DEVELOPMENT OF A METHOD FOR DETERMINING THE SIZE OF CLEARANCE IN SLIDING BEARINGS

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Abstract:

Introduction/purpose: The purpose of this paper is to present the importance of applying a new diagnostic method with a monitoring system and its capability to reliably determine when and where the problem will occur related to the wear of plain bearings in further operation of the plant, to offer a quality assessment of how the system will continue to function over time, to predict causes of failures and how to eliminate them as well as to provide time for planned maintenance of technical systems.

Method:The new method solves the problem of sliding bearing diagnostics by measuring the dynamic trajectories (orbit) of the sleeve in the sliding bearing. Modern methods of technical diagnostics based on the measurement of dynamic parameters and electrical quantities and their analysis enable the support of measuring systems and measurement sensors from different manufacturers.

Results: Measurements of the dynamic trajectories of sleeves in sliding bearings determine the values characterizing: normal condition, initial clearance size, further clearance increase, bearing clearance sizes, the moment when the state parameters are close to the upper limit of the allowable clearance, and the specific clearance size when further plant exploitation can cause system failure.

Conclusion: The new diagnostic method and monitoring systems can be widely applied in all technical fields: internal combustion engines, hydroelectric power plants, thermal power plants, process plants, and in many other areas. Many monitoring systems, compatibility of devices and equipment of different manufacturers are supported by hardware and software. The calculations of theoretical and experimental dynamic parameters were verified. The method has a wide range of possibilities of application.

Keywords: sliding bearing, bearing clearance, bearing wear, bearing sleeve, dynamic trajectory.

Introduction

The new procedure and measuring systems (hereinafter referred to as monitoring systems) for determining the clearance size, i.e. the degree of wear of sliding bearings, is a new diagnostic method in the technical field. It is is characterized by the following: it allows direct measurement of the movement parameters of the sleeve in the sliding bearing, thus enabling automatic quality control of plants in which sliding bearings are built into. The monitoring system has an almost unlimited lifetime, has no transferable and wear-exposed elements, does not depend on the speed of rotation and overload of a plant, in the case of internal combustion engines, and does not depend on the engine type and stroke. The new method is used as a precautionary measure to prevent possible system crashes. The monitoring system can be built into machinery during production, so there is no limit if it is installed on a plant already in operation. Monitoring systems can be implemented in the on-line version, i.e. as a constant monitoring system or in the off-line version, i.e. as an occasional monitoring system, depending on the plant type and the requirements of the plant user (Černej et al, 1986).

Theoretical settings, dynamics of sliding bearings, and the calculation of dynamic parameters

There is a range of impacts such as defective lubrication, dirty and diluted lubricating oil, incorrect bearing geometry due to manufacturing and assembly errors, or large bearing deformations caused by dynamic forces during plant operation, which causes damage to sliding bearings. High temperatures in sliding bearings are most often due to:

- transient mixed friction with a longer duration, mixed friction in a larger zone along the bearing range or due to mixed friction of higher intensity,
- lack of lubricating oil for a certain period of time or a permanent lack of oil in the sliding bearing.

The causes of mixed friction may be different or a combination of multiple impacts, such as (Ličen & Zuber, 2003):

- mistakes in making liners and sleeves,
- small and large bearing clearances,
- excessive mounting or working deformation of the bearing,
- clogging of oil channels with dirt in the lubrication system,
- small lubricant pump capacity,

- transmission of dynamic forces, and
- filtration and cooling of lubricating oil.

Fatigue of the material in the sliding bearing occurs if the limit of the dynamic endurance of the bearing material exposed to cyclically variable pressures in the oil film is exceeded. Today's limits, with regard to fatigue and bearing wear, are (Černej et al, 1986):

- minimum oil film thickness obtained by calculation - 2 μm,
- maximum pressure in the oil film up to 250 MPa,
- maximum temperature in the oil film up to 140°C, and
- hydrodynamic specific friction force in the bearing up to 0.15 W/mm^2.

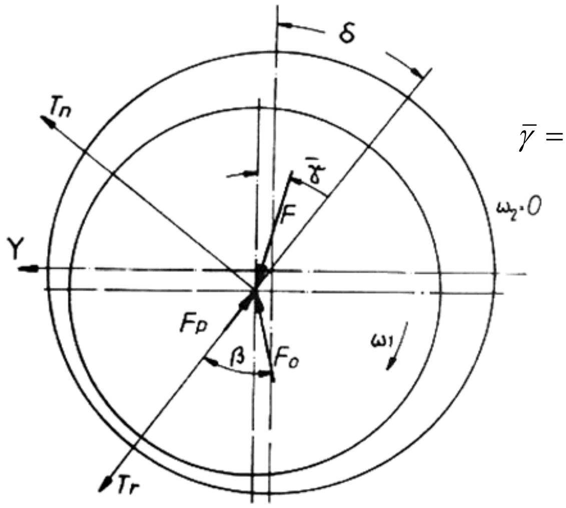
The bearing pressure of the bearing lubricating oil is determined by the bearing capacity characteristic and is in the range from 90° to 130° of the bearing circularity. In the rest of the clearance, the lubricating oil pressure is approximately equal to the oil supply pressure in the sliding bearing.

Knowledge of the parameters of the dynamic orbit of the sleeve in the sliding bearing leads to quality indicators about the relative performance of certain drive and design parameters such as: bearing dimension, the size of the clearance in the bearing, speed of rotation, viscosity of lubricating oil, and the like.

The dynamic orbit (trajectory) parameters of the sleeve in the bearing are determined by:

- the minimum thickness of the oil film, which is closely related to the bearing load,
- maximum pressure in the oil film,
- the friction magnitudes in the bearing (energy losses) depending on the bearing temperature regime, and
- the amount of oil flow in the bearing.

In order to theoretically obtain the dynamic trajectory of the sleeve in the sliding bearing, the input data requires a dynamic load force of the bearing and the angle at which it acts. The basis for obtaining the dynamic trajectory is based on the balance of the dynamic force (F) loading the sleeve and the hydrodynamic forces due to the pushing of the sleeve (Fp) and turning the sleeve in the bearing (Fo), (Figure 1), (Žegarac, 1989).



$$\bar{\gamma} = \gamma - 180$$

Figure 1 – Schematic presentation of a cylindrical sliding bearing and the equilibrium position of dynamic forces
Рис. 1 – Схематическое изображение цилиндрического подшипника скольжения и положение равновесия динамических сил
Слика 1 – Шематски приказ цилиндричног клизног лежаја и равнотежног положаја динамичких сила

Based on the balance of dynamic forces, the exact position of the sleeve motion in the sliding bearing is obtained at each moment of the plant cycle time.

For realistic operating conditions and different directions of rotation of the sleeve in the bearing in Figure 2, possible positions of the non-stationary loaded bearing (Lang & Steinhilper,1978) are shown.

From equation (1), the value for the dynamic force is obtained:

$$F_p = F\left[\cos(\delta-\gamma) - \frac{sign(\delta-\gamma)\sin(\delta-\gamma)}{tg\beta}\right] sign\left(\beta - |\delta-\gamma|\right) \tag{3}$$

From equation (2), the value for the dynamic force is obtained:

$$F_0 = F\frac{sign(\delta-\gamma)\sin(\delta-\gamma)}{\sin\beta} \tag{4}$$

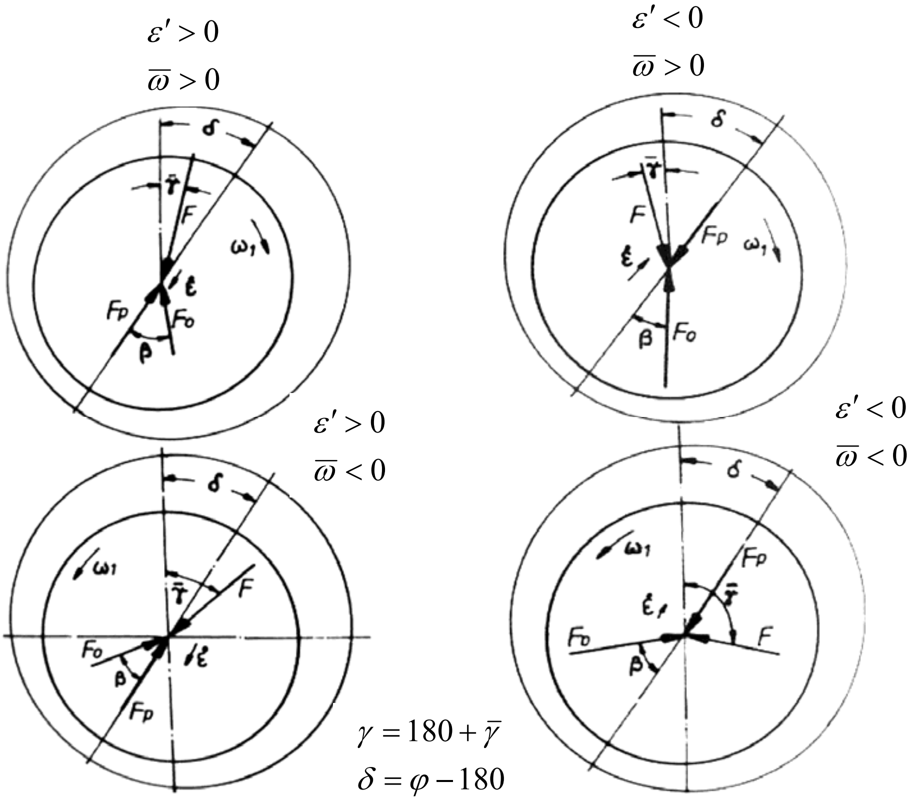


Figure 2 – Possible balance positions of the sleeve of a non-stationary loaded sliding bearing
Рис. 2 – Возможные положения равновесия рукава нестационарно нагруженного подшипника скольжения
Слика 2 – Могући положаји равнотеже рукавца нестационарног оптерећеног клизног лежаја

By introducing known ratios for Sommerfeld's bearing characteristics for components due to the rotation and thrusting of the sleeve, the parameters of the dynamic trajectory of the angular velocity of the sleeve in the bearing are obtained (Lang & Steinhilper,1978):

$$So_0 = \frac{F_0 \psi^2}{BD\eta|\overline{\omega}|} \tag{5}$$

$$So_p = \frac{F_p \psi^2}{BD\eta|\dot{\varepsilon}|} \tag{6}$$

The effective angular velocity of the sleeve $\left|\bar{\omega}\right|$ is given by equation (7):

$$\left|\bar{\omega}\right| = \left|\bar{\omega}^{*} - \omega_{sp}\right| = \left|\omega_1 + \omega_2 - 2\dot{\delta}\right| , \qquad (7)$$

where is:

ω_1 - angular velocity of the sleeve,

ω_2 - angular bearing speed (ω_2 = 0, in the case of the non-rotating main sleeve bearing),

$\left|\bar{\omega}^{*}\right|$ - relative angular velocity of the sleeve in the bearing,

$\omega_{sp} = -2\frac{d\delta}{dt}$ - is a hydrodynamic effect in the bearing, stopping the sleeve and causing it to rotate in the opposite direction,

$\left|\dot{\delta}\right|$ - angular velocity of the minimum thickness of the oil film,

ψ - relative bearing clearance calculated according to equation (8):

$$\psi = \frac{R - r}{r} = \frac{Z}{r} , \qquad (8)$$

where is:

R - bearing radius,
r - bearing sleeve radius, and
Z - radial bearing clearance.

Finally, the value for $\left|\bar{\omega}\right|$:

$$\left|\bar{\omega}\right| = \left|\bar{\omega}^{*} - 2\dot{\delta}\right| = \frac{sign(\delta - \gamma)F\sin(\delta - \gamma)\psi^{2}}{BDSo_{0}\eta\sin\beta} \qquad (9)$$

The value for the radial velocity of the sleeve center ($\dot{\varepsilon}$) is obtained from equation (10), (Lang & Steinhilper,1978):

$$\dot{\varepsilon} = \frac{F\psi^2}{BD\,\eta S_{op}}\left[\cos(\delta-\gamma) - \frac{\sin|\delta-\gamma|}{tg\beta}\right] \tag{10}$$

The value for the angular velocity of the minimum thickness of the oil film ($\dot{\delta}$), is obtained from equation (11):

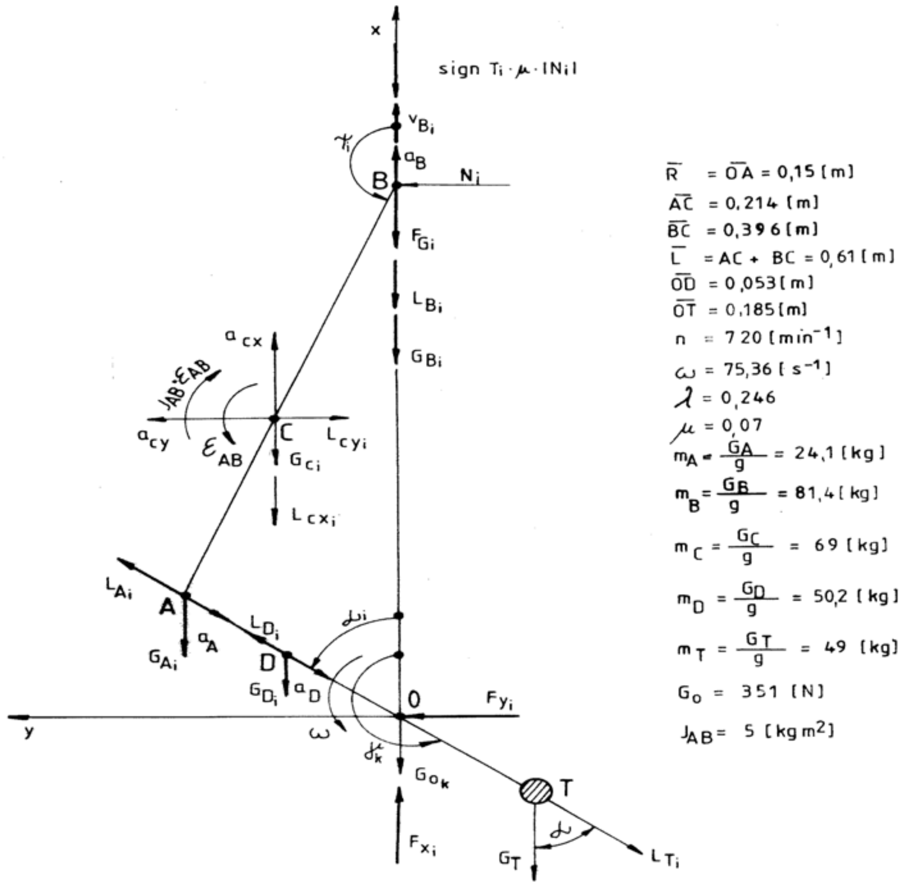
$$\dot{\delta} = \frac{1}{2}\left[\bar{\omega}^{*} - \frac{F\psi^2}{BD\,\eta So_0}\frac{\sin(\delta-\gamma)}{\sin\beta}\right] \tag{11}$$

An internal combustion engine of the type 6ASL-25D, manufactured by Jugoturbina - Karlovac, licensed by Sulzer company, Switzerland, was chosen as an example of modeling and calculating dynamic parameters in order to determine the size of clearance in the sliding bearing.

Determining clearance in sliding bearings of internal combustion engines is very complex. In this case, the main bearings of the engine crankshaft were the subject of investigation.

Figure 3 shows the kinetostatic model of the motor mechanism, on the basis of which the calculation of all dynamic parameters will be performed: the speed of the parts of the motor mechanism, acceleration, the dynamic forces that load the sliding bearing and the calculation of the parameters of the dynamic orbit, on the axis, which will determine the clearance size, i.e. bearing wear. A dynamic calculation was performed in a completely new way. The results of the calculations were obtained by the help of the software developed for this purpose (Žegarac, 1989).

Figure 4 shows the change in the intensity of the dynamic loading force (F) acting on the sleeve of the 1st main bearing, depending on the rotation angle of the crankshaft (α = 0° to 720°), during the engine operating cycles, (Žegarac, 1989).



$L_{A_i} = m_A \cdot a_A$ $\quad$ $L_{cx_i} = m_c \cdot a_{cx}$ $\quad$ $L_{T_i} = m_T \cdot R \cdot \omega^2$

$L_{B_i} = m_A \cdot a_A$ $\quad$ $L_{cy_i} = m_c \cdot a_{cx}$ $\quad$ $L_{D_i} = m_D \cdot a_D$

Figure 3 – Kinetostatic model of an internal combustion engine

Рис. 3 – Кинетостатическая модель двигателя внутреннего сгорания

Слика 3 – Кинетостатички модел моторног механизма са унутрашњим сагоревањем

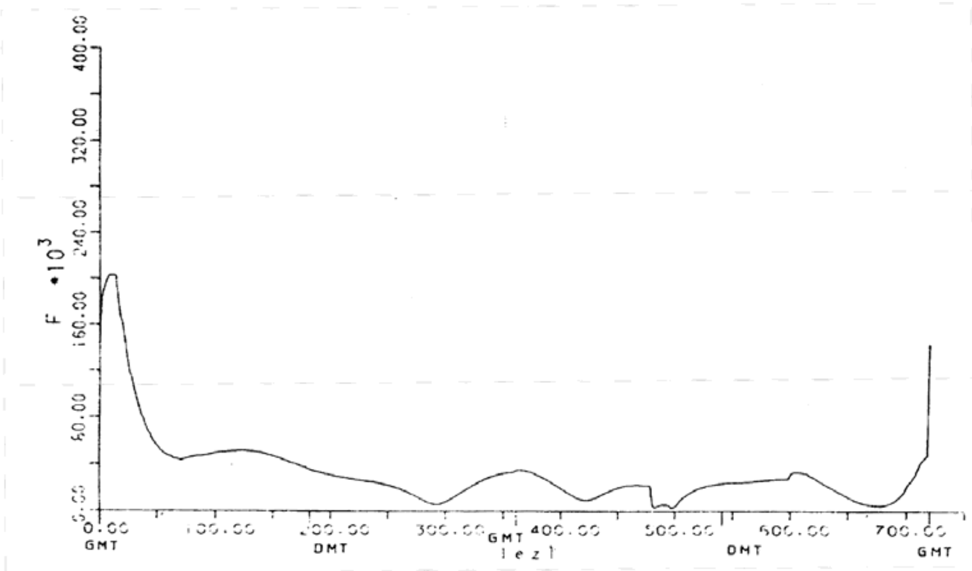


Figure 4 – Changing the intensity of the dynamic force (F) acting on the sleeve of the 1st main bearing depending on the angle of rotation of the crankshaft engine (α=0° to 720°)

Рис. 4 – Изменение интенсивности динамической силы (F), действующей на рукав первого главного подшипника, в зависимости от угла поворота коленчатого вала двигателя (α=0° до 720°),

Слика 4 – Промена интензитета динамичке силе (F) која делује на рукавац првог главног лежаја у зависности од угла заокрета коленастог вратила мотора (α=0° до 720°)

Figure 5 shows the change in the angle (γ) at which the load force (F) acts on the main bearing sleeve depending on the crankshaft rotation angle (α = 0° to 720°):

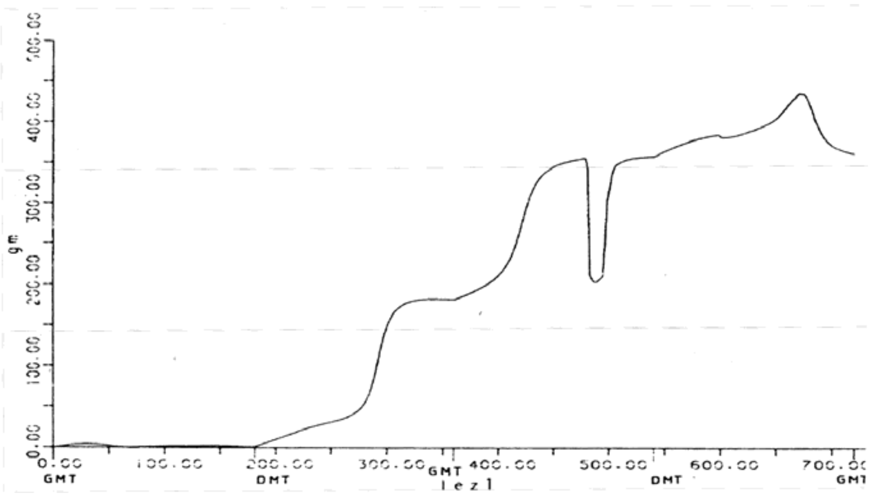


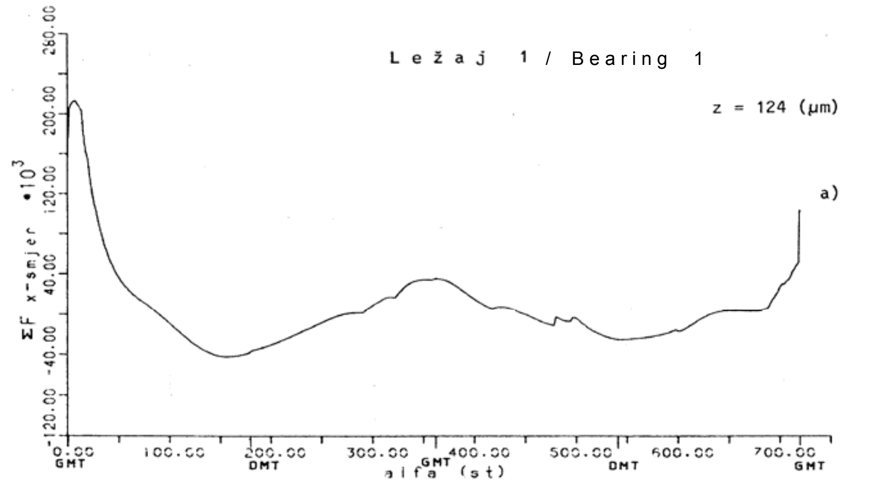
Figure 5 – Changing the angle (γ) at which the force (F) acting on the main bearing sleeve depends on the the crankshaft rotation angle (α=0° to 720°)

Рис. 5 – Изменение угла (γ), под которым сила (F), действующая на рукав главного подшипника, зависит от угла коленчатого вала двигателя (α=0° до 720°)

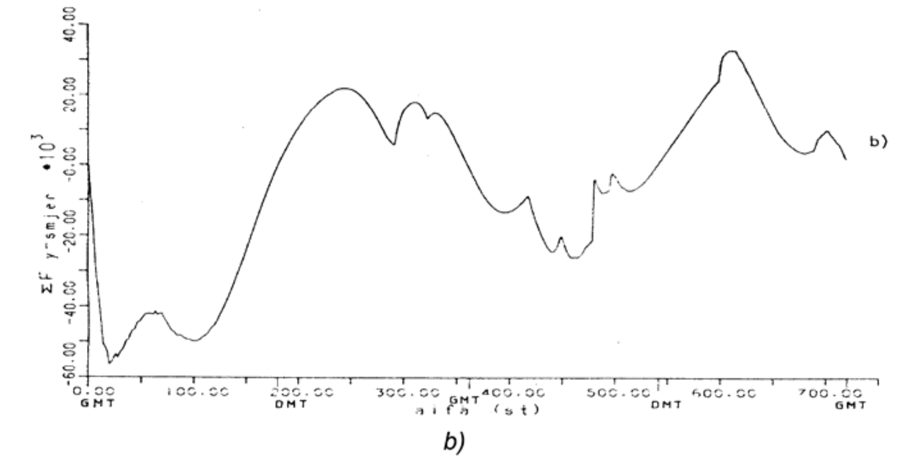
Слика 5 – Промена угла (γ) под којим сила (F) делује на рукавац главног лежаја у зависности од угла заокрета коленастог вратила мотора (α=0° до 720°)

Figure 6 shows the hydrodynamic forces (Fo) and (Fp) depending on the crankshaft rotation angle (α) on the 1st main bearing of the engine. It presents the sum of the forces (ΣFy), (ΣFx), the influence of the dynamic forces from all engine cylinders on the 1st main bearing of the engine in the horizontal axis (y) and the vertical axis (x).

The radial clearance in the bearing is Z = 124 μm, (Žegarac, 1989):



a)



b)

Figure 6 – Changing the intensity of the hydrodynamic forces ΣFx, ΣFy depending on the the crankshaft rotation angle (α = 0° do 720°)
a) vertical axis of the motor, b) horizontal axis of the motor

Рис. 6 – Изменение интенсивности гидродинамических сил ΣFx, ΣFy, в зависимости от угла поворота коленчатого вала (α = 0° do 720°)
а) вертикальная ось двигателя, б) горизонтальная ось двигателя

Слика 6 – Промена интензитета хидродинамичких сила ΣFx и ΣFy у зависности од угла заокрета (α = 0° до 720°) коленастог вратила мотора:
а) вертикална ос мотора, б) хоризонтална ос мотора

Figure 7 shows the calculated dynamic trajectory of the sleeve on the 1st main bearing of the crankshaft, at 100% engine load and an engine speed of n = 720 min^{-1} when the value of the radial clearance is Z = 124μm. In the dynamic trajectory, the crankshaft rotation angle (α = 0° to 720°) is indicated. The angle value (α = 0°) indicates the start of the expansion in the engine, (Žegarac, 1989).

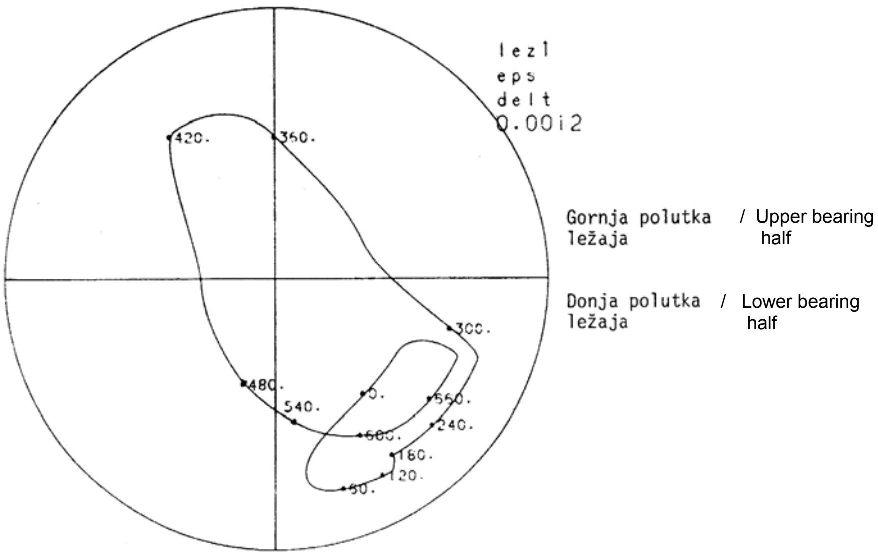


Figure 7 – View of the calculated dynamic trajectory of the main sleeve on the 1st main bearing on the engine crankshaft
Рис. 7 – Изображение измеренной динамической траектории главного рукава в первом главном подшипнике двигателя коленчатого вала
Слика 7 – Приказ прорачунате динамичке путање главног рукавца на првом главном лежају коленастог вратила мотора

Experimental research

The experimental studies were performed on a marine diesel engine type 6ASL-25D, manufactured by Jugoturbina - Karlovac, licensed by Sulzer company, Switzerland, at different engine modes and engine speeds up to a maximum speed of n = 720 min^{-1}. All the preparations for the measurement were made, the operating parameters of the engine were adjusted, the monitoring system was installed and the measurement system calibrated (Easy-Laser, 2020), (Žegarac,1993).

If the plant is in operation and there is a need to install a new monitoring system, the installation and centering of contactless encoders to measure the center displacement of the sleeve center can be performed with a laser centering system of the sleeve center relative to the bearing center (Žegarac, 2016), (Ličen & Zuber, 2003).

Dynamic trajectories were measured in the unladen mode, up to a maximum speed of n = 720 min^{-1}.

Dynamic trajectories at partial engine loads were also measured, up to a maximum engine speed of n = 720 min^{-1}.

Figure 7 shows the measured dynamic trajectories of the main sleeve on the 1st main bearing of the crankshaft engine at 100% engine load and a speed of n = 720 min^{-1}.

The size of the radial clearance in the sliding bearing is Z = 124μm.

The figure also shows the size (e) that represents the displacement of the sleeve center relative to the bearing center, i.e. the eccentricity of the sleeve center, which is directly related to the bearing clearance (Z).

In order to compare the calculated and measured sizes of the bearing clearance, the term of so-called relative eccentricity (ε) was introduced and determined by the equation:

$$\varepsilon = \frac{e}{Z} \tag{12}$$

Figure 8 shows the angle (δ) of the minimum thickness of the oil film.

On the basis of the measured values (e) and (δ), the clearance in the sliding bearing and the assessment of further plant exploitation are determined in order not to cause a plant failure.

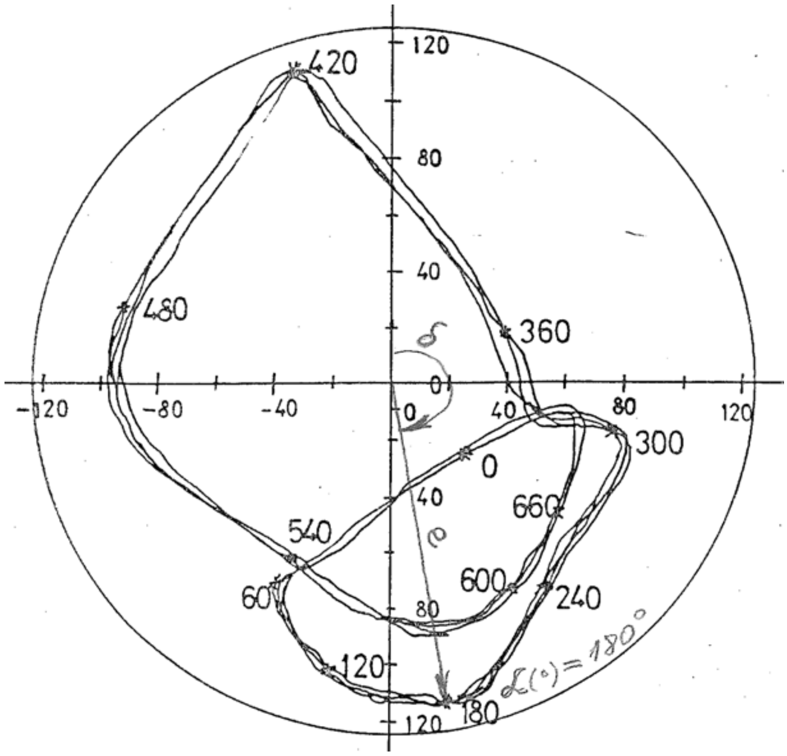


Figure 8 – Display of the measured dynamic sleeve trajectory on the 1st main bearing of the crankshaft, at the engine load of 100%, n = 720 min^{-1}, and the bearing clearance Z=124 (μm)

Рис. 8 – Изображение измеренной динамической траектории рукава в первом главном подшипнике двигателя коленчатого вала, при нагрузке на двигатель (100%), n = 720 мин$^{-1}$, зазор подшипника Z = 124 (μm)

Слика 8 – Приказ измерене динамичке путање рукавца на првом главном лежају мотора коленастог вратила мотора при оптерећењу мотора (100%), n = 720 min^{-1}, зазор лежаја Z=124 (μm)

Figure 9 displays the measured dynamic trajectory of the sleeve on the 1st main bearing of the crankshaft engine at the load of 0%,n = 720 min^{-1}, and the bearing clearance Z = 124(μm):

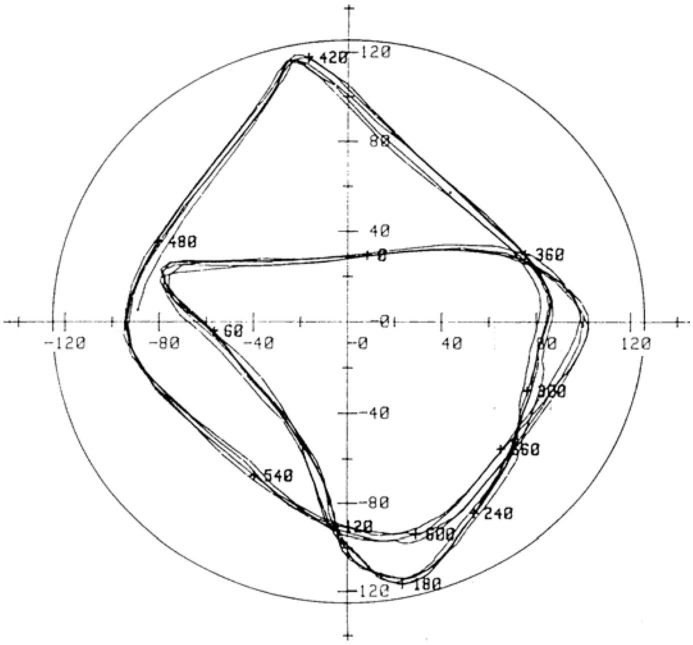


Figure 9 – Display of the measured dynamic trajectory of the sleeve on the 1st main bearing of the crankshaft , at the engine load of 0%, n = 720 min^{-1}, and the bearing clearance Z = 124 (μm)

Рис. 9 – Изображение измеренной динамической траектории рукава в первом главном подшипнике двигателя коленчатого вала, при нагрузке на двигатель (0%), n = 720мин$^{-1}$, зазор подшипника Z = 124 (μм)

Отображение измеренной динамической траектории втулки на 1-м главном подшипнике коленчатого вала при нагрузке двигателя 0%, n = 720 мин-1 и зазоре подшипника Z = 124 (μm)

Слика 9 – Приказ измерене динамичке путање рукавца на првом главном лежају мотора коленастог вратила мотора, при оптерећењу мотора (0 %), n=720min^{-1}, зазор лежаја Z=124 (μm)

Based on the results of measuring the displacement of the center of the sleeve (e) for this engine type, the dependence (Z) of (e) was determined (Žegarac, 1989):

$$Z = 1.01 \cdot e + 7{,}48 \pm 4 \tag{13}$$

The results are presented in Figure 10.

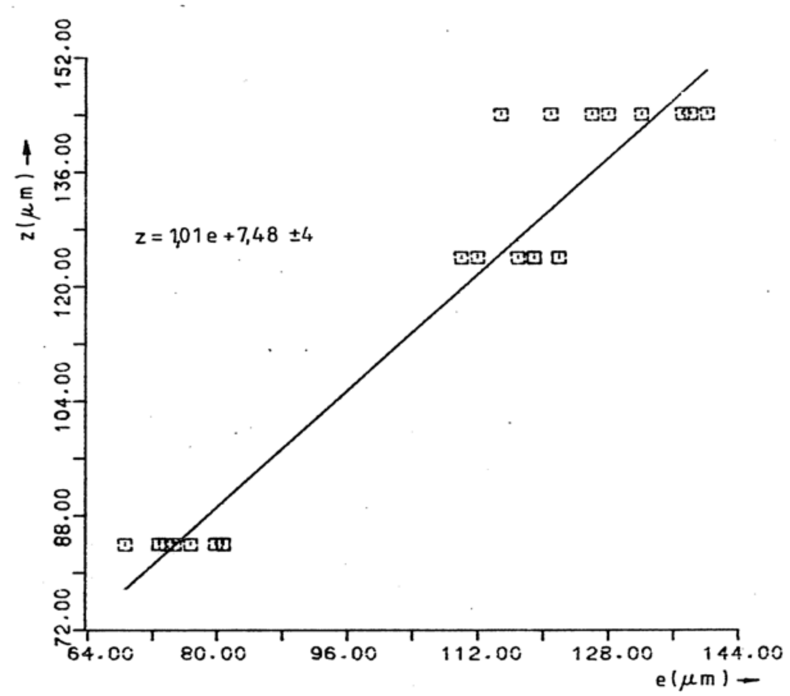


Figure 10 – Graphic depiction of the clearance size (Z) and the displacement of the center of the sleeve (e) for different rotation speeds and engine loads

Рис. 10 – Графическое изображение размера зазора (Z) и смещения центра рукава (e) для разных скоростей вращения и нагрузок на двигатель

Слика 10 – Графички приказ зависности величине зазора (Z) и помака средишта рукавца (e), за различите бризине вртње и оптерећења мотора

Comparison of the theoretical calculation results and the experimental research results

Figure 11 gives a graphic representation of the calculated and measured values of the sleeve eccentricity (e) depending on the angle of rotation of the crankshaft (α) for various bearing wear degrees, at a rotation speed of n = 720 min^{-1} and at 100% engine load, (Žegarac, 1993):

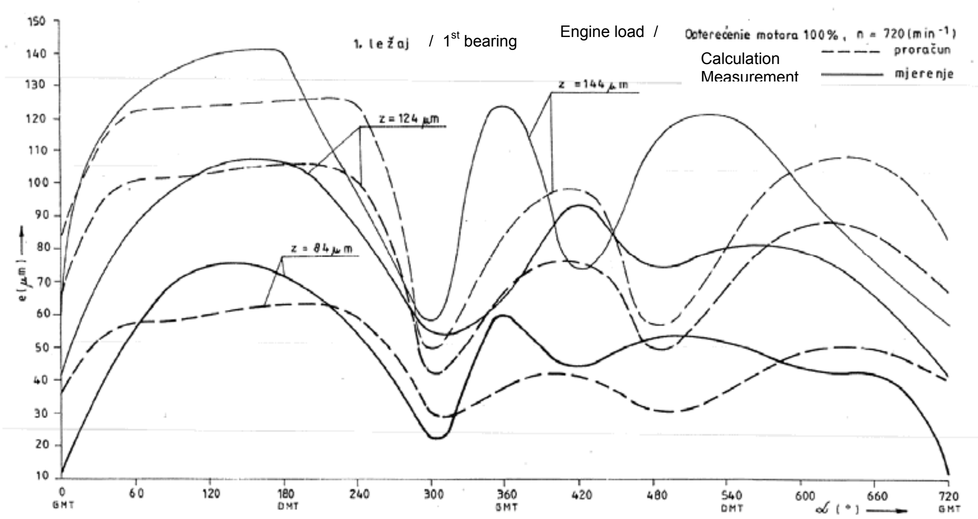


Figure 11 – Display of the calculated and measured eccentricities of the sleeve (e) on the 1st main bearing depending on the crankshaft rotation angle (α) for different sizes of bearing clearance Z=84 (μm), Z=124(μm), Z=144 (μm), engine speed n = 720 min^{-1} and 100% engine load

Рис. 11 – Изображение рассчитанных и измеренных эксцентриситетов рукава (е) первого главного подшипника, в зависимости от угла поворота коленчатого вала (α) при разных размерах зазора подшипника Z = 84 (μm), Z = 124(μm), Z = 144 (μm), частота вращения двигателя n = 720 мин$^{-1}$ и нагрузка на двигатель 100%

Слика 11 – Приказ прорачунатих и измерених ексцентричности рукавца (е) на првом главном лежају у зависности од угла заокрета коленастог вратила мотора (α) за различите величине зазора у лежају: Z=84 (μm), Z=124(μm), Z=144 (μm), брзини вртње мотора n =720 min^{-1} и оптерећењу мотора 100%

In Figure 12, there is a graphic representation of the measured values of the sleeve eccentricity (e) depending on the angle of rotation of the crankshaft (α) for various bearing wear degrees Z=84 (μm), Z=124 (μm), Z=144 (μm), at different speeds of rotation, without loading the engine.

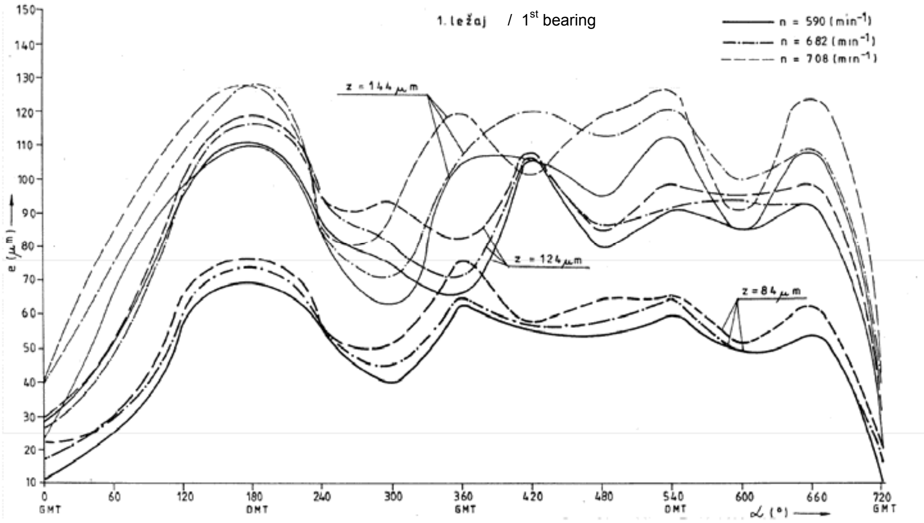


Figure 12 – Display of the measured eccentricities of the sleeve (e) on the 1st main bearing, depending on the crankshaft rotation angle (α) for different sizes of the clearance in the bearing Z=84(μm), Z=124 (μm), Z=144 (μm) and speed, no engine load

Рис. 12 – Изображение измеренных эксцентриситетов рукава (е) первого главного подшипника в зависимости от угла поворота коленчатого вала (α) при разных величинах зазора в подшипнике Z = 84 (μm), Z = 124 (μm), Z = 144 (μm) и скорость, без нагрузки на двигатель

Слика 12 – Приказ измерених ексцентричности рукавца (е) на првом главном лежају, у зависности од угла заокрета коленастог вратила (α), за различите величине зазора у лежају: Z=84(μm), Z=124 (μm), Z=144 (μm) и брзинама вртње, без оптерећења мотора

Figure 13 gives a graphic representation of the calculated and measured values of the eccentricity angle of the bearing hose (δ) depending on the angle of rotation of the crankshaft of the engine (α) for various bearing wear degrees, at 100% engine load and the engine speed of n = 720 min^{-1} (Žegarac, 1993):

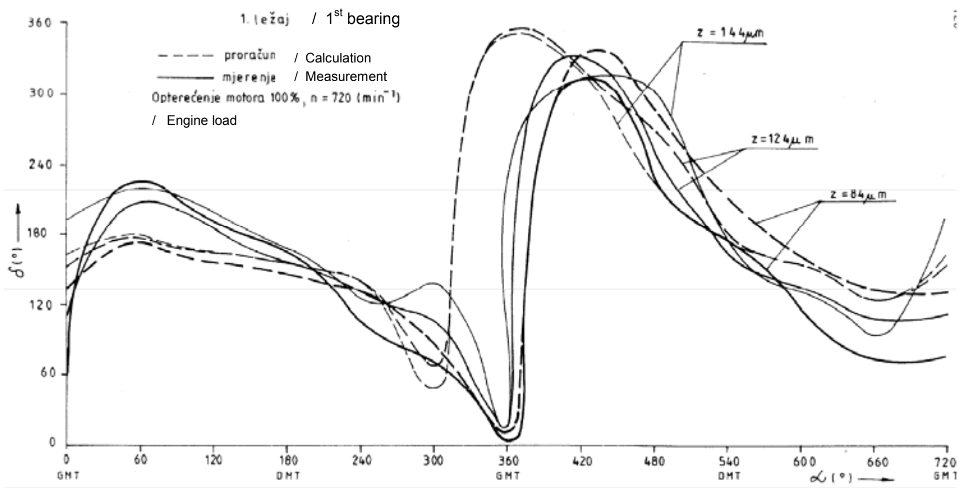


Figure 13 – Display of the calculated and measured values of the angle of eccentricity of the bearing hose (δ) depending on the crankshaft rotation angle (α) for various bearing wear degrees, 100% engine load, speed of rotation engine n = 720 min^{-1}

Рис. 13 – Изображение расчетных и измеренных значений угла эксцентриситета рукава подшипника (δ) в зависимости от угла поворота коленчатого вала двигателя (α) при различном износе подшипника, при 100% нагрузке на двигатель, скорости вращения двигателя n = 720 мин$^{-1}$

Слика 13 – Приказ прорачунатих и измерених вредности угла ексцентричности рукваца лежаја (δ) у зависности од угла заокрета коленастог вратила мотора (α) за разна истрошења лежаја, при 100% оптрећењу мотора, брзини вртње мотора n=720 min^{-1}

Based on the comparison of the calculated dynamic sizes and the measurement results, it can be concluded that the correspondence of the results of the calculations and measurements is in the domain of 5%, which is completely acceptable. The repeatability of the measurement values during the measurement was obtained.

The method of calculation and measurement of the functional parameters of the new diagnostic method has been completely verified.

Diagnostics of sliding bearings can be carried out for all modes, rotation speeds and plant loads.

The author of this scientific paper has spent many years of research on many projects related to technical diagnostics of machine and electrical plants, development, research, design and implementation of monitoring systems (Žegarac, 1994), (Žegarac, 2019).

Conclusion

The application of the new diagnostic method and the monitoring system gives an opportunity to reliably determine when and where the problem will arise related to the wear of sliding bearings in further plant operation, to offer a quality assessment of how the system will continue to function over time, as well as to predict the causes of failures and how to remedy them, and to provide time for routine maintenance of technical systems. Modern methods of technical diagnostics based on the measurement of dynamic parameters and electrical sizes and their analysis allow the support of measuring systems and measurement sensors from different manufacturers. The capabilities of a very advanced program configuration for system diagnostics and monitoring are presented. The values of the functional parameters were measured and software processed as well as their limit values and alarms displayed in different ways.

There are options for remote control and monitoring, shutting off particular parts from further exploitation to prevent damage to various plants where vital machine elements are installed. High accuracy and reliability of measurements of all relevant measuring sizes are ensured, on the basis of which the technical correctness of the plant can be qualitatively determined. Multi-channel, modular, software-hardware monitoring, control and protection systems allow the optimization of plant operating parameters. The new diagnostic method and monitoring systems can be widely applied in all technical fields: internal combustion engines, hydropower plants, thermal power plants, process plants, and in many other areas. Many systems for monitoring, compatibility of devices and equipment from various manufacturers are supported by hardware and software. Mechanical and electrical sizes measured on different types of plants are very important for further operations of drive systems. In this paper, a special attention is given to the diagnostics of sliding bearings.

The new method of diagnostics of sliding bearings, by measuring the dynamic trajectory of the sleeves in the bearing, does not depend on the plant type, and can be applied to all types of internal combustion engines and various other plants. The author of this very important project proposes that this new method and its accompanying monitoring systems be installed in drive systems. In the past, many important systems did not have similar monitoring systems in place, and many problems arose. For system installations used since the 1950s, simpler systems for monitoring basic functions were designed, which was not enough, especially in recent times, as the development of new techniques has progressed.

New plant monitoring systems have alarm functions. The alarm can be visual or accoustic or in the form of SMS notifications. The field of technical diagnostics is a very interesting field for many scientists (Braut et al, 2008), (Ličen & Zuber, 2003). This area is under-researched, as is the area of medical diagnostics.

References

Braut, S., Žigulić, R., & Butković, M. 2008. Numerical and experimantal analysis of a shaft bow influence on a rotor to stator contact dynamics. *Strojniški vestnik-Journal of Mechanical Engineering,* 54(10), pp.693-706 [online]. Available at: https://www.sv-jme.eu/article/numerical-and-experimantal-analysis-of-a-shaft-bow-influence-on-a-rotor-to-stator-contact-dynamics/ [Accessed: 7 April 2020].

Černej, A., & et al. 1986. Radni uslovi podmazivanja ležaja dizel motora. *Goriva i maziva*, 26(5-6), pp.229-236 (in Serbian).

-Easy-Laser. 2020. *Laser measurement & alignment systems. Technical documentation.* Damalini, Sweden: Easy-Laser.

Lang, O.R, & Steinhilper, W. 1978. *Gleitlager*. Berlin, Heidelberg: Springer-Verlag. Available at: https://doi.org/10.1007/978-3-642-81225-5.

Ličen, H., & Zuber, N. 2003. Vibrodiagnostics as an element for quality and reliabilty assurance: SPIDER 8 – universal measuring device, concept and application. In: *Quality 2003: 3rd Research/expert conference with international participation,* Zenica, Bosnia and Herzegovina, pp.299-306. November 13-14 [online]. Available at: http://www.quality.unze.ba/zbornici/QUALITY%202003/045-R-037.pdf (in Serbian) [Accessed: 7 April 2020].

Žegarac, N. 1989. *Dijagnostika kliznih ležajeva u dizel motoru* (in Serbian). Ph.D. thesis. Zagreb: University of Zagreb, Faculty of Mechanical Engineering and Naval Architecture.

Žegarac, N. 1993. *Postupak dijagnostike ležajeva merenjem dinamičkih putanja glavnih rukavaca, kolenastog vratila.* Serbian Patent number P-640/93.

Žegarac, N. 1994. Analysis of the wear of vital parts of turbocharged engines of high specific power. *Scientific Technical Review*, 44(6), pp.14-24.

Žegarac, N. 2016. Experience in the development of the invention from the aspect of scientific verification and market creation. *Vojnotehnički glasnik/Military Technical Courier*, 64(1), pp.92-109 Available at: https://doi.org/10.5937/vojtehg64-9315 (in Serbian).

Žegarac, N. 2019. The Application of Modern Diagnostic Systems on Machine and Electric Power Plants. *Tehnika*, 74(1), pp.73-87. Available at: https://doi.org/10.5937/tehnika1901073Z.

РАЗРАБОТКА МЕТОДА ОПРЕДЕЛЕНИЯ ВЕЛИЧИН ЗАЗОРОВ В ПОДШИПНИКАХ СКОЛЬЖЕНИЯ

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РУБРИКА ГРНТИ: 55.00.00 МАШИНОСТРОЕНИЕ; 55.45.00 Судостроение, 44.00.00 ЭНЕРГЕТИКА
ВИД СТАТЬИ: оригинальная научная статья

Резюме:

Введение/цель: Цель данной статьи заключается в представлении важности применения нового метода диагностики и системы мониторинга, возможности надежного определения времени и точного места возникновения проблемы, связанной с износом подшипников скольжения при дальнейшей эксплуатации установки, а также качественного прогноза при продолжении работы системы с течением времени, предусмотрение причин отказа и способов их устранения, а также планировки регулярного технического обслуживания системы.

Метод: Новый метод решает проблему диагностики подшипников скольжения путем измерения траекторий перемещения (орбита) вала в подшипнике скольжения. Современные методы технической диагностики, основанные на измерении динамических параметров и электрических величин и их анализе позволяют поддерживать измерительные системы и датчики от различных производителей.

Результаты: Измеряя траектории передвижения (орбиты) валов в подшипниках скольжения, определяется значения, которые характеризуют: нормальное состояние, начальную величину зазора, его дальнейшее увеличение, величину зазора подшипника, когда параметры приближаются предельно допустимой величине зазора и определяется величина зазора, в случае если дальнейшая эксплуатация установки может привести к отказу системы.

Вывод: Новый метод диагностики и системы мониторинга может иметь широкое применение во всех технических областях, таких как: двигатели внутреннего сгорания, гидроэлектростанции, тепловые электростанции, технологические установки и пр. Аппаратное и программное обеспечение поддерживает множество систем мониторинга, контроля совместимости устройств и оборудования от разных производителей. Проведена верификация расчета *теоретических и экспериментальных динамических параметров. Метод имеет широкий спектр применения.*

Ключевые слова: подшипник скольжения, зазор подшипника, износ подшипника, вал подшипника, траектория перемещения.

РАЗВОЈ МЕТОДЕ ЗА УТВРЂИВАЊЕ ВЕЛИЧИНЕ ЗАЗОРА У КЛИЗНИМ ЛЕЖАЈЕВИМА

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ОБЛАСТ: машинство, енергетика, бродоградња
ВРСТА ЧЛАНКА: оригинални научни рад

Сажетак:

Увод/циљ: У раду је представљена примена нове дијагностичке методе и мониторинг система. Приказана је могућност да се поуздано утврди када и где ће се појавити проблем који се јавља при трошењу клизних лежајева у даљој експлоатацији постројења. Поред тога, оцењује се како ће систем наставити да функционише током времена, предвиђају се узроци кварова и начин њиховог отклањања, као и време за планско одржавање техничких система.

Метода: Нова метода решава проблем дијагностике клизних лежајева мерењем динамичких путања рукавца у клизном лежају. Савремене методе техничке дијагностике, засноване на мерењу динамичких параметара и електричних величина и њихове анализе, омогућавају подршку мерних система и сензора за мерење разних произвођача.

Резултати: Мерењем динамичких путања (трајекторије) рукаваца у клизним лежајима утврђују се величине које карактеришу: нормално стање, почетну величину зазора, његово даље повећавање, величине зазора лежаја, када су параметри стања близу горње границе дозвољеног зазора, и утврђивање величине зазора када даља експлоатација постројења може проузроковати хаварију система.

Закључак: Нова дијагностичка метода и мониторниг система могу се широко применити у свим техничким областима: моторима са унутрашњим сагоревањем, хидроелектранама, термоелектранама, процесним постројењима и многим другим областима. Хардверски и софтверски подржавају се многи системи за надгледање и проверава компатибилност уређаја и опреме разних произвођача. Извршена је верификација прорачуна

теоријских и експерименталних динамичких параметара. Метода има широке могућности примене.

Кључне речи: клизни лежај, зазор лежаја, истрошење лежаја, рукавац лежаја, динамичка путања.

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