

Benefits of using titanium alloy as the construction material of the connecting rod and piston in the SI engine

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The overall efficiency of an internal combustion engine determines the ability to convert the energy generated during fuel combustion into useful mechanical work. One method to increase efficiency is to reduce internal losses. The component of the internal losses of an internal combustion engine is the inertia force of the parts in a reciprocating motion. In the work, the author presented the reduction of the inertia force in the reciprocating motion, obtained by reducing the masses of the connecting rod and piston. Reducing the masses of these elements was the result of the use of an innovative material: titanium alloy Grade 5. The paper presents the analytical determination of mechanical loads affecting the modified elements of the crank system. The process of designing a connecting rod and piston using finite element simulation tests is briefly described. Inertial force was determined analytically for two versions of crank systems: factory and modified. The obtained results were compared, and a significant reduction in inertial force is one of the advantages of using titanium alloy on selected elements of the crank system.

Key words: *inertial force, fuel saver, energy efficiency, connecting rod, piston*

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1. Introduction

One of the methods of increasing the overall efficiency an Internal Combustion Engine (ICE) is based on reducing internal losses. The component of internal ICE losses is the friction force, related to the inertia force of the parts moving in a reciprocating motion. Therefore, reducing the value of inertia forces should reduce friction losses in the engine, with the value of the friction coefficient remaining unchanged. The magnitude of the inertial force depends on the acceleration on the ICE components moving on the cylinder axle and on the mass of these parts. The acceleration value depends on the geometry of the crank system and the instantaneous rotational speed of the crank. In the article, the author presented the benefits of using the Ti6Al4V titanium alloy as a construction material, as a reduction in the masses of the crank system (piston and connecting rod). By reducing the masses of these components, compared to the factory components of the modified engine, the designated inertial forces in reciprocating and rotary motion were reduced.

There are publications describing the use of titanium alloys for crank system elements, such as the connecting rod [1]. In the article, the authors presented the reduction in connecting rod mass achieved by optimizing its shape and dimensions. Ti6Al4V titanium alloy was used as the construction material. ANSYS software was used to calculate static and fatigue strength.

The manuscript [4] presents the methodology for using CAE software to design a connecting rod. The result of the analysis is a change in dimensions and shape in subsequent iterations of the connecting rod model subjected to simulations. As a result, a reduction in weight and the maximum value of reduced stresses were achieved.

In the paper [9], the authors presented the results of simulation calculations of a connecting rod using the finite element method. The computer analysis was carried out to determine the cause of failure (breakage) of the connecting

rod in the existing design of a compression-ignition engine. Based on the results of the simulations, it was found that the main cause of failure of the connecting rod was a high level of stress near the bolt hole. In this zone, the stress pile-up is caused by high bolt preload. The use of the FEM method made it possible to determine the area of notch occurrence.

The manuscript [5] contains a description of the design process and numerical analyzes using ANSYS software to design the connecting rod of an ICE. The determination of the amount of static stresses was made using the Finite Element Method (FEM).

The paper [6] is concerned with the analysis of fatigue-damaged pistons. Also, static stress analysis is used to determine the stress distribution during the combustion on piston.

The manuscript [7] covers issues related to the design of an ICE piston with thermal load, for different materials. FEM were also used in simulation analyses for design the piston.

In their conclusions, the authors of the presented articles presented the possibility of using titanium alloy as a construction material for the connecting rod. Some of the works determined the weight reduction resulting from the use of titanium alloy in selected elements of the crank system. However, there is no determined reduced value of inertia forces resulting from the use of titanium alloy. This is the main goal of this article. Moreover, the author proposes the use of titanium alloy as a construction material for the piston of an ICE. The effect of simultaneous reduction in the mass of both the connecting rod and the piston (compared to the factory elements) will be presented in the form of a comparison of the course of inertia forces as a function of the crankshaft rotation angle, for the rotational speed corresponding to the maximum power of the modified ICE.

The object of modernization by the author of the paper is the ICE a single-cylinder, 4-stroke unit, powered by ethanol, with an eccentric crank system. The parts subject to modification are the connecting rod and the piston. The selected construction material is Ti6Al4V titanium alloy, some of its physical parameters are presented in Table 1.

Table 1. Physical properties of the Ti6Al4V alloy (source <https://asm.matweb.com>)

Parameter	Unit	Value
Tensile strength Rm	MPa	896
Yield strength Rh	MPa	827
Young's module	GPa	114
Density	g/cm ³	4.43
Thermal conductivity	W/m·K	6.7
Specific heat capacity	J/kg·K	526

The modified engine is used to power the vehicle running in the Shell Eco-Marathon (SEM) competition. The idea of the international competition for student teams is to build vehicles whose goal is to cover the greatest possible distance on one liter of fuel. Therefore, all possibilities of increasing the efficiency of the combustion engine are desirable and justify the use of an unusual construction material, such as titanium alloy, for the connecting rod and piston. The desirability of minimizing inertial force by designing new parts results from the use of an ICE.

Selected parameters of the drive unit are included in Table 2.

Table 2. Technical data of ICE

Parameter	Unit	Value	
Bore	D	mm	35
Crankshaft cranking	r	mm	21.9
Length of rod	l	mm	120
Cylinder axis shift	e	mm	1.5



Fig. 1. The vehicle taking part in the Shell Eco-Marathon competition

2. Connecting rod

2.1. Determining the forces acting on the connecting rod

From the theory of the structure of the combustion engine, we know that the forces acting on the connecting rod come from the pressure in the cylinder and the inertia of the parts in translational motion [2]. The effect of the resultant force from the combustion pressure and inertia leads to the

development of compressive and bending stresses on the connecting rod (buckling).

I-beam was chosen as the connecting rod cross-section. The cross-sectional markings used are shown in Fig. 2.

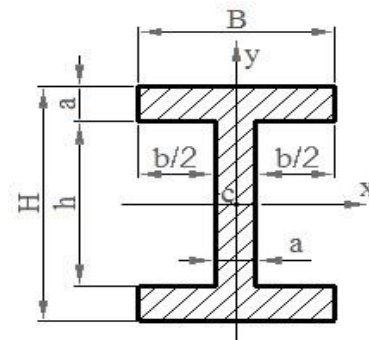


Fig. 2. Markings of the connecting rod head cross-section

The preliminary dimensions of the cross-section, selected to determine the masses and inertia loading the designed connecting rod, are as follows: B = 12 mm, b = 10 mm, H = 18 mm, h = 14 mm, a = 2 mm. Total mass of the preliminary connecting rod model was 53.5 g, mass of the rotating part 31.3 g and mass of the translational part 22.1 g. Mass of the complete stock piston, i.e. with rings, piston pin and protection devices is 59 g. The values of these masses were then used to determine the value of the inertia force loading the connecting rod.

Figure 3 shows the kinematic diagram of the crank system of the ICE being upgraded.

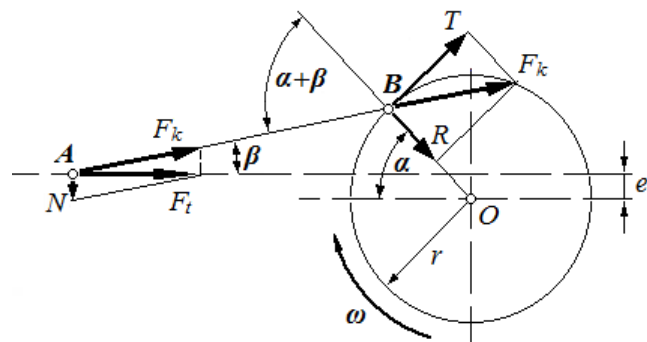


Fig. 3. Diagram of eccentric crank system

According to the simplification used in the study [2], we can present the acceleration in translational motion as equations (1)–(3). This simplification results from the development of an infinite trigonometric series, considering the first and second harmonics.

$$a = r\omega^2[(\cos \alpha + \lambda \sin \alpha) + \lambda \cos 2\alpha] \quad (1)$$

$$u = \frac{e}{l} \quad (2)$$

$$\lambda = \frac{r}{l} \quad (3)$$

where: a – linear acceleration, α – crankshaft rotation angle, ω – angular rotational speed, u – eccentric shift coefficient of the rod length, λ – relative slenderness of the connecting rod.

The calculations of the acceleration were carried out for constant the rotational speed, corresponding to the maximum power of the ICE, i.e. 5000 rpm.

The results of the acceleration value as a function of the crankshaft rotation angle are shown in Fig. 4. The maximum value of positive acceleration is 5939 m/s^2 and the minimum value of negative acceleration is -8591 m/s^2 .

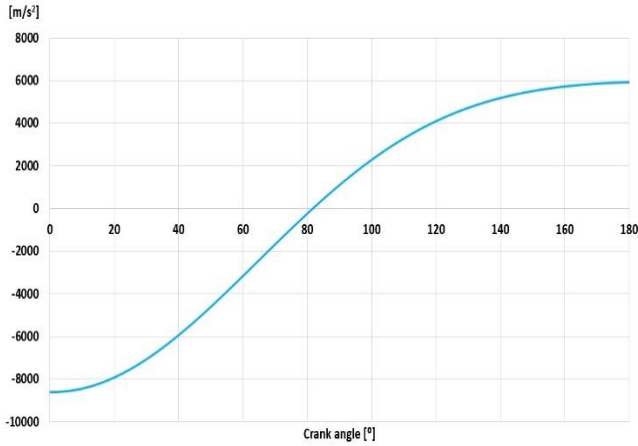


Fig. 4. Acceleration of translational moving parts as a function of the crankshaft angle

The inertia force for the reciprocating motion was determined from equation (4).

$$F_{mr} = \sum_{i=1}^n m_i \cdot a \quad (4)$$

where: F_{mr} – inertial force of all parts in sliding motion, m_i – mass of all crank system components in translational motion.

In ICE, the cylinder pressure as a function of the crankshaft rotation angle was determined by simulation using the AVL Boost program. The result in the form of a graph depending on the crankshaft rotation angle during the power stroke is shown in Fig. 5.

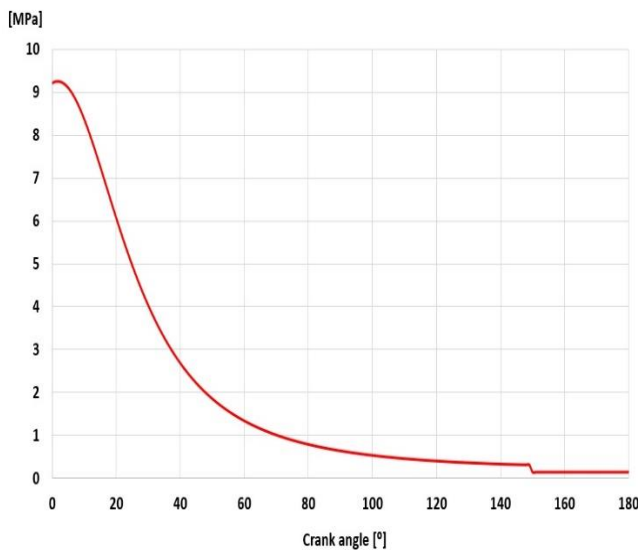


Fig. 5. Pressure in the cylinder during the combustion stroke

As we can observe in the graph above, the maximum pressure in the cylinder is 9.28 MPa and 8 degrees of crankshaft rotation are achieved after TDC.

The pressure force of the gases in the combustion chamber was determined from the equation (5).

$$F_g = p_g \cdot D^2 \cdot \frac{\pi}{4} \quad (5)$$

where: F_g – combustion force, p_g – combustion pressure.

The resultant force loading the connecting rod is determined from eq. (6) and (7), considering the sense of the determined forces.

$$F_t = F_g + F_{mr} \quad (6)$$

$$F_k = F_t \cos \beta \quad (7)$$

$$\cos \beta = \frac{r \cos \alpha - e}{l} \quad (8)$$

where: F_t – total force on the cylinder axis, F_k – total force on the rod axis, β – angle between cylinder and rod axis.

2.2. Numerical calculations of the connecting rod

The forces acting on the connecting rod determined in the previous subsection were used to perform the Finite Element Method (FEM) analysis. The software used to create the 3D model and static and fatigue simulation is SolidWorks

The boundary conditions of the analyzed connecting rod include support on the cylindrical wall of the connecting rod foot bearing, and receiving radial and axial displacement. Additionally, the piston pin axis is connected to a virtual wall, which corresponds to the physical contact of the piston leading part with the cylinder surface. The load was applied to the cylindrical wall of the piston pin sleeve, with a parabolic distribution corresponding to the unit pressures in the plain bearing.

The value of the safety factor was assumed to be 2.5. This means a maximum allowable static stress value of 350 MPa. If the stresses deviated significantly from the assumed level, the dimensions of the connecting rod head section were corrected and the simulation was repeated.

The final dimensions of the connecting rod head cross-section are as follows: $B = 11.25 \text{ mm}$, $b = 9.45 \text{ mm}$, $H = 13.05 \text{ mm}$, $h = 9.45 \text{ mm}$. The cross-section of the connecting rod at its foot has the following dimensions: $H = 26.10 \text{ mm}$, $h' = 22.50 \text{ mm}$. The distribution of reduced stresses according to Mises' theory for the performed static analysis is shown in Fig. 6.

Due to the variable loading conditions of the connecting rod, fatigue analysis was also used, in accordance with the guidelines for using FEM software for the calculation of crank system elements contained in [1]. The loading conditions are alternating (tension and compression), and uneven. The value of maximum compressive stresses was determined according to the static stress test. On the other hand, the value of tensile stresses was determined by knowing the maximum value of the inertia force, according to eq. (4).

The assumption of the fatigue tests used for FEM was to conduct a durability analysis for an assumed number of cycles of 100,000. The result is the number of cycles in which the tested item will not suffer damage. As you can see in Fig. 6, the entire connecting rod has a uniform red

color, corresponding to the assumed number of fatigue cycles.

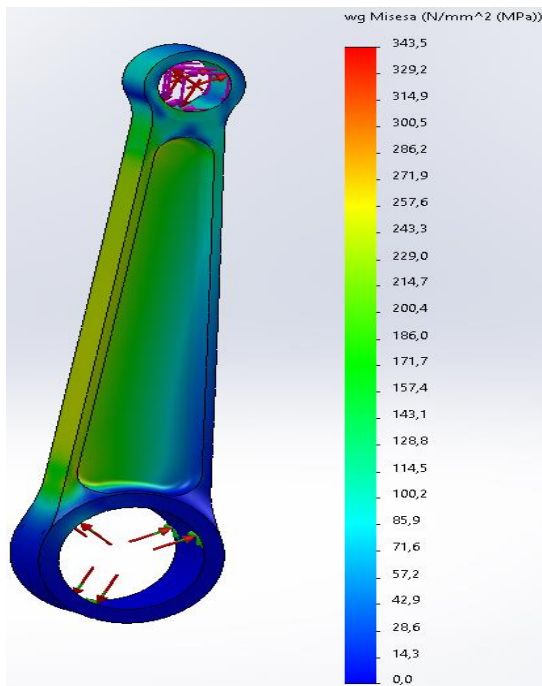


Fig. 6. Result of static analysis: reduced stresses according to Mises' theory

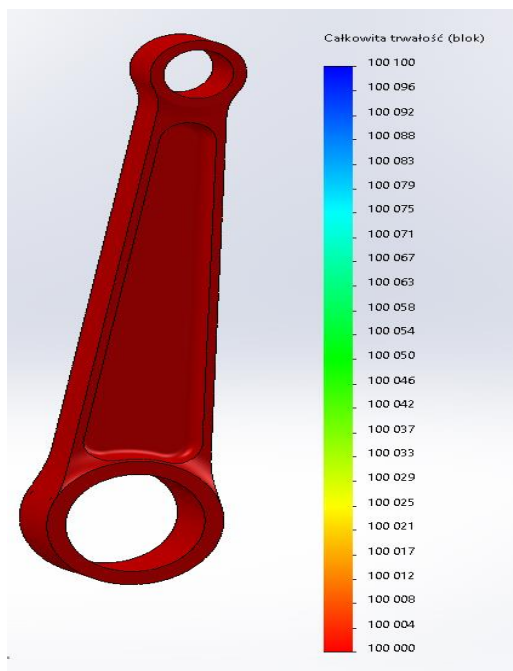


Fig. 7. Result of fatigue analysis

3. Piston

3.1. Determining the forces acting on the piston

The designed piston uses a single sealing ring and a loose pin fit, secured by two spring elements. The initial dimensions and shape of the piston were determined based on criteria given in [3].

The stresses in the piston result from the gas force load and the inertia of the piston itself. The gas force acting on

the bottom of the piston is equal to the force given in eq. (5). The inertia force of the piston, similarly to the inertia force loading the connecting rod, is determined from eq. (1) and (4). The difference is the mass of the component parts in eq. (4): in the case of piston load, we only take into account its mass and the piston rings.

In addition to the gas force and inertia, the piston is additionally loaded with stresses resulting from the thermal effect and the force pressing the leading part against the cylinder surface. The force pressing the leading part of the piston against the cylinder surface is determined from the eq. (9).

$$F_n = (F_g + F_{mp}) \sin \beta \quad (9)$$

where: F_n – force pressing the piston to cylinder, F_{mp} – inertia force of the piston with rings.

The value of the thermal load resulting from the impact of hot combustion products was determined according to the method included in [8]. From the known value of cylinder pressure (obtained in AVL Boost software), the average value was determined. The perfect gas equation was then used to determine the average duty cycle temperature (assuming no charge loss in the cylinder chamber). Assumptions for the calculations:

1) The medium is in thermodynamic equilibrium at all times. This is necessary so that equal values of the intensive parameters (pressure and temperature) for the entire combustion chamber can be assumed for calculations once the equations of classical thermodynamics are applied.

2) It is thermodynamically in a closed system (closed valves).

The temperature dependence as a function of the angle of rotation of the crankshaft was determined from eq. (10).

$$T_{(\alpha)} = \frac{P_{c(\alpha)} V_{(\alpha)}}{m_c R_c} \quad (10)$$

Due to the nature of the SEM competition, the operating time of the internal combustion engine is relatively short (up to 13 seconds), compared to the interruptions in its operation (more than 3 minutes) during track driving. Figure 8 is a graph of driving speed (green line) and engine speed (red line) during one vehicle start in the competition.

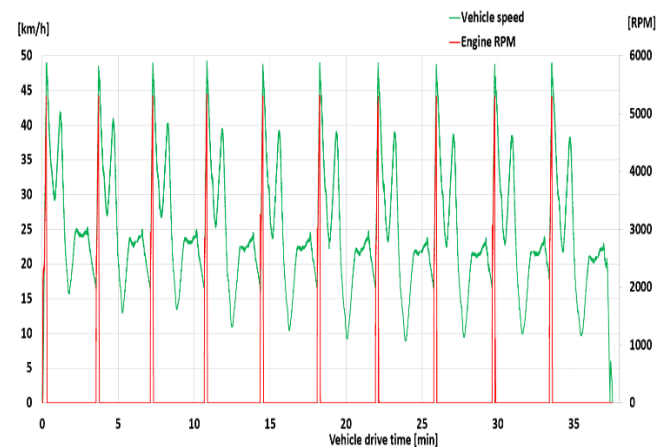


Fig. 8. Engine rotational speed and vehicle speed during one competition start

The effect of such a short engine run time is a low thermal load, due to the very short time for heat to affect the piston bottom. The effect of such a short engine operating time is a low thermal load, due to the very short time for heat to affect the piston bottom. The longest the internal combustion engine runs is during the start of the vehicle for the race. Other engine cycles do not last longer than 10 seconds. Figure 9 shows the engine speed over time, as recorded by the internal combustion engine controller.

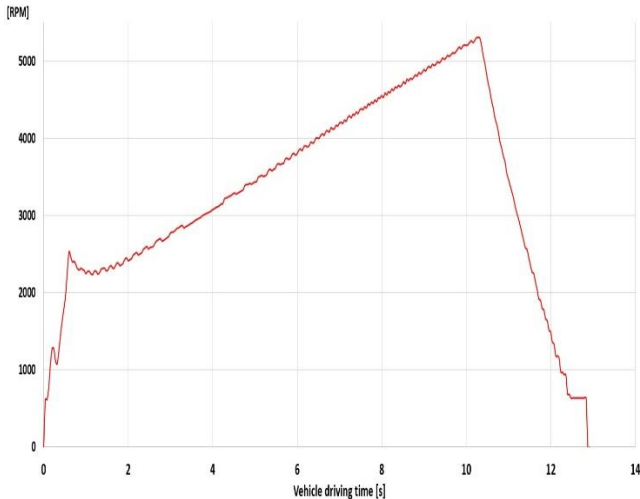


Fig. 9. Rotational speed of ICE as a function of time when starting the vehicle from a standstill

The heat transfer coefficient from the working medium to the piston bottom was determined from Woschni's experimental formula. The amount of heat transferred by radiation was ignored. The time of the transient thermal test was assumed to be the longest time of motor operation, i.e. 13 s. The initial temperature of the piston material was assumed to be 95°C. The engine is heated to this temperature with electric heaters before the start of the competition. The temperature distribution is shown in the following Fig. 10.

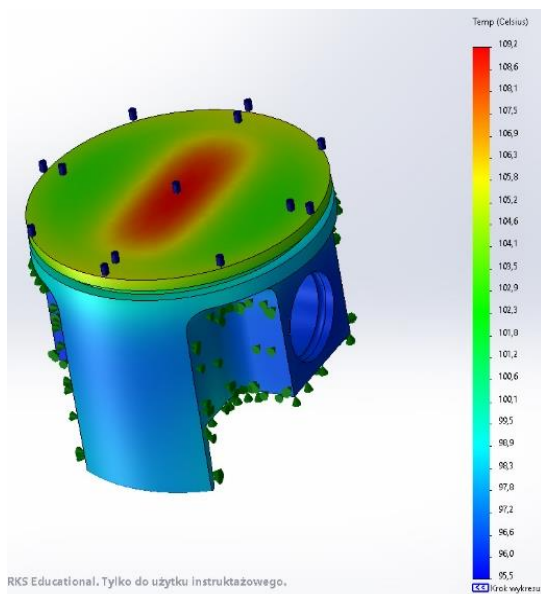


Fig. 10. Temperature distribution on the tested piston

The boundary conditions of the strength-analyzed piston include support on the cylindrical wall of the piston pin bore, accepting radial and axial displacement. In addition, the recesses under the piston pin protections received support from a movable support that receives displacement in the direction of the pin axis. The combustion pressure acts on the piston bottom and the lower surface of the sealing ring groove. The force of inertia of the piston is applied at the point of its center of mass.

The results of the FEM obtained in SolidWorks software are shown in Fig. 11.

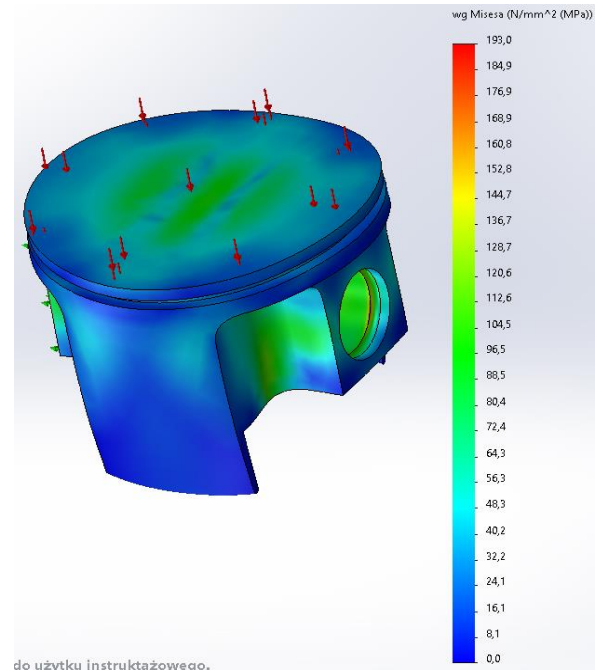


Fig. 11. Reduced stresses according to Mises theory

The criteria for evaluating the results of FEM simulations are the maximum values of reduced stresses and deformations of the piston face. The value of permissible reduced stresses was determined similarly to that for the connecting rod, with a safety factor of 2.5. In accordance with the recommendations of [3], a change of 0.006 mm in the diameter dimension was considered as a limit value, regardless of whether the diameter of the leading part increased or decreased. Such restrictive values of permissible displacements of the piston leading part force the use of thicker piston walls, which in turn leads to relatively small reduced stresses compared to the permissible ones.

4. Determination of inertia force

The value of the inertia forces depends on the magnitude of the accelerations acting on the engine components moving along the cylinder axis and on the mass of these parts. The eq. (1)–(3) determines the dependence of acceleration on the instantaneous angular position of the crankshaft. Table 3 is a summary of the masses values of individual elements of the crank system for the factory and modified versions.

Equation 4 was used to determine the value of the inertia force as a function of the crankshaft rotation angle, both for the standard and modified engines.

Table 3. Mass data of piston and rod

Parameter	Unit	Stock	Modified
Piston, rings, and pin rings	g	59	48
Rotating part of the connecting rod	g	93	36
Sliding part of the connecting rod	g	47	18

Figure 12 shows the dependence of the inertia forces of the original and modified crank system. These values were determined for a rotational speed of 5000 rpm.

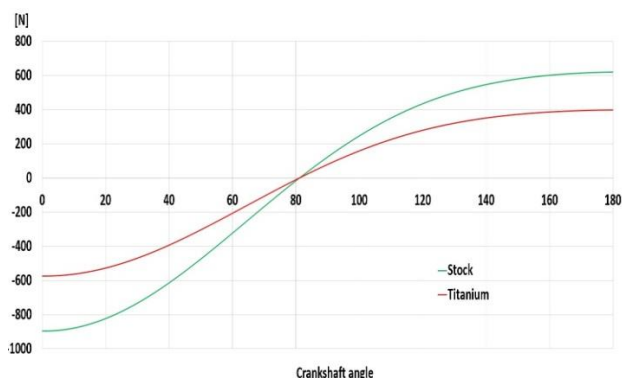


Fig. 12. Force of inertia of engine parts in reciprocating motion

5. Conclusions

The use of an innovative construction material in the form of titanium alloy Ti6Al4V allows to reduce the masses of the crank system components. A particularly noticeable difference concerns the connecting rod, for which the total mass reduction is around 53% from stock value. Such a large difference in mass is the result of twice the density of the titanium alloy compared to the construction material of the factory element. Greater strength allows for reduced

dimensions required by permissible stresses. In addition, the technology of making the connecting rod outline (removal machining) allows the design of a cross-section that allows the transfer of greater bending stresses while minimizing the weight. Obtaining small wall thicknesses with a large moment of inertia of the cross-section is an example of considering the properties of available manufacturing methods at the design stage.

In the case of the piston, the mass reduction is not so significant because the factory component is made of an aluminum alloy, a material with a lower density than the titanium alloy used. Moreover, during the FEM analysis, the criterion of permissible displacements was used, which results in low structural effort and, as a result, weight reduction. Reducing the mass of the piston is 22% of the original piston weight.

The largest difference in the negative inertia force is 322 N, which is a 35.8% change from the factory engine's inertia force value. The largest difference in positive inertia force is 222 N. This is also a 35.8% change from the factory engine's inertia force value. This is a significant reduction in the value of the force of inertia in progressive-return motion. At the same time, it should be noted that reducing the mass of the rotating part of the connecting rod will also reduce the moment of inertia of the entire crank system. In addition, a reduction in the masses of parts in forward and rotary motion, required a reduction in the masses of the crankshaft counterweights. As a result, the total reduction in the inertia of the entire crankshaft system (including the crankshaft after the reduction of the counterweights) will be greater than would result from differences in the masses of the modified parts alone.

Nomenclature

FEM finited element method
ICE internal combustion engine

SEM Shell Eco-Marathon
TDC top dead center

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