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# About I-beam versus H-beam connecting rod design using Inventor Autodesk 2018

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**Abstract.** Design method based on 3D modelling and finite element analysis is very convenient since minimizes materials consumptions and reduces considerably the manufacturing costs. That is why it is widely used in cars sector – industry and research. Based on this method, a frequently encountered problem of car engines, namely connecting rod stress, is analysed in the paper. The approach refers to the motor-sport cars and consists in the comparative analysis of stress in two cases described by two profiles of the connecting rod shank – I-beam profile and H-beam profile. Connecting rod modelling as well as the finite element analysis were carried out with Autodesk Inventor 2018. The analysis indicated H-beam profile as optimum technical solution.

## 1. Introduction

In attempt to break world records, the engines of the motor-sport cars are subjected to impressive forces. All the machinery component parts are pushed to the limit in these situations [1]. Many of the damages are caused by the bending or rupture of the connecting rods, which is one of the most vulnerable component parts of the engine. Several failure modes in connecting rods are analyzed in [2].

A case of damage caused by the destruction of the connecting rod during competition is indicated in figure 1. This is one of exemples presented in [3], which treats the diesel engines damages in the motor-sport world.

**Figure 1.** Connecting rod fail [3]

In the situation described above, the connecting rod comes from a standard Diesel engine. Aiming to produce more power, the dosing fuel system of the engine had been upgraded, as indicated in [4]. Besides, the ECU management was modified according to the aggressive programming principles. In situations like this, any component can fail, causing the total destruction of the engine.

In order to avoid the imbalance of the engine mechanism, the solution applied in the motorsport world is the replacement of the critical standard parts with more resistant components, having the same dimensions and weights. Thus, standard pistons and connecting rods are replaced by parts made of forged materials, with high mechanical strength; the powder forged rods developed as replacement parts for General Motors V8 [5] is just an example. Air and fuel supply circuits are oversized while the gas exhaust circuit is optimized accordingly. Due to all these changes, the power of the engine increases.

Currently, design method based on 3D CAD programs is widely used in cars industry and research sector. This is due to the fact that materials consumptions can be optimized and manufacturing costs can be minimized in this approach. Combining 3D CAD design method with finite element analysis, the fatigue analysis of engine component parts can also be performed. An analysis of fatigue stresses (carried out with ANSYS software) on a connecting road (modeled with CATIA software) is performed in [6]. An analysis on static, dynamic and thermal stress of two new sets of materials for piston, connecting rod and crank shaft is performed in [7] with ANSYS software. The present study is also based on the 3D CAD design method and finite element analysis. The aim of the study is calculation of forces in the engine mechanism and simulation of the working conditions for the connecting rods of a diesel engine. A difficult problem in the connecting rod design process is choosing the shank profile [8]. Two of the most common are I-beam profile and H-beam profile. These both profiles were analysed in the paper.

## 2. Calculation method

The study was performed assuming the physical parameters of a Volkswagen 2.0 liters TDI engine, of 102 kW, with a bore × stroke of 81 × 95.5 mm and a connecting rod length of 144 mm.

In order to perform the design of the connecting rod, the forces acting in the engine mechanism should be known. Forces acting on the connecting rod are function of stroke and function of crankshaft position. They are calculated according to [9] as presented below.

The gas force that acts on the piston was calculated as

$$F_g = \frac{\pi D^2}{4}\left(p_c - p\right) \quad [N], \tag{1}$$

where: D - cylinder bore, in mm;
$p_c$ - cilinder gas pressure, in MPa;
p - crankcase pressure, in MPa.

The inertial force was expressed as

$$F_i = -m_i a_i = 10^{-3} m_i r \omega^2 \left(\cos\theta + r\, l^{-1} \cos 2\theta\right) \quad [N], \tag{2}$$

where: $m_i$ - reciprocating mass, which is the sum of the piston mass and a percentage of the connecting rod mass, in kg;
$a_i$ - instantaneous acceleration of the piston, in m/s$^2$;
r - crank radius, in mm;
ω - crank angular velocity, in rad/s;
θ - crank rotational angle from top dead center, in °;
l - connecting rod length, in mm.

The resultant force acting in the cylinder axis was calculated as

$$F = F_g + F_i \quad [N]. \tag{3}$$

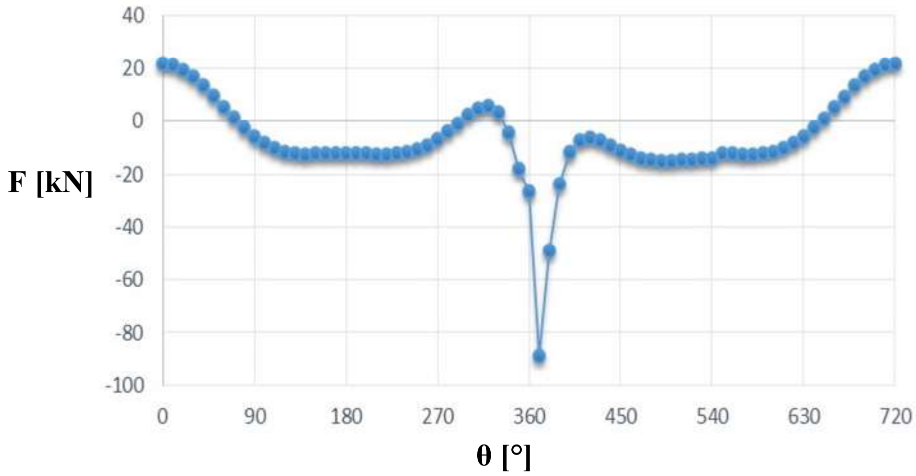



**Figure 2.** Resultant force function by crankshaft angle

Conventionally, it is assumed that resultant force acts in the piston pin axis. Dependency of this force as function of the crankshaft angle is presented in figure 2.

## 3. 3D modeling

Modelling of the engine parts and their assembling were carried out with Autodesk Inventor 2018 software. Theoretical calculations and setting of the gauge dimensions were followed by parametric assembling of the engine components. The component parts of the engine and the assembled engine block are presented in figure 3a and in figure 3b, respectively.

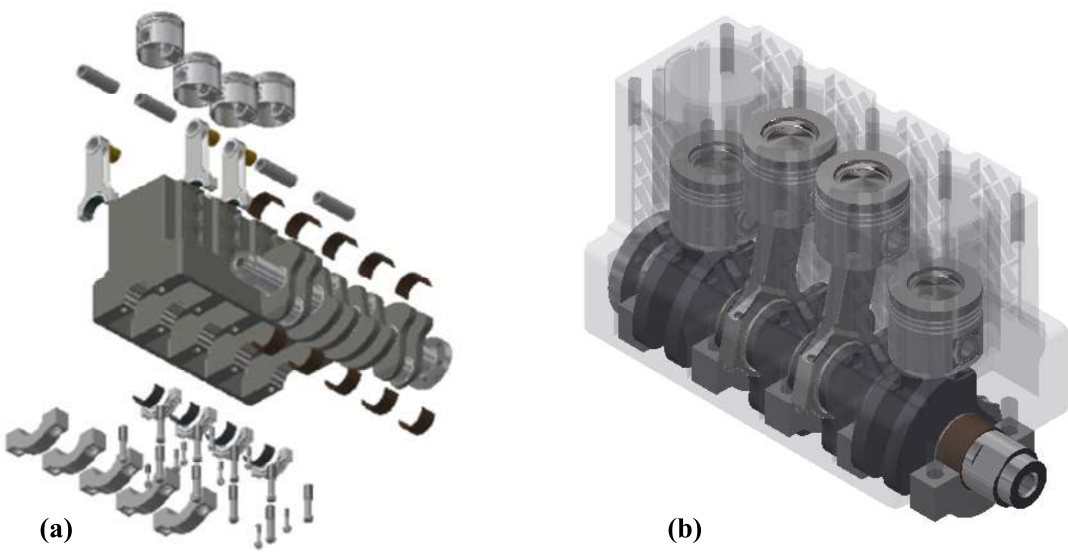



**Figure 3.** Component parts of the engine and engine block

## 4. Finite element analysis

### *4.1. Materials*

A high quality steel, namely AISI 4340 409 QT, was considered as material for connecting rods since engine must cope with peculiar operating conditions of competitions. The properties of this material are presented in figure 4, as offered by Autodesk database.

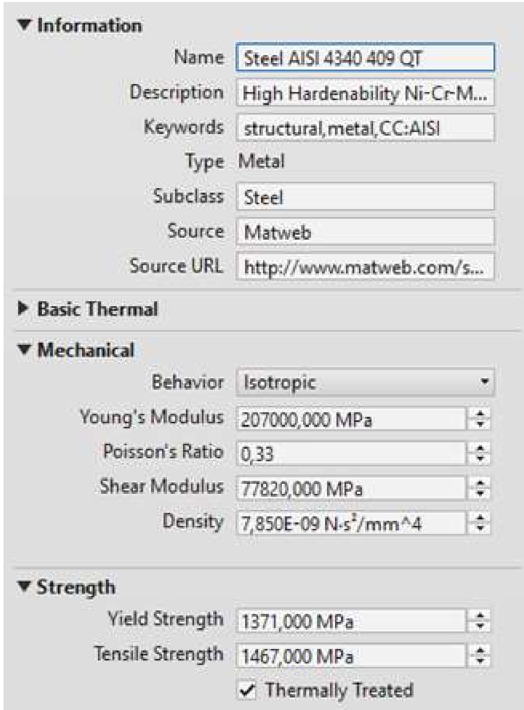
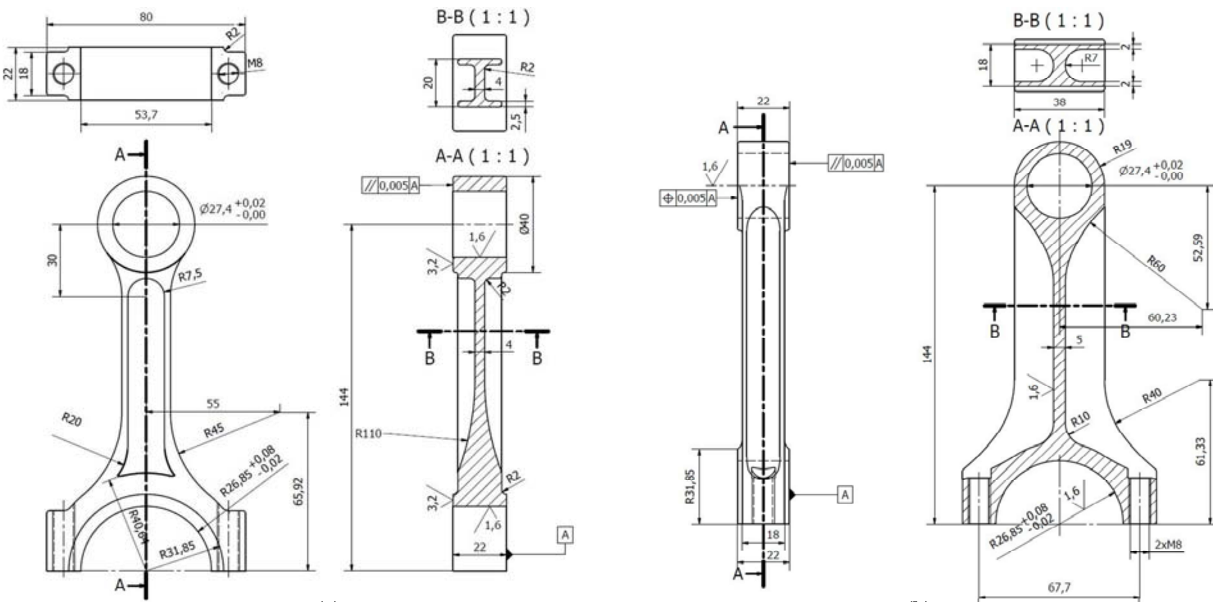



**Figure 4.** Properties of AISI 4340 409 QT steel according to Autodesk Library

A main objective of the study was to establish the optimum of two analysed profiles of the connecting rod shank, namely I-beam or H-beam. Dimensions of the two analyzed connecting rods are presented in figure 5. The mass of the connecting rod with I-beam profile is 820 g while the mass of the connecting rod with H-beam profile is 804 g (lighter by 1.95 %). The piston pins and bearings rod are identical in the two cases. They are made of the same material as the connecting rod, namely AISI 4340 409QT.




(a) (b)

**Figure 5.** 3D models of the connecting rods – H-beam profile (a) and I-beam profile (b)

### *4.2. Mesh generation*

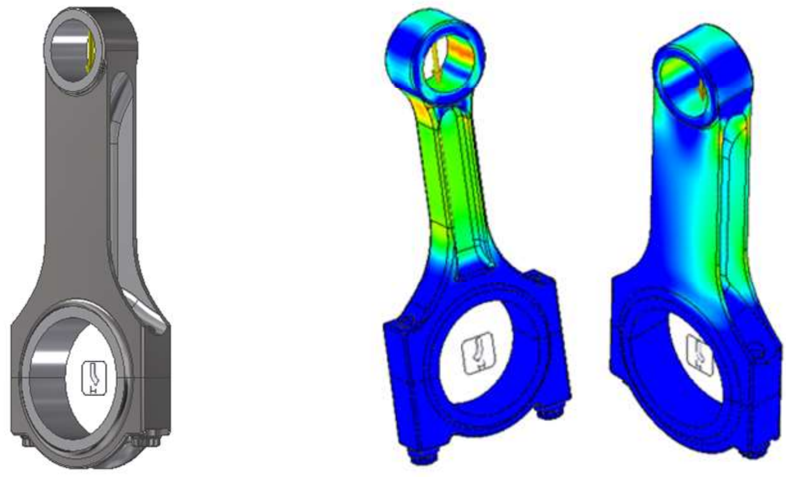
The automatic generation method with tetrahedral elements was used for meshing. The solid model of I-beam connecting rod has been meshed into 50309 elements and 82498 (figure 6a) while model of 3D H-beam connecting rod has been meshed into 59143 elements and 96700 nodes (figure 6b).


**(a)** **(b)**

**Figure 6.** Mesh generation

### *4.3. Constraints*

Constraints were applied according to the loadings in the connecting rod. Thus, connecting rod is mainly subjected to compression and bending – because of the gas pressure inside the cylinder – as well as to tensile load – because of inertia (inertia increases when rotation speed and mass of oscillating parts increase). Inertial load is ignored when calculate the compressive load. In this assumption, calculations are more safety since inertial load acts in opposite direction to the gas pressure load when the expansion in power stroke starts [9].



**(a)** **(b)**

**Figure 7.** Constraints of connecting rod and stress analysis

According to the aspects presented above, the big end was considered fixed (figure 7a). The force calculated with formula (3) was applied to the connecting rod in vertical direction. Analysis was performed by taking into consideration the crankshaft angle as well as the connecting rod angle relative to the vertical line [10], [11].

### *4.4. Static simulations*

Based on the theoretical data, several static analysis were performed during the 4 strokes (two complete rotations of the crankshaft). Function by stress conditions, location of stress concentrators can be identified, as indicated in figure 7b.

Depending on application, the stress concentrators can be removed or the connecting rod mass can be reduced in order to make the assembly lighter. Thus, parametric design allows the development of very economical and easy-to-achieve concepts.

### *4.5. Final results*

As result of the simulations performed, the charts indicating variations of Von Mises stress and displacement (deformation) function by crankshaft angle were drawn for both I-beam and H-beam profiles of the connecting rod shank. The charts are presented in figure 8.

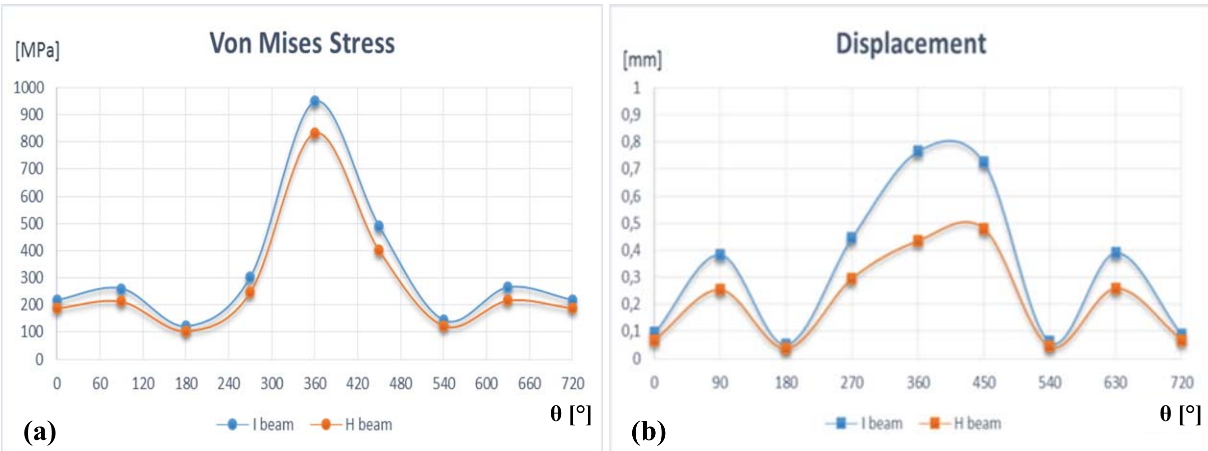



**Figure 8.** The evolution process of Von Mises stress and displacement function by crankshaft angle

It can be observed that Von Mises stress as well as displacement are lower in the case of H-beam profile of connecting rod shank. Thus, maximum Von Mises stress (when θ = 360°) is 12.3 % lower in the case of H-beam profile than in the case of I-beam profile (832.1 MPa versus 948.6 MPa) while displacement is up to 43.1 % lower (0.765 mm versus 0.435 mm, achieved when θ = 360 °). In average, the displacement is lower by 31.9 %.

## 5. Conclusions

Assuming the same material and the same acting forces in both analyzed cases – I-beam profile and H-beam profile for the connecting rod shank –, the study revealed that Von Mises stress is up to 12.3 % lower (15.7 % in average) when H-beam profile is used. Besides, stability is higher in the case of H-beam profile since displacement is lower – up to 43.1 %.

Consequently, it can be concluded that H-beam profile is a better option than I-beam profile. In spite of this, I-beam profile is used more often because implies an easier and less expensive manufacturing technology.

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