ISSN: 1231-4005
e-ISSN: 2354-0133
DOI: 10.5604/01.3001.0010.3129

DIAGNOSTIC OF THE PISTON ROD GLAND'S FAILURE OF MARINE SLOW SPEED MAIN ENGINE

Lech Murawski, Do Van Doan

Gdynia Maritime University
Faculty of Marine Engineering
Morska Street 81/87, 81-225 Gdynia, Poland
tel..+48-58-6901 481, fax: +48586901399
e-mail: lemur@wm.am.gdynia.pl
dodoan.vimaru@gmail.com

Abstract

The article presents untypical diagnostic of the slow-speed, marine main engine. The engine was newly made-up and checking during mounting process and during sea trial. However, a leak under piston chamber of second cylinder along the piston rod appeared few days after the sea trial. The stuffing box exchange (with the piston rod's regeneration) did not give expected results – the failure happened again. All geometrical and exploitation parameters was in acceptable range. The ship with main engine was fifth in the series of sister ships. The authors were asked for urgent expert opinion. Mix of different measurements was planned after formulation several hypothesis. Displacements of piston rod and cylinder, a stress level of main engine body, and vibrations level in different points of main engine was performed during short sea voyage. Not a single hypothesis can be considered as the main cause of the failure. It turns out, that a serious failure need not be caused by a single reason. According to snowball theory a sum of small effects, each of them affecting slightly the engine operation, can be a cause of serious failure. In the authors' opinion, the failure was caused by a sum of relatively slight effects. The probably scenario of the failure process was enunciated. The recommendation for the engine project was formulated: some of the geometrical tolerances should be changed.

Keywords: *marine propulsion systems, slow-speed main engine, failure of piston-crankshaft system, diagnostic based on vibrations, displacements and stresses level*

1. Introduction

The authors were asked for urgent expertise for finding the cause of failure of piston rod's gland (stuffing box) in the engine 7 S70 MC-C type [6]. The engine was installed in newly built container ship; the fifth in the series of sister ships. A leak in under piston chamber of second cylinder along the piston rod appeared few days after the sea trials during first commercial voyage. It has been found, as the result of visual inspection, that some material of the gland housing was pressure pad welded on the piston rod along the length of about 1.8 m. The piston rod was grinded off on the depth of 4 mm of the gland housing, at the exhaust (port) side of the engine. Since nominal radial clearance of the piston rod with respect to the gland, housing is equal to 2 mm [6]. It is mean that the displacement of the piston rod towards the exhaust side had to be about 6 mm!

Exchange of the stuffing box and regeneration of the piston rod by the ship crew did not give expected results – the failure happened again. After the successive exchange of defective parts, the piston-crank system (including the crosshead) and geometry of its motion were checked by rotation with use of the turning gear (quasi-static test). No deviations from technical documentation have been found; only cracks in the pads of side stopping blocks of the engine were found near cylinder No. 2, as well as displacement of the engine housing by 0.5 mm to port side. An independent expert opinion was asked, due to the lack of clear proof that the defect in engine foundation is the cause or result of the failure, as well as of doubts whether this defect has a significant effect for the engine operation parameters.

2. Hypotheses of the failure cause and its verification conceptions

Upon detailed analysis of the phenomenon, a number of the most probable reasons of the failure have been singled out [3] and then a program of measuring tests has been prepared for verification of advanced hypotheses. The following working hypotheses for piston rod gland failure have been assumed:

1. Cracked A-frame of cylinder system No. 2 and asymmetric foundation of crosshead slide bearing, activated at increased load of the engine [5].
2. Deformation of piston-crank system due to considerable difference between the tensions in side bracings of the engine body.
3. Defect or deformation of crosshead slide-ways.
4. Thermal displacement of the lower edge of cylinder liner due to its inhomogeneous structure.
5. Insufficient rigidity of ship's double bottom or engine foundation [4] due to cracks near cylinder system No. 2.

Standard scope of engine monitoring in running-in process includes check of main and big end bearings, valves, fuel valves (exhaust gas temperature) etc. [2]. It has been assumed that the failure could be due to a defect arisen during the ship construction or engine assembling or engine running-in process. All the more reason, the engine working point was set up at "L1" – maximum rating and rpm for the engine type in question [6]. Typical serious design error (such as bed foundation of the engine) has been excluded due to the fact that similar failures did not happen in a dozen or so engines of this type installed on similar ships.

It has been assumed that measuring tests would be carried out in the function of engine rpm, on the ship during sea voyage [1, 4]. Two series of measurements have been carried out with partly loosened bracings (former condition of the engine support) and with fully tight bracings (according to the manufacturer recommendation). The following measurements was planned to be performed:

- stresses (measured along the vertical axis) in the both A-frames (fore and aft) for cylinder No. 2 at the exhaust side [5];
- displacement of lower edge of cylinder liner for cylinders Nos. 2 and 6 (reference cylinder);
- displacement of piston rods near the glands of Cylinder Nos. 2 and 6;
- vibration velocity of the engine body at the height of cylinder heads in the first and last cylinder [4];
- vibration velocity of the engine crankshaft and foundation near the stopping blocks at six points along the engine envelope [4].

An arrangement of the measuring points is presented in Fig. 1 [6]. Engine rpm was measured at the same time as the above-mentioned values in order to enable accurate analysis of the measurement results. All the measured values were digitally recorded with use of analogue-digital card. The stresses were measured with use of strain gauge method by means of 16-channel bridge HBM. The displacements were measured by means of contact sensors HBM. The vibration velocity was measured by means of portable meter B&K with piezoelectric sensors.

The ship in question is equipped with right-handed main engine. Therefore, the port (exhaust) side of the engine is loaded more when the ship is going forwards. The strain gauges glued along vertical axis are enabling to measuring the bending stresses of the A-frames. The measured stresses would enable to verification of hypotheses Nos. 1 and 2. In case of broken A-frame a significant difference of stresses is expected (particularly their dynamic components) between the "fore" and "aft" A-frame. The significant differences between the dynamic components of measured stress, with loosened and tighten transversal bracings, would confirm hypothesis No. 2.

In the 7-cylinder engine, cylinder sets No. 2, 6 are symmetric, and their loads are similar. The measurement of displacements of piston rods for defective and correct operation conditions would enable assessment of the hazard of repeated failure after carried out repair. Moreover, this

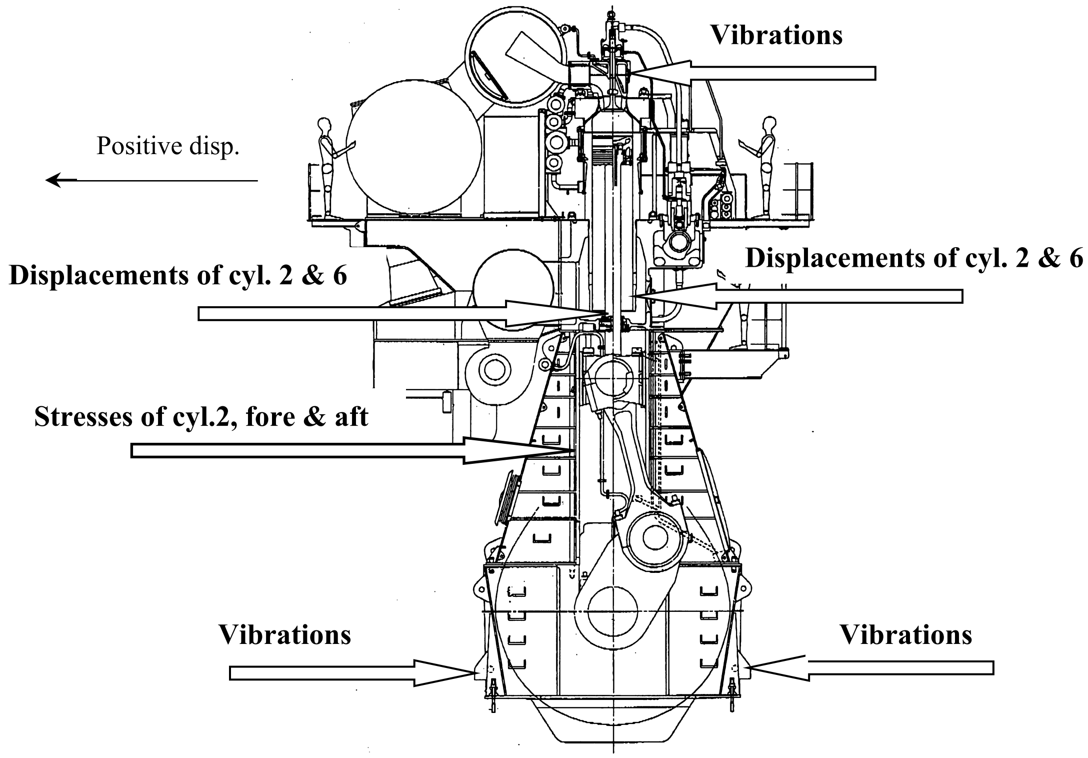

Fig. 1. An arrangement of the measuring points

measurement will show possible defect in the crosshead guiding – hypotheses Nos. 1-3. The measurement of dynamic displacement of cylinder liner would enable verification of hypothesis No. 4. Possible defects of engine foundation will be shown by the measurements of vibration velocity (hypothesis No. 5). Insufficient rigidity of the engine foundation will be revealed by excessive vibration level at the height of cylinder heads. Possible crack of double bottom, foundation or engine bed would be revealed by locally increased vibration near the engine stopper blocks. Dynamic micro displacements between the engine bed and its foundation (difference in vibration velocity) would allow to assess the significance of stopper block plays for operation of piston-crank system and to tell whether cracked pads of stopper blocks are the cause or result of the analyzed failure.

3. Analyses of the measurements

The diagram of exemplary stresses in the engine "aft" A-frame, in operation with loosened transversal bracings, is shown in Fig. 2. The measured values, marked with points, have been approximated by means of third degree polynomials. The presented values are defined by the equations (1) and (2). The measured values of type "*Ampl.*" are responsible for dynamic loads of the structure, whereas the values of type "*RMS*" are equivalent quasi-static load.

$$Ampl. = \frac{MAX - MIN}{2}, \tag{1}$$

where:
MAX – maximum value recorded for given rpm of the engine,
MIN – minimum value recorded for given rpm of the engine.

$$RMS = \frac{1}{n} \cdot \sum_{i=1}^{n} k_i , \tag{2}$$

where:
n – number of samples of the measured value,
k_i – value of measured signal.

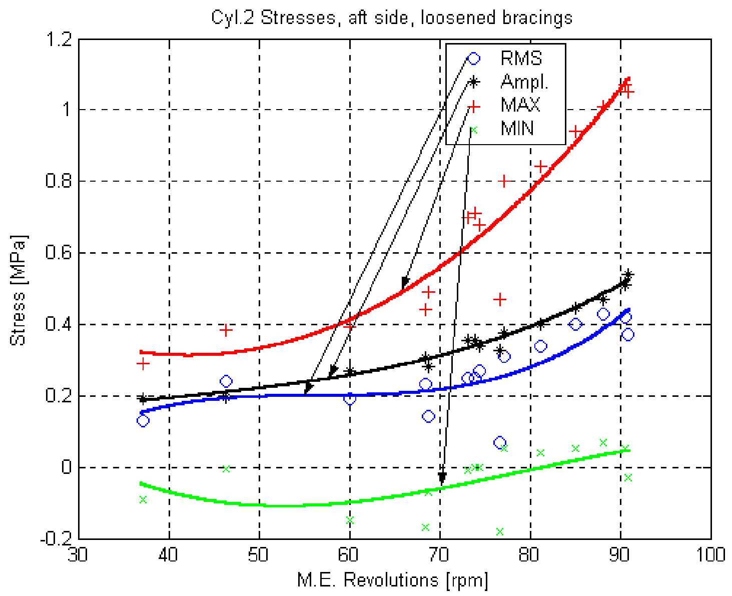

Fig. 2. Stresses in aft A-frame, in operation with loosened bracings

The results of stress measurements for the both measuring points ("fore" and "aft" of A-frame), as well as for the both series of measurements (upper engine bracings loosened and tight) are shown in Fig. 3 and 4.

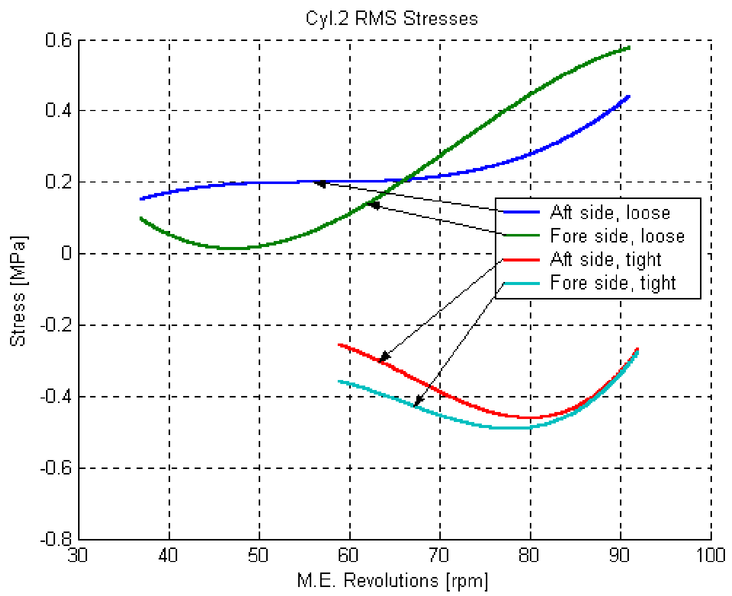

Fig. 3. RMS type stresses in the engine A-frames

The stresses level in the engine A-frames is low. The hazard of A-frame fissure may happen only in the event of serious material defect, defective welding process or serious assembling error. Such events are little probable for the engines made by recognized manufacturers and installed by recognized shipyards. Similar stresses level for the both measuring points for dynamic component and for RMS proves that the A-frames are not defective. The level of dynamic loads increases with

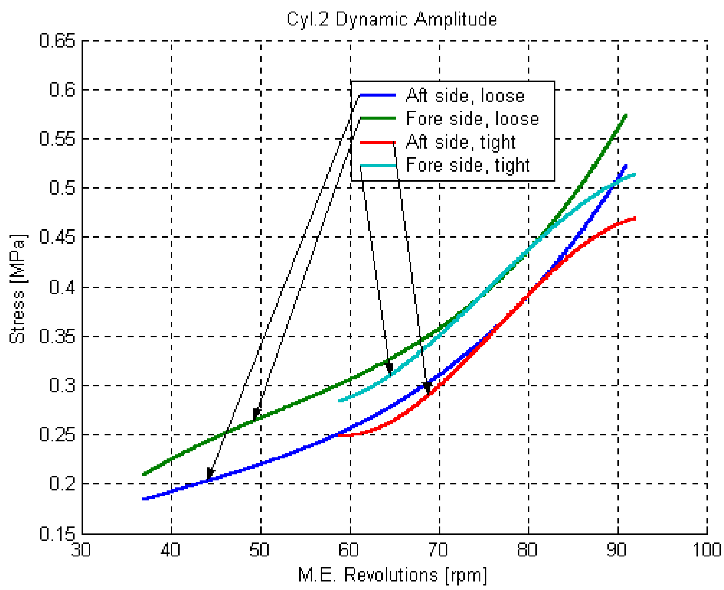

Fig. 4. Ampl. type stresses in engine A-frames

the increase of engine load, whereas the characteristics of RMS stresses are flat. No abnormal stress distribution has been observed in A-frames. Tensions in the transversal bracings are a source of small changes of quasi-static stresses in the engine body. These phenomena might be a cause of small changes in geometry of piston-crank system motion. However, it cannot be substantial cause of the analyzed failure, because the level of dynamic stresses in relation to the engine load does not change for tighten upper transversal bracings.

An exemplary (for cylinder No. 2 with loosened upper bracings) changes of displacements of cylinder liner lower edge is shown in Fig. 5. For each measurement, the displacements are small – not exceeding 0.3 mm. Constant value of dynamic component of the displacement, poorly dependent on engine rpm shows that the loads of piston-crank system are not transferred to the cylinder liners. All transverse loads are transferred by the crosshead system. The quasi-static displacements increase with increasing rpm of the engine. Displacements of the cylinder liners cannot be the cause of the analyzed failure.

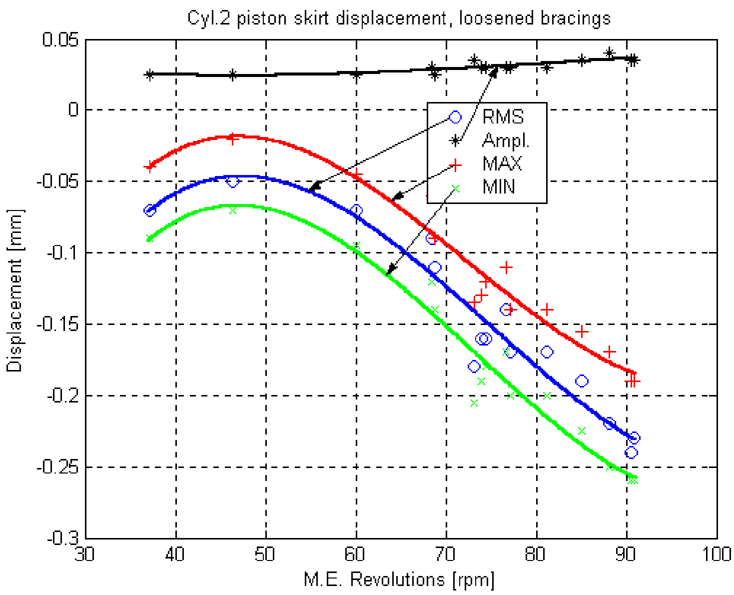

Fig. 5. The displacements of cylinder liner No. 2 (M.E. running; loosened bracings)

Maximum vibration velocity of the engine body at the height of cylinder heads (for all variants of measurement) does not exceed the following values: 7.0 mm/s for transverse direction, 4.2 mm/s for longitudinal direction and 3.4 mm/s for vertical direction. According to Germanischer Lloyd Rules the allowed vibration velocity amounts to 14 mm/s. Distribution of vibration velocity around the engine bed and foundation is constant and does not depend on the measuring point. The following maximum vibration velocities have been recorded: 2.5 mm/s for transverse direction, 1.3 mm/s for longitudinal direction and 2.0 mm/s for vertical direction. Allowed vibration velocity for this type of structure is 6.2 mm/s. Low vibration velocities of the engine body prove its sufficient rigidity and correct foundation within the double bottom. No micro-movements have been observed between the engine bed and foundation. There are no reasons for suspicion of defect in the engine bed or in double bottom structure. Cracks of stopping blocks and small displacement of the engine body are rather the effect and not the cause of the failure. Displacement of the engine body in accordance with direction of the greatest load of the crosshead and with location of the gland defect (to the port side) reduced slightly the effect of the failure.

The comparative measurements of piston rod displacement show that the results for failure-free cylinder No. 6 are two times worse than those measurements for cylinder No. 2 after replacement of the elements. The displacement of piston rod No. 6 was near to 1 mm (with tight upper bracings). The displacements changes of piston rod No. 6, with loosened upper bracings, are shown in Fig. 6. The dynamic displacements of the both analyzed piston rods with loosened upper bracings are shown in Fig. 7.

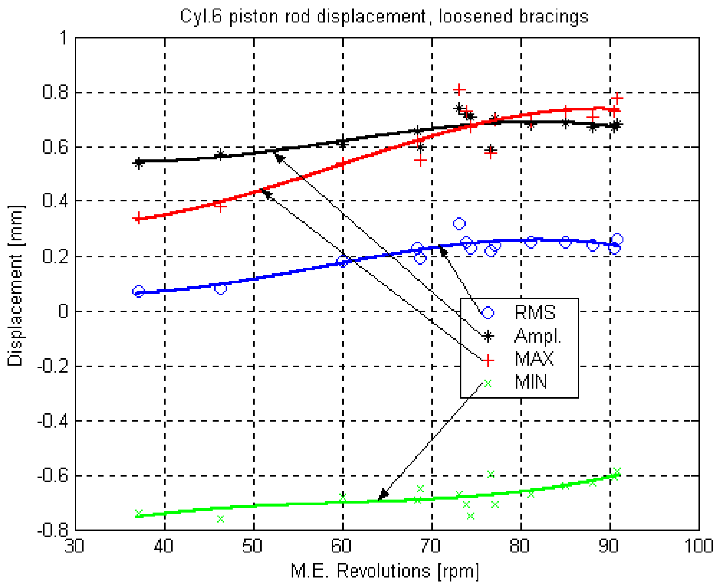

Fig. 6. The displacement of piston rod of cylinder No. 6 with loosened bracings

There are considerable transverse displacements of piston rods in the region of under piston space. Moreover, these displacements differ considerably from each other (cylinder No. 2 and 6). After replacement of gland and crosshead of cylinder No. 2 (by standard spare parts), the system condition became even better in comparison to that of free of damage system No. 6. In the present condition, there is no risk of failure and the engine can be operated without limits.

Exact analysis of dynamic displacements of the piston rods showed that the first harmonic component (once per revolution) is the dominant one. The displacements increase relatively slightly with increased engine load (Fig. 7). The values of analyzed displacements do not depend on the engine operational parameters. The decisive factor is summary of geometry of piston-crank system, including tolerances of manufacturing and assembling of individual parts. In correctly functioning system of cylinder No. 6, the displacements are going even to ±0.7 mm. The standard

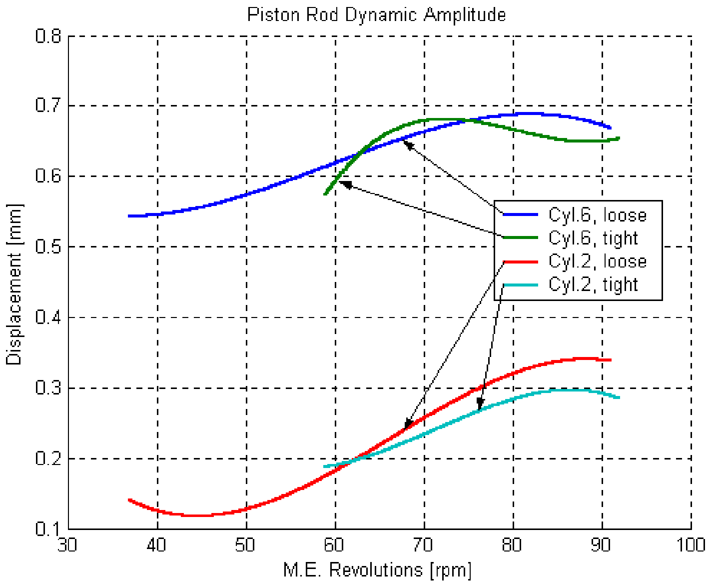

Fig. 7. The piston rod dynamic displacements

backlash between the piston rod and gland body is 2 mm. Unfavorable summary of tolerances of correctly made elements of piston-crank system is hardly probable, but possible. According to theory of great numbers, the probability of analyzed failure is real for suitably great number of manufactured engines.

4. Conclusions

No one of the presented diagrams shows the increases of the measured values due to torsion vibration resonance. That means that there is no strong coupling between the power transmission system and operational parameters of gland-piston rod system.

Not a single hypothesis presented in Section 2 can be considered as the main cause of the failure. It turns out that a serious failure need not be caused by a single reason. According to snowball theory a sum of small effects, each of them affecting slightly the engine operation, can be a cause of serious failure. In the authors' opinion, the failure was caused by a sum of relatively slight effects. It seems that design backlash between the gland body and piston rod (2 mm) should be increased. This would not affect significantly correct operation of the engine. The engine manufacturer says (after repeated check), that tolerance ranges of the defective crosshead are in accordance with the documentation. In the authors' opinion, the tolerance of perpendicularity of surfaces joining the crosshead with piston rod was probably at its limit value. The coincidence of manufacturing and assembling tolerances of other elements of piston-crank system could be also so unfavorable, that the global clearance of the gland became dangerously reduced. Loosening of the upper bracings of the engine bode did not increase displacement of the piston rod (see Fig. 7); it caused, however, a slight static deformation of the engine body (see Fig. 3) and reduced the clearance between the gland and piston rod. The clearance was additionally reduced as the result of thermal deformation of the cylinder liner (within allowed limits). Additional, slightly greater than standard values, deformations of crosshead slide-ways could be caused by relatively strong load of the engine (working point "L1"), as well as by relatively flexible ship hull (container carrier with engine room located in the aft part of the ship).

In the authors' opinion, the course of failure was as follows: after running down the clearance between the gland body and piston rod (for above-mentioned reasons), the gland body material (cast iron) became gradually worn out by steel piston rod. The heat produced as the result of this process increased the piston rod deformation and intensified wearing out of the material. Positive feedback of this phenomenon was amplified by flexural buckling of the piston rod (transverse displacement) caused by non-coaxial action of gas and mass forces. Additional factor amplifying the gland body wearing process was the change in piston rod geometry (increase of cross-sectional surface area) caused by pressure welding of the gland body material to the piston rod. The failure could by noticed by ship engineers only when a leak appeared in the under piston space (when the process was already considerably advanced).

References

[1] Charchalis, A., Grządziela, A., *Diagnosing the shafting of alignment by means vibration measurement*, ICSV Congress (electronic ver.), Garmisch-Partenkirchen 2000.
[2] Deven, A., *Marine Diesel Engines*, August 17, pp. 45-48, 2004.
[3] Matthias, A., Leo, S., Thomas, W., *Sulzer TriboPack*, Service Experience Report, Wärtsilä Corporation, May 2003.
[4] Murawski L., *Statyczno-dynamiczne charakterystyki pracy okrętowych układów napędowych i ich wpływ na drgania konstrukcji kadłubów i nadbudówek statków*, Zeszyty Naukowe Instytutu Maszyn Przepływowych Polskiej Akademii Nauk w Gdańsku, Nr 542/1501/2006, 191 str., Gdańsk 2006.
[5] Zhang, S., Yang, J., Li, Y., Li, J., *Identification of bearing load by three section strain gauge method: Theoretical and experimental research*, Measurement, Vol. 46, pp. 3968-3975, 2013.
[6] *Wärtsilä RT-flex96C and Wärtsilä RTA96C: Technology review and operation manual*, Wärtsilä Corporation, June 2009.