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BUILDING A MATHEMATICAL MODEL OF THE DESTRUCTION OF A CONNECTING ROD-PISTON GROUP IN THE CAR ENGINE AT HYDRAULIC LOCK

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This paper investigates the process of destruction of parts of the connecting rod-piston group of the engine due to hydraulic lock after the ingress of liquid into the cylinders of the engine. Comparing expert data on actual engine destruction due to hydrolock with existing estimation models has made it possible to identify a number of significant contradictions affecting the objectivity and accuracy of the destruction assessment.

To resolve the existing contradictions, a mathematical model for reconstructing the destruction of the connecting rod-piston group of the engine during a hydraulic lock has been improved. Unlike the existing ones, the model makes it possible to take into consideration not only the static deformation of the connecting rod but also to give a comprehensive assessment of the deformations of the connecting rod, piston pin, and piston at different volumes of hydrolock fluid.

Underlying the model is the hypothesis assuming that the deformation of the piston pin under excessive load caused by hydraulic lock leads to the emergence of tension and an increase in the friction in the mated pin-piston. The calculation from the condition of differential change in the amount of friction in the mated pin-piston produced a satisfactory result that does not contradict the practical data and has confirmed the working hypothesis.

By calculation, the onset of the destruction of engine parts during hydrolock at a pressure in the cylinder close to 17.3 MPa, at a crankshaft angle of about 346°, was revealed. In addition, it was found that in the case of violating the operating conditions, due to friction, the mated pin-piston is exposed to the lateral force on the skirt that reaches 17.2 MPa, which exceeds the permissible one, calculated according to known procedures, by 2.8 times.

The results reported here are confirmed by known practical data, which makes the devised model applicable to the practice of expert studies into the causes of engine malfunctions when violating the operating conditions of a car

Keywords: violation of operating conditions, hydrolock in the cylinder, connecting rod-piston group, deformation of parts

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1. Introduction

In the practice of designing internal combustion engines (ICEs), it is usually assumed that the load acting on the parts should not exceed some maximum permissible values determined from the condition of the strength of the part or the amount of its strain. In line with this approach, models and programs have been developed to simulate workflows, including part loads, within the scope of workloads.

However, the design approach and the models used for this do not fully correspond to the problems inherent in the operation of modern engines. Thus, many types of engine damage occur due to a violation of operating conditions, beyond the action of workloads, that is, during overload [1]. For example, when the engine overheats, the process of plastic deformation of compression of the piston skirt occurs, caused by the thermal expansion of the piston in the cylinder and the resulting significant compression stresses [2]. A similar process is characteristic of overloading due to exceeding the maximum permissible rotational speed. Accordingly, during hydrolock, when liquid enters the cylinder, excessive pressure occurs [3], at which plastic deformation (compression) of the piston along the skirt may also indicate overload.

Under such conditions, including during the automotive expertise, it is important to correctly establish the cause of the damage, for which it is necessary to determine the forces that caused the deformation of the parts. This requires calculation methods and algorithms applicable not only to operating modes but also in the field of operation of abnormal loads that occur when the operating conditions of the car are violated. To this end, it is necessary to investigate the loading processes of both individual parts and engine components as a whole, in this area.

Thus, the relevance of the topic is defined, on the one hand, by the need for expert practice in determining the causes of malfunctions, and on the other, by the need to investigate for this purpose the processes of damage to engine parts when the operating conditions are violated.

2. Literature review and problem statement

Papers [4, 5] outline the basic principles and techniques of engine design and report the results of research summarized from the experience of constructing engines of different types. In addition, manufacturers of engine components publish the results of modeling and experimental refinement of their products [6], including a detailed analysis of the forces acting on the parts [7]. At the same time, deviating the load beyond the operating range is considered unacceptable, so the calculations of the force effect on the parts and its analysis immediately stop as soon as the loads go beyond the permissible limits. The limitations are explained by the fact that for an engine operated under working modes, overload is not allowed, and any design estimation is carried out on the basis of such restrictions.

On the other hand, it is known that during operation overload is a fairly common cause of damage and failure of engines. For example, works [8, 9] provide a large amount of data on various damages caused by excessive loads in violation of operating conditions. Paper [2], in addition to describing various malfunctions, also reports methods for experts to determine the causes of damage caused by overloads.

It is clear that the degree of damage to the engine may depend on how much the effective load in operation exceeded the permissible level. However, there are no models in works [8, 9] that make it possible to obtain quantitative data on these processes. Then, when analyzing the causes of malfunctions, as, for example, in [10, 11], it is not possible to associate the damage to the part in the form of a deviation in the dimensions and condition of working surfaces with the amount of load that caused such changes.

There are several reasons why models have not yet been devised to link the quantification of damage to its cause. On the one hand, the task of quantifying operational damage, especially if they are associated with violations of operating conditions, is not set in the design of engines and, accordingly, is not solved as not meeting the requirements of production. On the other hand, the collection of quantitative data on engine damage in operation is not systematized, and when performing automotive technical expertise, generalizing studies are not carried out due to the disinterest of the parties, lack of time and money.

As a result, the quantitative analysis of the effect of excessive forces on the parts of the connecting rod-piston group is limited or very difficult. When conducting an automotive expertise of an engine malfunction, that does not make it possible to clarify the actual conditions under which the damage occurred, which makes it difficult to correctly determine its cause in expert practice [2].

However, for some cases and types of damage where parts experience significant overloads, missing data and a solution can be found. Such damage includes the deformation of parts during hydrolock from liquid ingress in the cylinder [12]. However, these data do not have a quantitative characteristic.

3. The aim and objectives of the study

The aim of this work is to build a mathematical model of deformation and destruction of the connecting rod-piston group due to hydrolock in the cylinders when the operating conditions of the car are violated. The use of the model in the practice of auto technical expertise of the technical condition of engines makes it possible to determine the causes of the malfunction by the nature and magnitude of the deformation of the parts.

To accomplish the aim, the following tasks have been set:

– to devise a model for determining the main loads and deformations acting in the connecting rod-piston group during hydraulic hydrolock;

– with the help of the model to investigate and determine the ranges of loads and deformations of the parts of the connecting rod-piston group during overload caused by hydrolock;

– to compare the simulation results with the known expert data on the actual damage to the engine due to hydrolock.

4. The study materials and methods

It is possible to determine the state of the parts of the connecting rod-piston group of a motor gasoline engine by numerical modeling of stresses and deformations. It is possible to simulate the operating conditions of the selected elements in the engine in different ways, while it is necessary to proceed from the requirements of expert practice in determining the causes of malfunctions when violating the operating conditions in terms of the time and complexity of work performed.

We simulated the stressed-strained state of parts using the ANSYS software, Student edition (USA, [13]). To establish boundary conditions, experimental data on the magnitude of the deformation of parts at known loads were used [14]. In addition, the Lotus Engine Simulation internal combustion engine cycle calculation program was used to determine the loads acting on the parts (England, [15]).

The initial data adopted for the simulation included the geometric dimensions and material characteristics of the main parts of the connecting rod-piston group. A steel connecting rod with a traditional I-beam rod profile of 20×12 mm with a base of 4 mm and shelves 3 mm thick was chosen [12]. In addition, a typical piston made of piston aluminum alloy of a car engine for a cylinder with a diameter of 83 mm and a steel piston pin with a diameter of 22 mm were taken for modeling [16].

Based on the known data acquired during the simulation of the loss of stability of the connecting rod [3, 12], a connection was established between the deformation of the connecting rod and the pressure force acting on the piston. After that, the deformation (ovalization) of the piston pin was determined by calculation. As the main hypothesis, the assumption was taken that the deformation of the pin causes it to burst the hole in the piston reels and the appearance of a friction moment when turning the piston on the pin near the upper dead point. In addition, it is accepted that the coefficient of friction when landing a pin in a hole with tension corresponds to dry friction.

When building a model of the deformation of the piston, a simplifying assumption was also adopted about the stationary landing of the pin in the connecting rod, which was considered possible due to the fact that the deformation of the pin in this section is maximum [4]. Further, the model of pin sliding with friction in the mating with the piston was used to derive a semi-empirical formula for calculating the lateral force acting on the piston skirt during overload caused by a violation of operating conditions.

5. Results of studying the deformation and destruction of the connecting rod-piston group due to hydrolock in cylinders

5. 1. Building a model for determining the main loads and deformations acting in the connecting rod-piston group

Using the ANSYS software, Student edition [13], the stressed-strained state of the piston was investigated under the distributed compression load of the skirt. When solving the problem, the boundary conditions were set so that the bottom was stationary, and sliding was allowed in the hole of the pin.

As a result of modeling, it was established that at a specific pressure on the skirt of 2.0 MPa, its elastic deformation at the lower edge is 0.069 mm per side or 0.138 mm per diameter. From experimental data and the experience of engine repair, it is known that pistons with a significant amount of elastic deformation, more than 0.50 mm per diameter, receive a permanent deformation [14]. If in the 1st approximation, we put this value corresponding to the achievement of the yield strength of the material, then the calculation gives a critical value of pressure on the skirt of the piston under study close to 7.2 MPa.

Therefore, it can be assumed that with a specific pressure on the skirt of 7.2 MPa, the elastic deformation of the skirt will be 0.25 mm per side (0.50 mm with symmetrical squeezing from 2 sides). In this case, the stresses will correspond to the yield strength of the material and the onset of the plastic deformation of the skirt. Thus, if after a hydrolock the piston skirt has a residual deformation [2], then the above data on specific pressure and deformation will allow us to assess the forces acting in this process.

To solve the problem, it is necessary to consider the scheme of the connecting rod-piston group and the main forces acting in it (Fig. 1). In order to simplify, the offsetting in the studied scheme of the crank mechanism is not taken into consideration, and the process under study is considered quasi-static.

In all classical works on the theory of internal combustion engines, it is assumed that the lateral force of the pressure of the piston on the cylinder arises only from the decomposition in the directions of the total force (pressure and inertia) acting on the mass of reciprocating moving parts [4, 5]. At a given mass and speed of rotation, the presence of the angle of deviation of the connecting rod from the axis of the cylinder, according to the theory, completely determines the amount of lateral force, which is the force of pressure on the piston skirt.

The diagram (Fig. 1) clearly demonstrates that the pressure force on the skirt should be equal to:

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

where $\beta=\arcsin(\lambda\sin\varphi)$ is the angle of deflection of the connecting rod from the cylinder axis, P_{Σ}, P, P_j is the total force, the force of pressure, and inertia.

To obtain quantitative data on the lateral force, it is necessary to consider the dependence of the pressure force in the cylinder on the angle of rotation of the crankshaft. Under the modes of low and medium rotational speeds of automobile gasoline engines, the pressure forces are much higher than the inertial forces. This directly follows from the results of the loop calculation performed using the Lotus Engine Simulation program [15]. When compressing with a liquid in a cylinder, the pressure in the cylinder is not just significant but many times higher than the usual level due to the replacement of part of the volume with an incompressible liquid [12]. In this case, the influence of inertial forces on the process will be even less, and they can be generally neglected.

Next, data on the loss of stability of the connecting rod of the type of engine under study should be used. For a typical connecting rod, from the simulation results, including using a finite-element method (Fig. 2), it is known [16] that a loss of stability occurs at a specific compressive force of about 700 MPa (Fig. 3).

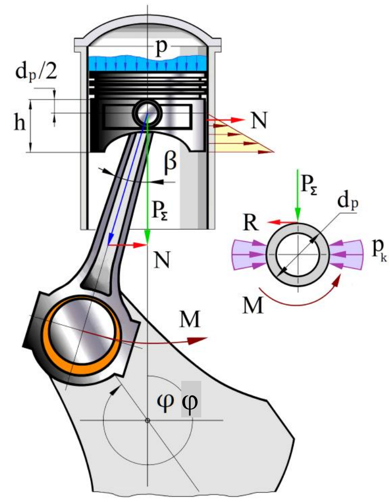


Fig. 1. Diagram of the connecting rod-piston group, showing the forces and moments acting on the parts during hydrolock: φ – the angle of rotation of the crankshaft from the upper dead point, M – torque, p – pressure in the cylinder, p_k – specific load from pulling the pin in the piston hole, N – lateral pressure force on the skirt, R – pin friction force

This means that when this value is reached during hydrolock, the axial force does not grow further, and in the narrow range of rotation angles of the crankshaft under consideration, it can be taken as constant.

Obviously, during hydrolock, excessive compression stresses in the rod, leading to a loss of stability of the connecting rod. The compression process in the cylinder is accompanied by an increase in pressure, the stronger the more fluid is trapped in the cylinder at the beginning of the compression stroke. To simulate the forces acting on the piston during hydrolock, the value of the relative amount of liquid in the cylinder (that is, related to the volume of the combustion chamber) equal to $\varepsilon_v=10$ [3, 12] is chosen as an example. This value reflects the average amount of liquid that most often causes hydrolock under actual operating conditions.

The specific compressive force acting on the rod is related to the pressure in the cylinder by the ratio:

$$\sigma = \frac{R}{A} = \left(p - p_0\right)\frac{F}{A}, \tag{2}$$

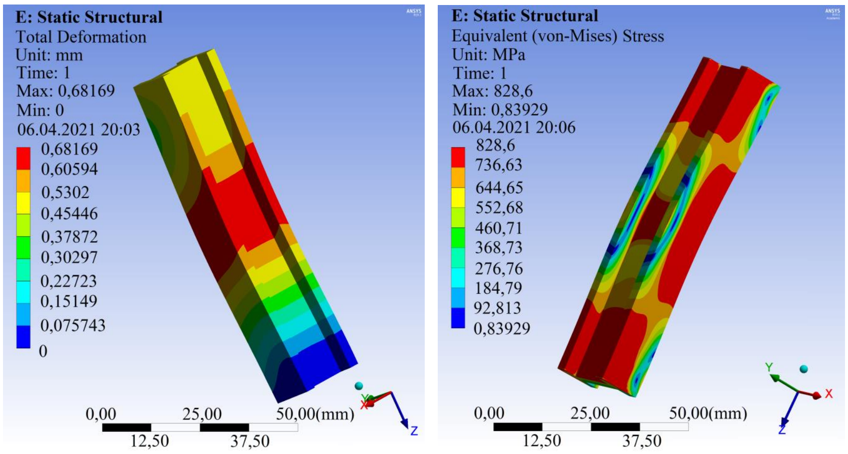
where F is the area of the piston, A is the cross-sectional area of the connecting rod, p_0 is the pressure in the crankcase.

For the selected value of ε_v, the calculation of the process of compression of air with liquid according to the devised model [9, 16] gives the change in pressure in the cylinder at the angle of rotation of the crankshaft, shown in Fig. 4.

In this case, formula (2) determines the maximum pressure in the cylinder, based on the condition of loss of stability of the connecting rod (after which the pressure in the cylinder no longer increases):

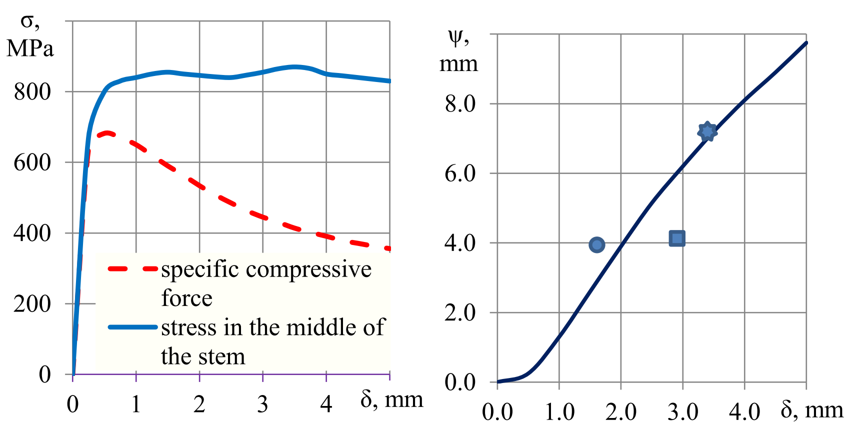
$$p_1 = p_0 + \sigma_{\max}\frac{A}{F}. \tag{3}$$

Given this, a limit (p_1) of 17.3 MPa can be applied to the pressure diagram at a crankshaft angle of 346°. When this angular position is reached and then the pressure in the cylinder is approximately constant due to the loss of stability and deformation of the connecting rod.



a *b*

Fig. 2. The results of modeling the loss of stability of the connecting rod during axial compression of the rod 0.5 mm [16]: *a* — total deformation; *b* — equivalent stresses according to von Mises



a *b*

Fig. 3. The results of modeling the loss of stability of the connecting rod during axial compression caused by hydrolock [16]: *a* — stress σ in the connecting rod; *b* — the dependence of the longitudinal bending of the connecting rod ψ on its axial deformation Δ with applied values for several deformed connecting rods

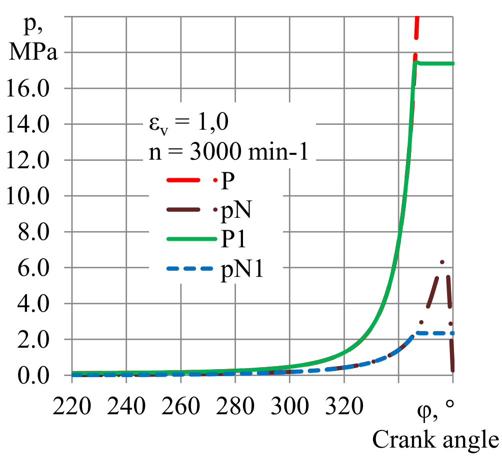


Fig. 4. Results of calculation of cylinder pressure and specific pressure on the piston skirt from lateral force

Usually, the description of the regularity of hydrolock by the deformation of the connecting rod is limited [1]. However, data on the pressure in the cylinder, as well as on the stresses in the rod of the connecting rod with loss of stability, can be used not only to analyze the deformation of the connecting rod itself. If we go beyond these limitations, it is possible to describe the processes of deformation of the parts mated with the connecting rod and thereby somewhat expand the model of a hydrolock to the piston group.

Accordingly, the data obtained above make it possible to calculate the lateral force on the skirt. To this end, the lateral force described (1) can be written as:

$$N = P\,\mathrm{tg}\beta = (p - p_0)F\,\mathrm{tg}\beta. \tag{4}$$

The area of the skirt that rests on the cylinder in the process in question (one side of the skirt), in the 1st, but a rather close approximation to the real one, can be taken to be equal to half the area of the piston. Hence, the specific pressure of the skirt on the cylinder will be:

$$p_N = 2(p - p_0)\cdot \mathrm{tg}\beta. \tag{5}$$

The limit on the loss of stability of the connecting rod, according to (5), will extend to the specific pressure, and it is this value (*pN*1 in Fig. 4) that is the purpose of the calculation.

5. 2. Determining and studying the ranges of loads and deformations of the parts of the connecting rod-piston group during overload caused by hydrolock

According to the result obtained, during hydrolock, the studied piston has a lateral specific pressure of only 2.3 MPa, which, following the data on the lateral load simulation (Fig. 1, 2), should give an elastic deformation of approximately 0.16 mm.

It turns out that the design specific pressure is still far from the critical, causing an excess of the yield limit, which would be achieved when reaching a critical specific pressure of 7.2 MPa and elastic deformation of the skirt of 0.5 mm.

Moreover, even if the connecting rod could withstand hydrolock, the pressure in the cylinder would become even greater (*p* curve in Fig. 4). Then the lateral force (*pN* curve) would still not be enough for the plastic deformation of the skirt, which remains below the critical one.

In other words, the result of the calculation contradicts the data from expert studies of hydrolock, where plastic deformation of the piston skirt is common [2, 17]. At the same time, it remains unclear how to explain this contradiction, and where the «missing» force is located, squeezing the skirt with an additional force corresponding to a specific pressure of 4.9 MPa. That is much greater than the lateral force determined by the pressure in the cylinder and the kinematics of the crank mechanism.

To answer the question, it is proposed to consider the ovalization of the piston pin under the load that occurs at the time of hydrolock. To simplify the task, it is assumed that the pin has a press fit in the piston head of the connecting rod. Due to the extremely high pressure in the cylinder, exceeding the maximum working pressure by several times, the frictional force on the upper part of the pin will act in the pairing of the pin with the piston. In addition, due to the ovalization and expansion of the pin in the horizontal plane, the additional force will appear associated with tension in the hole.

From a large number of sources, it is known [4, 5] that for the distributed load on the pin along the circumference according to the cosine law, the increase in diameter from ovalization in the horizontal plane is proportional to the load:

$$\Delta d_p = 0.09(p - p_0)B_\alpha, \tag{6}$$

the coefficient B_α is:

$$B_\alpha = \left(\frac{1+\alpha}{1-\alpha}\right)^3 \frac{FA_\alpha}{El_p}, \tag{7}$$

where $\alpha = d_0/d_p$, $A_\alpha = 1.5 - 15(\alpha - 0.4)^3$, d_p, d_0 is the diameter of the pin and the hole, respectively, l_p is the length of the pin, E is the modulus of elasticity of the material.

The calculation for a pin with dimensions of 22×14×60 mm gives, for example, at the maximum pressure in the cylinder corresponding to the hydrolock, the following result: at p=17.4 MPa Δd_p=0.083 mm. It follows that if the operating conditions are violated, the piston pin receives a significant deformation (Fig. 5).

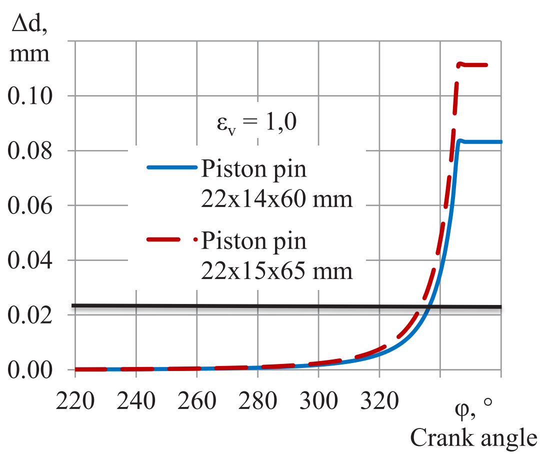


Fig. 5. Ovalization of the piston pin in the process of hydrolock

To test the result, the same pin at working pressure in the cylinder p=4.7 MPa gives an ovality of approximately Δd_p=0.02 mm, which, according to [4–7], is the maximum permissible. In accordance with this, with the ovalization of the pin, which is caused by the action of excessive pressure in the cylinder, the area of deformation with an increase in the size of the pin over 0.02 mm is taken as an area of abnormal operation.

Knowing the load, the stress in the middle of the pin can be calculated according to the well-known formula [4]:

$$\sigma_0 = 0.8(p - p_0)\frac{F(l_p + 0.5b)}{d_p^3(1-\alpha^4)}, \tag{8}$$

where b is the width of the piston head of the connecting rod.

The change in the stress in the process of hydrolock for a pin with dimensions of 22×14×60 mm is shown in Fig. 6.

It is understood that a significant increase in the size of the pin in the hole of the piston can cause a local tension in the piston hole, which can significantly change the conditions for pairing and interaction between the piston and the pin.

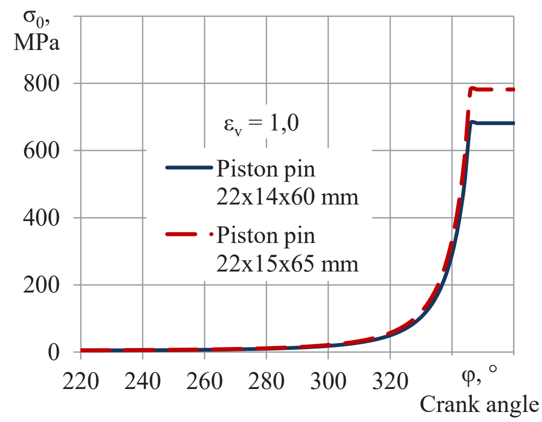


Fig. 6. Stresses in the middle of the piston pin during hydrolock

Upon further consideration of this process, a working hypothesis was adopted that excessive deformation of the pin under an abnormal load caused by a violation of the operating conditions creates significant friction in the hole of the piston reels. It affects the lateral force acting on the piston skirt near the top dead center.

It is known that the force (moment) of friction that occurs in the pairing of the pin with the connecting rod and piston is not taken into consideration in the classical theory in any way, they are neglected due to their smallness in comparison with other acting forces [18, 19]. The classical theory is actually built on the absence of friction in the conjugation of the pin [20, 21]. However, what is permissible under normal loads can give a big mistake when the operating conditions of the car are violated. Therefore, there is every reason to assume that it is the lack of consideration of the frictional force in the pin-piston pairing that makes the result of the calculation of the lateral force during hydrolock (5) not corresponding to the actual loads.

In fact, if the lateral force resulting from excessive pressure in the cylinder is completely insufficient to deform the piston skirt during hydrolock, then the opposite case can be considered. Thus, it can be assumed that due to excessive efforts, the piston pin is so strongly deformed that it jams in the connecting rod and piston. Then the connecting rod near the upper dead point will actually rotate relative to the axis of the upper (piston) head along with the piston (Fig. 1) and deform its skirt.

The proposed model makes it possible to calculate this deformation. To do this, set the angle φ_0 of rotation of the crankshaft, corresponding to the moment of jamming of the pin. In most automotive gasoline engine pistons, the top edge of the skirt roughly corresponds to the top edge of the pin opening. When turning the piston along with the connecting rod, one can put that the deformation of the skirt will occur only below the axis of the hole under the pin. Then the deformation of the lower edge of the skirt is determined by an approximate ratio:

$$\delta_0 = \left(h - \frac{d_p}{2}\right)\mathrm{tg}\beta_0, \tag{9}$$

where β_0 is the angle of deviation of the connecting rod from the vertical axis at the time of jamming of the piston pin, h is the height of the skirt.

This angle is related to the angle of rotation of the crankshaft φ_0 via [4]:

$$\sin\beta_0 = \lambda\sin\varphi_0. \tag{10}$$

Then expression (9) gives the following dependence of the deformation of the lower edge of the skirt on the angle of rotation of the crankshaft at the time of jamming the pin:

$$\delta_0 = \left(h - \frac{d_p}{2}\right)\frac{\lambda\sin\varphi_0}{\sqrt{1-\lambda^2\sin^2\varphi_0}}. \tag{11}$$

Our calculation according to formula (11) for the initial data corresponding to the common dimensionality of gasoline engines (h=44 mm, d=22 mm, λ=0.333) at an angle value of φ_0 equal to 15° (which approximately corresponds to the complete filling of the chamber with liquid) gives a deformation of the lower edge of the skirt=2.85 mm. This means that if the pin jammed completely, then the pistons after the hydrolock would not only be deformed but most likely would have a ruined skirt. However, this is not observed in practice [14, 17].

In other words, the assumption of complete jamming of the piston pin also does not correspond to the actual picture of the hydrolock, as well as the results of calculations according to the classical theory. It remains to be assumed that the state of the piston pin is somewhere between complete jamming and free sliding. This state corresponds to cranking with friction.

To describe the cranking of the pin with friction and to eliminate contradictions, consider the frictional force resulting from the actual appearance of tension in the connection of the pin with the piston and connecting rod (Fig. 1). Thus, the contact pressure p_R that occurs on the pairing surface is related to the moment of friction M by the ratio [22]:

$$p_R = \frac{2M}{kfd_p}, \tag{12}$$

where k is the coefficient of friction, f is the contact area.

When landing with tension, it is assumed that the parts are mated in the presence of a lubricant. For such conditions, a coefficient of friction of 0.15 can be assumed to pair the steel pin with the silumin piston [23].

It is clear that if the increase in diameter when the pin is ovalized is less than the gap, then there is no tension, the contact area is zero, and the pairing does not create any friction. That is, friction occurs when there is tension in the pairing, so, from expression (6), one can find a load (pressure in the cylinder) at which the increase in the diameter of the pin is equal to the gap Δ:

$$p_\delta = p_0 + 11.1\frac{\delta}{B_\alpha}. \tag{13}$$

It follows that the tension Δ of the pin in the conjugation can be taken with the following restrictions:

$$\Delta = \begin{cases} 0, \ \Delta d_p \le \delta, \ p \le p_\delta, \\ 0.09(p - p_\delta)B_\alpha, \ \Delta d_p > \delta, \ p > p_\delta. \end{cases} \tag{14}$$

In fact, this means that the beginning of the action of friction on the process occurs at the time of reaching such a pressure that ovalizes the pin until the opening of the reels bursts. Obviously, under normal operating modes of the engine, such a significant deformation is not allowed. However, this point determines the boundary beyond which the conditions of normal operation are violated, and instead of the classical theory, the operation of the mechanism with friction should be considered without taking into consideration friction. To this end, it is sufficient in the calculation formulas to use the boundary pressure p_Δ instead of the pressure p_0.

The pull of the pin, provided that the distribution is uniform along the circumference and length, and the specific pressure of the pin on the surface of the hole are related via the dependence [22]:

$$\Delta = p_R d_p C, \tag{15}$$

where coefficient:

$$C = \frac{C_d}{E_d} + \frac{C_p}{E_p}, \tag{16}$$

and E_d and E_p are the elastic modules of the pin and piston materials.

The coefficients C_d and C included in expression (16) are calculated using the formulas [22]:

$$C_d = \frac{1+\left(\dfrac{d_0}{d_p}\right)^2}{1-\left(\dfrac{d_0}{d_p}\right)^2} - \mu_d, \ C_p = \frac{1+\left(\dfrac{d_p}{d}\right)^2}{1-\left(\dfrac{d_d}{d}\right)^2} + \mu_p, \tag{17}$$

where μ_d, μ_p are the Poisson coefficients of the pin material and the piston, respectively, d is the conditional outer diameter of the enveloping part, close to the outer size of the piston reels.

From formulas (12) and (15), it follows that the moment of friction in the pairing of the piston pin is a function of its tension in the hole, that is:

$$M = \frac{1}{2}kfd_p p_R = \frac{1}{2}kf\frac{\Delta}{C}. \tag{18}$$

From (18), it can be seen that the connection of the pin, piston, and connecting rod with tension will be able to transmit the greater the torque, the greater the tension, the coefficient of friction and the contact area.

Next, we should consider where and how this moment is applied. From the scheme of action of forces (Fig. 1) it follows that near the upper dead point, the connecting rod makes a turn around the axis of the piston pin. The resulting moment is transmitted to the piston if there is friction in the connection of the pin. That is, the moment of friction increases the specific pressure on the skirt.

This moment does not affect the entire skirt but its part located below the axis of the pin (Fig. 1), while the force from the moment will act on the skirt depending on the height of its application, increasing from 0 to the lower edge of the skirt. Then, if the force is applied in the middle of the skirt, it can be associated with the moment via the following dependence:

$$N_k = \frac{2M}{h - \dfrac{d_p}{2}}. \tag{19}$$

The area of the skirt, previously taken as half the area of the piston, makes it possible to write the expression for the specific pressure on the skirt from the moment of friction in the pairing of the pin:

$$p_k = \frac{4M}{F\left(h - \frac{d_p}{2}\right)}. \qquad (20)$$

Or, taking into consideration formula (18), for the moment:

$$p_k = \frac{2kf\Delta}{FC\left(h - \frac{d_p}{2}\right)}. \qquad (21)$$

In (21), it is necessary to clarify the area of contact f. When ovalized, the pin deforms the hole, but not along the entire circumference, since it is originally installed in a hole with a small gap, which in the hot state is usually about Δ=0.020 mm.

It can be assumed [24] that the deformation of the piston under load from excessive pressure in the cylinder occurs in the direction opposite to the bending deformation of the pin (Fig. 7). This is difficult to take into consideration in the calculation but it can seriously enhance the effect of ovalization.

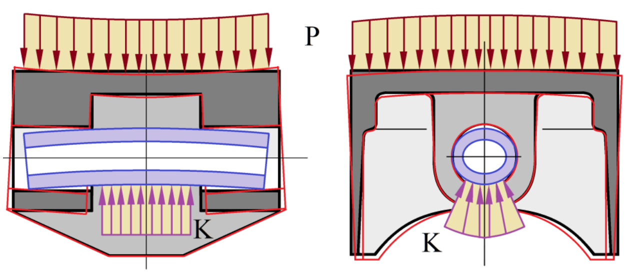


Fig. 7. The nature of the deformation of the piston and piston pin when loading the piston with excessive pressure (K – connecting rod reaction)

To calculate the landing at the ovality of one of the mating parts, estimation interaction models based on a finite-element method are known [25]. However, the complex spatial nature of the deformations of the piston and pin (Fig. 10) greatly complicates the construction of the model, its numerical solution, and the subsequent application of the results in expert practice. As a result, to obtain quantitative data, it is customary to use simpler models that do not require special tools for their implementation.

Taking into consideration the fact that the gap in the pairing is less than 0.1 % of the diameter, and the hole of the reels is also deformed to one degree or another from the current loads, it is advisable to define the contact area as:

$$f = \gamma\pi d_p l_b, \qquad (22)$$

where l_b is the total length of the part of the pin in the piston reels, in modern pistons usually close to 2/3 of the length of the pin, γ is a coefficient that takes into consideration the influence of various factors on the area and nature of the contact.

In this case, the coefficient γ takes into consideration how much of the area of the hole is in contact with the pin, as well as what effect the deformation of the parts has on the contact area and on the friction in the pairing of the pin with the piston.

With increasing loads and ovalization of the pin, the contact area also increases, but since there is a gap in the initial state in the pairing, the contact area is always smaller than the area of the hole itself. Given the approximate nature of the model under consideration, the change in area was not taken into consideration in further calculations, and the coefficient γ in the 1st approximation, before refinement with the help of finite-element models, was taken to be equal to 0.5. This may overestimate the specific pressure at the beginning of the process with a small ovalization of the pin but will not affect the result with a large ovalization at maximum pressures limited by the loss of stability of the connecting rod.

In accordance with this, expression (21) takes the form:

$$p_k = k(p - p_\delta)\frac{0.18\frac{\pi\gamma}{EC}\left[1 - 10(\alpha - 0.4)^3\right]\left(\frac{1+\alpha}{1-\alpha}\right)^3}{\frac{h}{d_p} - \frac{1}{2}}, \qquad (23)$$

wherein the coefficient C is calculated from formulas (16) and (17).

Then, taking into consideration expressions (3) and (5) for the total specific pressure on the piston skirt during hydrolock, the following formula can be derived:

$$p_{N\Sigma} = p_N + p_k = (p - p_0)2\text{tg}\beta +$$
$$+(p - p_\delta)k\frac{0.18\frac{\pi\gamma}{EC}\left[1 - 10(\alpha - 0.4)^3\right]\left(\frac{1+\alpha}{1-\alpha}\right)^3}{\frac{h}{d_p} - \frac{1}{2}}, \qquad (24)$$

in this case, friction in the formula begins to be taken into consideration only after the pressure in the cylinder reaches the p_Δ value – this means that the pin is deformed so much that it creates tension in the piston hole.

Fig. 8 shows the results of the calculation of specific pressures based on (24) with all the data and conditions specified above.

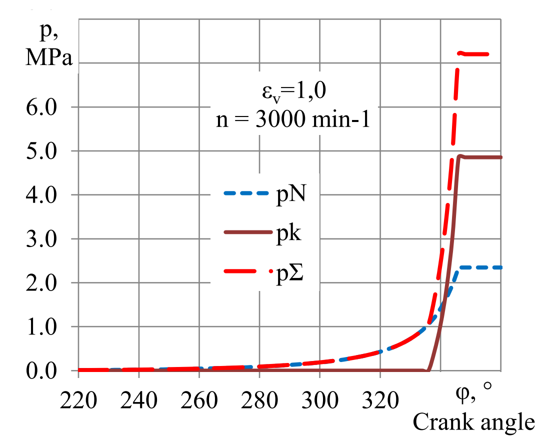


Fig. 8. Specific pressures on the piston skirt during hydrolock: from the component of the pressure force on the piston p_N, from the tension and friction in the pairing of the piston pin p_k and the total p_Σ

The result (Fig. 9) indicates that the resulting specific pressure on the piston skirt during hydrolock is much higher than the ICE theory gives for lateral force per piston.

5. 3. Comparing the simulation results with known expert data on actual engine damage due to hydrolock

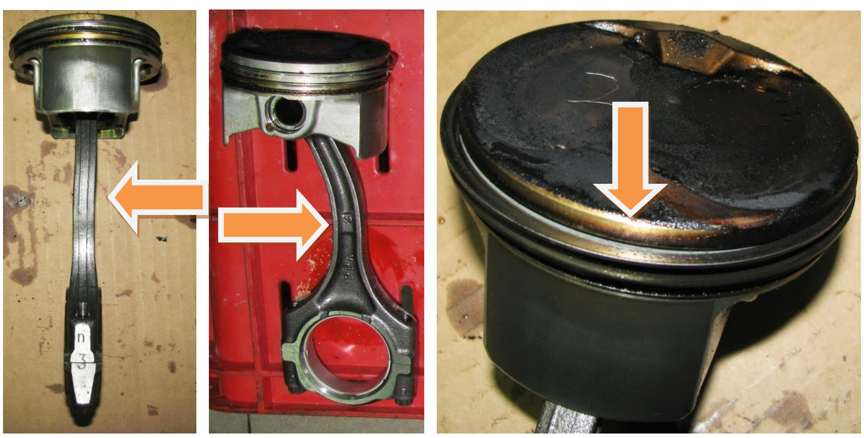
To verify the model, a comparison of the calculated values of the deformations of the piston skirt and piston pin with the data obtained during expert studies of the technical condition and causes of malfunction of automobile engines was performed.

For our study, a 4-cylinder gasoline engine for the Honda CR-V car with a working volume of 2.4 liters, damaged as a result of a hydrolock in two cylinders, was taken (Fig. 9). At the same time, in one cylinder, the longitudinal bend (transverse deformation) of the connecting rod was about 1.0 mm, and in the other – about 5.5 mm. By micro metering, a residual deformation of the piston skirt and piston pin was also revealed.

These data were used in modeling the loading of parts during hydrolock according to the procedure described above. Next, the pressure in the cylinder (Fig. 4), the stresses and deformation of the rod of the connecting rod (Fig. 4), as well as the stress of the piston pin (Fig. 6), were determined.

After calculating the specific pressure on the skirt using (24), the results obtained were compared with the limit value of the specific pressure on the skirt of 7.2 MPa given in chapter 4. Similarly, for the piston pin, the maximum stress was 700 MPa. The excess in the calculations of these values was taken as the appearance (presence) of their residual deformation, which can be calculated by a finite-element method with a relative error not exceeding 8–10 % (Fig. 3, *a*).

Thus, the devised model makes it possible to identify and explain the residual deformation of the piston pin and the piston skirt after a hydrolock (Table 1).



a *b* *c*

Fig. 9. Parts damaged by hydrolock: *a* — connecting rod with deformation up to 1.0 mm; *b* — connecting rod with deformation up to 5.5 mm; *c* — piston with traces of work with the shift in the cylinder due to the deformation of the skirt

Table 1

Evaluation of deformation of engine parts after hydrolock in the cylinder

Research method	Deformation, mm		
	connecting rod	piston skirt	piston pin
Examination (measurement) of engine parts	1.0–5.5	to 0.10 mm and larger	to 0.01 mm and larger
Calculation by known models	1.0–5.5	0	0
Calculation using the improved model	1.0–5.5	to 0.10 mm and larger	to 0.01 mm and larger

These data are of key importance for determining the causes of engine malfunctions in expert studies of their technical condition when operating conditions are violated. It is not possible to obtain similar results using estimation models of the kinematics of the crank mechanism from the ICE theory.

6. Discussion of the results of modeling the destruction of parts of the connecting rod-piston group in the engine due to hydraulic lock

We shall analyze the established dependence (24) and the results of our calculation (Fig. 8). In addition to the traditional lateral force, an additional force can act on the piston skirt under certain conditions. It is actually caused by friction of the piston pin in the hole of the piston – due to deformation under the influence of an abnormal load from excessively high pressure in the cylinder (Fig. 7). Moreover, the effect of friction in the pairing of the piston pin on the piston skirt will be the stronger the higher the coefficient of friction and pressure in the cylinder, the shorter the skirt and the larger the diameter of the piston pin.

It follows that the proposed model qualitatively corresponds to the studied process of loading parts with loads that are not provided for in the design of the engine and are caused by a violation of operating conditions. Such tasks are considered and solved in a set of expert research methods, including for other components and systems of engines, by developing models applicable under abnormal conditions and loads [26].

In addition, the proposed calculation procedure actually makes it possible to find the «missing» force of pressure on the skirt at excessively high pressures in the cylinder, characteristic of hydrolock. Indeed, even despite the assumptions made, the calculated value of the total specific pressure on the skirt (Fig. 8) is close to 7.2 MPa. This is about three times more than the lateral force in a crank mechanism without taking into consideration friction gives. In addition, this value for a given piston is close to the critical, presumably corresponding to the beginning of the plastic deformation of the skirt during compression (according to the simulation using data from [14]).

Since friction in the pairing causes deformation of the piston pin, the results of our calculation of the tension and deformation of the pin itself are of practical interest and require analysis (Fig. 5, 6).

Thus, Fig. 8 demonstrates that at the time of loss of stability of the connecting rod during a hydraulic lock, the stress in the middle of the pin reaches a dangerous level of 700 MPa. Since the common sizes of the pins were chosen for the study, their deformation (Fig. 6) at tension (Fig. 7) may not yet exceed the permissible limits. However, there are known designs of gasoline car engines with longer (up to 70 mm), and with thinner (thickness less than 4 mm) piston pins, for which the deformation will be greater. In this case, during hydrolock, it is possible for the stresses to exceed the yield limit of the material, when there is a permanent deformation of the piston pin known from practice after hydrolock [2, 14].

Our results establish a connection between external conditions and the nature of the current loads with damage to parts, which shows the difference between this study and the known ones. Thus, in [18, 21], the calculation of deformation is reduced to the state of the part itself, while the relationship of this state with the external conditions that caused it, as a rule, is not considered. In addition, unlike many studies limited only to a part with a visible deformation (connecting rod), this study examines the condition of other paired parts (piston, piston pin). As a result, a completer and more objective picture of the complex deformation and destruction of mated parts during hydrolock was obtained, which was absent in earlier studies.

At the same time, the proposed model, developed on the basis of expert studies of gasoline engines of cars, has not been tested for other types of engines, for example, diesel, which have different loading conditions and the nature of damage to parts. In addition, when building the model, in order to simplify it, the effect of the deformation of the piston pin on its mobility in the bushing of the piston head of the connecting rod was not taken into consideration. This assumption does not cause an error in the simulation for the press landing of the piston pin in the connecting rod, but for some engines with a floating pin landing, the calculation error may be noticeable.

As a consequence, further studies are expected to consider the effect of piston pin deformation on friction in conjunction with the bushing of the piston head of the connecting rod. In addition, it is required to test the model for diesel engines using available expert research data. This will make it possible to refine the model and extend the results obtained with its help to a wider range of automobile engines.

7. Conclusions

1. A mathematical model has been developed to determine the main loads and deformations acting in the connecting rod-piston group of the engine during a hydraulic lock in its cylinders due to violations of the operating conditions of the car. The model was studied for 4–8-cylinder gasoline engines with a working volume of 1.2–4.8 liters. With the help of the model, the parameters of the onset of deformation and the effect of destructive loads on engine parts during hydrolock were established – this is the pressure in the cylinder, close to 17.3 MPa at a crankshaft angle of about 346°.

2. The ranges of loads and deformations of the parts of the connecting rod-piston group during overload caused by hydrolock have been investigated and determined. Thus, with the help of the model, it was established that the amount of deformation of the piston pin during overload caused by hydrolock reaches 0.02 mm or more, which leads to the appearance of tension in the pin-piston pairing, with an increase in the stress in the middle of the pin to 700 MPa. This causes a sharp increase in the amount of friction in the pin-piston pairing and, as a consequence, an increase in the specific pressure on the piston skirt to 7.2 MPa, which is more than 2.8 times compared to normal and permissible pressure. The model gives a completer and more objective picture of the process of deformation and destruction of engine parts during hydrolock and makes it possible to explain the complex deformations and destruction of not only the connecting rod but also the skirt of the piston and piston pin.

3. The comparison of the simulation results with known expert data was performed from the condition of differential change in the amount of friction in the pin-piston pairing. The calculation yielded a satisfactory result that does not contradict the practical data and has confirmed the reliability of the model, which allows an objective assessment of the residual deformation of the parts of the connecting rod-piston group of the engine with a relative error not exceeding 8–10 %.

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