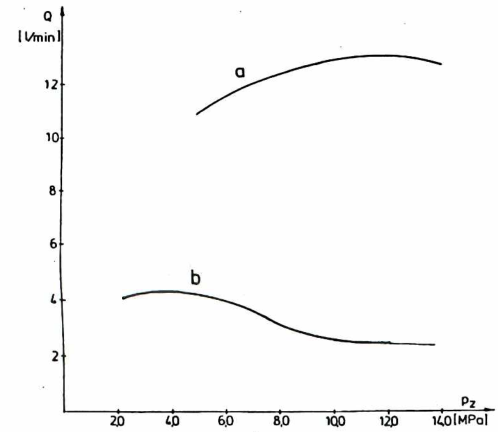


Fig. 11. Flow rate of the oil leaking out of the seal with the thick-walled sleeve a and that with the thin-walled sleeve b
Applied oil: Hydrol 30; shaft speed: n = 1.5 rev/s ; supply oil temperature: t_z *= 35 °C*

The investigations did not cover long operation periods of the seal with the thin-walled sleeve. The continuous operation time during consecutive measurement sessions was 70 min due to waiting for oil to get warm. The temperature of Hydrol 30, seal supply oil, reached 45°C. During all the measurment periods the stable, quiet operation of the seal was observed without any signs of scuffing or even unjustified raising of the electric current of the seal shaft driving motor. Each time, just after switching-off electric supply of the motor, the shaft kept rotating for some time until it gently stopped.

CONCLUSIONS

The investigation results clearly confirmed that the assumed direction of the reseach and development works on the seals in question was correct.

- The seal of the same design, the same design radial shaft-sleeve clearance, but with the thin-walled sleeve, operated in the same working conditions with the oil leakage four times smaller than that revealed by the version with the thick-walled sleeve. It should be mentioned that the transverse stiffness of the sealing sleeves, usually applied to the similar seal designs, greatly exceeds that of the investigated thick-walled sleeve, however a tendency to lower the stiffness can be noticed.
- The advantageous feature of the seals, expected already in [3], which practically eliminates the hazard of locking-up the sleeve on the shaft, if only the seal is correctly designed, was very clearly manifested here. It consisted in that the increase of oil supply pressure and - connected with that - sleeve elastic deformations towards sleeve locking-up on the shaft, caused an increase of oil choking and friction power as well as of amount of the heat generated in that interspace region. In result, the pressure distribution over the interspace turns to more convex one. The higher the pressure and the lower the sleeve

transverse stiffness, the more close to rectangular the interspace pressure distribution, i.e. to that applied to the sleeve outer surfaces, which can be observed in Fig.10.

● The increase of the heat amount generated in the region of the interspace constrictions causes an increase of thermal deformations tending to enlarge the interspace height, which is reflected by a convex inflection on the pressure distribution curve over the interspace end segment.

● The investigation results, however incomplete, allow to expect an additional important advantage of the seals in question, especially those with the thin-walled sleeve. It could consist in a relatively low influence of the design clearance on seal operation effectiveness, mainly at high oil supply pressure values. The influence is higher at lower pressures; it is justified and can be utilized for developing a new generation of radial seals.

● To make it possible, however, the initiated investigations should be continued within the earlier presented research scope as soon as possible to take advantage of the preparatory works already performed on the test stand. Their future results could make it possible to verify and upgrade design calculation methods of the seals.

● The results of the hitherto carried out research works on the seals in question have been put into practice, mainly to the CPPs manufactured by the ABB Zamech Engineering Works in Elbląg.

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

Current reports

TECHNICAL UNIVERSITY OF GDAŃSK
FACULTY OF OCEAN ENGINEERING
AND SHIP TECHNOLOGY

WCS - 300 microprocessor control system for ship trawlers

Fish trawling by the use of the twin trawl system is about 1.5 to 2 times more effective than that by using the single trawl at the same fishing ship propulsion power. The twin trawl system requires however an appropriate synchronization of laying-out and hauling-in the trawl lines. It can be attained if their instantaneous length is known and laying-out and hauling-in velocities are controlled. This is the task of the WCS - 300 system.

The device was designed by the team of the Chair of Ship Deck Equipment and Systems, Technical University of Gdańsk.

Trawl line length and its laying-out or hauling-in velocities are measured by counting the impulses from proximity detectors fixed on the trawling winches, with the characteristics of the installed lines and winches, saved in the device memory, accounted for. The device design makes the system applicable to any ship after easy and inexpensive installation of the transducers which are inherent elements of the device.

Additionally, the WCS - 300 system enables to determine current ship engine load and its fuel consumption rate by measuring the fuel rate and speed of the main engine. Any limit value exceedance is signalled by digital indicators.

The WCS-300 system is applicable to:

- optimizing the fish catching manoeuvre
- increasing the trawling line durability
- increasing catch effectiveness by trawl shape control
- optimizing main engine fuel consumption
- prevention against main engine overloading.

Roman Lutowski, M.Sc., and Bolesław Pempera, M.Sc., are the authors of the system.

TECHNICAL UNIVERSITY OF SZCZECIN
FACULTY OF MARITIME TECHNOLOGY
OCEAN AND SHIP TECHNOLOGY INSTITUTE

Software systems for preliminary design and parameter assessment of ships and off-shore units

Computer software systems for multi-variant parametric design analysis which make selection of the best solution possible are the basic tool used today by ship and off-shore unit designers.

Several such software systems were elaborated at the Ocean Engineering and Ship Design Department, Technical University of Szczecin. They can be shortly described as follows:

● WOWM software system - for preliminary assessment of ship seaworthiness. Input data are: ship parameters and wave spectrum of a sea area. It makes possible to calculate ship motion amplitudes for given waving parameters or analyze ship staying time in a given sea region

● MPJSZ program - design modelling of a mono-hull offshore supply ship. The program is based on the floatability, stability (acc. to IMO), freeboard, propulsion and powering and seaworthiness design criteria. It makes possible, apart from generating multi-variant design parameters, to calculate investment expenditures, operation costs and pay-back period

● ATPSZ program - for design parameter assessment of an off-shore semi-submersible supply unit. It comprises the floatability, stability, motion periods and propulsion power multivariant analysis of SWATH twin-hull, semi-submersible supply vessel

● MOST program - for design optimization of transport ships. It is suitable for preliminary design of special ore carriers, but its adjustment to universal bulk carriers is possible. The program comprises the floatability, stability (acc. to IMO), freeboard, propulsion and powering and capacity analysis. It makes also possible to calculate investment expenditures, operation costs and incomes.

The authors of the programs are: J. N. Siemionow, Prof., D. Sc., T. Szelangiewicz, D.Sc., K. Sanecka, M.Sc., and W. Chądzyński, M.Sc.