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PROBLEMS RELATED TO FUNCTIONALITY AND UNRELIABILITY OF AXISYMMETRIC JOINTS IN THE MECHANICAL ENGINEERING

Наведено завдання, пов'язані з випадком відхилень форми для пін-шарнірів. Базуючись на круглому і циліндричному відхиленнях, здійснюється перевірка для елементів, що злучаються, моделюються різні комбінації їх зв'язку. Використовуючи непрямий метод, було оцінено геометричний (контур) проміжну поверхню між держаком і раковиною. Продемонстровано, що відхилення хрестової секції і подовжньої секції є вирішальними для розміру контактної площі.

This paper presents problems associated with occurrence of form deviations for pin joints. Basing on roundness and cylindricality deviation tests carried out for mating elements, different compilations of them in connection were modelled. Using indirect method, the geometrical (outline) contract surface between the shaft and the hole was evaluated. It has been demonstrated that deviations of the cross-section and the longitudinal section are decisive for the size of the contact area.

Introduction

Geometric shape of the actual surface of an object in only approximately consistent with nominal shape of that surface. For example, the surface of a shaft can be tapered or barrel-shaped, whereas its cross-section made by a plane perpendicular to the shaft axis can be elliptic. Deviations from the nominal shape are called geometrical deviations. Geometrical tolerances are limited only by geometrical deviations of an actual object from its nominal counterpart. That is why they are included in the group of simple geometrical tolerances. In machine industry, actual objects, like rollers or balls of bearings, rarely have elementary shape, e.g. of a cylinder or a sphere. Geometrical shape of machine parts is usually more complex, e.g. a stepped shaft consists of several cylindrical surfaces, a wheel-case is a solid with several holes, etc. In such cases, apart from necessity to meet dimensional and geometrical requirements, it is also necessary to ensure proper orientation and location of individual components. For example, surfaces of a double-stepped shaft should be coaxial cylinders, whereas axes of holes in a wheel-case should be parallel. It is not easy to achieve that for technological reasons. In order to define acceptable limits for relative deviations of direction and position of those components, ISO standards define tolerances of orientation, location and run-out. Tolerance of orientation, location and run-out are limited both by actual object shape deviations and orientation or/and location deviations. In most cases, tolerances of orientation, location and run-out require specification of the base, that is why in the ISO 1101 standard they are classified as geometrical tolerances with reference element. The ISO 1101:1985 standard defines the geometrical tolerance as the area (the tolerance range), in which the surface or line of the actual object should be included. This area has a form of a cylinder or a circle, space between two parallel planes or straight lines, space between two coaxial cylinders or circles etc. Value of the form tolerance specifies respectively: the diameter of a cylinder or a circle, the distance between planes or straight lines or the difference of radii of cylinders or circles that limit the range of tolerance. There are no limitations for (simple) form tolerances as regards the location of the range of tolerance in space. Whereas for the orientation, location and run-out tolerance, the range of tolerance is specified in space by bases [1].

Selected problems related to functionality of axisymmetric joints.

Assumption of ideal geometrical properties of the connection components, as well as about the presence of plastic strains caused by roughness of the surface makes the distribution of contact stress more uniform. On that basis, it is possible to assume simplification of calculations. Uniform contact stress is assumed both at the ideal circumference and at nominal length of the contact between components of interference joint [3]. Taking those assumptions into account results in generation of several conditions of the load of the joint.

- Condition of load for connections that transmit torque moment M_s:

$$M_s = F_P\, d/2 \le T\, d/2 \tag{1}$$

where: F_P – transverse force, $F_P = 2M_s/d$; T – friction force oriented transversely; d – rated diameter of the shaft.

The value of the friction force is calculated from the following formula:

$$T = A\mu p_{dop} = \pi d l \mu p_{dop} \tag{2}$$

where: A – contact area between the shaft and the hole; μ - friction factor, l – length of the area of contact, . p_{dop} – allowable unit load over the shaft surface and hole surface due to negative allowance;

- Condition of the connection load with transverse force F_w:

$$F_w \le T = \pi d l \mu\, p_{dop} \tag{3}$$

- The connection loaded with the transverse force F_w and the torque M_s, - calculation of the transverse force F_P and the resultant force:

$$F_z = \sqrt{F_p^{\ 2} + F_w^{\ 2}} \tag{4}$$

The form of the condition of the connection loading is as follows:

$$F_z \le T = \pi d l \mu\, p_{dop} \tag{5}$$

Presented conditions do not take into account changes of the friction factor values in main directions of the equilibrium or movement within planes of axes x, y, z caused by form deviations of mating elements. The model that takes into account variability of the friction factor has been proposed (fig. 1).

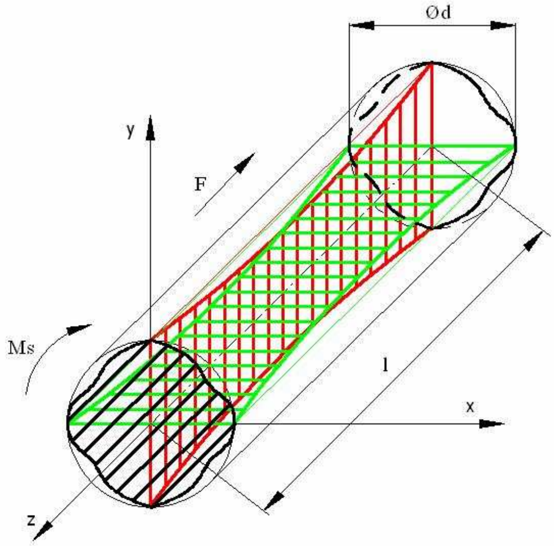


Fig. 1. The load model taking into account variation of the contact surface of the pin caused by form deviations: Quadric-lobing (plane x, y – black colour) and saddleback distortion (plane x, z – green colour; plane y, z – red colour); d – rated diameter, l – rated length of the contact surface.

- Plane (x, y) – load with torque moment M_s:

$$T = A_{rz} \cdot \mu_{xy} \cdot p_{dop} = \pi \cdot d_{rz} \cdot l_{rz} \cdot \mu_{xy} \cdot p_{dop} \tag{6}$$

- Plane (x, z) or (y, z) – load with transverse force F_w:

$$F_w \leq T = A_{rz} \cdot \mu_w \cdot p_{dop} = \pi \cdot d_{rz} \cdot l_{rz} \cdot \mu_w \cdot p_{dop} \tag{7}$$

- Plane (x, y) and (x, z) or (y, z) – load with torque moment M_s and the transverse force F_w:

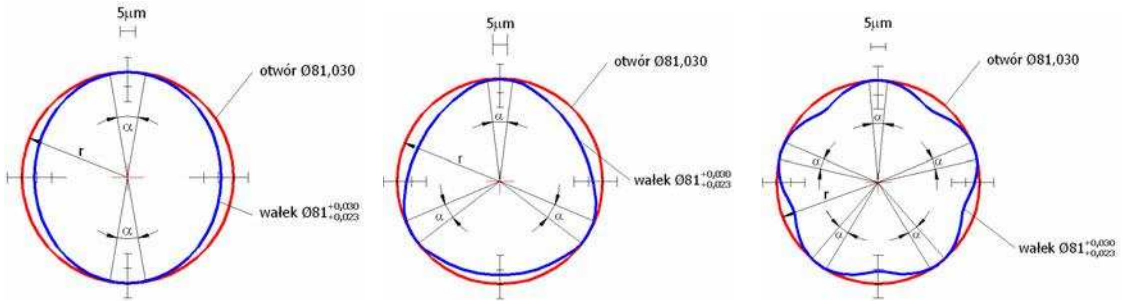
$$F_z \leq T = A_{rz} \cdot \mu_z \cdot p_{dop} = \pi \cdot d_{rz} \cdot l_{rz} \cdot \mu_z \cdot p_{dop} \tag{8}$$

Friction factor μ_{xy} and μ_w as well as μ_z are dependant on actual contour contact surface. It is impossible to explicitly calculate quality relationships of individual values of the friction factor depending on form deviations present in joints such as roundness and cylindricality. This is caused by variable values of deviations achieved in the manufacturing process. However, it is possible to indirectly evaluate geometrical (outline) contract surface of the mating elements.

Theoretic surface area that takes into account contact of elements that rub against each other over their entire geometric (outline) contact surface can be defined as follows:

$$A_t = \pi \cdot d_t \cdot l_t \tag{9}$$

Whereas actual mating occurs over small areas which transfer the load [2, 5]. In order to perform the quality analysis of the contact in the fit, as the subject of the research we assumed the pin, for which transition fit ϕ81H7/n6 occurs within that joint. Diameters of holes were measured and it was found that 32% of them were manufactured with a dimension of ϕ81,030 mm, and for pins we measured roundness and cylindricality deviations.



a) b) c)

Fig. 2. Samples of mating between an ideal hole □81,030 mm and a real shaft □ $81^{+0,030}_{+0,023}$ of the pin joint H7/n6 in cross-section: a) oval shaft, b) tri-angular shaft, c) five-angular shaft

The quality analysis of the joint was performed on the basis of measurements that were carried out. The hole ϕ81,030 mm was assumed to be ideal one and we analysed possible changes of the contact surface caused by deviations of roundness and cylindricality of the pin. Figure 2 a÷c shows cross-sections of often found patterns of mating parts.

The contact that occurs over the theoretical circumference of mating elements can be described as follows:

$$O_t = \frac{\pi \cdot r \cdot \alpha}{180^\circ} \tag{10}$$

where: r – radius of an ideal circle, *á* – angle of the contact between the shaft and the hole.

As there are different models of roundness deviations in cross-section (fig. 2 a÷c), the actual circumference of contact can be presented as follows:

$$O_{rz} = x \frac{\pi \cdot r \cdot \alpha}{180^\circ} \tag{11}$$

where: x = 2, 3,..., n (model of bi-, tri- ,..., n-angularity (lobing))

The relationship between the actual circumference of contact and the theoretical circumference in cross-section can be specified as follows:

$$\frac{O_{rz}}{O_t} = \beta \tag{12}$$

where: O_{rz} - actual contact circumference, O_t - theoretical contact circumference, β – the coefficient of change of the contact circumference between mating elements.

As a result of performed calculations for examples presented in figure. 1, the actual values of circumferential contact between mating elements were achieved. They are as follows, respectively: (fig. 2 a) – 11%, (fig. 2 b) – 12% and (fig. 2 c) – 14% of the theoretical circumference assumed for calculations.

The form deviations are present also in longitudinal section of the pin (fig. 3 a÷c) and those deviations influence the size of the contact surface.

Fig. 3. Parts selected for measurement: cylinder and piston.

Relationships between the actual length of the contact surface and the theoretical length of it (fig. 3 a÷c) are defined by the following formula:

$$\frac{l_{rz}}{l_t} = \gamma \tag{13}$$

where: l_{rz} – actual length of the area of contact, . *lt* – theoretical length of the area of contact, γ – factor of the contact surface length change.

As a result of performed calculations for presented examples (fig. 3), actual values of the length of contact surface between mating elements were achieved. They are as follows, respectively: (fig. 3 a) – 10%, (fig. 3 b) – 9% (fig. 3 c) – 17% of theoretical length of contact area between the shaft and the hole surfaces that was taken into account.

The following zones can be specified in the presented joints between the shafts and the hole: the zone of contact and the zone of no contact between surfaces. The ratio of the contact to no contact zones is respectively (fig. 3 a) – 10%, (fig. 3 b) – 9% (fig. 3 c) – 17%. Existing form deviations in the cross section and longitudinal section of the shaft determine the value of the contour contact surface, the area of which usually does not exceed a dozen or so percent. Whereas possible compilations of roundness and cylindricality deviations reduce its value to several percent of the nominal surface.

Similar analysis and assessment of the influence of design properties on the size of the contact surface should also be performed for holes of pin joints.

Unreliability of axisymmetric joints

Unreliability of axisymmetric joints results from many factors and can be systemized in many ways. First, the unreliability can be discussed at the stage of design, then at the stage of production and finally at the stage of measurement (control and identification). It determines the parameter connected with the measurement result that specifies the dispersion of the measured quantity value.

When measuring geometrical structure of the surface using instruments equipped with mapping point, there are numerous components of the measurement unreliability. They include environmental conditions, the measuring instrument, the software, the measured piece as well as influence of the mapping point itself. Plenty of conditions influence the measuring error during measurement.

The following measurements were carried out:

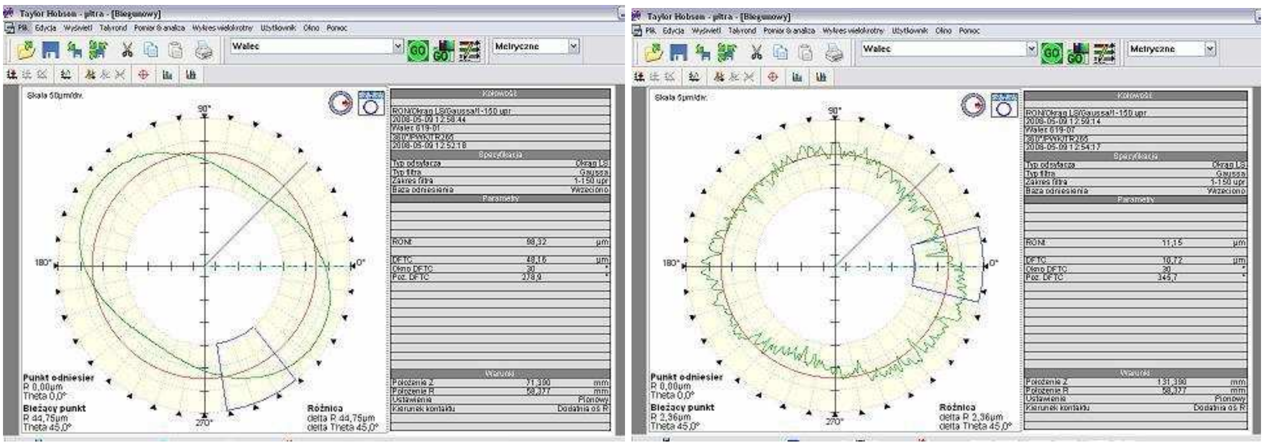
- cylinder diameter
- piston diameter

Diameters of elements selected for research were measured in two or three planes.

Measurement of the cylinder

The cylinder was measured by means of basic measuring instruments at marked cross-sections and by means of *Form Talysurf Series 2 instrument, manufactured by the company Taylor- Hobson*. The analysis of measurement results as well as calculations aimed at determining the roundness and cylindricality deviations shows that the roundness deviation can be classified as a specific case, i.e. the ovality deviation. Whereas the cylindricality deviation can be classified as the case of taper. To make sure, it is necessary to compare obtained results to measuring protocols of measurements carried out using special instruments for form deviation analysis.

Figure 4.a and b shows measurement records where the cylinder roundness deviation can be seen.



a) b)

Fig. 4. a) Roundness of analysed cylinder in plane I: -roundness deviation Δ_o=18,32 µm
b) Roundness of analysed cylinder in plane II: - roundness deviation Δ_o=11,15 µm.

Whereas figure 5 shows the measuring record obtained using the form deviation instrument, *Talyrond 265*, and it includes the value and the type of cylindricality deviation for the analysed cylinder.

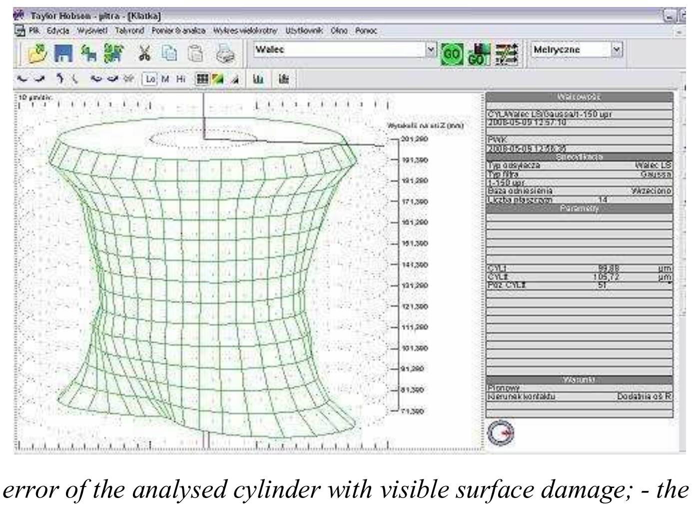


Fig. 5. Cylindricality error of the analysed cylinder with visible surface damage; - the value of cylindricality deviation Δ_w= 89,88µm

When we compare roundness and cylindricality deviation values obtained using “roundness gauge” to the results obtained from measurement carried out using callipers, one can conclude that those results differ from each other, especially as regards cylindricality deviation. This results from the fact that there were numerous damages to the cylinder. The character and size of the cylindricality deviation was determined only during measurement performed using specialized instrument. Its shape should be classified as a specific case, i.e. the saddleback distortion, unlike it had been determined earlier, as taper distortion.

Piston measurements

Measurements of the form deviations for the piston shown in the figure 5.10 were carried out using basic measuring instruments and by means of the roundness gouge Talyrond 265

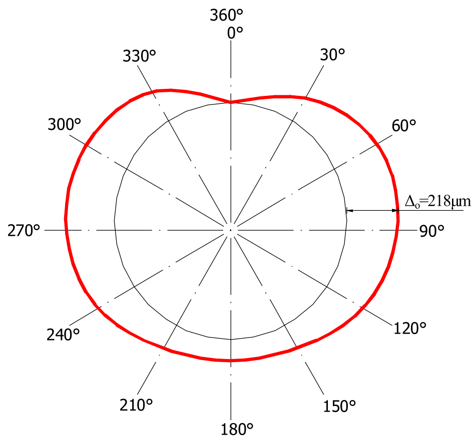
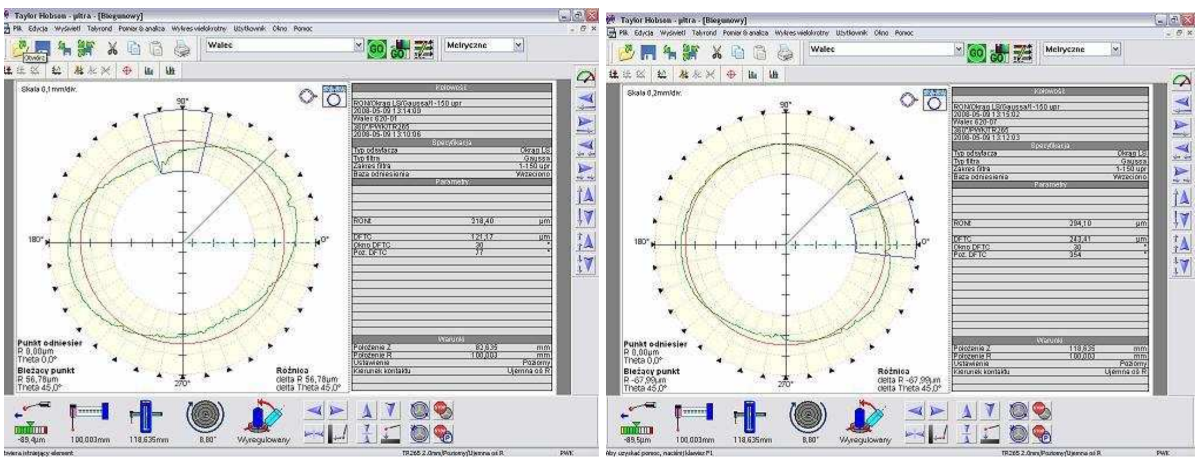


Fig. 6. Piston roundness diagram

Whereas figure 7 shows the measuring record obtained using the instrument *Talyrond 265*, manufactured by *Taylor- Hobson, and* it includes the value and the type of roundness deviation for the analysed piston.



a) b)

Fig. 7. a) Roundness of analysed piston in plane I: -roundness deviation Δo=218,40 μm
b) Roundness of analysed piston in plane II: - roundness deviation Δo=224,10 μm

Whereas figure 8 shows the measuring record obtained using the instrument *Talyrond 265*, manufactured by *Taylor- Hobson, and* it includes the value and the type of cylindricality deviation for the analysed piston.

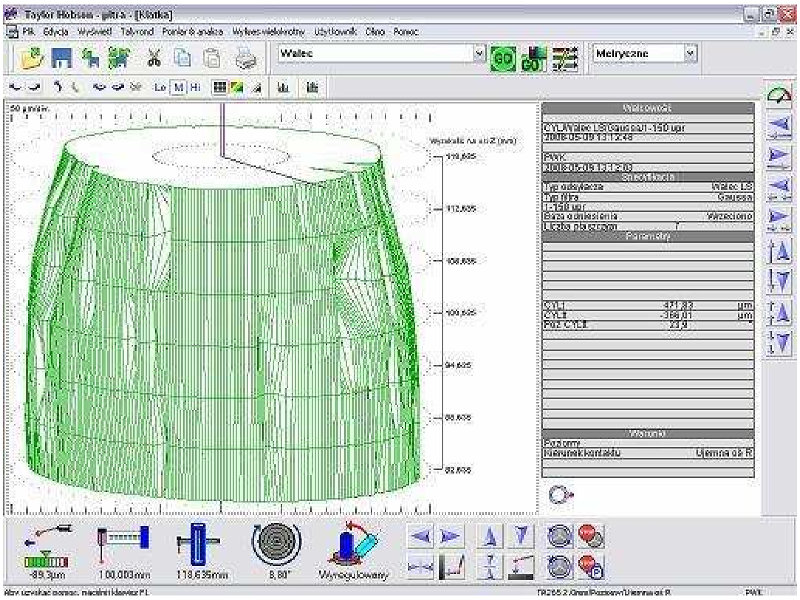


Fig. 8. Cylindricality error of the analysed piston with visible dent in the surface;
- cylindricality deviation Δw=471,33μm

When comparing the roundness deviation value determined using dial gauge and its value measured using special instrument, one can conclude that measurement in the repair workshop was performed correctly. However, if the value of the deviation calculated on the basis of measurement performed using extensometer was compared to the previous one, the measuring error would be about 20μm. Whereas when comparing the value of cylindricality deviation, one can notice very large spread of measuring results. This results from the type of the piston wear, because the terminal cross-section, that was characterised by largest wear, was not measured in the repair workshop.

Conclusions

The most popular way of the analysis of unreliability is determining the minimum and maximum value of the analysed parameter. However due to performance deviations of the axisymmetric joints, it is necessary to apply more complex methods. Actual functioning of those joints is associated with: reduction of the contour contact area, change of function (negative allowance, play), assembly stresses, stress concentration, friction factor values spread, which reduces ability of load transfer. Assessment of actual functionality of axisymmetric joints depends on appropriate selection of the fit, the shape, the geometric structure of the joint components and taking into account its complete structure in the XYZ co-ordinate system.

1. Białas S., Metrologia techniczna z podstawami tolerowania wielkości geometrycznych dla mechaników wyd. Ш popr. 2006r. 2. Bober A., Dudziak M.: Zapis konstrukcji, Wydawnictwo Naukowe PWN, 1999r. 3. Domek G., Kołodziej A.: Jakościowe aspekty wytwarzania połączeń w przemyśle maszynowym, Mechanik nr 12/2007r., str. 1034 – 1038. 4. Hebda M.: Procesy tarcia, smarowania i zużywania maszyn, Wydawnictwo Instytutu Technologii Eksploatacji – PIB, Warszawa 2007r. 5. Humienny Z.: Specyfikacje Geometrii Wyrobów (GPS) – wykład dla uczelni technicznych, Oficyna Wydawnicza Politechniki Warszawskiej, 2001r. 6. Kołodziej A.: Zagadnienia tarcia w wybranych połączeniach kształtowych – a bezpieczeństwo tych połączeń, Wydawnictwo IPBM PW, 2004r., str. 128 – 134. 7. Kołodziej A.: Projektowanie cech jakościowych kształtowych połączeń sworzniowych i kołkowych. Rozprawa doktorska. Poznań, 2006.