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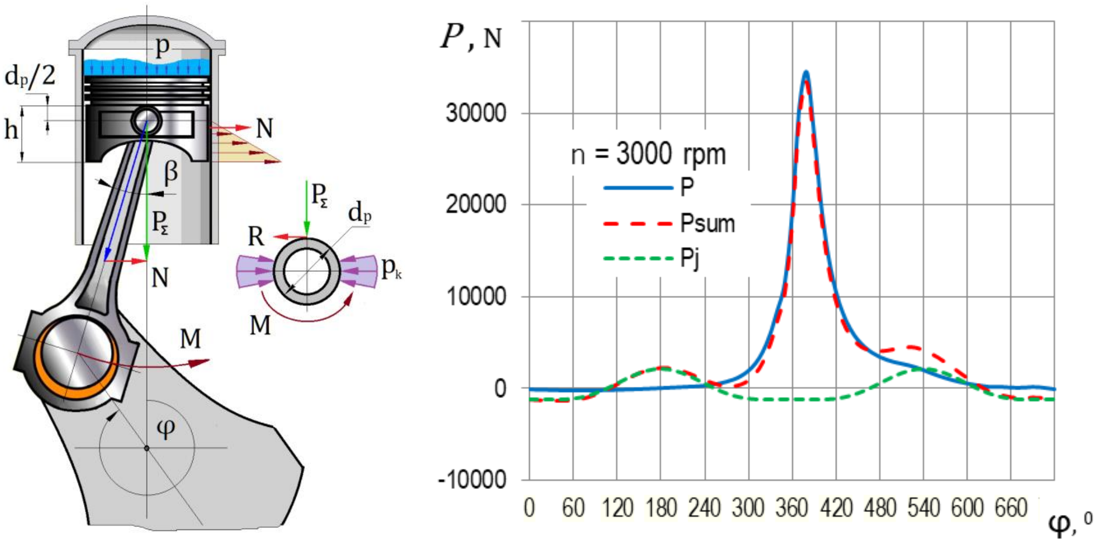
REGULARITIES OF PISTON-SIDE FORCE CHANGE IN CRANK MECHANISM UNDER EXCESSIVE LOADS CAUSED BY BREAKAGE OF OPERATING CONDITIONS

In classical works on the theory of internal combustion engines, it is believed that the lateral force of pressure of the piston on the cylinder arises only in the direction of the total force (pressure and inertia), which acts on the mass of reciprocating moving parts [1,2]. For a given mass and rotation speed, the presence of the angle of deviation of the connecting rod from the cylinder axis, according to theory, completely determines the magnitude of the lateral force that is perceived by the piston skirt. However, what is true for operating conditions may not correspond to abnormal loads when operating conditions are broken.

Let's consider the process of compressing air together with an incompressible liquid during hydrolock in the cylinder of a gasoline car engine. From the diagram (Fig. 1a) it is clear that the pressure force on the skirt should be equal to:

$$N = P_{\Sigma} tg\beta = \left(P + P_j\right) tg\beta\,, \tag{1}$$

where $\beta = \arcsin\left(\lambda \sin\varphi\right)$ is the angle of deflection of the conrod from the cylinder axis, P_{Σ}, P, P_j are the total, pressure and inertia forces acting on the piston.

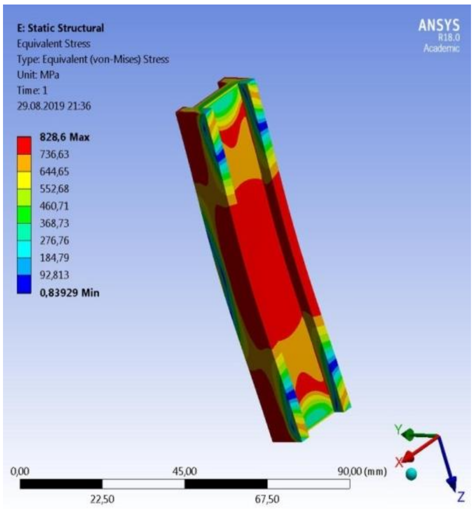


a b

Fig.1. Diagram of a conrod-piston group showing the forces and torques that act on parts during hydrolock (a) and a typical indicator diagram (b) of a gasoline automobile engine when operating at medium speeds (obtained using the Lotus Engine Simulation software).

The cycle calculation was performed using the Lotus Engine Simulation program [3] for a gasoline internal combustion engine with parameters typical for car engines (cylinder diameter 83 mm, piston stroke 80 mm, piston pin diameter 22 mm). It has been determined that at low and medium rotation speeds, the pressure forces near the top dead center far exceed the inertial forces (Fig. 1b). This made it possible to simplify the calculation and take into account only pressure forces.

Next, you need to use the known data on the loss of stability of the conrod stem during a hydraulic shock. A typical conrod stem was modeled by the finite element method (Fig. 2a) using the ANSYS software package in the Student version. As a result, it was found [4] that loss of stability occurs at a specific compressive force of about 700 MPa (Fig. 2b). When this load is reached, the axial force does not increase with further deformation. Therefore, in the narrow range of crankshaft rotation angles under consideration, it can be taken as constant.



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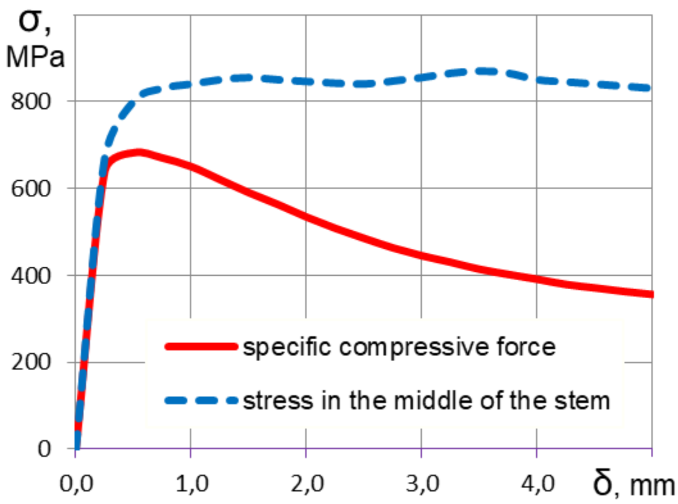

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Fig. 2. Results of modeling the loss of stability of the conrod stem: diagram of von Mises equivalent stresses with axial compression of the stem of 0.5 mm (a), stress and force with axial compression of the stem during a hydrolock in the cylinder (b).

The area of the skirt that rests on the cylinder in the process under consideration (one side of the skirt) can be taken to be approximately equal to half the area of the piston. Hence, the specific pressure of the skirt on the cylinder, according to classical theory, will be proportional to the pressure difference on the piston:

$$p_N = 2\left(p - p_0\right) \cdot tg\beta \,. \tag{2}$$

Restriction against loss of stability of the conrod stem, according to the formula (2) will also apply to the specific pressure on the skirt. Therefore, its value (pN1 on the graph in Fig. 3a) will have a characteristic “shelf”.

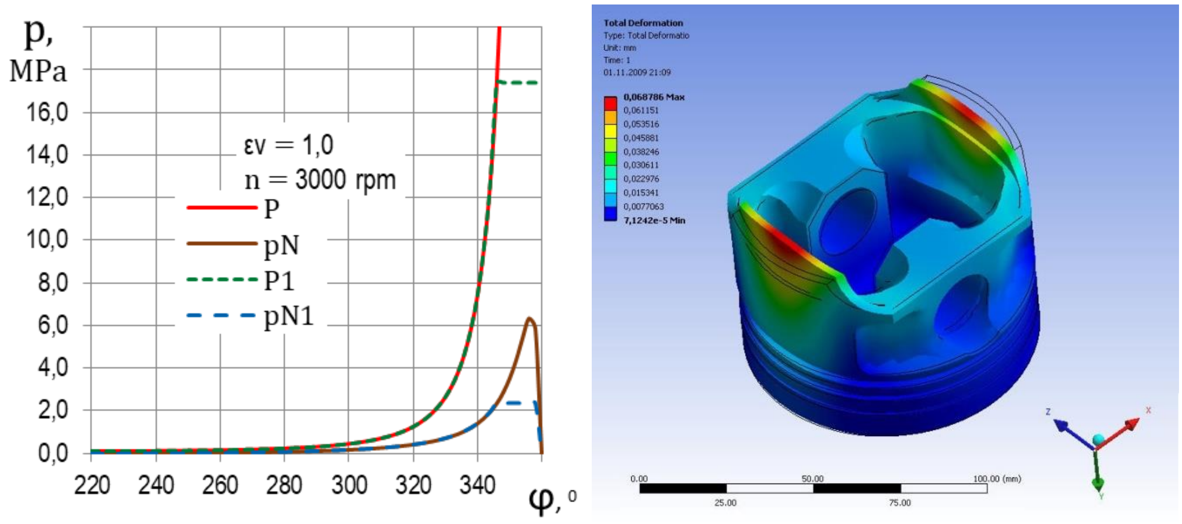

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Fig. 3. The results of calculating the specific pressure on the piston skirt from the lateral force during hydrolock according to the classical theory (a) and the results of modeling the deformation of the piston skirt from the distributed compression load to the skirt (b).

According to the result obtained, during a hydrolock, the piston under study has a lateral specific pressure on the skirt close to 2 MPa. This value requires evaluation, for which, using the ANSYS program in the Student version, the stress-strain state of a typical car engine piston was studied under such a skirt compression load (Fig. 3b).

As a result of the simulation, it was found that its elastic deformation is equal to 0.069 mm per side or 0.138 mm per diameter (Fig. 3b). However, it is known from repair experience that most pistons made of a similar alloy experience permanent deformation only after the skirt has been deformed by more than 0.50 mm per diameter [5]. If we take this value to correspond to the achievement of the yield strength of the material, then the calculation gives a critical value of pressure on the skirt of about 7 MPa.

It turns out that the calculated (within the framework of classical theory) specific pressure from the lateral force in the crank mechanism is still far from critical. And if the conrod was deformed during a hydrolock (curve *p* in Fig. 3a), then the lateral force (curve *pN*) would still not be enough for plastic deformation of the skirt.

To answer the question of where the “missing” force is located that compresses the skirt, a hypothesis has been proposed. According to it, excessive deformation of the piston pin under abnormal load at breakage of operating conditions creates significant friction in the hole of the piston bosses. This friction influences the lateral force that acts on the piston skirt near the top dead center.

To obtain quantitative dependencies, instead of the classical theory of the absence of friction in the joint of the pin, let us consider the opposite case – complete jamming of the pin. Then the connecting rod near the top dead center will actually rotate relative to the axis of the upper (piston) head together with the piston and deform its skirt.

This deformation is not difficult to calculate. To do this, you should determine the crankshaft rotation angle φ_0, which corresponds to the moment the pin jams. For most car gasoline engine pistons, the top edge of the skirt approximately corresponds to the top edge of the piston pin hole. When turning the piston together with the conrod, it can be assumed that deformation of the skirt will occur only below the axis of the pin hole. Then, by the deformation of the lower edge of the skirt, an approximate relationship can be obtained:

$$\delta_0 = \left(h - 0,5d_p\right)\lambda \sin\varphi_0 \Big/ \sqrt{1-\lambda^2 \sin^2\varphi_0}\,, \tag{3}$$

where d_p is the diameter of the piston pin, h is the height of the skirt, φ_0 is the crankshaft rotation angle, λ is the relative length of the connecting rod.

The calculation was carried out according to formula (3) for the common dimensions of gasoline engines (h = 44 mm, d_p = 22 mm, λ = 0.333) at an angle φ_0 equal to 15^0, which approximately corresponds to the complete filling of the combustion chamber with liquid [3, 4]. The calculation gave the value of deformation of the lower edge of the skirt δ_0 = 2.85 mm. This means that if the piston pin jammed completely, then the pistons would not only be deformed after a hydrolock, but most likely the skirt would be destroyed. However, this is not observed in practice [3, 4, 6]. In other words, the assumption that the piston pin is completely jammed also does not correspond to the real picture at hydrolock, as do the results of calculations according to classical theory. It remains to be assumed that the condition of the piston pin is somewhere between complete jamming and free sliding.

This state corresponds to the piston on the pin rotation with friction. Presumably, the beginning of the friction effect to the process occurs at the moment of reaching such a pressure in the cylinder that the piston pin ovalizes before the opening of the bosses begins to expand. Accordingly, it is possible to find the boundary beyond which the conditions of normal operation are broken, and instead of the classical theory without taking into account friction, the operation of the mechanism with friction should be considered [2].

After an interference Δ appears in the hole due to ovalization, the friction torque M in the connection will become an interference function:

$$M = 0,5kf\,\Delta/C\,, \tag{4}$$

where k is the friction coefficient (taken equal to 0.15 [7]), f is the contact area, C is a coefficient that depends on the ratio of the internal and external diameters of the finger [8].

The specific pressure on the skirt obviously depends on the friction torque at the pin joint:

$$p_k = 4M\left(h - 0,5d_p\right)\Big/F\ . \tag{5}$$

Where, with appropriate assumptions after transformations, it is possible to obtain the formula for the total specific pressure on the piston skirt during hydrolock in the form:

$$p_{N\Sigma} = p_N + p_k = (p - p_0)2tg\beta + (p - p_\delta)0{,}18k\frac{\pi\gamma}{EC}\left[1 - 10(\alpha - 0{,}4)^3\right]\left(\frac{1+\alpha}{1-\alpha}\right)^3 \Big/ \left(h/d_p - 0{,}5\right), \quad (6)$$

where E is the elastic modulus of the finger material, γ is a coefficient that takes into account the influence of various factors on the nature of the contact (taken equal to 0.5).

Fig. 4 presents the results of calculating specific pressures using formula (6) with all the data and conditions specified above.

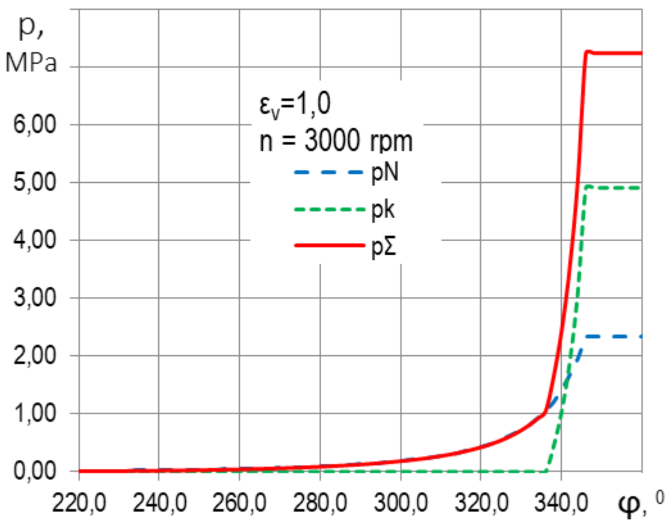

Fig. 4. Specific pressures on the piston skirt during hydrolock: from the component of the pressure force on the piston *pN*, from the friction of the pin *pk* and the total *pΣ*.

Even despite the assumptions made, the total specific pressure on the skirt, taking into account friction, immediately turns out to be close to 7 MPa that is, to the value that for a given piston is close to critical and presumably corresponding to the beginning of plastic deformation of the skirt.

Thus, the data obtained allows us to conclude that in addition to the traditional lateral force, under certain conditions, an additional force can act on the piston skirt, which is caused by friction of the piston pin in the piston bosses due to deformation under the influence of abnormal load from excessively high pressure in the cylinder.

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УДОСКОНАЛЕННЯ СИСТЕМИ ДІАГНОСТУВАННЯ ДВИГУНА АВТОМОБІЛЯ

Системі діагностування, обслуговуванню та ремонту двигунів приділяється постійна увага з боку інженерних розробок та наукових досліджень [1-9].

Як показує практика технічного обслуговування й ремонту автомобільних двигунів найбільш поширеними, доступними і достовірними залишаються методи технічної діагностики циліндро-поршневої групи (ЦПГ), що дають загальну оцінку герметичності надпоршневого простору з вірогідністю, яка не перевищує 50%, за допомогою різних засобів діагностування: компресометра, компресографа, мотортестера.

Проаналізована можливість реалізації діагностичної моделі на сучасному рівні із застосуванням цифрової техніки. Програма для обробки цифрової інформації забезпечує відтворення функції тиску в циліндрі двигуна й одночасно розраховує частоту обертання колінчатого валу виходячи з того, що