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Finite-element analysis of contact characteristics and friction modes of the "valve-guide" of the internal combustion engine

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Abstract

Modeling the performance of the "valve-guide" engine pair using modern software is an effective tool both for identifying weak points in the design and for predicting the behavior of the friction unit in operation. In this study, the method of finite element analysis was chosen as a tool to study the contact and antifriction parameters of the friction pair of the internal combustion engine "directional valve". Using the FEM application program, the raw data on the material, surface dimensions, loads, and motion kinematics are described. Based on the constructed finite-element model of the "valve-guide" conjugation, an analysis of the influence of determining tribological factors: sliding speed in contact, temperature, skew angle, friction coefficient on contact stresses both for each part of the friction pair and in the process of contact interaction was carried out. A consolidated matrix of the results of the numerical experiment was built, and the conclusions regarding the influence of each factor on the tribological characteristics were substantiated. Algorithms of influence on the design, technological and operational factors for prolonging the resource of the friction pair of the internal combustion engine "klpapn-napramna" are outlined.

Keywords: internal combustion engine, valve guides, finite element model, contact parameters, friction coefficient, stressed surface state

Introduction, analysis of research

Modeling the performance of the "valve-guide" engine pair using modern software is an effective tool both for identifying weak points in the design and for predicting the behavior of the friction unit in operation. This issue has received considerable attention in the scientific literature. In the article [1], filling materials are considered to reduce the weight of intake or exhaust valves of an internal combustion engine. Micro-computed tomography was used to reverse engineer the original component and assess the valve's internal geometry and material integrity. The valve has been redesigned using Finite Element Analysis (FEA) to select a lightweight weighted design that provides weight savings over the original equipment valve. The article [2] describes the concept of a non-invasive method of diagnosing the value of valve gaps in internal combustion engines, based on the analysis of engine surface vibration signals using artificial neural networks. The method uses as diagnostic signals the readings of vibration sensors, which record the acceleration of the engine head depending on the angle of rotation of the crankshaft, with pre-set values of the valve clearances measured in the cold state. In article [3], a study of the causes of engine intake valve damage was conducted, during which the intake valve heads were overheated and deformed as a result of material creep. On the example of a malfunction detected in the analyzed engine, it was established that traditionally known causes, such as a failure of the combustion process, cannot cause the described damage. The performed calculations showed that with an increase in the rotation frequency, the failure of the control system leads to an increase in temperature higher than recommended for the materials used. Based on the conducted research, the authors have developed recommendations for increasing the reliability of intake valves with variable gas distribution phases. In [4], a methodology for the analysis of valve wear of internal combustion engines is proposed, which is the result of the combined use of numerical and experimental methods. Numerical solutions are obtained using a specialized finite element method where a solution contact algorithm is used to model the flexible-flexible contact along with the adhesive wear law. Experimental results are obtained on a wear

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test rig specially designed to evaluate wear parameters under valve operating conditions. A good agreement was found between the experimental wear profiles and the numerical calculations of the wear on the contact surfaces. In [5], engine reliability was improved using Al-Sic composites for engine guide valves. Finite element analysis of Al-Sic composite with titanium alloy,copper-nickel silicon alloys and aluminum bronze alloy as an alternative material for the engine valve guide was carried out using Ansys 13.0 software. The finite element method is one of the most widely used methods for analyzing mechanical load characteristics in modern engineering components. The directional valve model was modeled as shown in Fig. 15. A finite element model was built to perform the analysis of the guide valve. The stress analysis of the engine valve guide at different pressures and temperatures was carried out. It was found that the stresses were significantly lower than the allowable for all materials, but the Al-Sic composites were found to be the most optimal. The purpose of the work [6] was to determine the main parameters affecting this wear. The approach was based on the tribological triplet and material flows within the contact, using both numerical and experimental approaches. A dynamic model and valve train test bench showed that wear flows can be activated by the architecture of the valve opening system. Therefore, limiting these flows can be achieved by controlling the geometry of the system without changing the properties of the materials. In the same way, the finite element model of the local response of the seat-valve contact emphasized the influence of the "local" contact geometry. As noted in [7], intake and exhaust valves are important engine components used to control intake and exhaust gas flow in internal combustion engines. They are used to seal the working space inside the cylinders and are opened and closed by means of a valve mechanism. The study is devoted to various types of failure of internal combustion engine valves: due to fatigue, exposure to high temperature, shock load. In works [8-9] it was shown that sliding in the valve sealing area is one of the main causes of wear. Sliding wear is expected to play an even more important role in modern engines. Experimental data obtained using a special technique on a test stand are presented. Experimental data are supplemented by FEM modeling. The simulation involves checking the sliding behavior of the valve seal area on a test bench and investigating how different parameters affect the sliding length. These parameters include combustion pressure, contact angle, contact length, valve head thickness, friction coefficient, run-in wear, and change in modulus due to temperature variations.that stresses were significantly lower than allowable for all materials, but Al-Sic composites were recognized as the most optimal. The purpose of the work [6] was to determine the main parameters affecting this wear. The approach was based on the tribological triplet and material flows within the contact, using both numerical and experimental approaches. A dynamic model and valve mechanism test bench showed that wear flows can be activated by the architecture of the valve opening system. Consequently, limiting these flows can be achieved by controlling the geometry of the system without changing the properties of the materials. In the same way, the finite element model of the local response of the seat-valve contact emphasized the influence of the "local" contact geometry. 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Output data for modeling and calculation

Valves in automotive and industrial applications are usually made of materials that can withstand high temperatures, pressures and active corrosive environments. The choice of material largely depends on the specific application and operating conditions. Stainless steel is widely used because of its resistance to corrosion and strength at high temperatures. Grades such as AISI 304 (UNS S30400) and AISI 316 (UNS S31600) are common in the chemical and food industries. For applications at higher temperatures, such as engine exhaust valves, heat-resistant stainless steels such as AISI 347 (UNS S34700) or AISI 321 (UNS S32100) are used.

Titanium-based alloys are alternative materials for the manufacture of valves - they are most often found in the cylinder heads of sports and "custom" car engines with increased requirements for reliability at high speeds and temperatures. The last indicator is primarily influenced by the coefficient of thermal expansion, which is a measure of how much the material expands for one degree of temperature increase. A specific example of a titanium alloy commonly used for high-performance engine valves is Ti-6Al-4V, also known as grade 5 titanium. This alloy is known for its excellent combination of strength, light weight and corrosion resistance, making it a popular choice in aerospace, automotive and the medical industry.

Valve guides, which are critical to maintaining proper alignment, positioning, and clearance of the valve stem as it moves through the cylinder head, are typically made from materials that offer high wear resistance and improved thermal conductivity. The choice of material for guide valves largely depends on the operating conditions and requirements of the engine or mechanism. Manganese bronze is a group of high-strength, hard bronzes that are typically used in assemblies that require a combination of high strength, wear resistance, and corrosion resistance. Such alloys provide excellent durability and heat dissipation properties.

The object of research is a prefabricated solid model of the valve together with a guide (Fig. qe a, b) as part of the head of the Brodix 10X cylinder block (Fig. qe c, d, g). The 10X modification belongs to the line of high-performance cylinder heads produced by Brodix, which is well-known in the field of automotive auto parts, including heads, blocks and manifolds. Brodix 10X cylinder heads are designed for installation in high-performance and frequently modified Chevrolet (General Motors) “Small-Block” racing gasoline engines. Introduced back in 1955, the Chevrolet Small-Block V8 engine has become a staple of GM vehicles thanks to its versatility and performance, and remains the most widely produced in the world. Displacement can vary between 262–400 cubic inches (4.3–6.6 L). Brodix 10X heads have a modified valve mechanism: stronger valve springs and rocker arms for reliable high-rpm operation, valves made of titanium-containing stainless steel (such as AISI 321 steel with the formula X6CrNiTi18-10) or directly titanium alloys such as Ti-6Al- 4V. Such heads have gained particular popularity among tuning enthusiasts for street racing (Drag Racing) or professional wheel racing, but remain reliable and adaptable enough for daily use. The Ansys model of the valve guide is made of C86300 bronze.

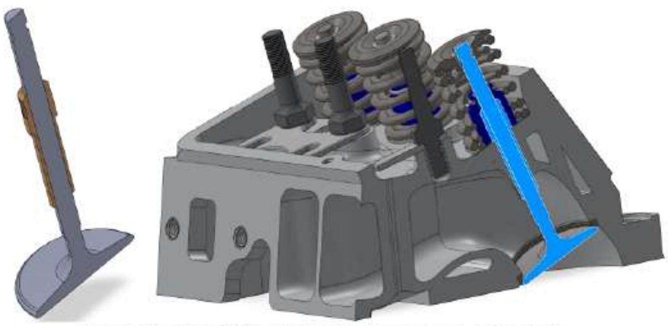
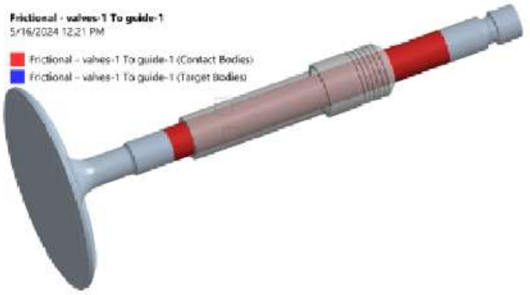
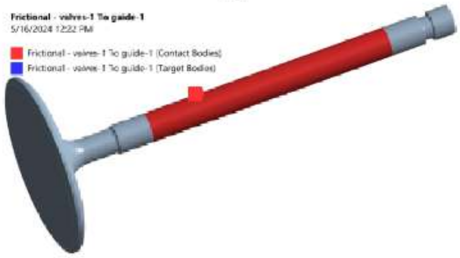


Fig. 1. Brodix 10X guide valve and head assembly

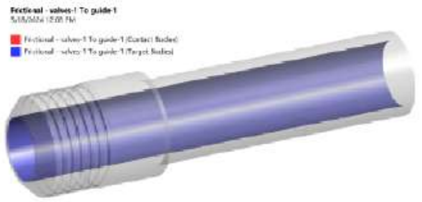
Additionally, for the contact surfaces between the valve stem and the guide, the detailing of the grid has been increased with the maximum element length Element size = 1.0 mm. The type of contact is Frictional with successive application of different friction coefficients: = 0.1 and 0.2 according to the boundary conditions, the summarized data for which are presented below. To achieve convergence of forces in the contact area, the Normal Stiffness > Factor parameter is set to 0.1 for some of the test modes μ_v .



a)



b)



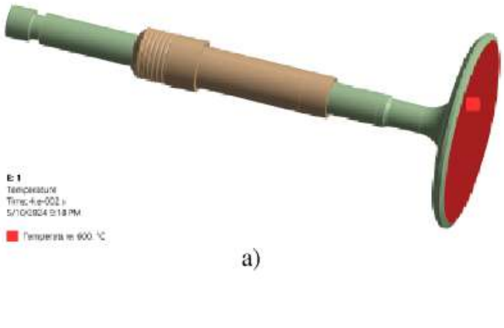
in)

Fig. 2. Frictional contact: a) valve stem with a guide; b) contact surface of the valve stem; c) target (Target) surface of the guide.

Statistics of the grid of finite elements (Fig. 3): 153399 nodes; 91621 elements.

Fig. 3. FEM mesh of the valve with a guide (Ansys Coupled Field Static)

The surface temperature of the exhaust valve can reach 600-900°C, while the intake valve usually operates at lower temperatures as it is cooled by the incoming air or fuel mixture. During the fourth and final exhaust stroke, the valve opens to allow the exhaust gases to leave the combustion chamber– it is during this period that the highest temperature on the surface of the valve head (Valve Surface Temperature) of the Ansys model is reached (600°C in Fig. 4). The following calculations will consider both temperature cases: 600 and 900°C T_{vs}



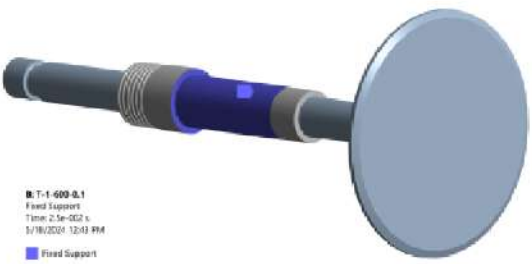
a)

b)

Fig. 5 Temperature model a) and application of convection b)

For the Chevrolet V8 engine, which is a typical liquid-cooled engine, the valve stems are primarily cooled by engine oil and the surrounding material of the cylinder head with additional heat transfer to the coolant. The convective heat transfer coefficient of the valve guide where the stem operates will depend on the oil film (or lack thereof) that lubricates the contact area, typically ranging from 100 to 500 W/(m²·K). Apply the value of 300 W/(m²·°C) to our calculation model. Other surfaces are set to a convection value of 100W/(m²·K) for the same temperature loads on the valve surface:600 or 900°C (depending on the mode).

In order to fix the model, we will apply restraints with restriction of movements and rotations relative to all three axes (hard clamping, corresponding to the type of Fixed Support in Ansys) to the following surfaces of the valve guide in accordance with its attachment in the body of the cylinder head (Fig. . 6).



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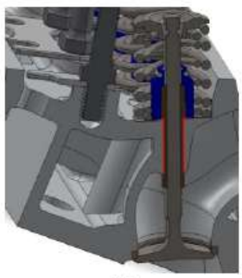
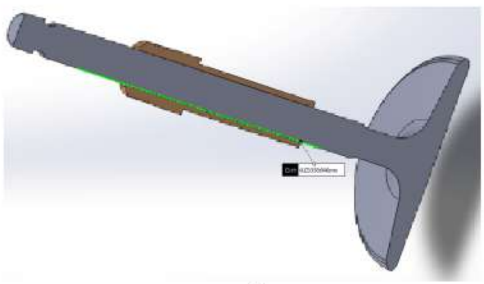


b)

Fig. 6. Attaching the Fixed Support type yokes to the valve guide: a) selected surfaces; b) cross-section of the block head with marking of guide mounting surfaces (red lines)

To form the boundary conditions of the kinematics and dynamics of the movement of the "valve–directional" in Ansys, we will analyze the main processes of the exhaust stroke): at the beginning or just before the start of the stroke, the exhaust valve is opened, controlled by the camshaft, which has cams designed to press on the valve pushers, effectively opening them at the right moment.

The exhaust valve moves down as the cam presses against the valve tip. The kinematics of this process is determined by the valve guide– a cylindrical part that ensures smooth and precise movement of the valve stem without lateral movement or rotation. In fact, there is a gap between the valve stem and the inner cylindrical surface of the guide, which in the studied model of the Brodix 10X block head is 0.03358 mm (Fig. 7) and is symmetrical about the longitudinal axis of the valve.



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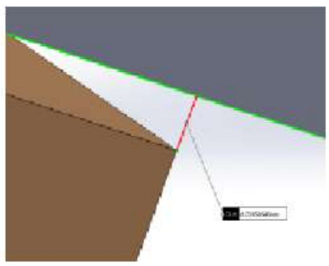


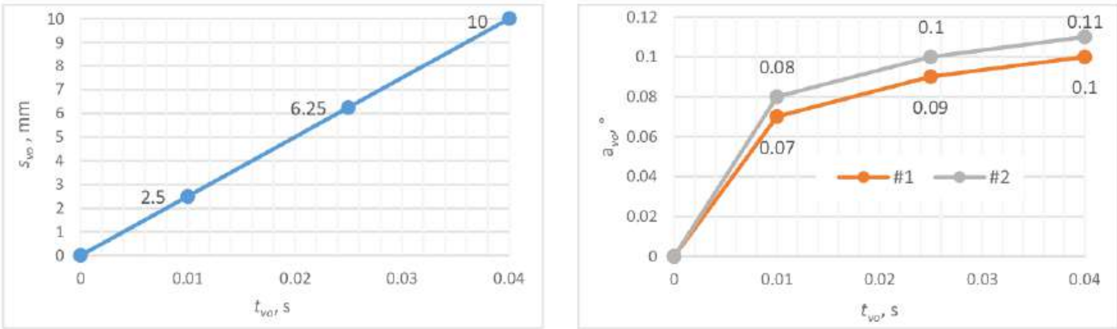
b)

Fig. 7. The gap between the valve stem and the guide: a) longitudinal section; b) enlarged view of the Solid model

In the next step, we will form the boundary conditions regarding the kinematics of the valve movement:

- the valve opening duration is = 0.04 s, which approximately corresponds to the idle speed of a low-speed enginet_{vo}“Small-Block” Chevrolet V8 (600-800 rpm);
- the stroke of the rod is linear from 0 to 10 mm within = 0.04 s (Fig. 8);$s_{vo}t_{vo}$

- the skew angle during the stroke of the rod is stepwise and is analyzed for two variants of its intensity (curve #1 and #2 in Fig. 8)$a_{vo}s_{vo}$
- step-by-step application and (Number of Steps = 3): 0 – 0.01 c; 0.01 - 0.025 c; 0.025 - 0.04 c;$s_{vo}a_{vo}$



a) b)

Fig. 8. Kinematics of valve movement: a)stroke of the rod; b) skew angles$s_{vo}a_{vo}$

It should be noted that the misalignment angle occurs as a result of the pressure of the cam on the valve tip: turning, the camshaft in addition to the normal forces that actually ensure the strokea_{vo}rod, also creates tangential ones. The stabilizing factors of the valve are the upper support plate of the spring and directly the valve guide with a gap. At the same time, the lower part of the valve (the base) is in a relatively free position during the opening stage (out of the seat) and exhibits oscillations during the opening time, which will be demonstrated below in the discussion of the research results. We consolidate the boundary conditions of the Ansys model in Fig. 8 and in table. qg and consider the following designations using T-1-600-0.1-0.04 as an example:$s_{vo}t_{vo}$.

- T–valve manufacturing material (T–Ti-6Al-4V or A–AISI 321);
- 1– skew angle curve number (#1 or #2 according to the graph in Fig. 8);a_{vo}
- 600– temperature on the surface of the valve head (600 or 900°C);T_{vs}
- 0.1– coefficient of friction between the rod and the guide (0.1 or 0.2);μ_v
- 0.04– valve opening duration (0.04 or 0.01 s).t_{vo}

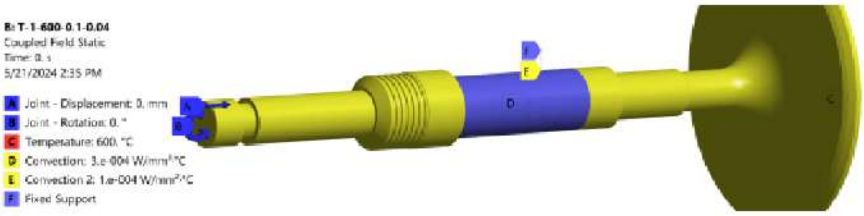


Fig. 9. Consolidated boundary conditions on the example of mode T-1-600-0.1-0.04

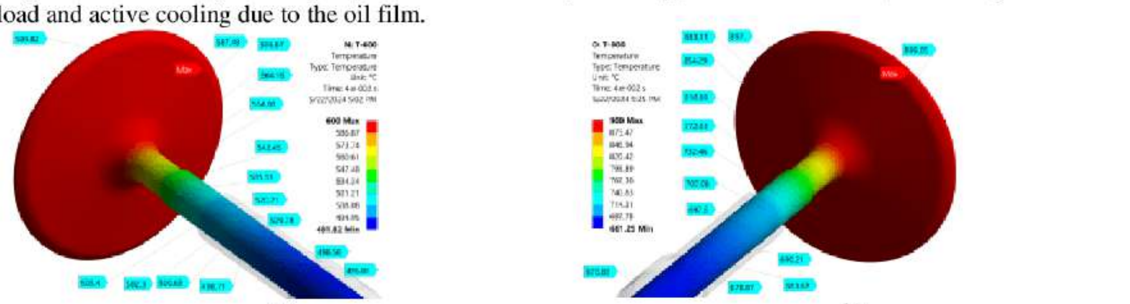
Table 1

Legend of calculation modes of the Ansys model

Name of the mode	Material	a_{vo}, #	t_{vo}, c	T_{vs},°C	μ_v	The purpose of the regime's tasks
T-1-600-0.1-0.04	Ti-6Al-4V	1	0.04	600	0.1	influence of the material
A-1-600-0.1-0.04	AISI 321	1	0.04	600	0.1	
T-2-600-0.1-0.04	Ti-6Al-4V	2	0.04	600	0.1	influence of the angleskew
A-2-600-0.1-0.04	AISI 321	2	0.04	600	0.1	
T-2-900-0.1-0.04	Ti-6Al-4V	2	0.04	900	0.1	effect of temperature
A-2-900-0.1-0.04	AISI 321	2	0.04	900	0.1	
T-2-900-0.2-0.04	Ti-6Al-4V	2	0.04	900	0.2	the effect of the coefficient friction
A-2-900-0.2-0.04	AISI 321	2	0.04	900	0.2	
T-2-900-0.1-0.01	AISI 321	2	0.01	900	0.1	influence of engine revolutions
A-2-900-0.1-0.01	AISI 321	2	0.01	900	0.1	

Temperature distribution by model

Application of temperature load$T_{vs}$to the surface of the valve head(Fig. 10) made it possible to obtain a map of the temperature distribution on the surface of his body. The dynamics of the temperature drop along the length of the valve significantly depends on the intensity of the applied convection. So, for an engine under low load and active cooling due to the oil film.



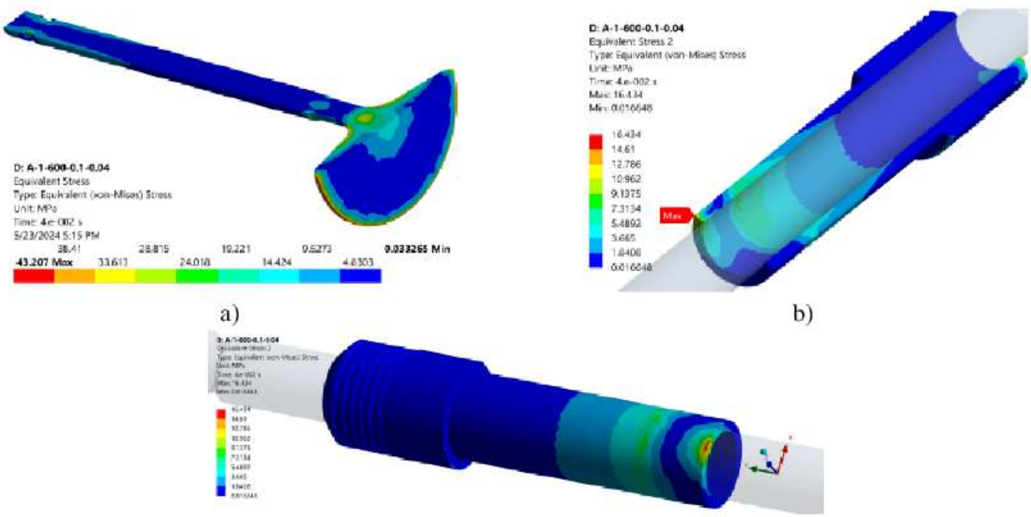
a) b)

Fig. 10. Temperature distribution along the length of the titanium valve (Ti-6Al-4V): a) = 600T_{vs}°C; b)T_{vs}= 900°C

The temperature near the end of the rod (valve tip) can drop to 150-200°C. Under the conditions of increased operating revolutions (over 2500-3000 rpm) and weaker cooling, the temperature can rise to 450-720 °C.

The influence of the material

We will begin the analysis of the stress state of the model with modesT-1-600-0.1-0.04 and A-1-600-0.1-0.04in accordance withtasks formedintable 1. "Equivalent (von-Mises) Stress" stress maps in Ansys of valves and guides (cross section in Fig. 11):



a) b)

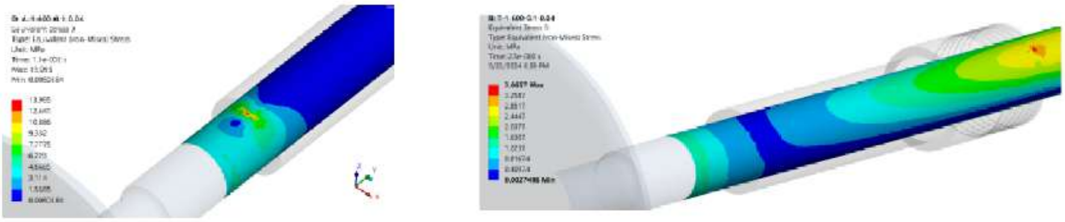
c)

Fig. 11. Mises stress maps of the valve and guide

As of = 0.04 c, the maximum and average stresses are: 22.46 MPa in the titanium valve and 43.21 MPa in the steel valve. In both cases, they are "thermal" and fixed on the edge of the valve head, where no mechanical interactions occur according to the set boundary conditions. The result shows that the steel sample is almost twice as heat-resistant, which is also reflected in the results (6.83 vs, 3.34 MPa for the titanium valve). guides are 13.98 MPa in combination with a titanium valve and 16.43 MPa – with a steel valve, respectively. The regularities here are as follows: 1) higher valve stresses provoke higher guide stresses (combination$t_{vo}\sigma_{max}\sigma_{ave}\sigma_{ave}\sigma_{max}$AISI 321–Bronze C86300); 2) guide demonstrated on the inner surface (zone of contact with the titanium valve), and when in contact with the steel one - on the outer surface; 3) as a result of valve misalignment, the maximum stresses in the body of the guide are observed along the diagonal of its opening. The average stresses in the body of the guide are 0.745 and 0.398 MPa when interacting with a steel and titanium valve, respectively.$\sigma_{max}\sigma_{ave}$.

Let's analyze the situation with the contact surface of the valve stem (Fig. 12). The more ductile steel sample of the valve received significantly higher stresses at the moment of first contact = 0–0.0005 s, where sliding occurred, and showed a peak stress of 16.09 MPa (= 0.00031). At that time, the titanium valve received only 2.73

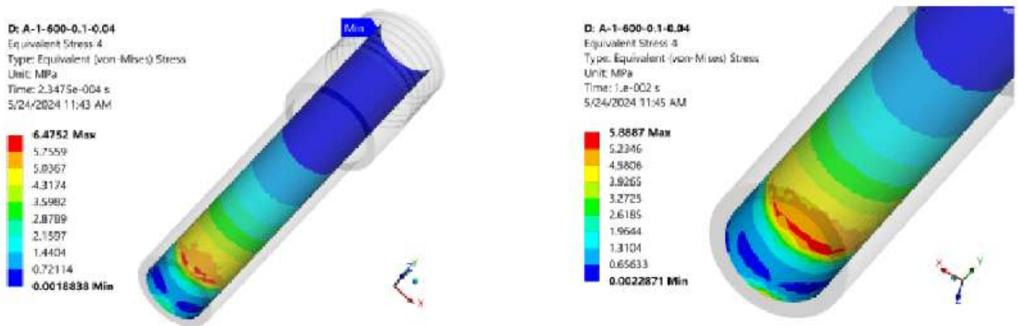
MPa (= 0.00043 s). After the end of the first step of distortion (angle) at = 0.013 s, the steel valve receives the next peak of maximum stresses = 13.99 MPa, which is clearly visible on the stress map in Fig. 12. In measure$t_m t_m t_m a_{vo} t_m \sigma_{max}$of the stroke of the stem and changes, the titanium valve changes the side of contact with the guide closer to the tip (the steel valve shows a similar behavior). As of the end of the experiment, titanium and steel valves acquire values of 5.21 and 9.41 MPa, respectively.$s_{vo} a_{vo} \sigma_{max}$



and) b)

Fig. 12. Mises stress map of the contact surface of the rod: a) steel (c); b) titanium (c)$t_m = 0.013$ $t_m = 0.025$

Let's examine the indicators and the contact surface of the guide (mode$\sigma_{max} \sigma_{ave}$A-1-600-0.1-0.04). At the moment of time = 0.00023 s, the first contact of the rod with the guide occurs - it is 6.47 MPa (Fig. 13 a). Thus, having a free end (according to the design of the engine head in Fig. 12 $t_m \sigma_{max}$), the guide received an impulse and shows jump-like stresses. Further, the contact stabilizes and as the angle increases by = 0.01 s, the maximum stresses are 5.88 MPa (Fig. 13b). At the end of the experiment (= 0.04 s), the stresses move to the edge of the contact surface and increase to 12.35 MPa.$a_{vo} t_m t_m$



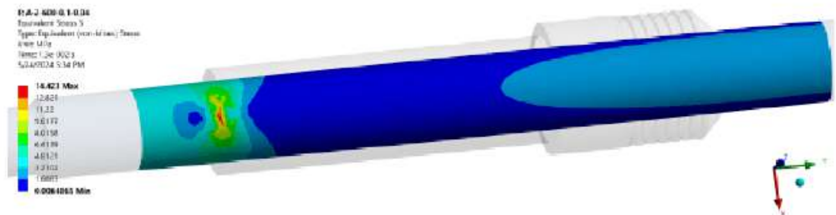
a) b)

Fig. 13. Stress map of the contact surface of the guide at the moment of time: a) 0.00023 s; b) 0.01 st_m

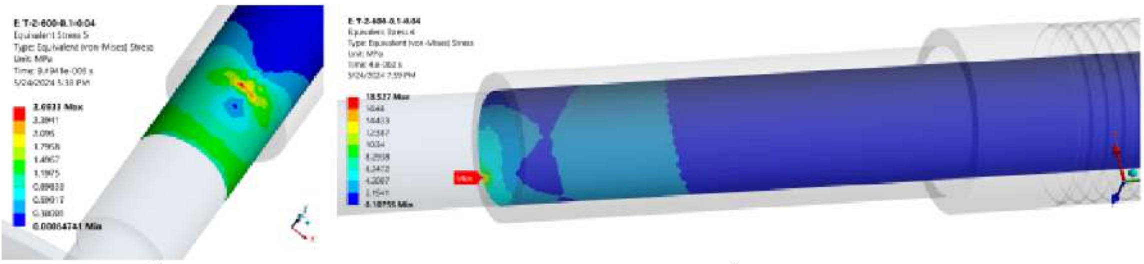
The behavior of the guide in the conditions of the regimeT-1-600-0.1-0.04 is very close to the steel valve model.

Effect of angleskew

According to the boundary conditions of the table. qg the following two modes were simulated, similar to those described above, but with a different graph of the skew angle. In fact, the difference between the skew angle is 0.01° at each of the 3 steps. Thus, curve #2 is more intense and in theory should cause higher stress values in the modesa_{vo}T-2-600-0.1-0.04 and A-2-600-0.1-0.04. INthe influence of the angle change in the case of a steel rod begins to appear from c, where the curves of maximum stresses diverge and, as of the end of the experiment, the difference is 3.63 MPa (9.41 and 13.1 MPa in the mode of curve #1 and #2, respectively). In contrast to mode A-1-600-0.1-0.04, where oscillations with the corresponding peak stress of 16.09 MPa were recorded in the first 0-0.0005 s, a more intensive change in angle stabilizes the valve better. The situation with the titanium valve is more obvious - a systematic growth is observed until the end of the experiment = 0.04 s, when the stresses are 7.14 and 5.21 MPa for modes with curve #2 and #1, respectively.$t_m = 0.025 \sigma_{max} a_{vo} \sigma_{max} t_{vo}$. Voltage at characteristic moments of time is shown in fig. 14. Of interest is the change in the shape of the contact spot in a very short period of time (s) and the change in stress from 2.69 to 2.87 MPa.$t_m t_m = 0.0094 - 0.01$.



a)



b) c)

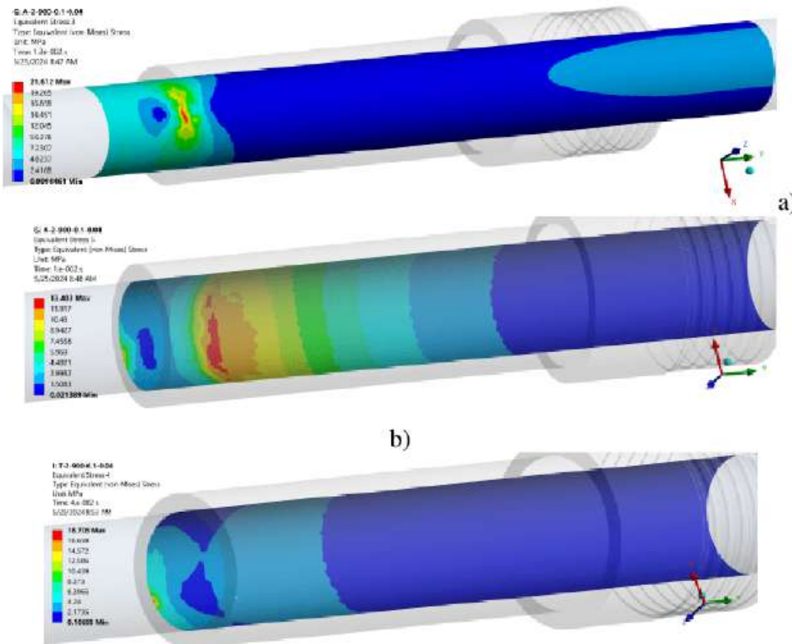
Fig. 14. Stress maps in the regimes with the skew angle curve #2: a) steel valve in c; b) titanium valve in c; c) guide (mode$t_m = 0.013$ $t_m = 0.0094$ T-2-600-0.1-0.04) at the moment$t_m = 0.04$with

Thus, the preliminary hypothesis regarding the growth of stresses attransitions from curve #1 to #2 were confirmed. For example, (increase in average stresses) is 24-30% for the valve surface and 28-41% for the guide.$\Delta\sigma_{ave}$

Effect of temperature

The next iteration of calculations is devoted to the effect of temperature on the stress state of the models. So, to the previous modes with curve #2 (T-2-600-0.1-0.04 and A-2-600-0.1-0.04) apply a temperature of 900°C (to the surface of the valve head - Fig. 14) and analyze the obtained results. Titane models almost coincide with each other in values (7.14 and 7.16 MPa at 600 and 900σ_{max}°C, respectively), which are fixed as of$t_m = 0.04$with. The largest difference in maximum stresses between them is 1.57 MPa and was recorded at the time of$t_m = 0.014$Thus, it can be stated that the titanium alloy is resistant to temperature loads. The situation is different with a steel valve–= 50.3% (increase from 14.42 to 21.67 MPa) at the peak moment c, which came earlier than that of the titanium model (c). Steel turned out to be more sensitive to temperature changes in the valve. The bronze guide when interacting with the titanium valve almost did not feel changes in its temperature during stress analysis$\Delta\sigma_{max}t_m = 0.013t_m = 0.014$ σ_{max}increased from 18.5 to 18.7 MPa.

This testifies to the positive influence of the titanium alloy on the thermodynamics of the guide and its working conditions– in fact, the titanium valve is as gentle as possible to the surface of the guide when heated.The steel valve as a result of the experiment (= 0.04 s) creates lower maximum stresses in the guide (14.02 and 14.5 MPa at 600 and 900t_{vo}°C, respectively), than titanium. WITHordinary tensionsσ_{ave}guide in combination with a steel valve are higher by more than 24% and when it is heated from 600 to 900°C show fluctuations.



a)

b)

c)

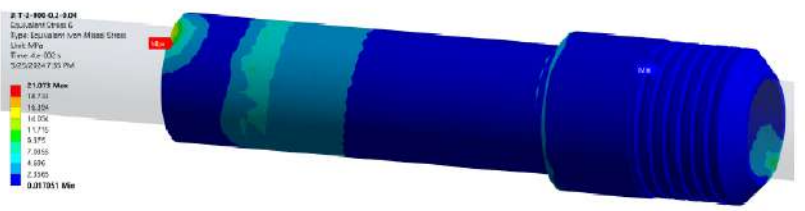
Fig. 15. Stress maps at key moments of time: a) 0.013 s – steel rod; b) 0.01 s – guide in a complex with a steel valve; c) 0.04 s – a guide in a complex with a titanium valvet_m

The influence of the coefficient of friction

The next step and change the coefficient of friction in the previous modes$\mu_v$from 0.1 to 0.2. The modes will receive the following designations, respectively: T-2-900-0.2-0.04 and A-2-900-0.2-0.04. Magnification $\mu_v$from 0.1 to 0.2minimally reducedσ_{max} steel valve: 21.67 vs, 21.42 MPa as oft_m= 0.013 s. At the end of the experiment(= 0.04 s) the mode with greater friction showed a 25% higher value: 16.56 vs 13.19 MPa. The titanium sample is much less sensitive to changes due to higher surface hardness – the curves of the graphs almost coincide, and at the final moment = 0.04 s the difference is only 12.2% (7.16 vs, 8.03 MPa in the regimes with = 0.1 and 0.2, respectively),$t_{vo}\sigma_{max}\mu_v t_{vo}\mu_v$.

The maximum stresses of the guide body increased from 16.43 MPa (regσ_{max}A-1-600-0.1-0.04 on Fig. 16) up to 26.26 MPa (Fig. 16 a) when interacting with a steel valve, which in turn received 67.4 MPa as a result of temperature loading (edge of the head in Fig. 16 b). Titanium valve in current modeT-2-900-0.2-0.04 showed only 34.97 MPa (Fig. 16 b), and the corresponding guideline is 21.07 MPa.

a)



b)

Fig. 16. Mises stress maps of valve and guide models in the mode: a)A-2-900-0.2-0.04; b) T-2-900-0.2-0.04

As a result, we get the following paradox: in general, the guide model receives lower maximum stresses when interacting with a titanium valve than in a complex with a steel one, but its contact surface, on the contrary, is more stressed precisely in the mode of interaction with a titanium valve.σ_{max}

Effect of engine speed

We will perform the next iteration of the calculation - we will determine the effect of engine revolutions (valve opening time) on the stress state of the model in modes T-2-900-0.1-0.01 andA-2-900-0.1-0.01 ($\mu_v = 0.1$), where the last component notation is just responsible forvalve opening duration =0.01 s. We will approximately calculate the frequency of rotation of the crankshaft for the given value.$t_{vo}t_{vo}$

The exhaust valve in a 4-stroke engine usually starts to open before reaching bottom dead center (BDC) on the intake stroke and finishes opening when it reaches top dead center (TDC) on the exhaust stroke. The total angle to which the exhaust valve is open is about 240-280° of crankshaft rotation, since it opens before the start of the exhaust stroke (about 40-60° before TDC) and closes after it ends (later 10-20° after TDC).

So, to determine the engine speed at which the valve opening time for 130° is 0.01 s, we will use the following approach: determine how long it takes for 1 full revolution of the crankshaft (360°); calculate the engine speed in revolutions per minute (rpm).

The time for a 130-degree rotation is 0.01 s. Let's determine the time for one complete revolution (360°): ; s, where: – time for one complete revolution, c.

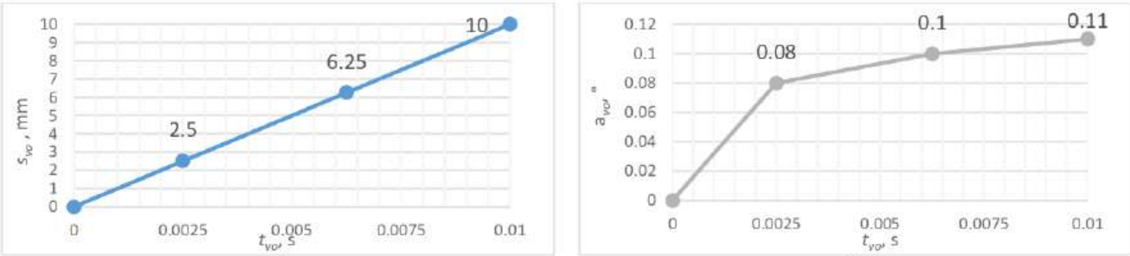
$$\frac{130°}{360°} = \frac{0.01}{t_{ft}} t_{ft} = 0.0277 t_{ft}$$

Engine speed (rpm):

$$rmprmp = \frac{1}{t_{ft}} \cdot 60 = \frac{1}{0.0277} \cdot 60 = 2166 \frac{1}{min}.$$

So, the engine rotates at a frequency of approximately 2166 rpm, if the valve opening time (corresponding to 130° rotation of the crankshaft) is =0.01 s.t_{vo}

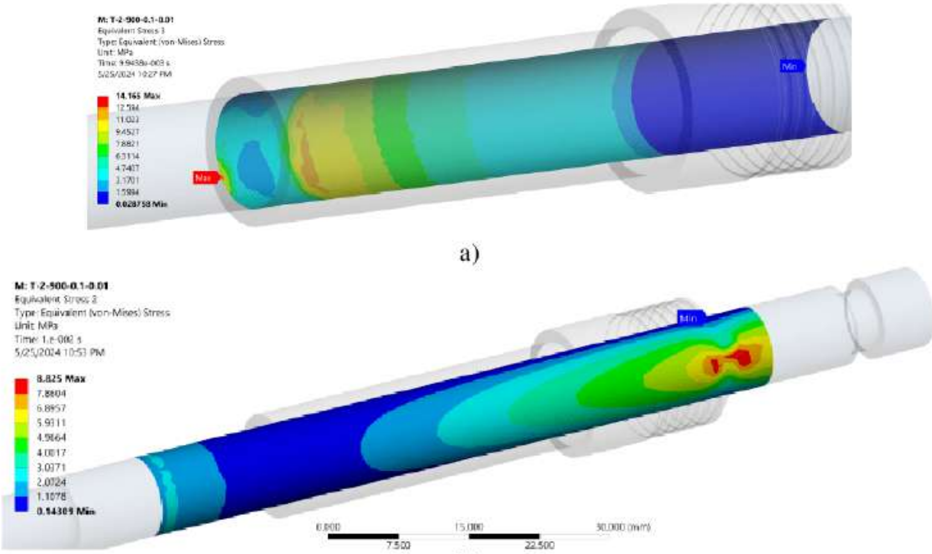
We update the boundary conditions - we update the graphs in Fig. 17, reducing the time along the abscissa axis to 0.01 s: graph$t_{vo}$of the stroke of the rod and the skew angle correspond to the step-by-step of the previous modes, only the values themselves are reached 4 times faster $s_{vo}a_{vo}$



a) b)

Fig. 17. Kinematics of valve movement at: a)$rmp = 2166\frac{1}{min}$stroke of the rod; b) skew angle$s_{vo}a_{vo}$

All values of □□□□ and □□vereached at the end of the experiment =0.01 s. The exception is the modet_{vo}T-2-900-0.1-0.01, at whichσmaxin the guidefixed as of =0.0099 s and ist_m14.16 MPa.σmax and σaveat =0.01 s were always higher than the compared regimest_{vo}A-2-900-0.1-0.04 and A-2-900-0.1-0.04 witht_{vo}= 0.04 s except for the case with the guide in contact with the titanium valve: □□□□decreased from 18.71 to 14.16 MPa with decreasing time$t_{vo}$from 0.04 to 0.01 s (Fig. 18 a). The dimensions of this location on the edge of the guide are so small (0.3 mm) that the titanium valve literally "flies" past the contact spot when the engine revs increase. At the same time, □□vein the guide grew on171.5% (with1.44 to 3.91 MPa). The situation is similar in the A-2-900-0.1-0.01 mode –$\Delta\sigma_{ave}$guide folded111.9%, and 74.7%. Thus, the overall behavior of a titanium valve with a bronze guide is unique - increased revolutions go to the lower contact surface of the guide due to the reduction of maximum stresses and almost a 2-fold difference$\Delta\sigma_{max}\sigma_{ave}$(3.91 MPa in mode T-2-900-0.1-0.01and 7.5MPa in A-2-900-0.1-0.01).



a)

b)

Fig. 18. Map of stresses according to Mises in modes z = 0.01 s: a) directionalt_{vo}T-2-900-0.1-0.01; b) titanium valvet_m= 0.01 s

Consolidated data

Let's summarize the array of results obtained above for each of the test modes in the table. 2: and – maximum and average stresses; and – stress difference in percentage between the current and previous mode of the same material;$\sigma_{max}\sigma_{ave}\Delta\sigma_{max}\sigma_{ave}t_{\sigma_{max}}$and is the time to reach the maximum and average stresses and .$t_{\sigma_{ave}}\sigma_{max}\sigma_{ave}$.

Table 2

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
T-1-600-0.1-0.04	5.21	–	0.04	1.38	–	0.04
A-1-600-0.1-0.04	16.09	+208.8	0.00031	2.79	+102.2	0.04
T-2-600-0.1-0.04	7.14	+37.0	0.04	1.80	+30.4	0.04
A-2-600-0.1-0.04	14.42	-10.4*	0.013	3.47	+24.4	0.04
T-2-900-0.1-0.04	7.16	+0.3	0.04	1.91	+6.1	0.04
A-2-900-0.1-0.04	21.67	+50.3	0.013	3.99	+15.0	0.04
T-2-900-0.2-0.04	8.03	+12.2	0.04	1.91	0	0.04
A-2-900-0.2-0.04	21.42	-1.2	0.013	4.01	+0.5	0.04
T-2-900-0.1-0.01	8.82**	+23.22	0.01	1.92	+0.5	0.01
A-2-900-0.1-0.01	24.21**	+11.7	0.01	4.29	+7.5	0.01
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
T-1-600-0.1-0.04	13.98	–	0.04	0.95	–	0.04
A-1-600-0.1-0.04	12.35 p.m	- 11.7	0.04	2.23	+134.7	0.04
T-2-600-0.1-0.04	18.52	+32.5	0.04	1.34	+41.1	0.04
A-2-600-0.1-0.04	14.02	+13.5	0.04	2.85	+27.8	0.04
T-2-900-0.1-0.04	18.71	+1.0	0.04	1.44	+7.5	0.04
A-2-900-0.1-0.04	14.50	+3.4	0.04	3.54	+24.2	0.04
T-2-900-0.2-0.04	21.076	+12.7	0.04	2.32	+61.1	0.04
A-2-900-0.2-0.04	17.01	+17.3	0.04	4.81	+35.9	0.04
T-2-900-0.1-0.01	14.16**	-24.3	0.0099	3.91	+171.5	0.01
A-2-900-0.1-0.01	25.33**	+74.7	0.01	7.50	+111.9	0.01

Conclusions

Mode A-2-600-0.1-0.04 demonstrated$\Delta\sigma_{max} = -10.4\%$relativelyA-1-600-0.1-0.04, where oscillations with a corresponding peak stress of 16.09 MPa were recorded in the first 0-0.0005 s. The next stress peak = 13.99 MPa in mode A-1-600-0.1-0.04 (curve #1) occurs already at the moment of c. Thus, without taking into account the specified jump, the value +3.1% will be more balanced, as the difference between 13.99 and 14.42 MPa.$\sigma_{max}t_m = 0.013\Delta\sigma_{max} =$

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Голенко К.Е. Вичавка А.А., Диха М.О., Дитинюк В.О. Скінчено-елементний аналіз контактних характеристик і режимів тертя пари «клапан-напрямна» двигуна внутрішнього згорання

Моделювання працездатності пари двигуна «клапан-напрямна» за допомогою сучасних програмних засобів є дієвим інструментом як для виявлення слабких місць в конструкції, так і прогнозуваня поведінки вузла тертя в експлуатації. В даному дослідженні для дослідження контактних і антифрикційних параметрів пари тертя двигуна внутрішнього згорання «клапан-напрямна» як інструмент вибраний метод скінчено-елементного аналізу. За використання прикладної програми FEM описані вихідні дані по матеріалу, розмірам поверхонь, навантаженням, кінематиці руху. На основі побудованої скінченоелементної моделі спряження «клапан-напрямна» проведений аналіз впливу визначальних трибологічних факторів: швидкості ковзання в контакті, температури, кута перекосу, коефіцієнту тертя на контактні напруження як для кожної деталі пари тертя, так і в процесі контактної взаємодії. Побудована консолідована матриця результатів чисельного експерименту, обгрунтовані висновки щодо впливу кожного фактора на трибологічні характерстики. Окреслені алгоритми впливу на конструкторсько технологічні і експлуатаційні фактори для подовження ресурсу пари тертя двигуна ДВЗ «клпапннапрямна».

Ключьові слова: двигун внутрішнього згорання, напрямні клапанів, скінчено-елементна модель, контактні параметри, коефіцієнт тертя, напружений поверхневий стан