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SOME ASPECTS OF INFLUENCE OF THE CONNECTING ROD DESIGN ON THE OUTPUT PARAMETERS OF HIGH-SPEED INTERNAL COMBUSTION ENGINES

A comparative analysis of the stress-strain state of the connecting rods with a traditional I-beam and H-beam rod profiles was performed in order to compare various designs of the connecting rods and identify the factors that affect the main parameters of the internal combustion engine. Using the finite element simulation, it was found that the H-beam conrod has a much greater transverse bending rigidity, but is inferior to the I-beam profile conrod in terms of its mass. The calculations also determined that the frequency of natural oscillations of the H-beam profile conrod is significantly higher as well, what was confirmed by the testing on high-speed engines. The obtained data on piston operation with the H-beam conrod concluded that a more rigid connecting rod makes it possible to exclude transverse oscillations, reduce mechanical losses and improve the characteristics of the engine.

Keywords: internal combustion engine, ICE, connecting rod, H-beam, piston, rigidity

Introduction. As is well known, the vast majority of internal combustion engines use conrods of the traditional and generally accepted design that has been developing rather well for decades in all aspects, including the materials, the form and size of elements, production technology, etc. [1; 2; 3].

However, it should be noted that even such a well-developed system might experience changes. Thus, several years ago manufacturers of some types of engines (in particular, passenger car gasoline engines) set a massive transition to the fracture splitting (FS) technique of the conrod [1; 4], in which the splitting of the conrod big end is done after deep cooling.

The described change of technology took place due to the manufacturers' urge to reduce the cost of mass production – the "splitting" technology not only significantly reduces the number of operations, but also simplifies the design.

The given example demonstrates that the change in the technical requirements of such a well-established and widely used engine part as the connecting rod can be the reason for their very serious modification.

There are a large number of studies on the simulation of conrod stresses and strains [5; 6]. Most of them are devoted to conrods of traditional designs [7; 8], and only a small number of works study other conrod designs, including the H-beam profile conrods [9; 10]. However, these studies mostly concern specific designs, including the analysis of their fatigue strength, and do not consider any general trends. Subsequently, many studies do not normally give due attention to the direct comparison of the properties of the conrod types in question, however, if a comparison of some characteristics is performed [9], no clear explanation is given as to which design is recommended to be used in what type of engine.

Consequently, the objective of this work is to analyze conrod designs and to identify the factors that influence the basic parameters of the engine.

Features of Modern Internal Combustion Engines. Even a superficial analysis allows us to conclude that further improvement of the automotive ICE has taken the

path to increasing specific power. If the standard power at the end of the past century was 50 hp per 1 litre of volume, while 70 hp /litre suggested some kind of exclusivity or even some special application of the engine, currently the level of 90-100 hp /litre is no longer the limit even for mass designs.

All these processes are accompanied by an increase in the maximum rotation speed. Since the 1970s, as the widespread introduction of the constructions with over-head camshafts had begun, the maximum rotation speed was persistently kept at around 6000 rpm. However, the engines with the maximum rotation speed of 6500-7000 rpm gradually became more widespread.

It is clear that such serious changes could not have happened without affecting most of the main engine parts. For example, low height pistons with thin (1.0–1.2 mm) piston rings and 18–19 mm diameter piston pins, valves with a stem diameter of 5–5.5 mm, etc., have become widespread [10; 12]. However, the connecting rod was less affected by these changes, which lets one hypothesize on possible trends for further development of the connecting rod design.

Comparison of Various Conrod Designs. It is clear that the conrods of various designs can find their application in internal combustion engines of different types. In order to determine the effect the conrod design has on the engine parameters, it is necessary to compare some features of these conrods.

As is known, the conrod of the traditional design has an element that connects the big and the small end in the shape of the I-beam stem (Fig. 1).

Fig. 1. A typical traditional conrod with the I-beam stem.

Over the past few decades, this design scheme has undergone minimal changes, which, in addition to the aforementioned "split" end, mainly concerned downsizing, accompanied by the thinning of the stem section.

As a result, many of today's gasoline engines conrods have fairly openwork de-signs that, even at the most superficial glance, do not have any outstanding stiffness characteristic [13].

In fact, the I-beam profile of the stem can sustain the loads well enough in the longitudinal (circumferential) direction, which cannot be said about the transverse one. However, the conrod gets no apparent load in the transverse direction, therefore this design continues to remain dominant in mass production [14].

At the same time, another conrod stem design is known, mostly used in sports engines. These are the so-called H-beam connecting rods [4, 10], in which the I-beam stem is deployed 90 degrees around the longitudinal axis of the conrod (Fig. 2). In the past, such connecting rods were also used in some aircraft piston engine designs.

It is easy to notice that this type of the connecting rod differs rather seriously from the traditional one. It evidently has a different stem profile, however it is not en-tirely clear at first glance why such a special profile is needed or what it is good for.

Fig. 2. Typical H-beam connecting rod.

Load Simulation of Stems of Various Types of Connecting Rods. Obviously, in order to perform a comparison of conrods, it is necessary to simulate the loading of the stems of these designs under the same conditions. A part of the study has been carried out before on the conrod design fine-tuning for sports engines [4; 15]. For the purpose, two beams were chosen that imitated conrod stems – the traditional I-beam and the studied H-beam (Fig. 3).
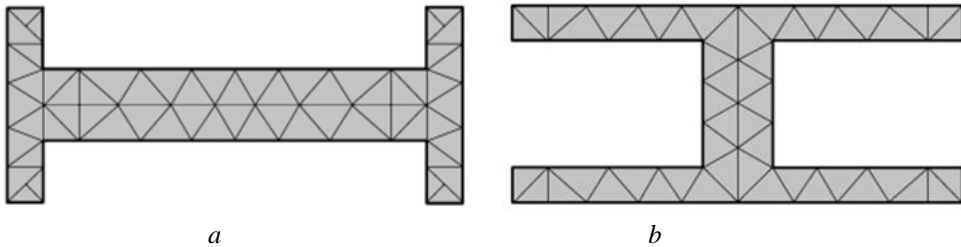

a   *b*

Fig. 3. The cross sections of the conrods under study: the traditional I-type (*a*) and the H-type (*b*) beams.

To equalize the conditions, the cross-sectional area of the beam was evenly distributed over the structural elements and in both cases the value was 128 mm^3. To simplify the task, the rounded edges were intentionally not provided.

The design scheme for the simulation was set as follows [15]: both beams had a fixed seating on one edge and the other edge was left free, but the equal transverse force of 1000 N was applied to it. This force was modeled as a load distributed over a small section (Fig. 4).
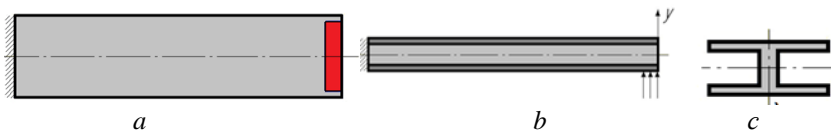

a   *b*   *c*

Fig. 4. The design scheme of the conrod stem with transverse loading:
a – bottom view, *b* – side view, *c* – stem shape.

Then, using the ANSYS software [16; 17], a computational experiment was conducted to determine the stress-strain state of the beams, for which the finite element method (FEM) was used. The resulting finite element model consists of approximately 6 thousand elements, for which the program calculated stresses and strains.

The calculation results are presented in the form of the so-called contour plots (Fig. 5), which make it possible to see not only the state of the part under load, but also its initial position. At the same time, for greater clarity, the scale of the beam deformation was increased by 5 times on the graphs [15].
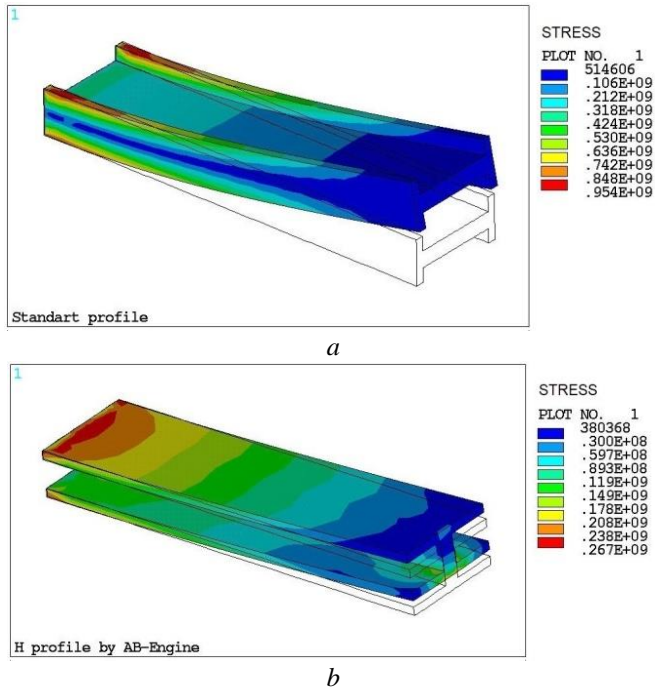
Fig. 5. The nature of variations of the transverse deformation and von Mises stress on the stem
of traditional design (*a*) and the H-beam stem (*b*) with the same load.

The obtained data allowed us to draw the following conclusions. When testing the traditional beam profile, the maximum lateral deformation (along the y axis) under the given load was 2.69 mm, while, when testing the H-beam profile, the deformation was 3.5 times less and that is 0.763 mm. Similarly, with the same load of 1000 N, the maximum von Mises stresses in the traditional profile beam are 3 times higher and amount to 946 MPa instead of 316 MPa for the H-beam one (Fig. 6).
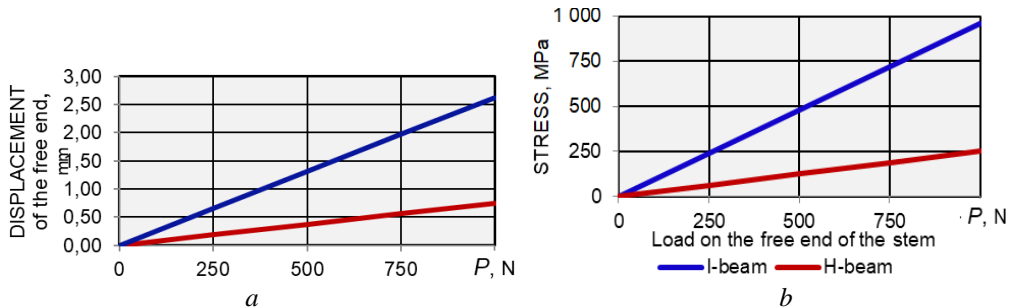

Fig. 6. Deformation as the free end displacement (*a*) and stresses (*b*) in the conrod stems of the analyzed designs in dependence on the cross load.

Thus, the conrod with the H-beam stem provides 3.5 times more transverse rigidity than the conrod with the standard profile stem, with the cross-sectional area being the same, and even requires a slightly less durable material for the same mass. However, in the longitudinal direction (in the rotation plane) both types of the connecting rods demonstrated little difference in terms of rigidity (determined by the thickness of the horizontal elements in Fig. 3).

Load Simulation of the H-beam conrod. This type of research helps identify cross sections with increased stresses under normal loading of the part, which may be of

practical importance. Examples of such studies are known [5; 6; 7] but they normally deal with purely specific tasks [8; 9] and do not perform comparison with the conrods of the traditional design.

Indeed, since the shape of the H-beam conrod is significantly different from the traditional design, it is desirable for practical purposes to check it for critical sections. For this purpose, a 3-dimensional model of the connecting rod (Fig. 7) was built, which was then used to create the finite element model (Fig. 8).

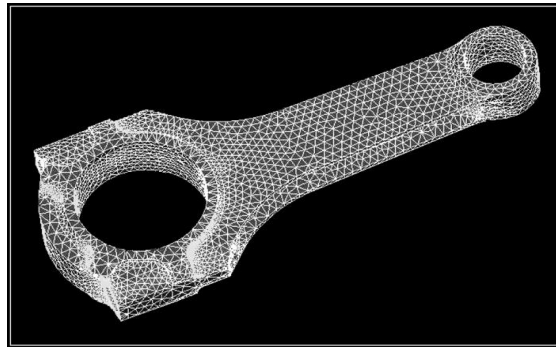Fig.7. 3-dimensional model of the conrod.

Fig. 8. Splitting the conrod model into the finite elements (mathematical model).

Further, when simulating the loading of the connecting rod loading under tensile and compressive forces, it was found that, in case of the H-beam profile conrod, unlike that of the traditional design, the attention should be paid to the ribs of the stem profile in the lower part near the transition to the big end. With the thickness of the stem profile ribs being constant, these places are subject to increased stress (Fig. 9, *a*), which can be reduced mainly only by increasing the thickness of the ribs.
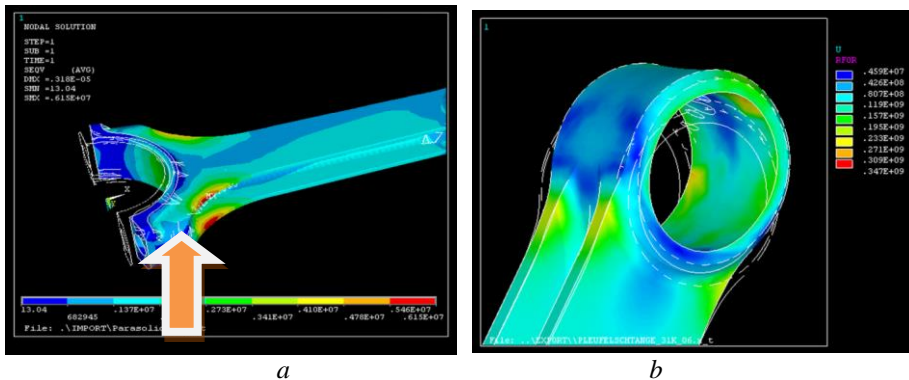

a

b

Fig. 9. Simulation of the compression of the connecting rod indicates the presence of potentially dangerous areas on the outer surface of the ribs near the big end (*a*) and the zone of high stresses at the small end of the conrod as indicated by the simulation (*b*).

Other areas of the ribs that require attention are located near the conrod small end. Conrods of traditional design have them close to the critical areas. However, while the critical points of the traditional conrod design are considered to be the cross section of the transition from the stem to the small end, the H-beam conrods have the increased stresses localized a little lower at the edge of the ribs (Fig. 9, *b*).

Despite the above-mentioned advantages, it is necessary to point out the drawback of the H-beam scheme, which is related to the so-called scale factor. When considering the cross-sectional shape of the traditional and the H-beam conrod types (Fig. 3), one can see that the area will be the same only if the ribs of the H-beam conrod are two times thinner than the base of the traditional I-beam one.

It is impossible to achieve that in practice, since the ribs of the H-beam stem would have to be extremely thin – 1.5–2.0 mm, which would make the practical application of such a design seem doubtful. As a result, the H-beam conrod, with all the other conditions being the same, is at least 10-15% heavier than the traditional one.

Nevertheless, even the overmass is actually acting towards increase in rigidity, which suggests that with the engine working at close loads, the H-beam conrod will deform many times less than the traditional one, especially in the transverse direction. It remains only to address the following questions: why is it so important as well as in what cases, how and what can the increased mass of the conrod affect.

The Study of the Conrods Working Features by Various Indirect Signs, including Abnormal Piston Swing inside the Cylinder. Despite rather large capabilities, the load simulation does not provide complete information on the working features of various conrod designs. Therefore, collecting and analyzing experimental data becomes an important stage of research.

However, before conducting any sort of analysis, it is necessary to understand which data are required and how they can be used.

The practice of investigating the causes of internal combustion engine faults shows [18] that one of the indicators of normal operation of the ICE is the piston – by the nature of its contact with the other parts and elements, it is possible to determine the damage or the defect, as well as its exact place or point.

So, in the well-known phenomenon – the change of direction of the piston movement at the dead centers – the piston swing occurs in the rotation plane and is limited by the piston skirt, therefore with a normal clearance in the cylinder and the corresponding skirt profile the swing is insignificant. In this case, the skirt that makes up the piston guide area, fully determines the position of the piston inside the cylinder. The way the piston will then look like is shown in Fig. 10.

Fig. 10. A normally working piston always has the top land area evenly covered with soot around the entire circumference.

However, in the case of, for example, large wear of the skirt and/or cylinder surface, the piston swing at the dead centers will become significant and may lead to the piston touching the cylinder with its upper area (top land). This is the cause for the appearance of the corresponding traces on the piston top land above the skirt such as the scuffed (polished) areas or areas free from carbon deposits (Fig. 11). A piston knock may begin too [18], especially at cold start [19; 20].
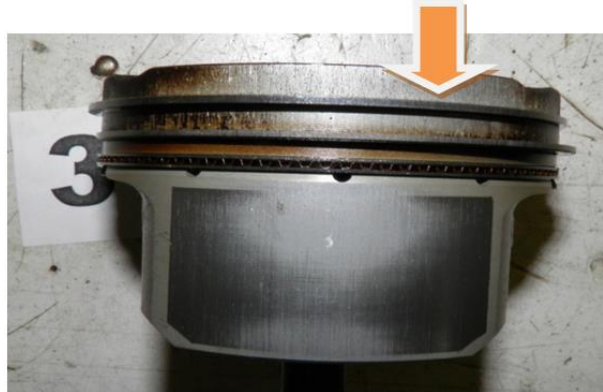

Fig. 11. A worn piston does not only lead to a large clearance inside the cylinder, but also to a significant piston swing, which can easily be seen from the traces of contact of its top land.

The above-mentioned signs of the contact traces (the touch and the impact) reveal the abnormal (i.e. absent in normal operation) interaction of the piston with the cylinder, which is important as it directly affects the engine output parameters.

Indeed, as is known, the piston group accounts for the main share of mechanical losses of the engine – up to 50% [2; 3]. This means that any abnormal contact of parts will inevitably cause an increase in mechanical losses and will lead to deterioration in the basic parameters of the engine. And precisely here, as experience shows, one should look for the difference between conrods of various designs.

The Cross Loads that Act upon the Conrod. It is commonly assumed that the connecting rod is not subject to cross loading. Indeed, the axes of the big and the small ends of the conrod are defined by the technology of its machining and are strictly parallel, as are the axes of the conrod and the main journals of the crankshaft. Additionally, the axis of the cylinder is strictly perpendicular to the axis of the crankshaft and is also defined by the machining technology, just like the axis of the piston is perpendicular to the axis of the piston pin hole.

Under such conditions, there are no additional transverse forces that could affect the conrod stem, while various kinds of deviations in the mutual arrangement of surfaces as well as the manufacturing defects and/or operational damage are, as they say, force majeure and are not related to the normal engine operation.

It is clear that if one makes an assumption that there are no transverse loads, then the conrod requires no additional transverse rigidity – the rigidity of the traditional I-beam conrod would appear sufficient.

However, the assumption that no transverse forces act upon the conrod can be applied to ideal conditions only. In general, such an assumption can be disproved by the known experience in engine operation and repair [18; 21], which suggests that the real working conditions of engine parts are far from the ideal ones. In order to under-

stand the source of the problem it might be necessary to investigate the signs that could appear if some kind of transverse load began to affect the conrod at some point.

Symptoms and Causes for the Piston Working with Cross-Skew. Even if one does not conduct a detailed analysis of the causes and features of the transverse forces that may arise, it is easy to imagine the consequences of such an effect: the piston is tilted inside the cylinder in the direction perpendicular to the rotation plane, i.e. in the vertical plane that passes through the crankshaft axis.

The piston inclination in the cylinder in this direction is not limited by the skirt the same way as in the rotation plane – simply because modern petrol engines have pistons that have no skirt on the sides of the pin hole (Fig. 10). In fact, the skirt is not needed there, since the piston is connected to the conrod small end with the help of a piston pin. And since during normal operation the clearances in this connection are measured in hundredths of a millimeter, any piston transverse swing in the cylinder is possible only together with the conrod small end.

In other words, the transverse inclination of the piston can only be caused by a bend in the conrod [21; 22]. Then it becomes clear what consequences the skew of the piston in this direction may lead to – the exact same polished areas, as shown in Fig. 12, will appear on the piston top land, but above the piston pin holes. That is, when the connecting rod is bent in the cross direction, the piston will inevitably touch the cylinder with its top land. This is nothing more than an abnormal contact of the parts, which can cause an increase in mechanical losses and a decrease in the maximum engine power.

It should be noted that the transverse deformation of the connecting rod is not an uncommon occurrence in the ICE operation [18; 22; 23], its main causes being hydrolock (which is the result of various liquids entering into the cylinder), piston impact damage (valve breakage, foreign objects) as well as poor-quality repair of the engine in general and the connecting rod in particular (Fig. 12).

Fig. 12. A typical and clearly marked one-sided wear of the piston top land over the pin hole is caused by the work of the piston with an inclination.

With this kind of damage the piston has a pronounced one-sided skew, as a result of which the contact area of the piston top land with the cylinder is only on one side (besides this, there are also other symptoms of the piston operating with the deformed conrod). But all these cases are associated with either operational damage or defects in repair production.

Symptoms of Transverse Bending Oscillations of the Connecting Rod. High-speed highly powered sports engines have displayed another type of abnormal piston

operation (Fig. 13), with the scuffing of the piston top land observed not from one, but from two sides at once.

a

b

Fig. 13. Small two-sided marks of scuffing over the pin holes on the top land of sports engine pistons – from a slight touch (*a*) to a wide contact area (*b*).

It was clearly a symptom of the piston operating with inclination, but it was unclear how and why the inclination occurred in both directions at the same time. Even after examining several engines it was not easy to figure out the cause of the phenomenon – presumably, the observed nature of the scuffing was not associated with the piston thermal expansion, since the pistons that exhibited the two-sided scuffing were of different geometry and design.

Results and their discussion. It turned out possible to resolve the contradiction with the piston inclination occurred in both directions at the same time by assuming that the cause of the two-sided scuffing of the piston top land lies in the transverse oscillations of the connecting rod. Indeed, such oscillations are possible and lead to the bending of the conrod [24], which is confirmed by known studies [25; 26].

To simplify, if the "conrod-piston" system is represented as a beam clamped on the crank side with a load (the piston with the pin) applied to the center of the small end of the conrod, the lowest frequency of the natural transverse oscillations of the system can be approximately calculated, according to [27], by the simple equation:

$$\omega = \sqrt{3EI_y/[l^3 (M + 0{,}235\,M_0)]}\,,$$

where E is the elastic modulus of the conrod material, I_y is the moment of inertia of the conrod stem in transverse bending, l is the length of the stem, M,M0 are the masses of the piston (with the pin) and the conrod, respectively.

The moment of inertia I_y depends on the geometrical characteristics of the stem, which determines the dependence of the resonance frequency not only on the type, but

also on the dimensions of the stem profile, in particular, on its relative width h ∕ b, equal to the ratio of the width h of the profile to its thickness b.

The calculation shows (Fig. 14) that for automotive engine conrods with the I-beam stem the natural transverse oscillation frequency is close to 7000-7500 rpm, which may be the cause of increase in mechanical losses when the rotation speed approaches resonance. At the same time the H-beam stem profile immediately increases the natural frequency by 1.5-1.6 times, up to 11000-12000 rpm, especially with an increased width of the profile.
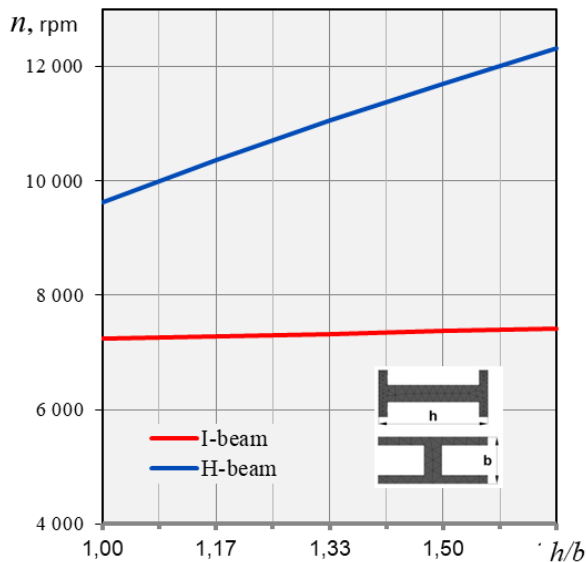

Fig. 14. The influence of the conrod stem type and size on the rpm corresponds to the natural frequency of the transverse oscillations of the conrod stem.

To test this hypothesis, similar connecting rods were made for several 1.6-liter engines, but with the H-beam conrods [28]. When testing the engines, it turned out that the transition to the H-beam profile resulted not only in the disappearance of scuffing on the piston top land above the pin, but also in a simultaneous increase in power characteristics, mainly due to the possibility of a significant increase of the maximum rotation speed.

The best result was obtained on a sports 2.0-liter atmospheric 4-cylinder engine with the cylinder diameter of 81.8 mm and piston stroke of 95 mm: with the H-beam conrods it could easily go up to 9000 rpm (at higher rpm power blockage appeared). At such rotation speed, the average piston speed reached 29 m/s, significantly exceeding the values of this parameter in similar engines, normally within 22-24 m/s [11; 29; 30].

It is characteristic that a slightly higher mass of the H-beam conrod did not manifest itself in any way during the study. On the one hand, the excess weight of the conrod could cause an increase in the load on the piston skirt and, consequently, lead to higher mechanical losses. However, in all likelihood, the mechanical losses due to friction of the piston top land at high rotation speed were much greater, and their reduction completely compensated for a certain increase in the friction losses of the skirt.

It is clear that the effect of the use of the H-beam conrods was observed only at high rotation speed, above 6000-6500 rpm, below which no changes in one direction or the other could be identified.

Conclusions. The traditional design of the I-beam connecting rod still used in the vast majority of gasoline engines, has practically exhausted its high-speed reserves. At rotation speeds above 6000-6500 rpm, signs of bending transverse oscillations of the connecting rod are found, which lead to an increase in mechanical losses and deterioration of the basic parameters of the ICE, especially in small-size high speed modern engines of different purposes. The obtained results indicate that a possible solution to the problem is the transition to the connecting rods with the H-beam stem profile, which ensures significant stress reduction, many times greater cross rigidity, thus reducing mechanical losses and improving the output parameters of the internal combustion engine.

The plans for future studies involve the simulation of transverse oscillations of the connecting rod as well as the estimation of the influence of the stem geometric characteristics on the amplitude of such oscillations and on the engine parameters.

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НЕКОТОРЫЕ АСПЕКТЫ ВЛИЯНИЯ КОНСТРУКЦИИ ШАТУНА НА ВЫХОДНЫЕ ПАРАМЕТРЫ ВЫСОКООБОРОТНЫХ ДВИГАТЕЛЕЙ ВНУТРЕННЕГО СГОРАНИЯ

Проведен сравнительный анализ напряженно-деформированного состояния шатунов с традиционным двутавровым и Н-образным профилями с целью сравнения различных конструкций шатунов и выявления факторов, влияющих на основные параметры внутреннего сгорания. Было отмечено, что несмотря на общеизвестность Н-образного профиля шатуна, в настоящее время практически отсутствуют работы, в которых бы подробно рассматривались особенности, возможные преимущества и рекомендации по применению Н-образных шатунов в двигателях различных типов. Используя моделирование методом конечных элементов, было установлено, что Н-образный шатун имеет гораздо большую жесткость поперечного изгиба, но уступает шатуну с обычным двутавровым профилем по массе. Кроме того, было обнаружено, что, в отличие от стандартных шатунов, у Н-образного шатуна наиболее нагруженными элементами являются ребра в нижней части вблизи перехода к кривошипной головке. Расчеты также показали, что частота собственных колебаний Н-образного стержня значительно выше, чем у обычного двутаврового, у которого частота собственных колебаний может оказаться близкой к рабочей на высоких частотах вращения и вызвать рост механических потерь. Испытания спортивных бензиновых двигателей помогли получить новые данные о том, как конструкция шатуна влияет на механические потери и мощностные характеристики двигателя. Так, было выявлено, что при высоких скоростях вращения поршень может работать в цилиндре с поперечным перекосом, что, вероятно, вызвано поперечными колебаниями шатуна, которые можно устранить с помощью перехода на Н-образные шатуны. Полученные данные о работе поршня с шатунами разных типов позволили сделать вывод о том, что более жесткий шатун помогает подавить поперечные колебания, снизить механические потери и улучшить характеристики двигателя.

Ключевые слова: двигатель внутреннего сгорания, ДВС, шатун, Н-образный стержень, поршень, жесткость.

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ДЕЯКІ АСПЕКТИ ВПЛИВУ КОНСТРУКЦІЇ ШАТУНА НА ВИХІДНІ ПАРАМЕТРИ ВИСОКООБОРОТНИХ ДВИГУНІВ ВНУТРІШНЬОГО ЗГОРЯННЯ

Проведено порівняльний аналіз напружено-деформованого стану шатунів з традиційним двотавровим і Н-подібним профілями з метою порівняння різних конструкцій шатунів і виявлення чинників, що впливають на основні параметри внутрішнього згоряння. Було відзначено, що всупереч загальновідомості Н-подібного профілю шатуна, в даний час практично відсутні роботи, в яких би детально розглядалися особливості, можливі переваги та рекомендації щодо застосування Н-подібних шатунів у двигунах різних типів. Використовуючи моделювання методом кінцевих елементів, було встановлено, що Н-подібний шатун має набагато більшу жорсткість поперечного вигину, але поступається шатуну зі звичайним двотавровим профілем по масі. Крім того, було виявлено, що, на відміну від стандартних шатунів, у Н-подібного шатуна найбільш навантаженими елементами є ребра в нижній частині поблизу переходу до кривошипної голівки. Розрахунки також показали, що частота власних коливань Н-подібного стрижня значно вище, ніж у звичайного двотаврового, у якого частота власних коливань може виявитися близькою до робочої на високих частотах обертання і викликати зростання механічних втрат. Випробування спортивних бензинових двигунів допомогли отримати нові дані про те, як конструкція шатуна впливає на механічні втрати та потужності характеристики двигуна. Так, було виявлено, що при високих швидкостях обертання поршень може працювати в циліндрі з поперечним перекосом, що, ймовірно, викликано поперечними коливаннями шатуна, які можна усунути за допомогою переходу на Н-подібні шатуни. Отримані дані

про роботу поршня з шатунами різних типів дозволили зробити висновок про те, що більш жорсткий шатун допомагає придушити поперечні коливання, знизити механічні втрати та поліпшити характеристики двигуна.

Ключові слова: двигун внутрішнього згоряння, ДВЗ, шатун, Н-подібний профіль, поршень, жорсткість.

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