Numerical analysis of the use of different lattice designs and materials for reciprocating engine connecting rods

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Abstract

Nowadays, the use of lattice structure designs in metallic parts to produce lightweight systems has gained importance owing to advances in additive manufacturing technology. On the other hand, vehicle manufacturers are constantly looking for new ways to reduce weight due to the depletion of fossil fuels, the demand for vehicles having higher performance, global warming and increasingly stringent emission standards. In this study, in order to reduce the weight of the internal combustion engine, different lattice designs were made in the connecting rods. Four different 2.5D lattice designs, hexagonal, octagonal, square and triangular, were created in reference connecting rod body. The dimension of the lattice designs was 10x10x12 mm with the wall thickness of 1.5 mm. The fatigue behaviors of the connecting rods as well as mechanical properties under static conditions were analyzed using finite element approach. Three different materials were used in the analyzes: AISI 4140, Inconel 718 and Ti6Al4V. As a result, it was seen that weight reduction of up to 15.75% was possible in the connecting rod thanks to the lattice designs and the maximum stresses were below the yield stresses of the materials. Moreover, connecting rods with lattice design had satisfactory safety factor values.

Keywords: Weight reduction; Lattice design; Connecting rod; Mechanical properties; Finite element; Fatigue analysis
1. Introduction

The use of lightweight materials in air and land vehicles is an important method to improve performance [1–3]. Therefore, improvements in the performance of vehicles are closely related to advances in materials science and manufacturing techniques. New manufacturing techniques enable lightweight designs in engine materials, and so it is possible to achieve an important reduction in engine mass compared to conventional designs. This makes a significant contribution to reducing the overall weight of vehicle [4]. Nowadays, due to environmental concerns such as global warming and strict policies on these issues, vehicle manufacturers who make great efforts to reduce fuel consumption are constantly looking for ways to reduce weight even more [5]. For a more concrete understanding of the relationship between fuel economy and mass, the following example can be given. If the weight of a 1500 kg vehicle is reduced by only 150 kg (or 10%), a reduction in fuel consumption of around 7% take place [6]. Moreover, even the smallest detail in the designs made to reduce the weight of the mechanical parts used in the engine affects the fuel consumption and life of the parts [7].

One way to make lightweight designs is to use lattice structures in the metallic engine parts [8]. High strength and low weight are important advantages of the lattice structure [9]. Moreover, these lattice structures eliminate wastage of the material [10]. At the same time, it is possible to reduce weight of the material without a significant decrease in its mechanical properties. Thanks to the latest developments in additive manufacturing technology, it is possible to produce very precise lattice structures at high resolution. Since additive manufacturing is a near net shape manufacturing technology, post-production processes are not required. In addition, thereby this environmentally friendly technology, the properties and qualities of the materials are higher compared to traditional methods [11–14]. Flores et al. [15] showed that the use of lattice designs in the production of lightweight parts with additive manufacturing techniques can be decreased the unit cost up to 70.6%, manufacturing time up to 71.1% and weight up to 54.3% compared to traditional solid infill designs.

On the other hand, connecting rods are one of the important part of reciprocating engines [16,17]. They are mechanical parts that transmit the variable axial movement in the form of the push and pull of the piston to the crankshaft [18]. Thus, a connecting rod converts the reciprocating motion of the piston into the rotational motion of the crankshaft [19,20]. With the movement of burnt gases and inertial components in the engine, pressures occur on the connecting rod, which create compression and tensile stress, respectively [21]. Reciprocating engines have at least one connecting rod depending on the number of cylinders [22]. Until
today; many studies have been carried out on the connecting rod, such as stress, deformation, fatigue and the use of different materials. Usually the connecting rod material is made of low alloy steel such as AISI 4140. Saravanan et al. [23] analyzed that the existing connecting rod material AISI 4140 steel and AL S355 (aluminum) cast alloy by using Ansys software. The reason why AL S355 casting alloy is used is that it has very good strength and much less density than steel. However, due to the poor mechanical properties of Al 355 alloy compared to AISI 4140, it is not suitable for use in high compression ratio engines where connecting rods made of AISI 4140 material are used. Many materials such as titanium alloy, aluminum alloy, glass fiber composite can be used to reduce weight instead of AISI 4140 steel in the connecting rod [24,25]. However, considering the actual load produced by the piston resulting from combustion and the fatigue life under these load conditions, it is understood that the use of alternative materials is not possible in most cases. On the other hand, titanium alloys such as Ti6AI4V are used in mechanical parts due to their light weight and high strength properties [26]. Ajayi-Oluwaseun et al. [27] discussed the effect of shape optimization on Ti6AI4V used as connecting rod material in a 500 cc engine. Thanks to the optimized model, 11.7% reduction in weight, lower deformation and stress distributions have been obtained. Loga and Ku [28] made a design and fatigue analysis using C70S6 by steel alloy (having relatively poor mechanical properties) and titanium alloy in the Ansys software. Titanium alloy connecting rod gave better results than steel alloy in terms of static mechanical and fatigue life properties. Lattice structures in different designs are used to reduce weight and increase strength in connecting rods. Rosso et al. [29] tested the lattice structure on whole body of the connecting rod. As a result, they showed that it was possible to use of lattice design in connecting rod for some applications. Shanmugasundar et al. [30] designed a different structure by drilling five circular holes on the body of a connecting rod. According to the results, a weight reduction of about 3.5% took place in connecting rod and there was no deterioration in mechanical properties.

The main purpose of this study is to reduce the weight of connecting rods by using different lattice designs and to determine numerically the effect of these designs on the mechanical properties of the connecting rod. Although there are limited (a few) studies on the use and analysis of 3D lattice designs in connecting rods, no studies have been found in the literature on the use and analysis of 2.5D lattice designs in connecting rods. The reference connecting rod design belonged to the Antor 3 LD 510 single cylinder diesel engine. Four different 2.5D lattice designs, hexagonal, octagonal, square and triangular, were made in this reference
connecting rod body. The dimension of the lattice designs was 10x10x12 mm with the wall thickness of 1.5 mm. In the analysis, three different materials were used, namely AISI 4140, Inconel 718 and Ti6Al4V, as they were used both as a connecting rod in some engines and were suitable for production by additive manufacturing methods. The load on the connecting rod was obtained from the experimental setup to which the Antor 3 LD 510 engine was connected. As a result, it was understood that lattice designs could be used in connecting rods and these designs significantly reduced the weight. There has been no such a study in the literature, and we think that this study will be an important resource for both vehicle manufacturers and researchers working on this subject.

2. Material and Methods

2.1. Lattice designs for connecting rod

In this study, five different lattice designs were made to reduce the weight of an internal combustion engine connecting rod (Figure 1). The connecting rods having these designs are suitable for production with additive manufacturing methods. Firstly, the original connecting rod were drawn by using SolidWorks 2017 software. As seen in Figure 2 (a), the connecting rod model was consisted from 4 main parts: rod (part no 1), cup (part no 2), bolts (part no 3) and nuts (part no 4). The geometry of the original connecting rod was compatible with the geometry of the connecting rod used in the Antor 3 LD 510 single-cylinder diesel engine, where we obtained the maximum in-cylinder pressure values for use in this study. The length, rod thickness, small end diameter and big end diameter of the original connecting rod were 145 mm, 12 mm, 15 mm and 48 mm, respectively. The 2.5D lattice designs in the connecting rod were carried out by Creo Parametric 7.02 Software (Student Edition). As can be seen in Figure 1, the dimension of all lattices was 10x10x12 mm with the wall thickness of 1.5 mm.

2.2. Meshing and material properties

The designed connecting rods were exported to ANSYS 2020 software. In order to obtain more accurate results in the finite element numerical modeling study of connecting rods, the average mesh quality value was provided to be 0.8 and above. To achieve this, tetrahedron meshes have been used. The connecting rod was divided into three parts and different mesh methods were applied. Since each design has lattices with different geometries, the mesh quality was adjusted with the "patch independent" option in the body of connecting rod and
cap. The meshing conditions and results were given in Table 1. The finite element model of the connecting rod after meshing process was presented in Figure 2 (b). In this study, three different materials were selected as connecting rod material and designs were analyzed by using these materials. First material was AISI 4140 steel. In that, this material is currently used as the connecting rod material of many internal combustion engines. In addition, this was the material used in the connecting rod of the Antor 3 LD 510 model single cylinder diesel engine from which we obtained the in-cylinder pressure values (Figure 3 (a)). Second material for connecting rod was selected as Inconel 718 alloy. Inconel alloys had high strength properties and resistance at elevated temperatures. This feature is especially important for aerospace applications. Also, the Inconel 718 material has not been studied as the connecting rod material to date, and for the first time, analyzes were made in this study. Ti6Al4V alloy was chosen as third material for connecting rod. Because the density of this material is low (4.41 g/cm³) and it is used as a connecting rod material in the internal combustion engines of automotive and aircraft industry.

In this study, since the connecting rods having lattice design were suitable to be produced by the additive manufacturing methods, the mechanical properties of the materials were used which produced by the additive manufacturing methods in the literature. Damon et al. [31] produced the low alloy AISI 4140 steel by using selected laser melting technique. They achieved maximum young’s modulus, yield strength, ultimate strength and ductility of 208 GPa, 1290 MPa, 1325 MPa and 7.2 %, respectively. In another study carried out by Wang et al. [32] young’s modulus, yield strength, ultimate tensile strength and ductility of INCONEL 718 (IN718) alloy produced by selective laser melting + heat treatment process were 201 GPa, 1161 MPa, 1358 MPa and 22%, respectively. In the case of production of Ti6Al4V by selective laser melting + heat treatment process, maximum young’s modulus, yield strength, ultimate tensile strength and ductility were measured as 115.5 GPa, 1118 MPa, 1223 MPa and 14.06 %, respectively [33].

2.3. Load on connecting rod

The load acting on the connecting rod is the most important factor determining the mechanical behavior of the connecting rod [34]. Moreover, since determining the actual load affecting the connecting rod in the numerical modelling studies is essential to obtain accurate results. In this study, an experimental setup having Antor 3 LD 510 model direct injection diesel engine was used to obtain accurate load on the designed connecting rod. This test
engine is used for research purposes in many studies today [35,36]. This single cylinder, four stroke and direct injection diesel engine was controlled by electrically dynamometer and to create harsh conditions on the connecting rod the experiments were fulfilled at full load and engine speeds in between 1200 - 2000 rpm. The cylinder volume, bore x stroke, compression ratio, maximum power and maximum torque of the engine were 510 cm$^3$, 85x90 mm, 17.5:1, 8.8@3000 kW and 32.8@2000 Nm, respectively. The schematically explanation of the engine experimental setup were given in Figure 3 (a).

As seen in Figure 3 (b) which is graph of pressure change versus the crankshaft angle at full loads and different engine speeds, maximum cylinder pressure was obtained as 92.31 bar at the engine speed of 1200 rpm. This gas pressure value was converted into newton units using Equation 2.2. In order to calculating the net force ($F$) acting on connecting rod (Equation 2.1); force due to gas pressure (Equation 2.2), force due to inertia of the connecting rod and reciprocating mass (Equation 2.3), and force due to friction of the piston rings and of the piston (Equation 2.4) were calculated. As the weight of the connecting rod changes in each material and design, the inertia force also changes. Therefore, the maximum net force acting on the connecting rod was calculated for each material and design, and they were given in Table 2.

$$F = F_{gas} + F_{inertia} - F_{friction}$$  \hspace{1cm} \text{(Equation 2.1)}

$$F_{gas} = \frac{\pi \ast d^2}{4} \ast P_e$$  \hspace{1cm} \text{(Equation 2.2)}

Here, $P_e$ and $d$ were explosion pressure and bore diameter, respectively.

$$F_{inertia} = M * \omega^2 * r * \left(\cos \theta + \frac{r \ast \cos \theta}{l}\right)$$  \hspace{1cm} \text{(Equation 2.3)}

$M$ was mass of \textit{(piston and rings + piston pin + $\frac{1}{3}$ rd of the connecting rod)}, $\omega$ was angular speed (rad/s), $r$ was crank radius (mm), $l$ is length of the connecting rod (mm) and $\Theta$ was crank angle.

$$F_{friction} = h \ast \pi \ast d \ast i \ast P_r \ast \mu$$  \hspace{1cm} \text{(Equation 2.4)}

Here, $h$ was axial width of rings, $i$ was number of seals, $P_r$ was pressure of seals and $\mu$ was coefficient of friction.

\textbf{2.4. Boundary conditions and analysis}
As seen in Figure 2 (c), to perform numerical analysis of designed connecting rods, they were fixed from the big end bearing surfaces. On the other hand, the forces generated during operation of engine are effected by weight of connecting rod, the maximum net compression forces given in Table 2 were applied from the small end bearing surface of the connecting rod. In numerical analysis the bonded contact was used for all parts of the connecting rod. During the engine operation, the ambient temperature of the connecting rod was determined to be 95 °C, so the ambient temperature was set to 95 °C during the analysis. By applying these boundary conditions, it was aimed to simulate the effects of different materials and designs on mechanical properties such as stress, deformation, strain and fatigue that occur in the connecting rod during the actual operation of the engine. Firstly, the weight savings achieved by the different designs were evaluated. Then, nonlinear static structural and fatigue analyzes were carried out in ANSYS software. At the maximum loads applied, maximum deformation amounts (mm), maximum stresses (equivalent von-misses (MPa)), fatigue lives, safety factors and damage conditions according to the Goodman stress theory and fully reversed loading conditions were evaluated. In fact, thermal fatigue is an important factor for metallic materials operating at high temperatures [37]. However, the operating temperature of the connecting rods in internal combustion engines is around 95°C, and the temperature difference between regions is not very high. In addition, the connecting rods are not exposed to a cycling thermal effect during operation. Therefore, thermal fatigue was not considered for connecting rods with lattice designs in this study.

3. Results and discussions

3.1. The effect of connecting rod designs on weight reduction

By making lattice designs in the body of connecting rod, the amount of material had been saved and more importantly, a significant weight reduction was achieved in connecting rod. The values in grams of the change in weights according to designs and materials were given in Table 2. In addition, the percentage change in weight of the designs relative to the reference material (without lattice structure) was shown in Figure 4. As can be understood, the most weight reduction (15.75 %) was in the “square” design. Furthermore, the least weight loss (11.55 %) was in the “triangle” design. The reason for this, in the given dimensions (Figure 1), the tightest pecked structure was formed in the triangular design. Based on this, it is possible to say that lattice designs provided a significant weight loss in the connecting rod. Considering that there are four connecting rods in the engine of a normal passenger vehicle,
there will be serious savings in terms of fuel economy. Moreover, there was a material saving at the same rate as the weight reduction percentage. However, the effect of these weight reductions on the mechanical properties should also be taken into consideration (see section 3.3). Because as a natural consequence of weight reduction by creating lattice designs on the connecting rod body, it is very likely that the mechanical properties will be weakened due to the locally reduced cross-sectional area of the connecting rod body. At this point, the maximum equivalent stress value that occurs in the connecting rod under the maximum load transmitted from the piston to the connecting rod must be lower than the yield value of the material of the connecting rod. It is also necessary to achieve certain minimum values for the stress factor of safety and fatigue life discussed in sections 3.2 and 3.3.

3.2. Static Structural Analysis

The maximum stress and deformation values obtained under maximum load in the static-structural modeling of the connecting rods were analyzed. As seen in Figure 5 (a), equivalent (Von-Mises) stress values for each design were close to each other. This was due to the close mechanical properties of 4140, IN718 and Ti6Al4V. As expected, the minimum tensile values chancing between 271.07 and 273.15 MPa belonged to the reference designs. The maximum stress values in connecting rods having lattice design of hexagonal, octagonal and square and they varied between 511.1 and 563.06 MPa. However, connecting rods with triangular lattice design had lower maximum tensile values (ranging from 414.73 to 420.68 MPa). Because in the triangle design, the applied load was transferred equally along the sides of the triangle to the non-lattice-formed part of the connecting rod. However, in other designs, the applied load was more difficult to transmit due to the vertical or wide-angle joints of the lattice walls. The maximum equivalent Von-Mises stress value must be smaller than the yield strength (σ₀.2) value of that material in order to avoid permanent deformation in the material. In other words, for static conditions, the value of safety factor (the ratio of the yield strength of the material to the stress value measured in design) must be greater than 1. When examined from the literature [31–33], it was seen that the yield strength values of AISI 4140, IN718 and Ti6Al4V (1540, 1161 and 1118 MPa, respectively) materials produced by additive manufacturing methods were higher than the maximum Von-Mises stress values obtained. As seen in Table 3, safety factor of all lattice designs for static loading conditions were between 2.02 and 3.71. These values were higher than the static safety factor determined by Pan and Yang as 1.77 for the standard connecting rod made of T6-7075 material [38]. Thus, it was concluded that
connecting rods having designed lattice structures can remain rigid without permanent deformation at the maximum load which was obtained from the combustion chamber pressure of the engine, and safe to use as connecting rod. However, it was very important to determine how these designs would respond mechanically to cyclic load application and how many cycles their lifetime would be under this cyclic load. These facts related to fatigue were discussed in section 3.3. In Figure 5 (b-f), the stress distributions of the connecting rod materials from which the maximum Von-Mises stress was obtained for each design were given. As seen in Figure 5 (b), maximum stress occurred in the piston pin region of the reference connecting rod. However, in the lattice structured designs, the maximum stresses were in the lattice structure regions and they were especially concentrated in the joining corners of the lattice walls (see focused figures on Figure 5 (b-f)). Because the cross-sectional area of the lattice walls was narrower than the cross-sectional area of the main structure, and also, the direction of the stress flowing linearly along the lattice walls had to suddenly change at the joining corners. Therefore, higher stress concentrations occurred in these regions. It may be possible to reduce the stresses in these areas by chamfering the joining corners of the lattice walls. In this study, the stresses occurring on the designed connecting rods were not higher than the yield strength of the materials, and so there was no need to make optimizations such as chamfering in the designs.

Additionally, as seen in Figure 6 (a), deformation value of the designed connecting rods changed between 0.08853 and 0.33919 mm. As expected, the amount of deformations was minimal in the reference connecting rods without lattice structure. When evaluated in terms of material, it was seen that the maximum deformations occurred in designs with Ti6Al4V material. Accordingly, young's modulus value of Ti6Al4V material produced by additive manufacturing method (115.5 GPa) was lower than 4140 (210 GPa) and IN718 (201 GPa) [31–33]. Therefore, deformation values of AISI 4140 and IN718 were approximately close to each other for the each connecting rod design. The deformation distributions in the connecting rod for Ti6Al4V material, which has the hexagonal lattice design where the highest deformation (0.33919 mm) occurred were given in Figure 6 (b). Because in the connecting rod having hexagonal design, the hexagonal geometry behaved like a spring and was deformed more elastically. As it is understood, the maximum deformation occurred at the end of the pin associated with the piston. In all other designs, since the area in the pressure distribution was very small, the maximum deformation was in the same place.
3.3. Fatigue analysis

Under the action of static conditions, the safety factors of the connecting rods having lattice designs were above 2 which is more than 1, which meets the static requirement. However, while the engine is operating, the piston does continuously reciprocating movement, and a cycling stress occurs on the connecting rod. Damage takes place as a result of long-term periodic repetition of this stress, which does not cause damage to the connecting rod under static conditions. This phenomenon is called as fatigue. In the fatigue analysis performed in this study, Goodman equation was used to determine the cycling life of connecting rod designs and to solve the fatigue safety factor. As seen in Table 3, the cycling fatigue life of all of connecting rods were above $1 \times 10^9$, except for hexagonal, octagonal and square designs made of Ti6Al4V material. From this point of view, it was possible to say that all connecting rods, except hexagonal, octagonal and square designs made of Ti6Al4V material, could operate for at least 13,889 hours at 1200 rpm where the maximum in-cylinder pressure was achieved at maximum load. In the assessment of the fatigue life of the connecting rod made under full load and maximum engine pressure conditions, it must withstand at least $10 \times 10^6$ cycles [39]. Therefore, designs with a cycling fatigue life of more than $10 \times 10^6$ could be said to have infinite lifetimes. In the present study, it was determined that the cycling fatigue lives of hexagonal, octagonal and square designed Ti6Al4V connecting rods were $5.84 \times 10^6$, $51.8 \times 10^6$ and $15.6 \times 10^6$, respectively. Thus, these connecting rods could work up to 81.1, 719.4 and 216.7 hours, respectively, at full load and 1200 rpm.

When the fatigue analysis results were evaluated in terms of the cycling safety factor, it was seen that the reference connecting rods had the highest minimum cycling safety factor values varying between 1.77 and 2.37. It was understood that the minimum cycling safety factor value was between 1.14 and 1.56 in all designs except the connecting rod with Ti6Al4V material, which has hexagonal, octagonal and square designs. Therefore, it had been understood that these designs with a minimum cycling safety factor more than “1” have no problem in their reliability in terms of fatigue. Because the cycling safety factor for connecting rods had to be between “1” and “2”. Otherwise, a very high safety factor was known to lead to negative consequences such as weight, excessive material use, and design difficulties. When the cycling safety factor values of the connecting rods with Ti6Al4V material having hexagonal, octagonal and square designs were examined, it was seen that they remained below “1” in accordance with their cycling fatigue life values. This was due to the relatively low fatigue strength of the Ti6Al4V material. Especially for this material, risk of
failure was very high in the hexagonal design. In Figure 7 (a-c), safety factor distributions of connecting rod designs having the lowest fatigue life were given. As expected, the regions with the minimum cycling safety factor were where the static stresses discussed in Section 3.2 were most intense. This situation was the same in all other designs (as seen in Figure 7 (d)). The overall cycling safety factor of the connecting rod is likely to be improved by increasing the thickness of the lattice wall or chamfering at the joining corners of the lattice walls. However, the negative effect of these procedures on weight should also be taken into account. Because increasing the thickness and chamfering at the joining corners of the lattice walls will mean more material input, these procedures will cause an increase in the weight of the connecting rod having lattice designs. As a general result, it can be said that triangle designed connecting rods meet sufficient mechanical requirements in all materials.

4. Conclusions

In this study, different lattice designs were created on the connecting rod of an internal combustion engine. The main purpose of the study was to reduce the weight of the connecting rods without causing a significant deterioration in mechanical properties. For this purpose, four different lattice designs, hexagonal, octagonal, square and triangular, were made on the connecting rods and analyzed with three different materials (AISI 4140, Inconel 718 and Ti6Al4V). The load on the designed connecting rod was experimentally obtained from the engine test setup. As a result of the analysis made with the finite element approach, the following results have been obtained.

1. It is possible to obtain a significant reduction in weight by making lattice designs on the connecting rod body. By using square, hexagonal, octagonal and triangular cage designs, weight reduction was achieved by 15.75%, 14.82%, 14.48% and 11.55%, respectively.

2. The maximum stress values of the connecting rods having lattice designs varied between 414.73 and 563.06 MPa, and these values were below the yield stress value of the materials. Therefore, the safety factors for static conditions varied between 2.02 and 3.71. The highest safety factor belonged to the triangular design and the lowest safety factor belonged to the hexagonal design.
3. Deformation value of the designed connecting rods changed between 0.08853 and 0.33919 mm. The maximum deformation occurred at the Ti6Al4V material having hexagonal lattice design.

4. Except for hexagonal, octagonal and square designs made of Ti6Al4V material, the cycling fatigue life and cycling safety factor of all of connecting rods were above $1 \times 10^9$ cycles and 1, respectively.

5. In general, it can be said that triangular designed connecting rods meet sufficient mechanical requirements in all materials. If AISI 4140 or Inconel 718 is chosen as the connecting rod material, all designs can be used safely.

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Figure Captions

Figure 1. Connecting rod and lattice designs.

Figure 2. Main parts of the connecting rod (a), finite element model of the connecting rod (b) and definition of mechanical loads (c).

Figure 3. (a) Schematically explanation of engine setup to obtain load on connecting rod: 1) air filter, 2) fuel tank, 3) engine, 4) dynamometer and 5) control panel. (b) Cylinder pressure graphs at full load and different engine speeds.

Figure 4. The percentage change in weight of the designs relative to the reference design of the connecting rod.

Figure 5. Static analysis results of connecting rods having different designs and materials. (a) Column graph showing maximum Von-Mises stress. Stress distributions on (b) Reference-4140, (c) Hexagonal-4140, (d) Octagonal-4140, (e) Square-4140 and (f) Triangle-IN718.

Figure 6. (a) maximum deformation graph and (b) deformation distributions on the hexagonal 4140 connecting rod.

Figure 7. Cycling safety factor distributions in connecting rods having lowest fatigue life cycle (a) Hexagonal-Ti6Al4V, (b) Octagonal-Ti6Al4V, (c) Square-Ti6Al4V and (d) Triangle-Ti6Al4V.
Table Captions

Table 1. Meshing parameters of finite element

Table 2. Connecting rod materials and mechanical properties belonging these materials produced by additive manufacturing process, weights of different designs, maximum net forces acting on connecting rod for each design.

Table 3. Safety factors for static conditions and fatigue conditions, and fatigue life.
Figure 1.
Figure 3.
Figure 4.

![Bar chart showing weight reduction and material savings for different designs.](chart)

- **Hexagonal**: 14.82%
- **Octagonal**: 14.48%
- **Square**: 15.75%
- **Triangle**: 11.55%
Figure 5.
Figure 6.
### Table 1.

| Design   | Part No | Algorithm      | Method               | Element Size (mm) | Nodes  | Elements  |
|----------|---------|----------------|----------------------|-------------------|--------|-----------|
| Reference| 1       | Patch Independent| Tetrahedrons         | Max. 3 Min. 1.5    | 163340 | 110520    |
| Hexagonal|         |                |                      | Max. 4 Min. 2     | 164722 | 107142    |
| Octagonal|         |                |                      | Max. 4.5 Min. 2   | 193566 | 127187    |
| Square   |         |                |                      | Max. 3.5 Min. 2   | 190770 | 126227    |
| Triangle |         |                |                      | Max. 3 Min. 2     | 202560 | 132219    |
| For all designs | 2 | Automatic | Tetrahedrons+Hexahedrons | Max. 5 Min. 3 | 21559  | 13765     |
| For all designs | 3, 4 | Automatic | Tetrahedrons+Hexahedrons | Max. 5 Min. 3 | 35377  | 9233      |
Table 2.

| Material | Design  | Weight (g) | Max. Force (N) | Young’s Modulus (GPa) | Yield Strength (MPa) | UTS (MPa) | Poisson Ratio |
|----------|---------|------------|----------------|-----------------------|----------------------|-----------|--------------|
| AISI 4140| Reference | 906.92     | 210            | 53077                 | 1540                 | 2073      | 0.29         |
|          | Hexagonal| 772.48     |                |                       |                      |           |              |
|          | Octagonal| 775.62     |                |                       |                      |           |              |
|          | Square   | 764.11     |                |                       |                      |           |              |
|          | Triangle | 802.14     |                |                       |                      |           |              |
| Inconel 718| Reference | 949.56     | 201            | 53087                 | 1161                 | 1358      | 0.3          |
|          | Hexagonal| 808.80     |                |                       |                      |           |              |
|          | Octagonal| 812.08     |                |                       |                      |           |              |
|          | Square   | 800.04     |                |                       |                      |           |              |
|          | Triangle | 839.85     |                |                       |                      |           |              |
| Ti6Al4V  | Reference | 508.88     |                |                       |                      |           |              |
|          | Hexagonal| 433.45     |                |                       |                      |           |              |
|          | Octagonal| 435.21     |                |                       |                      |           |              |
|          | Square   | 428.75     |                |                       |                      |           |              |
|          | Triangle | 450.09     |                |                       |                      |           |              |
Table 3.

| Material   | Design     | Safety factor for static conditions | Fatigue life (cycles $\times 10^6$) | Safety factor for fatigue conditions |
|------------|------------|-------------------------------------|-------------------------------------|-------------------------------------|
| **AISI 4140** | Reference  | 5.64                                | $> 1000$                            | 2.37                                |
|            | Hexagonal  | 2.74                                | $> 1000$                            | 1.15                                |
|            | Octagonal  | 2.96                                | $> 1000$                            | 1.24                                |
|            | Square     | 2.89                                | $> 1000$                            | 1.21                                |
|            | Triangle   | 3.71                                | $> 1000$                            | 1.56                                |
| **Inconel 718** | Reference  | 4.26                                | $> 1000$                            | 2.27                                |
|            | Hexagonal  | 2.07                                | $> 1000$                            | 1.11                                |
|            | Octagonal  | 2.24                                | $> 1000$                            | 1.20                                |
|            | Square     | 2.18                                | $> 1000$                            | 1.16                                |
|            | Triangle   | 2.76                                | $> 1000$                            | 1.47                                |
| **Ti6Al4V**  | Reference  | 4.12                                | $> 1000$                            | 1.77                                |
|            | Hexagonal  | 2.02 $= 5.84$                       |                                    | 0.87                                |
|            | Octagonal  | 2.19 $= 51.8$                       |                                    | 0.94                                |
|            | Square     | 2.10 $= 15.6$                       |                                    | 0.90                                |
|            | Triangle   | 2.66                                | $> 1000$                            | 1.14                                |
Biographies

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Mustafa Guven Gok graduated from Firat University department of Metallurgical Education with a bachelor and master of science degrees in 2008 and 2010, respectively. After, he graduated from Istanbul Technical University department of Metallurgical and Materials Engineering in 2015 with Ph.D. degree. Dr. Gok joined to the Materials Science and Engineering Department of Hakkari University in 2015 and he has been working in Hakkari University since 2016. His study fields include: Plasma Spray Coating and Spark Plasma Sintering (SPS) Processes, Thermal Barrier Coatings, Self-Healing Ceramics, Biomaterials, Finite Element Analysis and Materials of Internal Combustion Engine.

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Omer Cihan born in 1986, studied Mechanical Training. He completed his MSc degrees in department of Mechanical Training from Firat University, Elazığ, Turkey, in 2011. Then, He completed his PhD degrees in Mechanical Engineering from Istanbul Technical University, Istanbul, Turkey, in 2017. He has been worked as a research assistant at University of Istanbul Technical, Mechanical Engineering Department, Turkey, from 2010 to 2017. He is currently working at Hakkari University, Turkey, as Assist. Professor. The primary topics of his scientific work are such as Engine materials, Friction, Wear, Engine testing, Engine
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