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MEASURES AND SYMPTOMS OF THE WEAR MARGIN IN FUNCTIONAL UNIT NODES OF PRODUCTION SYSTEM ITEMS

Key words: wear margin, functional unit, thermal node, hydrodynamic node, tribological node, coefficient *k*, resistance coefficient, Sommerfeld number.

Abstract: Remedial actions aimed a production system items should be taken after unacceptable differences occur between the desired and actual values of item functional unit characteristic measures. The author justifies the need to use the term 'wear margin of a functional unit node'. Three basic models of nodes are presented, and selected models for each node are described taking into account their kinematics, the shape of solids, and the physicochemical properties of fluids. Substitute measures of the wear margin and measures of diagnostic symptoms of hydrodynamic, thermal, and tribological nodes are described and justified.

Miary i symptomy potencjału eksploatacyjnego węzłów zespołów funkcjonalnych obiektów systemu produkcyjnego

Słowa kluczowe: potencjał eksploatacyjny, zespół funkcjonalny, węzeł cieplny, węzeł hydrodynamiczny, węzeł tribologiczny, współczynnik k, współczynnik oporu, liczba Sommerfelda.

Streszczenie: Działania sanacyjne na obiektach systemu produkcyjnego powinny być następstwem wystąpienia nieakceptowalnych różnic pożądanych i rzeczywistych wartości miar cech zespołów funkcjonalnych obiektów. W pracy uzasadniono potrzebę posługiwania się pojęciem „potencjał eksploatacyjny węzła zespołu funkcjonalnego". Przedstawiono trzy podstawowe modele węzłów i dla każdego opisano wybrane modele szczegółowe uwzględniające kinematykę, kształt ciał stałych i właściwości fizykochemiczne płynów. Opisano i uzasadniono miary zastępcze potencjału eksploatacyjnego i miary symptomów diagnostycznych węzłów: hydrodynamicznego, cieplnego i tribologicznego.

Introduction

If we assume that an enterprise by itself defines desired values of that enterprise's characteristics, then quality characteristics of a production system item and the values of these characteristics measures should result from the decomposition of the characteristics and desired values of production system characteristic measures. Actual values of characteristics may differ from desired values. The differences may be due to manufacturing errors and wear caused by the use of items. When the differences become unacceptable, remedial actions have to be taken in relation to such items. In [1, 2], algorithms of remedial actions are discussed, along with the introduction of such concepts as *functional unit wear margin* and *functional unit node*.

Production system items are products of other production systems. The life cycle of a product understood as an item of a production system in the social market economy consists of the design, manufacture, operation, and scrapping stages. Actual characteristics and characteristic measure values of an item are defined at the stage of development (design) of an item as a product, and desirable values of machine characteristic measures should stem from the desired quality characteristics of the enterprise company.

Designing of an item consists in preparing the item's specification in the form of design and construction documentation. The specifications define the following:
– The ranges of characteristic measure values describing the item's load (torque, rotary speed);
– The desired ranges of characteristic measure values of the structural material of the item's elements;
– The desired ranges of characteristic measure values of the item's elements (macrostructure, form and dimensions of elements);
– The desired ranges of characteristic measure values for built-in units and elements of an item (backlash, preload of bolts, press fit, tightness, unbalance, etc.); and,
– The desired value ranges of the characteristic measures of the working fluid (e.g., viscosity of oil).

It is assumed that if all those measures have a value in the desired range, then the item as a whole will receive desired characteristics with values falling within the desired range.

Characteristics values rendered to an item in manufacturing change due to wear, the result of using an item. The effects of wear are incorporated in the characteristics describing the production system. Such characteristics are not listed in item specification. Not all the characteristics features of an item are subject to change. Therefore, it is convenient to use the concept of 'wear margin of an item'. The wear margin of an item can be defined as a set of item's characteristic measures, whose values
- are the result of item manufacturing,
- decrease due to wear during their use, and
- can be restored during an overhaul or maintenance.

The problems with using the term 'wear margin of a machine' are the following:
– Machine specifications do not include on-line measurable characteristics directly describing the wear margin.
– Many characteristics are necessary to describe the item's wear margin.

Due to a large number of characteristics necessary to describe wear margin, it is purposeful to establish substitute measures, i.e. measures that can be determined on the basis of characteristic measures of built-in elements, working fluids, load, or relative motion. The problem may be solved by using the term 'wear margin of a functional unit' of an item, for the following reasons:
– The wear margin of an item is the resultant of wear margins of its functional units.
– *Presence* in the item of functional units implementing partial functions means that the rate of degradation of wear margins of each functional unit may be different and may require a separate remedial action.
– Combining items into aggregates is done by joining elements of selected functional units. 'New' functional units are created that are not listed in the specifications.

A prerequisite to use the concept of wear margin of functional units is the decomposition of desired values of characteristic measures of an item into desired values of functional unit wear margin measures.

Direct measurement of the values of wear margin measures of operating functional units generally is not possible. It is necessary to make indirect measurements using signals emitted by functional unit elements. Elements of a functional unit form *nodes*. A functional unit can be represented as the sum of nodes, where one element of a functional unit may be a part of several nodes. In the simplest case, a functional unit can be a single node. There are three basic models of functional unit nodes:
1. Hydrodynamic node: two solids that constitute a channel separated by a fluid. The fluid is affected by hydrodynamic load, which forces and determines fluid motion relative to the solids.
2. Tribological node: two solids are separated by fluid. At least one of the bodies is subjected to mechanical load, which forces and determines the relative motion of the solids.
3. Thermal node: two fluids are separated by a solid. The fluids are at least under thermal load. Heat is transferred from one fluid to another fluid through a solid.

The literature on the subject provides many detailed models of the above basic models, incorporating the shape of solids and physicochemical properties of fluids. It is purposeful to analyse existing physical and mathematical descriptions and to indicate those quantities that can be adopted as substitute measures of the wear margin of a given node and identify associated diagnostic symptom measures.

1. Measures and symptoms of hydrodynamic node wear margin

A specific model of the node 'loaded fluid acting on two solids' is developed in fluid mechanics as 'flow of fluids in pressure pipes/flow of fluids in a round cross-section pipe'. Fluid flow in pipes is modelled as a *one-dimensional* flow: values of quantities describing the flow depend on one coordinate of position. According to [3], the concept of one-dimensional flow is quite well verifiable for a fully developed turbulent flow, because the velocity profile is then relatively flat and global quantities, such as the flux, volume, or kinetic energy, virtually do not depend on velocity distribution.

A pipeline section can be composed of straight pipe sections and desired built-in elements: elbows, branched pipes, cross-section changes, and instruments. Built-in components in terms of hydrodynamics are contractions, flown-round bodies, or a combination of both and, due to energy losses, are regarded as obstacles. Obstacles may also be undesired obstructions created due to manufacturing and installation errors (e.g., welding of pipelines) and as a result of wear. Characteristics of a straight section stem from characteristics of elements making up that section and characteristics obtained after building them in: inside diameter d [m], length l [m], absolute roughness of internal surface k (peak height of asperities) [µm], and relative roughness k/d. In the case of embedded elements, their characteristics are quantities describing the shape and dimensions of the element. Fluid can be a liquid or a gas. Incompressible fluid is a fluid whose density in a specific range of temperature and pressure is negligibly small [3]. Real fluids with Mach number up to 0.3 (for the air up to 100 m/s = 360 km/h) are considered as incompressible fluids [4]. In considerations of incompressible fluids, Bernoulli's equation is important. Fluid characteristics include kinematic viscosity v [m^2/s] and density ρ [kg/m^3]. Characteristics of fluid load and motion are fluid flow rate w, mean flow rate w_m [m/s], and geodetic height g [m].

During the flow in a pipeline, energy losses occur. Energy losses are the result of friction in the fluid and friction between the pipeline wall and the fluid. The mechanism of losses between a fluid and a wall [4] are the thermal movement of molecules at velocity w_b in the direction perpendicular to the direction of the flux, which causes fluid molecules to collide with each other. Molecules in the valleys of material asperities lose velocity w. Molecules diffusing back to the flux must be accelerated again to the speed of fluid flux w. Energy losses in a pipeline are presented as pressure losses between two cross-sections 1 and 2 [4], Fig. 1.

Pressure losses in a straight section of a pipeline are described by Darcy-Weisbach formula:

$$\Delta p = \lambda \frac{l}{d} \rho \frac{w_m^2}{2} \tag{1}$$

The number λ is called the *linear resistance coefficient* or coefficient of friction. It depends on the Reynolds number Re = wl/v, Table 1.

Table 1. The value of the linear coefficient of resistance depending on the type of fluid flow

Re value	Formula
≤ 2320	$\lambda = \frac{64}{\mathrm{Re}}$; $\lambda = f(\mathrm{Re})$
2320 < Re < 10^5	$\lambda = \frac{0,316}{\sqrt[4]{\mathrm{Re}}}$; $\lambda = f(\mathrm{Re})$
65 d/k < Re < 1300 d/k	$\lambda = \frac{1}{\left[2\log\left(\frac{2,51}{\mathrm{Re}\sqrt{\lambda}}\right) + \frac{0,27}{d/k}\right]^2}$; $\lambda = f(\mathrm{Re}, k/d)$
Re > 1300 d/k	$\frac{1}{\sqrt{\lambda}} = 1,74 - 2\log\frac{2k}{d}$; $\lambda = f(k/d)$

Source: Authors on the basis of [3, 4].

Losses on obstacles are taken into account by means of coefficient ζ. The coefficient ζ comprises additional losses compared to the losses of a straight section of the pipeline [4].

$$\Delta p_p = \zeta \rho \frac{w_m^2}{2} \tag{2}$$

The values of coefficient ζ are calculated experimentally.

Disturbances of the flux occur generally on the obstacle as well as within sections before and after it (after the obstacle – in a section 10 to 30d in case of turbulent motion) [4], Fig. 1.

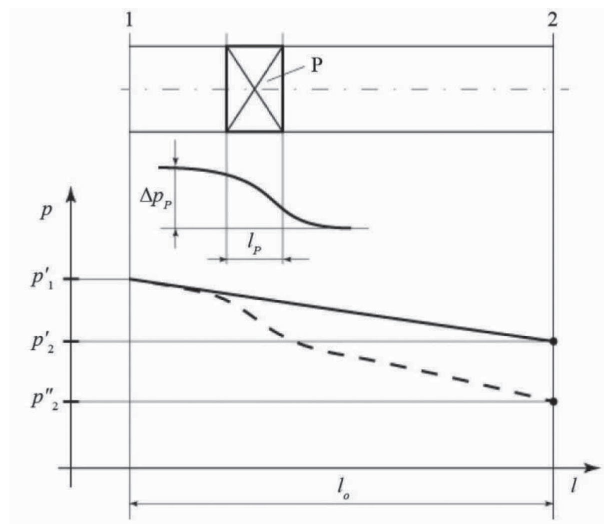

Fig. 1. Pressure loss between cross sections 1 and 2: P – obstacle, l_P – the length of the obstacle, Δp_P – loss at the obstacle (loss focused locally), (p'_1–p'_2) – loss at the straight section (loss evenly distributed), (p'_1–p''_2) – the total loss at the section l_o

Source: [5].

When the flow is laminar, in areas after any of the two types of obstacle, (contraction and a solid) turbulences are formed, characterised by small and large eddies [4], Fig. 2.

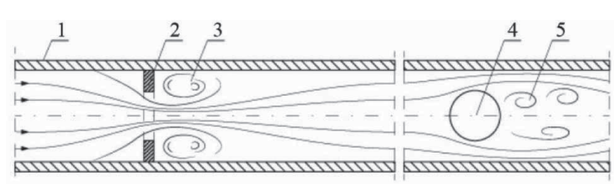

Fig. 2. Areas of turbulence after an obstacle in laminar flow: 1 – pipeline wall, 2 – orifice, 3 – turbulence (eddies) behind the orifice, 4 – flown-round solid body 5 – turbulence (eddies) after a flown-round body

Source: [5].

According to [6], every non-stationary process in a gas or liquid is a source of turbulences that spread in a wave motion. Each obstacle, therefore, is a source of noise – continuous acoustic emission.

When a liquid flows through an orifice, the pressure in the orifice can theoretically take any low value. Practically, however, the pressure value is limited and may not fall below the pressure of evaporation (boiling) corresponding to the temperature of the flowing liquid. The drop of pressure to values close to evaporation pressure is accompanied by the emission of gases and vapours from the liquid, the *phenomenon known as cavitation* [3]. Cavitation causes wear and acoustic emission.

In the case of flown-round bodies, at adequately large Reynolds numbers, vortex shedding can occur. We can determine the frequency of vortex shedding experimentally. The results are presented in the form of a Str – Re diagram [4]. Strouhal number (St, Str) is a dimensionless number (dimensionless time) that describes unstable flows in which accelerations have a dominant role:

$$\mathrm{Str} = \frac{d \cdot f}{w_m} \tag{3}$$

where f – the frequency of vortex shedding.

The phenomenon of vortex shedding causes burst acoustic emission.

The publication [6] contains a semi-empirical relation determining total acoustic power N_o [W] generated by an obstacle which is passed by a medium surrounding a pipeline:

$$N_o = k \frac{\Delta p^3 d^2}{\rho^2 c^3} \tag{4}$$

where Δp [Pa] – total pressure loss caused by a body in the pipeline; k – constant (for air $k = 2.5 \cdot 10^{-4}$); c – speed of sound in the surrounding medium [m/s].

2. Measures and symptoms of thermal node wear margin

If the principal load of a node consisting of a solid separated by two fluids is thermal load, such a node can be called a thermal node. Heat flux is a measure of thermal load, $\dot{Q}$. Heat flux is defined as the ratio of elementary amount of heat dQ to time length dt of the heat transfer $\dot{Q} = dQ/dt$ [W].

A solid body/barrier can have a multilayer structure, e.g., two cylinders connected by press fit. On the macro scale, a solid body in the form of machine element material is non-homogeneous; it has a core, surface layer, and films. The barrier surface may be of different shape. In the known models of thermal nodes the following are distinguished [7, 8]:

- A plane/flat barrier/wall (e.g., of a heat exchanger, slide bearing guide, thermal insulator, etc.; and,
- Cylinder (cylinder surface)/cylindrical barrier: heat exchanger 'tube-in-tube', shell and housing of a slide bearing etc.

Heat transfer from a hot source through a barrier to a cold source takes place by conduction, convection, and radiation. The participation of each phenomenon is specific to a particular node. Each of the above phenomena is characterised by a different drop in temperature in the direction of the heat flow. Figure 3 presents the temperature drop for the node, in which, in Zones 1 and 3, heat transfer is mainly through convection.

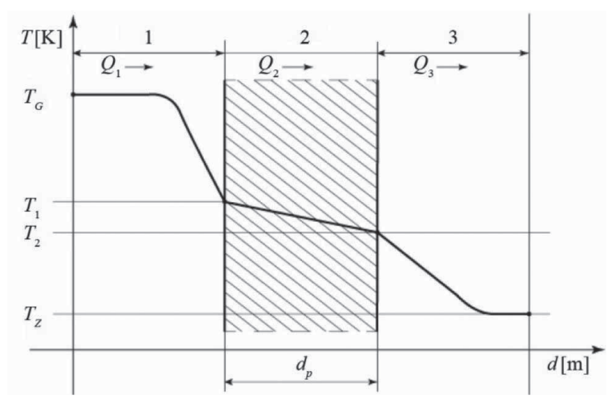

Fig. 3. The temperature drop for the node liquid–barrier–liquid: d – heat transfer path, d_p – barrier thickness, T_G – hot source temperature (temperature in the fluid flux), T_z – cold source temperature, T_1, T_2 – temperature at liquid-solid interface, 1, 3 – convection, 2 – heat conduction

Source [9].

Heat flux passing through a flat barrier, Fig. 3, is modelled [7, 8] as one-dimensional, fixed temperature heat conduction, since temperature changes in only one direction, local temperature is constant in time.

$$\dot{Q} = \frac{\lambda}{d_p} A \left(T_1 - T_2\right) \tag{5}$$

where λ – proportionality factor called heat conduction coefficient or thermal conductivity [W/mK] (inclination of the line $T = f(d)$ is proportional to the value of λ); A – surface area of the barrier measured perpendicular to heat flow direction [m²]; $(T_1 - T_2) = \Delta T$ temperature difference on both sides of the barrier [K].

For a multilayer wall, at a steady flow through a flat barrier, the following applies:

$$d_p = d_1 + d_2 + d_n \tag{6}$$

$$\dot{Q} = A\frac{\Delta T}{\sum_{i=1}^{n}\frac{d_i}{\lambda_i}} \tag{7}$$

The literature on the subject provides appropriate formulas for other shapes of the barrier.

The coefficient λ [W/mK] is a quantity characterizing a given medium/material in terms of the ability to conduct/insulate heat. It is a material property, which is a constant determined experimentally.

The process of heat transfer from the flux to the barrier (and vice versa) is called *convective* heat transfer, described by Newton's law [7, 8]:

$$\dot{Q} = A\alpha\Delta T \tag{8}$$

where ΔT temperature difference between the fluid and the wall (for example, from Figure 3, respectively: $\dot{Q}_1 = A\alpha_G\left(T_G - T_1\right)$; $\dot{Q}_3 = A\alpha_Z\left(T_2 - T_Z\right)$);

α – convective heat transfer coefficient [W/m²K].

Heat transfer coefficient results from the definition of Nusselt number [7, 8]:

$$\alpha = \frac{N_u\lambda_p}{L} \tag{9}$$

where λp – thermal conductivity of stationary liquid; L – characteristic length of a thermal node (in the case of channels – hydraulic diameter, in the other cases – length of the fluid flux); N_u – Nusselt number (dependent on other numbers of similarity: Reynolds, Prandtl, Grashof, Froud).

It follows from definitions describing Nusselt number that heat transfer coefficient α depends on the following:

- The physical properties of barrier and fluid materials;
- The geometries of the barrier and the fluid flux;
- The properties of the barrier surface (profile); and,
- The fluid velocity relative to the barrier.

The process of heat transfer from the fluid flux of higher temperature through a barrier to the flux of a lower temperature is called *heat penetration*, or *transmission*. Heat flux penetrating through flat barriers is described as follows [7, 8]:

$$\dot{Q} = Ak\Delta T \tag{10}$$

where k – overall heat transfer coefficient [W/m²K].

For example, from Figure 3:

$$\Delta T = T_G - T_Z \tag{11}$$

$$k = \frac{1}{\frac{1}{\alpha_G} + \sum_{i=1}^{n}\frac{d_i}{\lambda_i} + \frac{1}{\alpha_Z}} \tag{12}$$

The temperature on the barrier surface at a steady heat exchange is not equal, and the temperature distribution is dependent on the direction of the flux transferring heat. In tube-in-tube nodes, the temperature changes along the cylinder generatrices. It is advisable to know the temperature along the relevant generatrix on both surfaces of the barrier. The average temperature difference ΔT is determined. The logarithmic mean temperature difference ΔT_m is a universal measure of the average temperature ΔT. The method for determining the average logarithmic temperature difference is presented after [7], in Figure 4, as an example of a counter-flow heat exchanger.

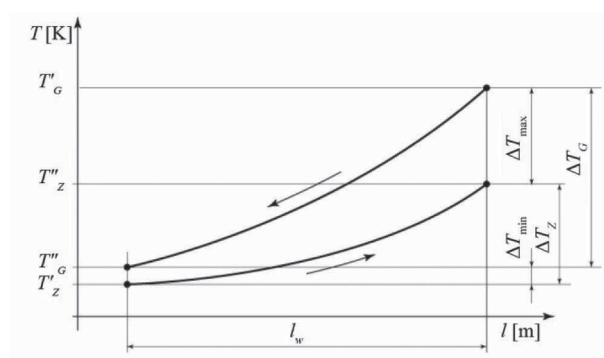

Fig. 4. The temperature as a function of the fluid flux path in a node with fluid counter flows under adiabatic conditions: l_w – heat exchanger length; T'_G, T'_Z – inlet temperature of the hot and cold flux; T''_G, T''_Z – hot and cold outlet fluid temperatures

Source: [9].

$$\Delta T_m = \frac{\Delta T_{\max} - \Delta T_{\min}}{\ln\left(\frac{\Delta T_{\max}}{\Delta T_{\min}}\right)} \tag{13}$$

The average logarithmic temperature difference ΔT_m is, according to [8], independent of flow direction. $\Delta T_{\max}$ and $\Delta T_{\min}$ are differences between fluid temperatures on a given side of the exchanger.

The study [10] indicates a possibility of an experimental determination of k – overall heat transfer coefficient. It follows from the heat/energy balance of a thermal node that

$$\dot{Q}_1 = \dot{Q}_2 = \dot{Q}_3 = Ak\Delta T_m \tag{14}$$

because heat flux in a fluid flow are

$$\dot{Q}_1 = \dot{m}_G c_{vG} \Delta T_G \tag{15}$$

$$\dot{Q}_3 = \dot{m}_Z c_{vZ} \Delta T_Z \tag{16}$$

Therefore,

$$k = \frac{\dot{m}_G c_{vG} \Delta T_G}{A\ \Delta T_m} = \frac{\dot{m}_Z c_{vZ} \Delta T_Z}{A \Delta T_m} \tag{17}$$

where

m_G, m_Z – flux of hot and cold fluid,
c_{vG}, c_{vZ} – specific heat at constant volume of hot and cold fluid.

3. Measures and symptoms of wear margin of tribological nodes

Deliberately constructed tribological nodes serve to efficiently and safely transfer forces and/or moments, and enable the motion of one mechanically loaded element relative to another element, where the elements are separated by liquid. Load excites the motion of one or both elements and has a component normal to the direction of motion. There is relative motion along the gap and/or perpendicular to the gap. Separation by a liquid occurs as the result of the following:

– The lubricating wedge between two elements and/or
– The effect of liquid extrusion from the gap between elements.

In terms of purpose/function:

– A stationary element of the trobological node may support a journal of rotor shaft or rotor disc (e.g., of a piston).
– Both moving elements may carry the moment from one rotor to the other (they are elements of rotor discs).
– A functional node may contain many nodes connected in parallel and/or series (roller bearing, a pair of gears).
– The effect of oil wedge occurs in nodes where the following exists [4]:
– A narrow clearance, or gap, exists between solid bodies.
– Along the gap, there is relative velocity of walls u.
– The walls are geometrically convergent.

In a convergent gap, Fig. 5, the pressure builds up [4]. According to [11], oil, thanks to the phenomenon of wetting solids by liquids, is taken by an element with greater speed and transported to the gap, which narrows in the direction of movement. As a result of super-pressure formed, the separating liquid flows out of the gap in the direction of movement and sideways (at the beginning and end of the sliding surface atmospheric pressure must prevail). Such a node is a combination of a friction pump and pressure accumulator with an overall efficiency not exceeding 1/3 (much lower than positive-displacement pumps).

In the literature on the subject, e.g. [12]:

– The coefficient of friction μ decisive for friction work W_R is given as dependent on the minimum thickness of the gap h_o and substitute roughness R_{az}.
– The minimum thickness of the gap (min. thickness of oil film) is considered to be proportional to Hersey number H_e.

$$\mu = f\left(h_o, R_{a_z}\right),\ h_o \sim H_e,\ H_e = \frac{\eta u}{q} \tag{17}$$

where η – dynamic viscosity of separating liquids; $R_{a_z} = \sqrt{R_{a_1}^2 + R_{a_2}^2}$, R_{az} – substitute measure of surface roughness of interacting walls; q – unit load of tribological node; R_{a1}, R_{a2} – arithmetic means of the profile deviation from the mean line of Walls 1 and 2, making up the tribological node.

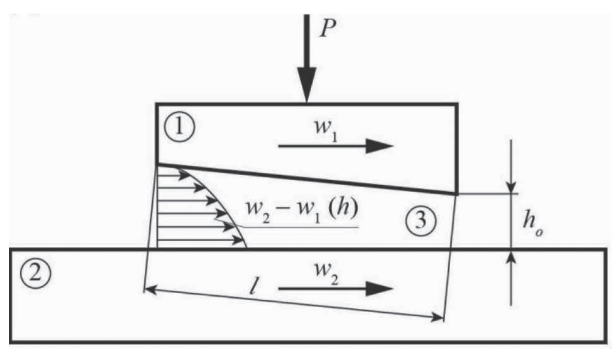

Fig. 5. A convergent gap of the node and distribution of fluid velocity in the gap: 1, 2 – node forming elements moving at speeds w_1 and w_2, 3 – separating liquid, l – gap length, h_o – minimum thickness of the gap, P – normal force loading the node

Source: Author on the basis of [4].

A convergent gap may be formed by the following:

– The elastic strain of the mating surfaces under the operating load;
– The self-aligning fixing of one element (slide base); or,
– The eccentric positioning of the journal and a shell.

As a result of local deformations caused by large Hertz pressure, elements nominally incongruent become congruent in the contact area, and due to high pressure, the viscosity increases significantly. The result is that the product of $u \cdot \eta$ becomes large enough to create a very thin lubricating film separating the mating surfaces.

The gap need not be designed as congruent, because in longitudinal bearings, the sliding surface settles (within the gap) automatically. Maximum pressure p_{max} in this bearing is proportional to the quotient $\eta lu/h_o^2$ [4].

The convergent gap arises in radial slide bearings with an off-centred shell and journal, Fig. 6.

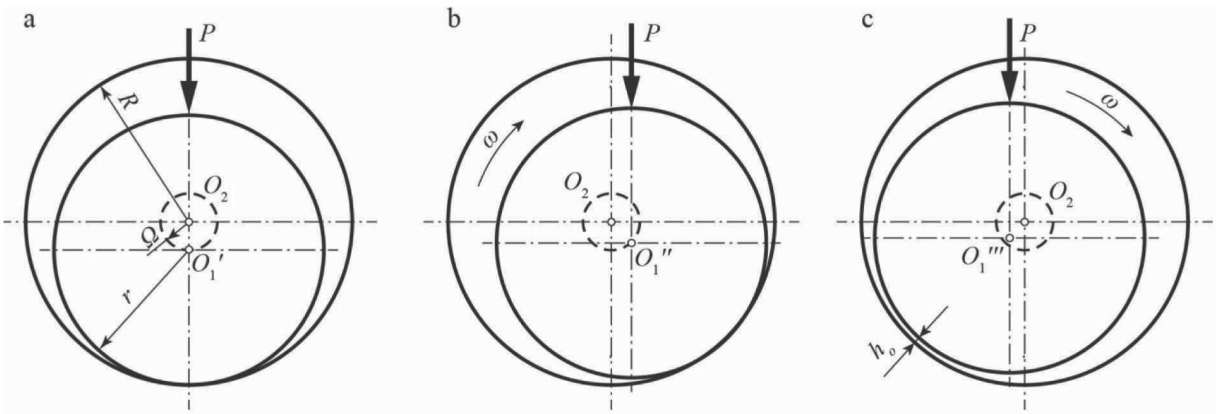

Fig. 6. Radial slide bearing: a – stationary journal, b – slow turnover of the journal at start-up, c – journal position at fluid friction, O_1 – journal centre, O_2 – shell centre, P – load, ω – angular velocity, $\Omega = R - r$ radius of the backlash circle, h_o – minimum thickness of the oil film

Source: Author on the basis of [13].

With constant values of load, bearing backlash and lubricant viscosity, as the rotary speed increases the following occur [13]:

– The journal moves in the direction opposite to the turning element, then breaks off from the shell, Fig. 6b.
– After break-off, the journal centre initially moves in the direction of rotation, then having achieved a certain rotary speed, starts to lift towards the shell centre, Fig. 6c.
– As the speed increases, the journal moves on a roughly circular path. When the angular velocity approaches infinity, the journal centre coincides with the shell centre, Fig. 7.

The position of journal centre relative to the shell depends on the equilibrium between the load and reaction force of the oil wedge. The equilibrium position is influenced mainly by the dimensionless Sommerfeld number *So*, Table 2:

$$So = \frac{P\psi^2}{bd\eta\omega} \tag{18}$$

The Sommerfeld number is a similarity criterion of hydrodynamic slide bearings working on the oil wedge principle. When bearings are similar, i.e. have the same relative length *b/d* and the same journal/shell angle, and the same Sommerfeld number, then their working conditions are similar, i.e. they show the same journal position in the shell and the same coefficient of friction [13].

Table 2. Characteristics and measures of the wear margin of a radial slide bearing

Characteristics of	Absolute and relative measures	Dimensionless quantity
shape of node elements	internal diameter *D* of the shell journal diameter *d* backlash $s = D - d$ relative backlash $\psi = s/D$ bearing length *b*	$So = \frac{P\psi^2}{bd\eta\omega}$
movement	angular velocity ω	
fluid	kinematic viscosity η	
load	force *P* unit load $q = P/bd$	

Source: Author.

Various authors use relative quantities for modelling radial bearings. The dependence of relative eccentricity ε and relative oil film thickness (1 – ε) on Sommerfeld number *So* for various *b/d* are presented in the form of diagrams [14].

For constant value and direction of the load and for a given rotary speed, the trajectory of the journal centre is a point, if the following is true:

– The resultant load is a harmonic force, and the trajectory becomes a circle, Fig. 7.

- The harmonic load, when the shaft or bearing are anisotropic, makes the trajectory become an ellipse.
- A periodic non-harmonic load produces a more complex shape of the trajectory.
- The trajectory radius depends not only on load, but also on the mode of the whole rotor and a position of the bearing relative to the rotor vibration nodes.

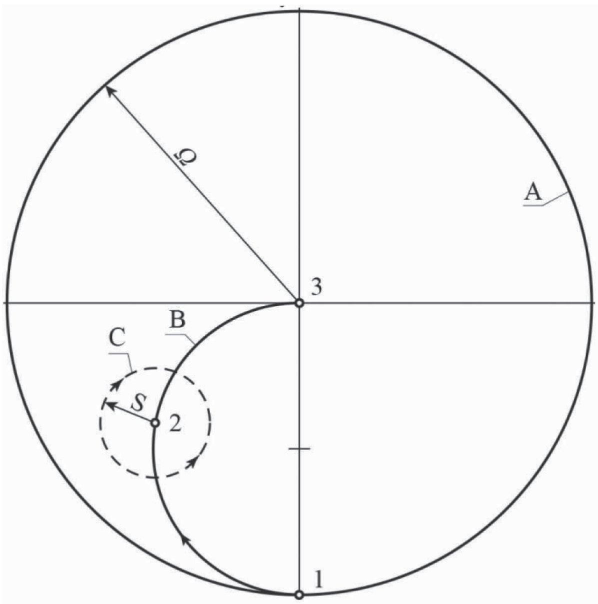

Fig. 7. A trajectory of journal centre: Ω – radius of backlash circle, A – backlash circle, B – path of average values of journal centre, C – path of variable component of trajectory, 1 – detachment of the journal from the shell 2 – average trajectory for a specific speed, 3 – theoretical position of journal centre for speeds towards infinity

Source: Author.

For bearing calculations, it is assumed that the flux in a narrow gap of the bearing is laminar, particles get relatively low accelerations, and flux lines are determined by the gap shape. At high speeds or low viscosities, additional effects occur, causing the formation of vortices and, consequently, turbulent flux. This leads to an apparent increase in viscosity, an increase in oil film thickness, and a higher coefficient of friction [15].

The efficiency of machine bearings capable of working in a wide speed range is a function of rotary speed. The dependence of the coefficient of friction in a bearing on rotary speed is described by the Stribeck curve.

A modified Stribeck curve is presented in Figure 8. It indicates a decrease in the coefficient of friction during use at constant nominal speed (section 3–4) and an increase in the coefficient of friction at zero rotary speed (6–1). The fall in section 3–4 is due to oil temperature rising to nominal bearing temperature and the resulting decrease in viscosity (change of Sommerfeld number). An increase in the friction coefficient at section 6–1 is explained by the fact that total extrusion of oil from the gap takes place after a specific time after movement is stopped [16].

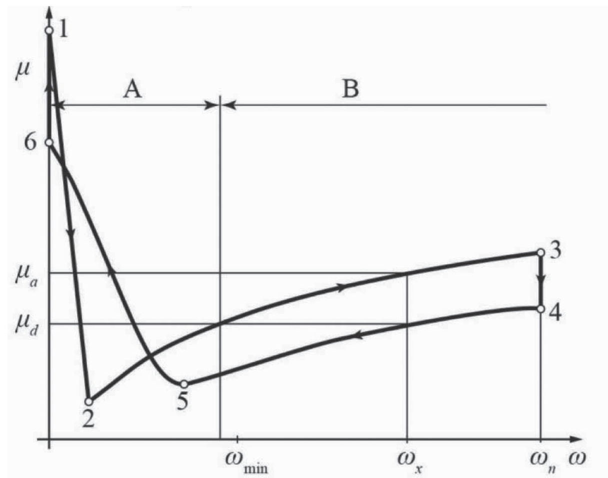

Fig. 8. The coefficient of friction as a function of rotary speed during runup and rundown: 1–2–3 runup of a bearing-supported machine; 3–4 work till constant temperature; 4–5–6 machine rundown; 6–7 extrusion of oil from non-working bearing; A – mixed friction; B – fluid friction; ω_{min} – minimum speed; ω_n – rated speed; μ_a – coefficient of friction during the run-up for ω_x; μ_d – coefficient of friction during the rundown for ω_x

Source: Author on the basis of [16].

Slide bearing wear results in the following:
- Increased backlash and/or
- A decrease in viscosity.

Increased backlash leads to an increase in Sommerfeld number. A decrease in viscosity increases Sommerfeld number. The inverse of the Sommerfeld number $1/So$ is a substitute measure of the radial slide bearing wear margin [17].

The impact of wear margin on the minimum thickness of oil film can be represented in the form of a diagram containing a collection of curves for constant values of backlash, load, and viscosity, Fig. 9. Limits of the minimum oil film thickness h_o may be established as follows:
- $h_{o\,min}$ due to a risk of bearing seizure, and
- $h_{o\,max}$ due to friction losses.

The trajectory of the journal centre is measurable. Systems for measuring trajectories, described in standards [18], enable direct oscilloscope observations of the trajectory [19] and measurements of varying trajectories. It is possible to use such systems for measuring the distance (radius) from the shell centre to the average trajectory for a given speed. However, this requires a special calibration of the system and a special algorithm combining an increase in distance with a change of the node wear margin.

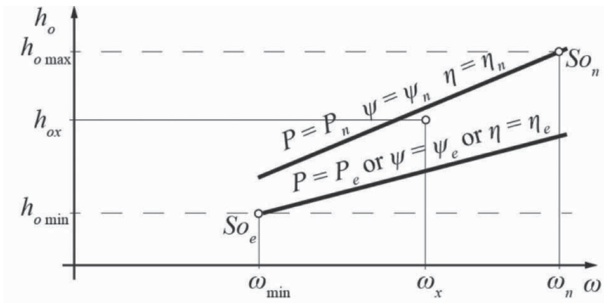

Fig. 9. The relation between thickness of the oil film and bearing wear margin: So_n – nominal value of Sommerfeld number, So_e – limit value of Sommerfeld number, index n – nominal values, index e – limit (extreme) values

Source: Author.

Vibrations of node elements can be a symptom of bearing wear margin. The pressure buildup in a liquid between elements of a tribological node means that both elements are affected by opposite forces, i.e. the product of the average pressure buildup and the surface area occupied by the 'buildup'. If the node load is variable, the pressure and forces resulting from the existence of pressure are variable. Variable forces acting on the element generate its vibrations. Values of element vibration measures depend on the following:

– The values of pressure measures, and
– Transmittance between the point of vibration measurement and the point at which a force from pressure buildup is applied.

At a variable load, besides the oil wedge effect, the effect of extrusion takes place. The pure effect of extrusion occurs when a loading force changes its sign [20].

The transition from fluid friction to mixed friction brings about an additional symptom – acoustic emission.

Summary and conclusions

An analysis of the existing physical and mathematical descriptions and of available research results permits one to indicate those quantities that can be adopted as substitute measures of the wear margin of a given node and identify related diagnostic symptom measures.

A pipeline section is an example of a hydrodynamic node. The wear margin of a pipeline reduces due to processes such as erosion, cavitation erosion, corrosion, particulate matter deposits on the inside walls.

The pipeline wear leads to changes in pipeline diameter, increased roughness of the pipes, and changed profiles of the built-in elements. This causes changes of values of wear margin substitute measures: Reynolds number, relative roughness and, consequently, coefficients of resistance.

Coefficients of resistance are diagnosable. The symptoms are pressure drops between two pipeline cross-sections and mean flow rates. Pressure drops are associated with the dissipation of pressure energy to heat energy and elastic energy of waves. The dissipation of energy takes place in areas of turbulence: areas of small vortices are formed and large vortices are shed away. The density of dissipated energy distribution is the greatest on fluid flux obstacles. The obstacles can be a result of local wear. Vortex formation and shedding is accompanied by acoustic emission. The level of acoustic emission is measurable. The increased level of emissions is due to a decrease in the wear margin of a pipeline or as a result of new obstacles formed in the pipeline.

A heat exchanger is an example of thermal node. A substitute measure of a heat exchanger, overall heat transfer coefficient k, depends on the characteristics of materials and fluids, the shape of built-in elements, the quantities describing motion and load. The overall heat transfer coefficient can range in value from k_1 to k_2:

$$k_1 \leq k \leq k_2$$

where

k_1 – 100% of node wear margin,
k_2 – 0% of node wear margin.

Thermal and erosive wear results in a change of characteristics decisive for the wear margin of a thermal node. The actual value of the overall heat transfer coefficient k changes.

The coefficient k as a substitute measure of the wear margin of a thermal node is diagnosable. The measure of symptom of thermal node wear margin is the product of hot or cold fluid flux $\dot{m}_{G/Z}$ and the ratio $\Delta T_{G/Z} / \Delta T_m$, changes of the temperature ΔT of hot or cold fluid, respectively, to the average logarithmic temperature difference ΔT_m.

A radial slide bearing is an example of a tribological node. In radial slide bearings, the following are true:

– The location of the journal centre relative to the shell and minimum thickness of oil film in a bearing depend on the dimensionless quantity known as Sommerfeld number So.
– Quantities determining Sommerfeld number are measures of the wear margin characteristics of the slide bearing.
– Wear increases Sommerfeld number, and the inverse of Sommerfeld number $1/So$ decreases along with increased backlash and load and reduced viscosity.
– The inverse of Sommerfeld number $1/So$ is a substitute measure of the radial slide bearing wear margin.
– As dependent on Sommerfeld number, the distance between the journal centre and the shell centre can be a measure of the symptom of slide bearing wear margin (In this case, it is necessary to calibrate the measuring circuit).

– If a node is under periodically variable load, the symptoms may be vibrations of a rotating or non-rotating element of the node.

For specific solutions to structural functional units, we can create models that are a combination of specific models. The wear margin of a functional unit can be described by different criteria of similarity/coefficients of proportionality related to symptoms of varying forms of energy and various shapes of the measured signal. One and the same signal can be a carrier of many wear margin symptoms.

References

1. Bielawski P.: Identyfikacja obiektów technicznych systemów produkcyjnych. Akademia Morska w Szczecinie, Szczecin 2014.
2. Bielawski P.: Miary i wartości graniczne potencjału eksploatacyjnego maszyn systemów produkcyjnych (Measures and limits of machine wear margin of production systems). (Problemy Eksploatacji) Maintenance Problem. 2016, 1, 129–159.
3. Jeżowiecka-Kabsch K., Szewczyk H.: Mechanika płynów. Oficyna Wydawnicza Politechniki Wrocławskiej, Wrocław 2001.
4. Böswirth L., Bschorer S.: Technische Strömungslehre. Vieweg + Teubner Verlag, Wiesbaden 2012.
5. Bielawski P.: Diagnozowanie rurociągów. Przegląd Spawalnictwa. 2016, 88, 10, 72–76.
6. Engel Z.: Ochrona środowiska przed drganiami i hałasem. PWN, Warszawa 1993.
7. Sperlich V.: Grundlagen der Technischen Thermodynamik. Universität Duisburg, Essen 2007.
8. Domański R., Jaworski M., Rebow M., Kołtyś J.: Wybrane zagadnienia termodynamiki w ujęciu komputerowym. Wydawnictwo Naukowe PWN, Warszawa 2000.
9. Bielawski P.: Diagnozowanie potencjału eksploatacyjnego węzłów cieplnych. Systemy wspomagania w inżynierii produkcji. Gliwice 2017, 6, 1, 7–16.
10. Moeck E., Strickert H.: Technische Diagnostik an Schiffsmaschinenanlagen. Verlag Technik, Berlin 1982.
11. Spiegel K., Fricke J.: Bemessungs- und Gestaltungsregeln für Gleitlager: Optimierungsfragen. Tribologie + Schmierungstechnik 50. Jahrgang, 2003, 1, 5–14.
12. Barwell F.T.: Łożyskowanie. WNT, Warszawa 1984.
13. Kozłowiecki H.: Łożyska tłokowych silników spalinowych. WKiŁ, Warszawa 1974.
14. DIN31652 Hydrodynamische Radial-Gleitlager im stationären Bereich.
15. Spiegel K., Fricke J.: Bemessungs- und Gestaltungsregeln für Gleitlager: Turbulenz und Instabilitäten. Tribologie + Schmierungstechnik 55. Jahrgang, 2008, 5, 16–24.
16. Spiegel K., Fricke J.: Bemessungs- und Gestaltungsregeln für Gleitlager: Anlagewinkel, An- und Auslauf, beanspruchung der Gleitflächen. Tribologie + Schmierungstechnik, 54 Jahrgang, 2007, 3, 5–16.
17. Bielawski P.: Diagnostyka techniczna obiektów systemów produkcyjnych – aktualne potrzeby i rozwiązania, [in] Wybrane zagadnienia diagnozowania i użytkowania urządzeń i systemów. Dąbrowski T., Radkowski S. (Red.) Wojskowa Akademia Techniczna, Warszawa 2015.
18. PN-ISO 7919-1. Drgania mechaniczne maszyn z wyłączeniem maszyn tłokowych. Pomiary drgań wałów wirujących i kryteria oceny. Część 1: Wytyczne ogólne.
19. Bielawski P.: Diagnostics of marine propeller shafts. Journal of Polish CIMAC. 2011, 6, 2, 31–40.
20. Bielawski P.: Elementy diagnostyki mechanizmów tłokowo-korbowych maszyn okrętowych. Studia Nr 39, WSM, Szczecin 2002.