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Modeling of contact stresses and evaluation of wear life of a valve mechanism guide with lubricating grooves

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Abstract

Valve guides are crucial for maintaining proper alignment, positioning, and valve stem clearance as it moves in the cylinder head. Valve bushings are subjected to load and sliding friction; with excessive wear of the guides, the engine begins to consume oil and the valve mechanism becomes noisy. This paper proposes the use of a special knurling to restore and increase the wear resistance of guide bushings. The profile tool is designed to restore valve guides by rolling a spiral groove on the inner surface of the sleeve. After applying this technology, a spiral oil-retaining profile remains on the surface of the sleeve bore. . Using the Solid model, the effect of changing the geometry of the guide bore by the lubricating grooves on the maximum and average stress indices in the guide-valve contact was analyzed. It was found that the maximum stresses of the model with grooves are lower than those of the model with a smooth guide surface. Based on the finite element model, the durability of the valve-guide pair with oil-retaining grooves was analyzed and it was determined how many cycles the contact surface of the guide can withstand. It was established that by reducing the contact pressure, the actual resource of the guide with grooves increased.

Keywords: guide bushing, valve, lubricating grooves, finite element analysis, contact pressure, resource.

Introduction

Valve guides, which are critical for maintaining proper alignment, positioning, and clearance of the valve stem as it travels within the cylinder head, are typically made of materials that provide high wear resistance and improved thermal conductivity. Valve bushings are subject to friction loads. Lateral forces act on the valve stem caused by changes in geometry in the valve mechanism, wear of the rocker cam or rocker arm. When the guides are heavily worn, the engine begins to consume oil and increased noise of the valve mechanism appears. This applies to both the intake tract (vacuum in the cylinder) and the exhaust tract (Venturi effect).

In this work, the use of a special knurling tool is proposed to restore and increase the wear resistance of guide bushings. The tool is designed to restore valve guides by rolling a spiral groove on the inner surface of the sleeve. After applying this technology, a spiral oil-retaining profile remains on the surface of the sleeve hole, which:

• increases the oil capacity of the surface, and therefore, improves the lubrication conditions in the friction pair "valve-guide";

• creates a gas labyrinth seal in the connection, which prevents oil from entering the combustion chamber;

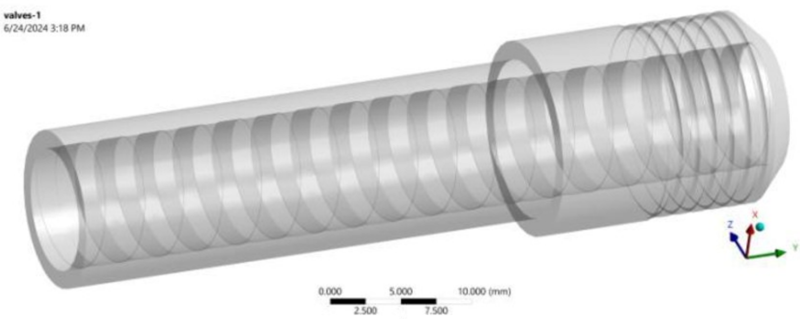
• strengthens the surface of the sleeve hole due to surface sealing (hardening effect).

The creation and study of the tribological properties of the lubricating profile on the inner surface of cylindrical sliding guides has been given attention in scientific papers [1-9].

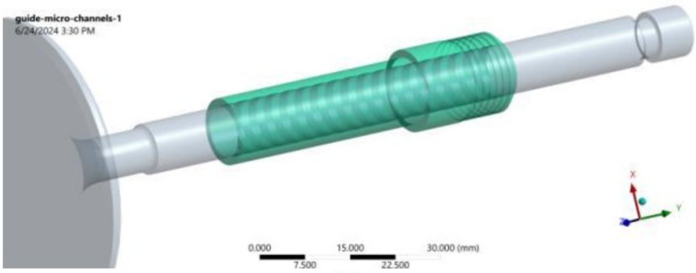
In this work, computer modeling of the performance of a valve guide with spiral grooves was carried out using the lubricity criterion.

The geometry of the oil retaining profile and the lubricity of the profile

Let's analyze the effect of changing the geometry with lubrication grooves of the guide hole on the indicators and . Using the method of extrusion of the body of the solid model, we draw a channel with a width of 1.5 mm and a depth of 0.05 mm in steps of 3 mm (Fig. 1).$\sigma_{max}\sigma_{ave}$



a)



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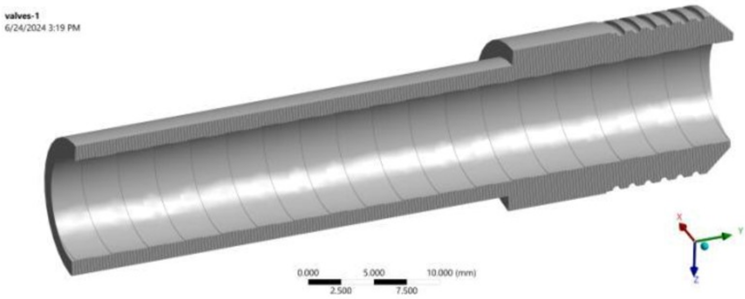


c)

Fig. 1. Solid model of a guide with channels: a, b) guide separately and assembled with a valve; d) cross section of the guide

The gap between the valve flap and the guide remains uniform and unchanged relative to the boundary conditions of the previous modes: 0.03358 mm in the lower points of the spiral (protrusions) and increased by 0.05 mm in the upper ones (channel depressions).

Consider the hypothesis that the presence of channels in the valve guide improves the lubrication regime, thereby reducing the coefficient of friction between the valve stem and the guide.μ_v

Factors of the lubricity of the oil-retaining profile of the grooves:

-Improved oil flow: channels in the guide can improve oil distribution and flow, increasing the formation of a lubricating film between surfaces.

-Hydrodynamic lubrication: Improved oil flow can change the lubrication regime from marginal or mixed lubrication to hydrodynamic lubrication, where a full film of lubricant separates the contact surfaces, greatly reducing friction.

-Reduced metal-to-metal contact: Channels help maintain a continuous oil film by reducing direct metal-to-metal contact, thus reducing the coefficient of friction.

The Reynolds equation describes the pressure distribution in a thin film of oil between two surfaces. For a simplified one-dimensional flow in the presence of channels, this can be expressed as:

$$\frac{\partial}{\partial x}\left(h^3 \frac{\partial p}{\partial x}\right) = 6\mu U \frac{\partial h}{\partial x},$$

where: – film thickness, mm; – pressure in the film, Pa; – dynamic oil viscosity, Pa·s; – relative speed between surfaces, m/s; – spatial coordinate along the length of the contact, mm.hpμUx

The force of friction () in the hydrodynamic mode of lubrication can be estimated using:F_f

$$F_f = \tau A = \frac{\mu U A}{h},$$

where: – shear stress in oil, Pa; – contact area, m²;τA

The coefficient of friction is determined by the ratio of the force of friction to the normal load ():F_f

$$\mu_f = \frac{F_f}{F_N} = \frac{\mu UA}{hF_N}$$

For a typical guide without channels, the film thickness can be less uniform and thinner, resulting in more friction due to more frequent metal-to-metal contact.h

For a guide with channels, the film thickness is more uniform and thicker due to improved oil flow, resulting in reduced friction.h_c

Assuming that the channels increase the effective thickness of the film by a factor (where):$\alpha\alpha > 1$

$$h_c = \alpha h,$$

the coefficient of friction with channels can be expressed as:μ_{fc}

$$\mu_{fc} = \frac{\mu UA}{\alpha hF_N}$$

Due to the increase in the thickness of the lubricating film and the homogeneity through the channels, the effective coefficient of friction decreases by the value of , confirming that the presence of channels can lead to a decrease in the coefficient of friction. In the modes studied below with a profiled guide (with channels), the value of the coefficient of friction is set.$\alpha\mu_v = 0.05$

Analysis of the stress state of the guide and valve with an oil-retaining profile

Taking into account the increase in the volume of space in the contact pair, it was empirically established that for similar constructions of the guide with channels, the valueconvection heat transfer increases to 400-600 W/(m²·°C). Let's apply the value of convection equal to 450 W/(m²·°C).

The difference in the boundary conditions between the previous and the current studied regime is only in growthconvection from 300 to450 W/(m²·°C), which is the result of modification of the geometry of the contact surface of the guide (adding channels according toFig. 1)

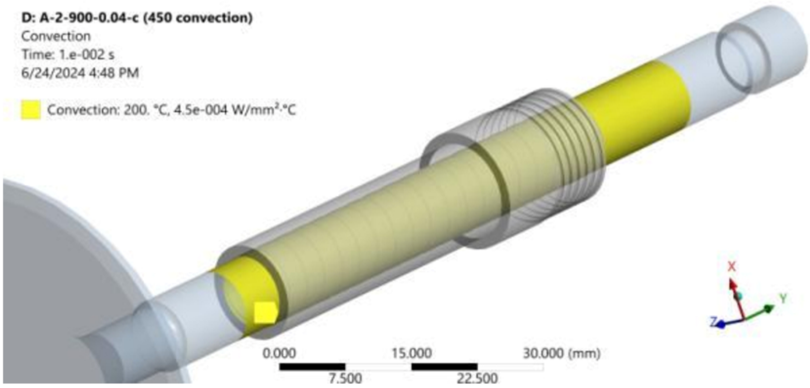


Fig. 2. Application of convection 450 W/(m²·°C) to the guide-valve contact pair

Let's analyze the influence of the channels in the guide on the resulting valve stresses (Fig. 2):

- the contact surface of the valve was subjected to23.5% lower maximum stresses in the channel modelσ_{max}compared to mode (smooth guide hole). The extremum is fixed at a moment in timet_m=0.002 s. The graph has a jumpy character, which is caused by the relief surface of the guide during the beginning of the contact.Unlike the "smooth" mode, where growth is recorded$\sigma_{max}$to 21.67 MPa followed by a decrease to 13.19 MPa at the end of the experiment (blue curve at 4.32), the uniformity of the curve(red curveon) clearly favors the surface of the valve valve. Tensions range from: 14.27 MPa as oft_m=0.0072 s (Fig. 4.23 b), 14.84 MPa (t_m=0.0145 s - fig. 4.23 c) and 13.6 MPa (t_m= 0.0244with);

- the average stress value is 4.24 MPa, which is 6.27% higher thanσ_{ave}mode without grooves.The explanation for this result lies in the "point-likeness" of stress transfer from the protrusions of the guide to the smooth surface of the valve.Both extremes occur at the end of the experimentt_{vo}= 0.04 s.

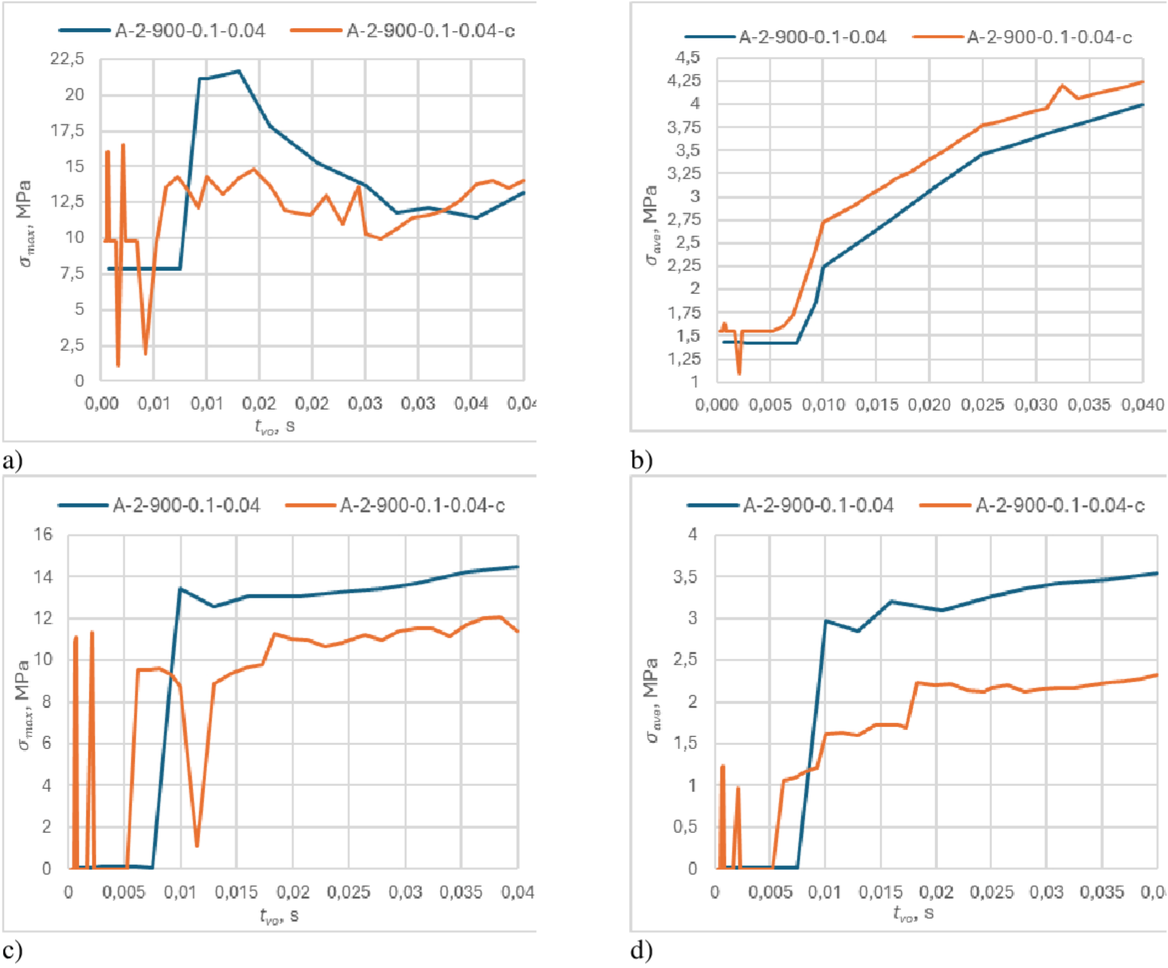


Fig. 3.The effect of the presence of channels on the stress during: a, b) and on the contact surface of the valve; c, d) and on the contact surface of the guide, respectively$t_{vo}\sigma_{max}\sigma_{ave}\sigma_{max}\sigma_{ave}$

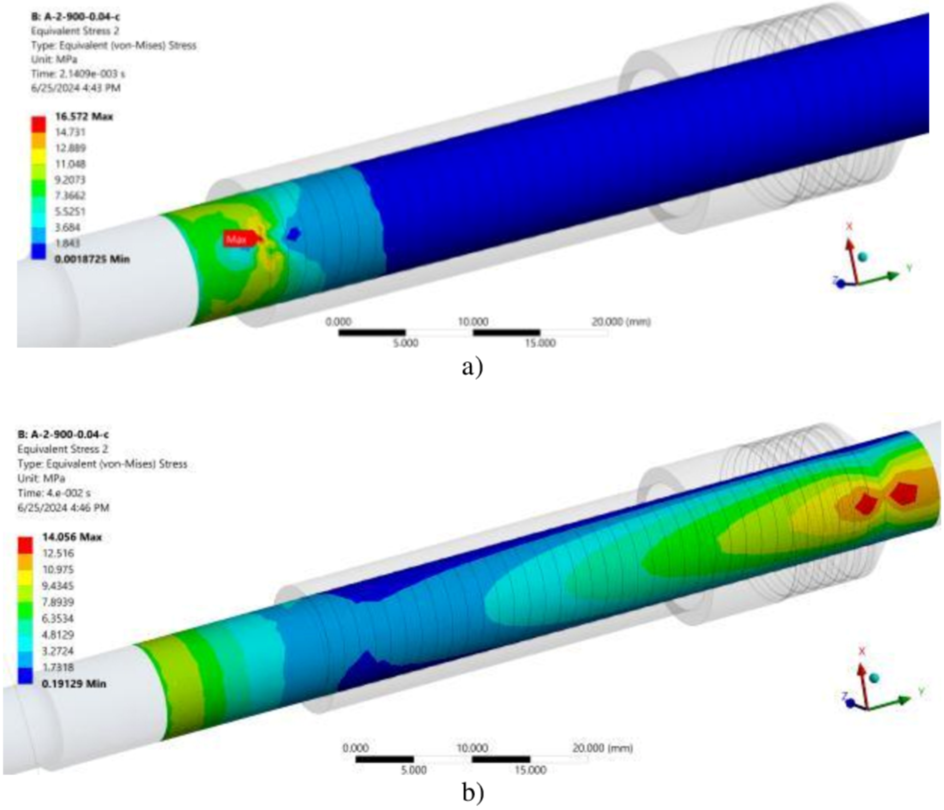
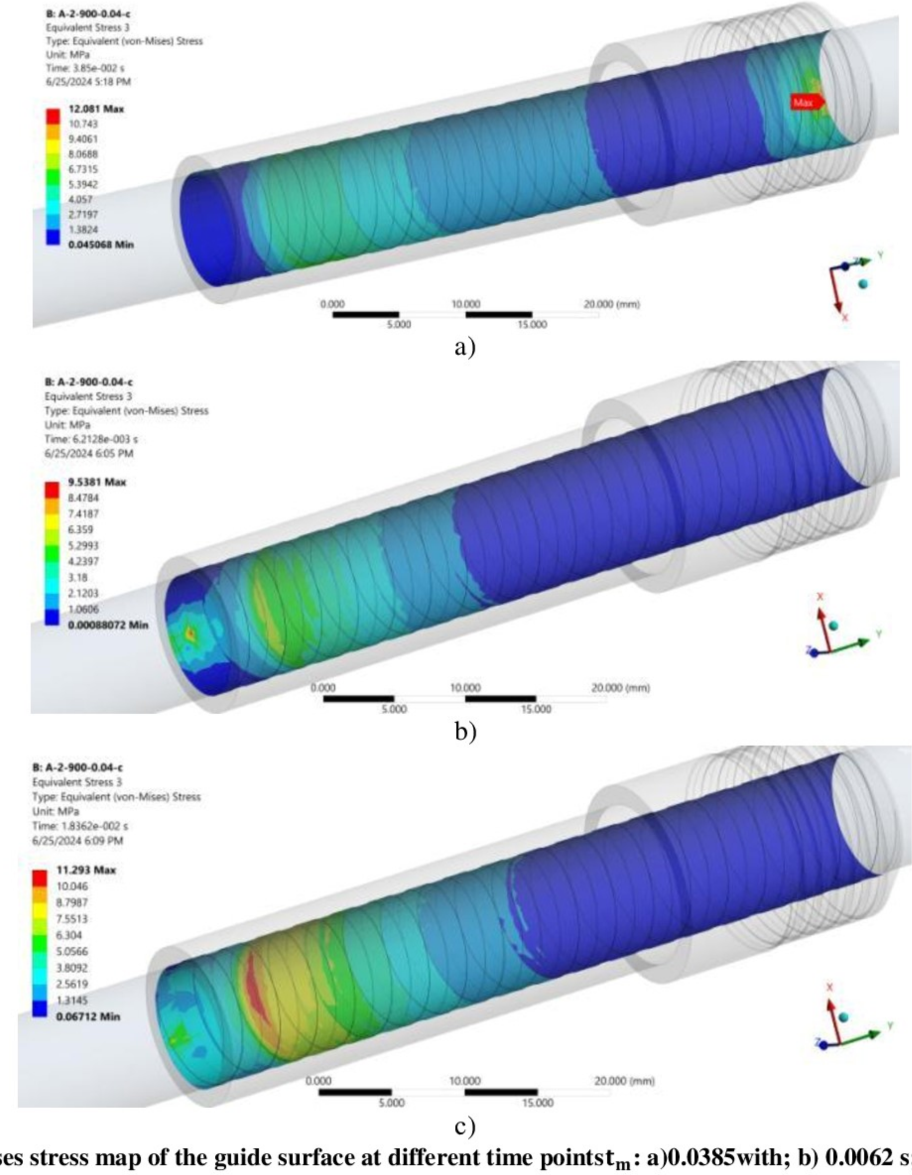


Fig. 4. Mises stress map of the valve surface at different time pointst_m: a) 0.0021 s; b) 0.04 s

The results on the contact surface of the guide are as follows:

- maximum stresses of the model with the channel$\sigma_{max}$are 12.08 MPa, which is 16.7% lower than the mode with a smooth guide surface. Valueσ_{max} achievedat the moment of time $=t_m$0.0385 s (Fig. 4.34 a). Locationσ_{max}changes during the duration of the experiment, for example, as of $=t_m$0.00214 s (Fig. 4.34 b)$\sigma_{max}$is 11.34 MPa and is observed at the edge of the opposite opening of the guide, and then shifts towards the center of the guide: =9.54 MPa at the moment of time $=\sigma_{max}\sigma_{max}t_m$0.00621 s (Fig. 4.34 c) and 11.29 MPa int_m=0.0183 s. During the first 0.012 s, oscillations were recorded until the contact between the valve and the guide was stabilized. Further, the stress curve is roughly parallel, but lower by 2.0-2.5 MPa. Thus, throughout the experiment, the valueσ_{max}in the first mode with the channel is lower, which positively affects the wear of the guide surface;
- unlike the smooth surface of the guide, the Mises stress map has clear contours within the spiral of the channel. This effect is of practical benefit - by selecting the desired geometric parameters of the channel, for example, the step, it is possible to control the distribution of stresses on the surface of the guide;
- the average stress value is 2.32 MPa, which isσ_{ave}-34.5%lower than the firstmode without grooves.The explanation for this result lies in the transfer of stresses from the guide to the smooth surface of the valve.σ_{ave}By analogy withσ_{max}average stresses also exhibited fluctuations during the first milliseconds of contact. Both extremesσ_{ave} come at the end of the experimentt_{vo}= 0.04 s.

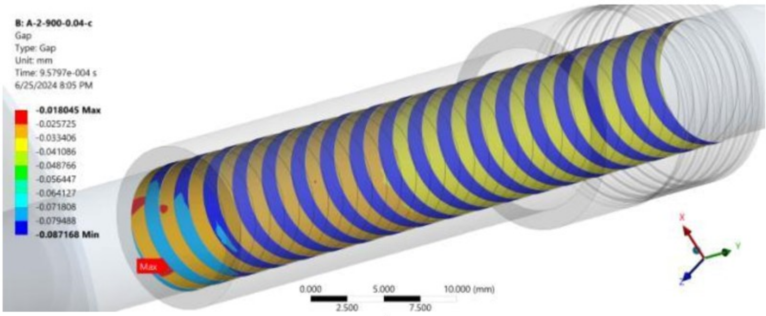


a) b) c)

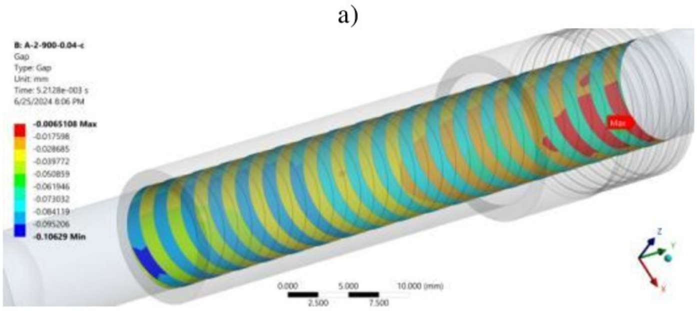
Fig. 5.Mises stress map of the guide surface at different time pointst_m: a)0.0385with; b) 0.0062 s; c) 0.0183 s

Study of the thickness of the lubricating layer (gap) in the "valve-guide" pair

To confirm that there is no contact between the valve and the recesses of the guide channels, use the Contact Tool > Gap tool. In fig. Figure 4.35 shows the value of Gap at different moments of time - the gap in the depressions remains throughout the experiment, as evidenced by the blue color of the scale.



a)



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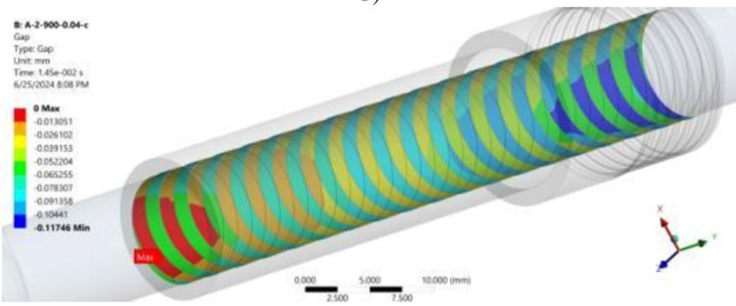
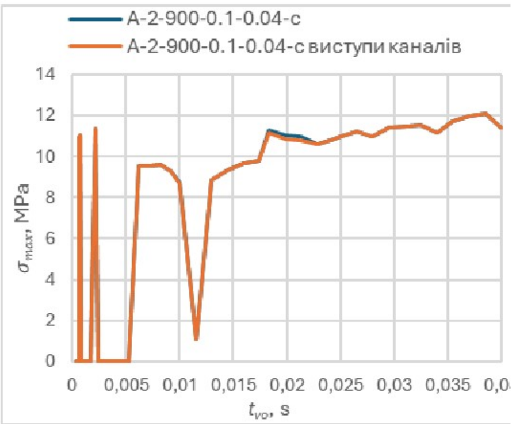


c

Fig. 6. Map of the gap (gap) in the contact area at the moment of time: a) 0.00096 s; b) 0.0051 s; c) 0.0145 st$_{m}$

Let's investigate the situation with stresses on the surface of channel protrusions, comparing the results with the entire surface (valves together with protrusions). The curves of the maximum stresses coincide throughout the experiment (Fig. 6a), except for the period 0.018–0.021 s, where it is even higher for the entire surface (together with depressions). This is quite unexpected, because the protrusions are always the first to perceive contact, which was confirmed based on the evaluation of the size of the gap (gap). The situation with average stresses is more unambiguous - it is lower by 2-3% in the case of only the surface of the protrusions during the entire experiment = 0.04 s (Fig. 6b). This indicates that the depressions concentrate higher stresses on themselves. As a rule, they appear in the corners of depressions, where theoretically the greatest bending moment occurs (the protrusion acts as a cantilever beam).$\sigma_{max}\sigma_{ave}t_{vo}$.



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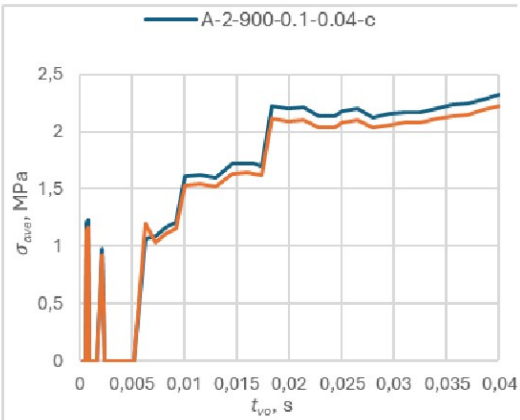
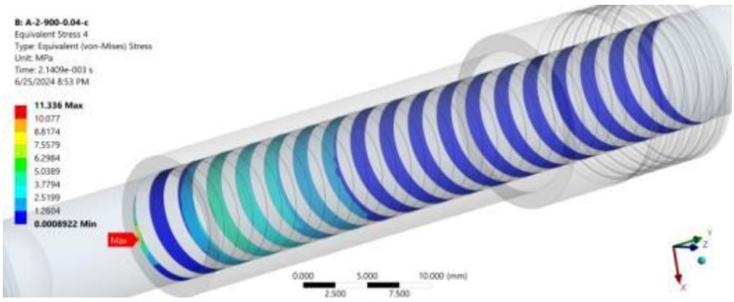


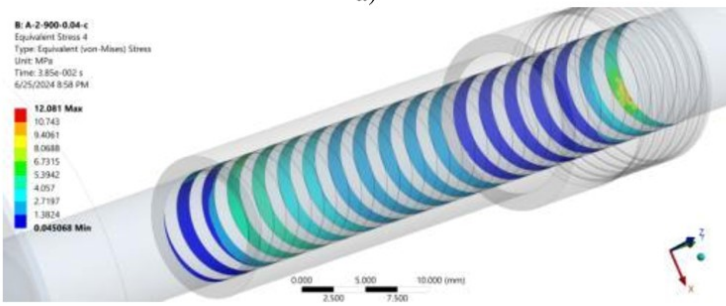
b)

Fig. 7. Comparison of stresses on the surface of channel protrusions with the entire surface of the guide hole: a) ; b)$\sigma_{max}\sigma_{ave}$

Intermediate stress states exclusively on the surface of the protrusions are shown in Fig. 8. Consolidated stress data are summarized in table. 1.



a)



b)

Fig. 8.Mises map of stresses exclusively on the surface of the protrusions of the guide at different moments of timet_m: a)0.0021with; b) 0.0385 s

Table 1

Summary results for stresses and for Ansys model modes$\sigma_{max}\sigma_{ave}$

Contact surface of the valve						
Regime	σ_{max}, Mpa	$\Delta\sigma_{max}$, %	$t_{\sigma_{max}}$, p	σ_{ave}, Mpa	$\Delta\sigma_{ave}$, %	$t_{\sigma_{ave}}$, p
A-2-900-0.1-0.04	21.67	+50.3	0.013	3.99	+15.0	0.04
A-2-900-0.05-0.04-c	16.57	-23.5	0.002	4.24	+6.27	0.04
The contact surface of the guide						
Regime	σ_{max}, Mpa	$\Delta\sigma_{max}$, %	$t_{\sigma_{max}}$, p	σ_{ave}, Mpa	$\Delta\sigma_{ave}$, %	$t_{\sigma_{ave}}$, p
A-2-900-0.1-0.04	14.50	+3.4	0.04	3.54	+24.2	0.04
A-2-900-0.05-0.04-c	12.08	-16.7%	0.0385	2.32	-34.5%	0.04
A-2-900-0.05-0.04-c*	12.08	0%	0.0385	2.23	-3.9%	0.04

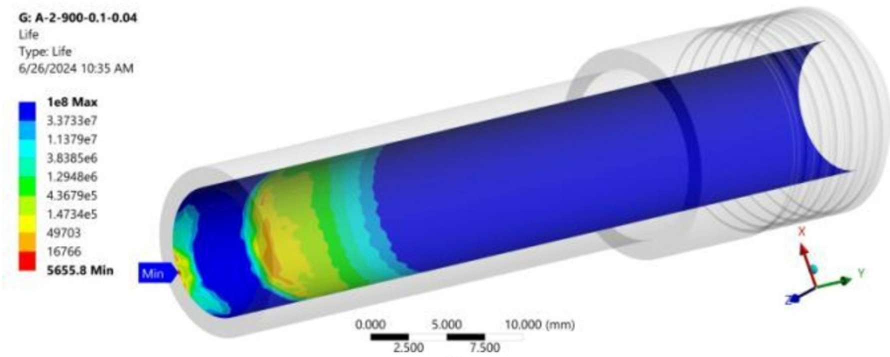
* the results of channel protrusion stresses

Durability of the "valve-guide" pair with oil retaining grooves

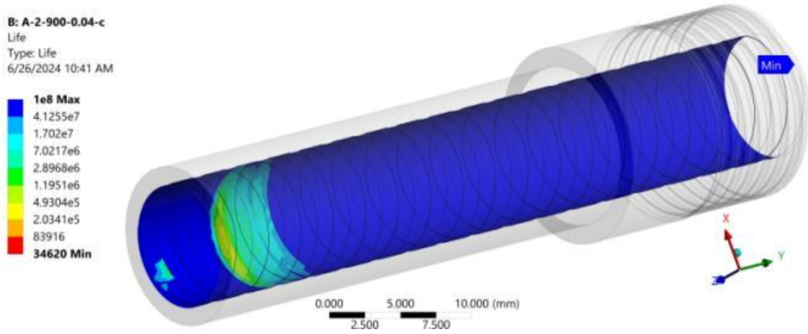
Let's set the "Scale Factor" value to 25 and measure how many cycles (Fatigue Tool > Life) the contact surface of the guide can withstand with a 25-fold increase in each of the modes:$N_c\sigma_{max}$

mode A-2-900-0.1-0.04 receivedσ_{max}at a point in time t_m=0.04 s and demonstrated =5655.8 cycles (Fig. 9 a);N_c

mode A-2-900-0.1-0.04-cshowed =34620 cycles (Fig. 9 b) according to at the moment of time$N_c\sigma_{max}$ t_m=0.0385.



a)



b)

Fig. 9. Maps of wear resistance based on the tool4 Tool Fatigue Tool > Life: a) regimeA-2-900-0.1-0.04 int_m=0.04 s; b) regimeA-2-900-0.1-0.04-c int_m=0.0385 s

Thus, by reducing the valueσ_{max}on16.7% (table 4.3), the actual resource of the guide with channels increased by more than 6 times.

Conclusions

1. Using the Solid model, the effect of changing the geometry with lubrication grooves of the guide hole on the indicators of maximum and average stresses in the "guide-valve" contact was analyzed. It was established that the maximum stresses of the model with grooves are 12.08 MPa, which is 16.7% lower than the mode with a smooth guide surface.

2. Based on the finite-element model, the durability of the "valve-guide" pair with oil retaining grooves was analyzed. It was determined how many cycles the contact surface of the guide can withstand with a 25-fold increase in each of the modes. It was found that due to the reduction of the value by 16.7%, the actual resource of the guide with grooves increased by more than 6 times.

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Голенко К.Е. Вичавка А.А., Диха М.О., Дитинюк В.О. Моделювання напружень в контакті і оцінка ресурсу по зносу напрямної клапанного механізму із змащувальними канавками

Направляючі клапанів мають вирішальне значення для підтримки правильного вирівнювання, позиціонування таформування зазору штока клапана під час його руху в головці блоку циліндрів. Втулки клапанів піддаються навантаженню і тертю ковзання, при великому зносі направляючих двигун починає витрачати масло і з'являється підвищений шум клапанного механізму. У даній роботі запропоновано використання спеціальної накатки для відновлення та підвищення зносостійкості напрямних втулок. Профільний інструмент призначений для відновлення напрямних клапанів шляхом накочування спіральної канавки на внутрішній поверхні гільзи. Після застосування цієї технології на поверхні отвору гільзи залишається спіральний маслоутримуючий профіль. За допомогою моделі Solid проаналізовано вплив зміни геометрії мастильними канавками направляючого отвору на показники максимального та середнього напружень у контакті «направляюча-клапан». Встановлено, що максимальні напруження моделі з канавками нижче, ніж у моделі з гладкою напрямною поверхнею. На основі кінцево-елементної моделі проаналізовано довговічність пари «клапан-направляюча» з маслоутримувальними канавками і визначено, скільки циклів витримує контактна поверхня напрямної. Встановлено, що за рахунок зменшення значення контактного тиску фактичний ресурс направляючої з канавками збільшився.

Ключьові слова: напрямна втулка, клапан, мастильні канавки, скінчено-елементний аналіз, контактний тиск, ресурс