

RESEARCH ARTICLE

The impact of bioethanol fuel on the fatigue life of the connecting rod using dynamical analysis

Hojjat Ashouri

Department of Mechanic, VaP.C., Islamic Azad University, Varamin, Iran

Abstract - The complex loading the connecting rod tolerates mandates an adequate solidity to withstand the applied forces. Therefore, it is necessary to accurately evaluate the stresses applied to the connecting rod and predict its fatigue life in the engine's actual operational conditions. The present study aimed to predict the high-cycle fatigue (HCF) life of the connecting rod by changing the fuel from gasoline to bioethanol using the results of the connecting rod dynamic analysis. For this purpose, the stress was predicted using finite element analysis (FEA) and ANSYS software. The nCode Design Life software was applied, along with the Goodman criterion, to estimate HCF life. The mechanical analysis results showed that using bioethanol instead of gasoline would increase the connecting rod stress by about 3.46%. According to the HCF life analysis, the minimum fracture cycles of the connecting rod in gasoline and bioethanol states were 2.744×10^9 and 2.356×10^9 cycles, respectively, observable in the area between the shank and the small end. Therefore, the HCF life of the connecting rod in the bioethanol state would be about 0.388×10^9 cycles, or 14.14% reduced. As shown by the HCF safety factor, the connecting rod had no critical area. 800-hour durability test showed no rupture in any part of the connecting rod. According to the 800-h durability test, the connecting rod will not rupture in any sections.

Article History

Received : 29 August 2025

Revised : 22 June 202

Accepted : 29 July 2026

Published : 25 September 2026

Keywords

Bioethanol effect

Finite element analysis

High cycle fatigue life

Connecting rod

1. Introduction

The connecting rod connects the piston (reciprocating motion) to the crankshaft (rotational motion). Since the connecting rod is subjected to axial tensile and compressive forces due to inertia and combustion, as well as the bending moment resulting from rotational inertia, this element should be designed to be highly reliable. While this component should be strong enough to withstand the applied forces, it should also be light enough to reduce the inertial force due to mass [1-3]. In recent years, the growing focus on improving engine efficiency and reducing fuel consumption has imposed greater thermal and mechanical stresses on engine components, presenting a new challenge for engine designers. Such conditions make the analysis and optimization of the connecting rod crucial, and require precise examinations of the stresses applied to the connecting rod and its fatigue life in the engine's real operational conditions [4], [5]. Depletion of fossil fuel resources, along with the expansion of their environmental footprint and increasing demands, has shifted researchers toward alternative approaches to reduce engine emissions, such as substituting fossil fuels with bioenergy ones. In this regard, alcohols are considered one of the best options to replace (or be combined with) gasoline [6], [7]. Ethanol fuel is one of the most favorable alcohol fuels, used as an additive to gasoline. Globally, the most prevalent mixture of ethanol and gasoline consists of 10% (or less) alcohol and 90% gasoline. Such a mixture, known as gasohol, can be used in lightweight vehicles, with no specific necessary modification to the engine and fueling system [8], [9]. The bioethanol condition produces higher in-cylinder pressure than gasoline due to its oxygenated molecular structure and higher flame speed, which enhance combustion [7], [10]. Consequently, the connecting rod experiences greater loading and stress, leading to a reduction in fatigue life.

Various studies have been conducted to analyze, predict, and optimize the stress and fatigue life of the connecting rod. For instance, Alam and colleagues analyzed the fatigue failure of a diesel engine connecting rod, indicating that improper machining and drilling of the oil hole can cause tensile residual stress, ultimately leading to fatigue failure [2]. Moreover, Nuthalapati and Malkapuram's study on composite application in the production of connecting rods showed that composite connecting rods can be considerable substitutes for aluminum connecting rods due to their lighter weights. However, composite connecting rods are not competitive with forged steel ones [11]. In a study on the high-cycle fatigue analysis and optimization of the connecting rod, Silva and Bittencourt reported a reduced mass of the connecting rod (by about 3%) as the result of its optimization [12]. Using high-strength steel, Kaiho and colleagues increased the connecting rod's strength by 10%. Notably, their resulting connecting rod was 8% lighter than the primary one [1]. Sathish and colleagues analyzed and modeled the connecting rod made from various aluminum alloys using the finite element method (FEM). Their findings suggest lower stress tolerance in the connecting rod made of AA2014 aluminum alloy [13]. According to the results of Witeka and Zelekb's study on stress analysis and stall of the connecting rod in a turbocharged diesel engine, the connecting rod failure is predominantly caused by high stress due to the bolts' preload [14]. By predicting the connecting rod's high-cycle fatigue life, Kolayath and colleagues reported the minimum fatigue life between the shank and the small end [3]. Interestingly, Gandhul and Shriwastava documented a new prediction method for connecting rod fatigue life using Goodman's fatigue criteria [15]. Moreover, by analyzing the connecting rod fatigue stresses, Mohammed Mohsin and Mohamed observed the minimum fatigue life in its shank area [16]. Using Ansys

software, Godara and colleagues studied the design and analysis of the connecting rod, indicating the area between the shank and small eye as a critical area, where the maximum von Mises stress occurs [17]. In a mechanical analysis and topological optimization performed by Bhamdare and colleagues, the optimization of the connecting rod decreased its weight approximately 11% [18]. Another approach, presented by Xu and colleagues, to predict the connecting rod fatigue life by using the Palmgren-Miner cumulative damage model, predicted the minimum fatigue life near the small end (at its joint with the shank). As their findings suggest, the variation and reduction of the connecting rod thickness caused critical points in such an area [19]. Another study by Jia et al. evaluated the impact of forging marks on the strength of connecting rods using FEM. Based on the results, such marks should be avoided in critical areas of the connecting rod due to their adverse effect on the strength [20]. The study on structural optimization of a diesel engine's connecting rod by Muhammad et al. demonstrated an increase in high-cycle fatigue safety factor by approximately 19% [21]. Rakić and colleagues analyzed the diesel engine connecting rod fracture, reporting insufficient machining, lack of finishing, and maximum stress in the failure zone as the causes of the connecting rod fracture [22]. Baldini and colleagues developed a new analytical method to evaluate stress in the connecting rod cap, validated by FEM [23]. The connecting rod fatigue life was predicted by Savaripour and colleagues, using the outcomes of combustion tests, in which the minimum safety factor of high-cycle fatigue was calculated as 1.586 in the critical area between the shank and small end [24]. The mechanical analysis of the connecting rod by changing its material was performed by Seralathan et al., suggesting that the A356-5%SiC-based connecting rod tolerates lower stresses than other materials [25].

The topological optimization and analysis of the connecting rod by Shanmugasundar and colleagues showed the area between the shank and small eye as critical, with the maximum stress [26]. Also, Rao et al. predicted fatigue life by altering the engine's compression and varying the connecting rod's material. The results of fatigue analysis demonstrated that the fatigue life of the titanium alloy-based connecting rod is approximately 100 and 10 times that of the forged steel and the aluminum alloy, respectively [27]. Honarvar and colleagues studied the impact of shot blasting on the connecting rod fatigue life and reported that shot blasting can increase the connecting rod's fatigue life from 194000 cycles to approximately 2 million cycles [28]. Juarez and colleagues investigated the fracture of a diesel generator connecting rod used for electric energy production and found that improper machining in the lubrication channel can lead to its fracture [29]. Zhu and colleagues' fracture analysis on a reciprocating compressor connecting rod's cap and bolts demonstrated that high-cycle fatigue causes fracture of the cap and bolts [30]. In another study, Pan and Zhang optimized and predicted the fatigue life of the connecting rod in a piston engine, reporting a minimum fatigue safety factor of 0.88, which would increase to 1.32 after optimization [31]. The connecting rod fatigue life was also predicted by Zheng and colleagues and showed a minimum fatigue safety factor of 1.5 [32]. Through their optimization of the connecting rod to increase engine power (up to 17%), Ismail and colleagues found that slight alterations in connecting rod geometry can reduce the stress (up to 25.5%) [33]. Strozzi and colleagues studied different regular and irregular failures of internal combustion engine connecting rods by a stress-level approach, examining failure interpretations with traditional calculations and more advanced analytical patterns using FEM predictions [34]. In a study, Kandareegula and colleagues analyzed the stress and predicted the fatigue life of the connecting rod, showing a very good fit between experimental and simulated results [4]. Ketmuang and Wongthong examined the impact of fatigue behavior on the gasoline engine connecting rod failure, reporting fatigue as the reason for its failure [35]. The fatigue analysis of an optimized connecting rod by Cecchel and colleagues indicated that the fatigue cracks initiate from the point with maximum tensile stress. Moreover, the starting point of the fatigue failure corresponded well with the simulated results [36]. Rahmatinejad et al. employed static analysis and fatigue life prediction using FEM, evaluating a minimum fatigue life of 10^8 cycles [37]. Similarly, the evaluation of the impact of design parameters on the stresses at the small end by Elmaihy et al. revealed a reduction in stress by increasing the cylinder diameter [38]. Using FEM and experimental data, Kuldeep et al. developed a composite connecting rod with improved tensile and compressive load-bearing capabilities [39]. Furthermore, decreasing the shank thickness from 3 to 2 mm showed no structural damage [40]. Zheng et al. demonstrated that topological optimization can reduce the weight of the connecting rod and its applied stresses by approximately 15.62% [41].

Hence, while the stress analysis and fatigue life prediction of the connecting rod in gasoline and diesel-powered engines have been extensively studied, the impact of bioethanol fuel on connecting rod stress and fatigue life in bi-fuel engines has not been explored. Growing environmental concerns over fossil fuel consumption and the depletion of their reserves have intensified research into renewable and sustainable fuels. Bioethanol has emerged as a renewable fuel, particularly when blended with gasoline. Previous studies have reported improved engine performance, including brake torque, brake power, and volumetric efficiency, alongside decreased carbon monoxide and unburned hydrocarbon emissions [6], [7], [10]. However, the effects of bioethanol on stress distribution and fatigue life of engine components, especially the connecting rod, remain mainly unexplored. Nowadays, various vehicles use bioethanol fuel as a replacement for gasoline to improve engine performance and reduce emissions. Considering the mentioned benefits of bioethanol fuels over gasoline, there is a lack of precise and comprehensive investigation on the impact of bioethanol fuel on stress distribution and the fatigue life of the connecting rod in bi-fuel engines. The present investigation studied the impact of bioethanol fuel as an alternative to gasoline on the connecting rod fatigue life. In the mechanical analysis, the stresses applied to the connecting rod can be evaluated using two static and dynamic methods. While the latter leads to more correlation with the actual results, various references employed the former due to its simplicity and time-saving nature. However, according to [5] and [42], there is a significant difference between the results of static and dynamic analysis [5], [42]. This research used the dynamic method for mechanical analysis and prediction of the connecting rod fatigue life. This study then aimed to assess the effect of bioethanol gas fuel on the high-cycle fatigue life of the connecting rod using dynamic analysis results. For this purpose, the connecting rod was first modelled in SolidWorks. The boundary

conditions in mechanical analysis were extracted from the engine's one-dimensional (1D) simulation in GT-Power and MATLAB. Then, its mechanical stresses were analyzed in two gasoline and bioethanol conditions using Ansys software. Eventually, the high-cycle fatigue life and the impact of altering the fuel to bioethanol were predicted and evaluated, respectively, using Ansys nCode Design Life.

2. Materials and Methods

2.1 The Finite Element Model and Behavioral Model

Simulation methods are cost-effective and time-efficient, enabling the conduction of several tests. The FEA can be used to evaluate the effectiveness of different designs [4], [5], [13]. The connecting rod was first modelled in SolidWorks. ANSYS Workbench was used to identify the piston stress distribution. The HCF life of the connecting rod was then studied by inserting the mechanical results into the ANSYS nCode Design Life. Figure 1 displays the connecting rod used. The engine and material properties are presented in Tables 1 and 2, respectively. The piston pin and crankshaft were also modelled to increase the reliability of the mechanical analysis and high-cycle fatigue life prediction of the connecting rod. The finite element model comprised 74,984 elements and 213,392 nodes to increase accuracy. The three-dimensional Tet10 elements were used to mesh the connecting rod. This element is tetrahedral with 10 nodes.

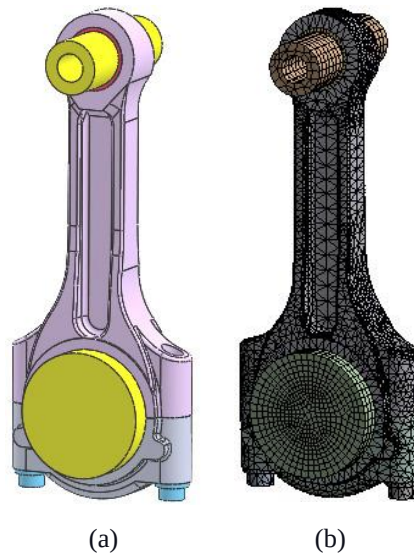


Figure 1. (a) Connecting rod assembly and (b) finite element model of the connecting rod

Table 1. The intended engine Characteristics

Parameter	Value
Cylinder radius (mm)	39.3
Crankshaft radius (mm)	42.5
Engine volume (cc)	1650
Peak torque (N-m)	155@4000rpm
Peak power (hp)	113@6000rpm
No. of valve	16
Fuels	Gasoline, Bioethanol(E10)

Table 2. Material properties of the connecting rod assembly

Part	Young's modulus (GPa)	Density (kg/m ³)	Poisson's ratio (-)
Piston pin	206	7850	0.292
Bush	115	2700	0.324
Connecting rod	210	7250	0.330
Crankshaft	221	7833	0.300

2.2 The Boundary Conditions of Mechanical Loads

The connecting rod undergoes complex motion, linking the reciprocating piston to the rotating crankshaft. Through the slider-crank mechanism, it transmits combustion-induced loads from the piston to the crankshaft [43]. The slider-crank mechanism was used to analyze the applied forces on the connecting rod, as shown in Figure 2, in which the 2, 3, and 4 components represent the crankshaft, connecting rod, and piston, respectively.

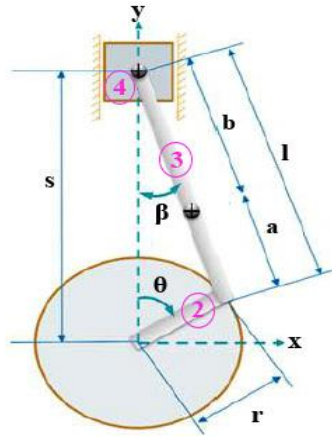


Figure 2. Slider-crank mechanism

According to geometrical relations, we have [43]:

$$s = r \cos \theta + l \cos \beta \quad (1)$$

$$l \sin \beta = r \sin \theta \quad (2)$$

Regarding Figure 2, r , l , θ , and β express the crankshaft radius, the connecting rod length, the crankshaft angle, and the connecting rod angle, respectively. The angular velocity and angular acceleration of the connecting rod can be obtained by deriving Eq. 1, according to the following equations [43]:

$$\dot{\beta} = \frac{r}{l} \frac{\cos \theta}{\cos \beta} \dot{\theta} \quad (3)$$

$$\ddot{\beta} = \frac{-r \dot{\theta}^2 \sin \theta + r \dot{\beta}^2 \sin \beta}{l \cos \beta} \quad (4)$$

By deriving Eq. 2, the piston's velocity and acceleration would be expressed as follows [43]:

$$v_p = \dot{s} = -r \dot{\theta} \sin \theta - l \dot{\beta} \sin \beta \quad (5)$$

$$a_p = \ddot{s} = -r \dot{\theta}^2 \cos \theta - l \ddot{\beta} \cos \beta - l \dot{\beta}^2 \sin \beta \quad (6)$$

The forces acting on the piston and connecting rod were determined using Equations (3)-(6), which describe their respective velocities and accelerations along with the corresponding free-body diagrams. According to such kinematic relationships, the acceleration of the connecting rod's center of mass is given by the following equation [43]:

$$\vec{a}_G = [b \ddot{\beta} \cos \beta - b \omega^2 \sin \beta] \vec{i} + [b \ddot{\beta} \sin \beta - b \omega^2 \cos \beta - a_p] \vec{j} \quad (7)$$

Regarding the piston free body diagram, shown in Figure 3(a), the piston equilibrium equations are as follows [43]:

$$F_{34x} = F_{14} \quad (8)$$

$$F_{34y} = F_{\text{gas}} + m_p a_p \quad (9)$$

where; F_{gas} shows the gas force resulting from the combustion, and m_p and a_p are the piston's mass and acceleration, respectively.

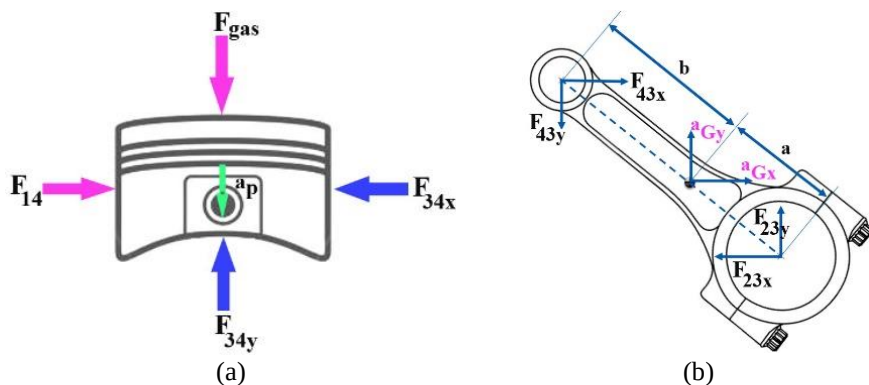


Figure 3. Free body diagram of the (a) piston and (b) connecting rod

The F_{34x} piston's lateral force can be calculated regarding the connecting rod free body diagram shown in Figure 3(b). The balance equations of the connecting rod regarding Figure 3 are as follows [43]:

$$F_{43x} = F_{23x} + m_c a_{Gx} \quad (10)$$

$$F_{23y} = F_{43y} + m_c a_{Gy} \quad (11)$$

$$F_{23x} a \cos \beta + F_{43x} b \cos \beta - F_{43y} b \sin \beta - F_{23y} a \sin \beta - I_G \ddot{\beta} = 0 \quad (12)$$

where; m_c represents the connecting rod mass, a_{Gx} and a_{Gy} respectively show the acceleration of the center of gravity in the x and y directions, I_G represents the moment of inertia, and b and a show the distance of the center of gravity from the small and big ends, respectively. The applied forces of the connecting rod were obtained using Equations (13)-(16) [43]:

$$F_{43y} = F_{34y} \quad (13)$$

$$F_{23y} = F_{43y} + m_c a_{Gy} \quad (14)$$

$$F_{43x} = \frac{-I_G \ddot{\beta} - F_{23y} a \sin \beta - F_{43y} b \sin \beta - m_c a_{Gx} \cos \beta}{a \cos \beta + b \cos \beta} \quad (15)$$

$$F_{23x} = \frac{-I_G \ddot{\beta} - F_{23y} a \sin \beta - F_{43y} b \sin \beta + m_c a_{Gx} \cos \beta}{a \cos \beta + b \cos \beta} \quad (16)$$

The forces applied to the connecting rod were calculated and applied to the connecting rod using the program written in MATLAB software. The crankshaft was constrained during the mechanical analysis, while the forces obtained from the connecting rod dynamic analysis were applied at the piston pin. Frictional contact was defined at the interfaces between the components, with a friction coefficient of 0.1.

2.3 Modelling and Simulation of the Engine

Engine was modelled and simulated using GT-Power due to its versatility and extensive library (Figure 4).

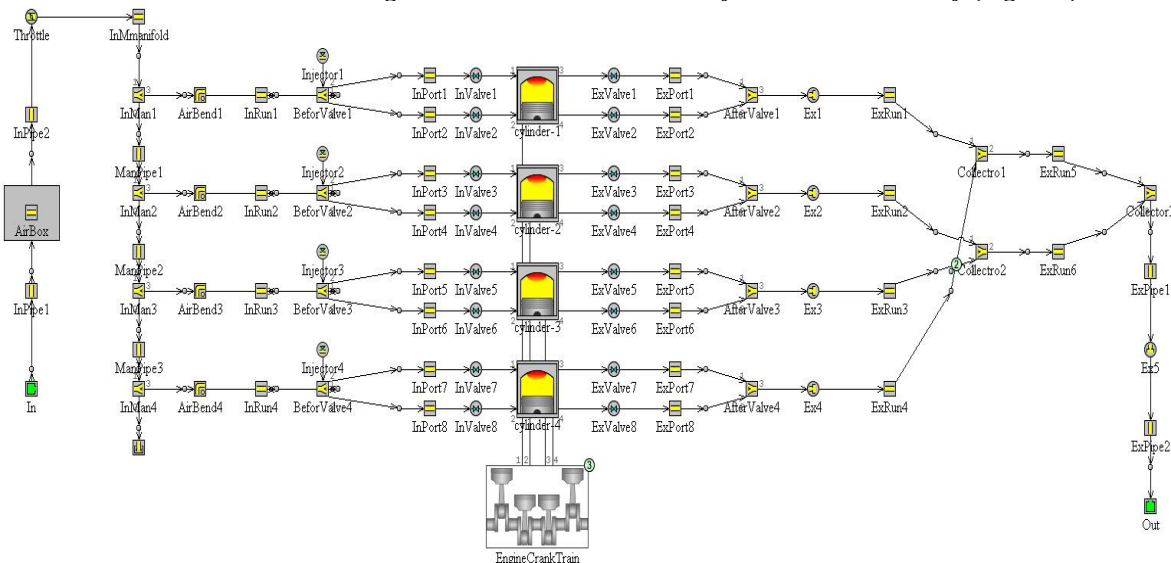


Figure 4. GT-Power model of engine

The present study applied the EngCylComSIWiebe combustion model in GT-Power, using the Wiebe function to determine the mass fraction burned at each crank angle. The in-cylinder pressure was then calculated based on the injected fuel mass and its chemical energy. To minimize discrepancies between the simulated and experimental results, the model was calibrated by comparing brake torque, brake power, and in-cylinder pressure for gasoline and bioethanol and adjusting the relevant model parameters accordingly (Table 3).

Table 3. Gasoline and bioethanol characteristics [8], [10], [44]

Fuel property	Gasoline	Bioethanol
Density at 15°C (kg/m ³)	750.8	755.4
Kinematic viscosity at 15°C (mm ² /s)	0.494	0.572
Vapor pressure at 38°C (kPa)	63.98	68.88
Latent heat of vaporization (MJ/m ³)	308	1103
Lower heating value (kJ/kg)	44000	26900
Stoichiometric air–fuel ratio	14.7	8.97

2.4 Model for HCF Life Prediction

The fatigue life is crucial due to exposure to repetitive forces and rupture susceptibility due to fatigue. The fatigue life of materials exposed to mechanical loads is low-cycle if their stress reaches the plastic region. However, their fatigue would be high-cycle if they remain under the plastic region [45]. Goodman criterion is used to estimate the HCF fatigue life [15, 31, 42]. The Goodman criterion is given by equation [45]:

$$\frac{\sigma_a}{\sigma_e} + \frac{\sigma_{\text{mean}}}{\sigma_{\text{ult}}} = \frac{1}{n} \quad (17)$$

where σ_a is the alternating stress, σ_e is the endurance limit, σ_{mean} is the stress (average), σ_{ult} is the ultimate tensile strength, and n is the safety factor. The S-N curve was obtained from the nCode Design Life material library (Figure 5). For the HCF analysis, the connecting rod was assigned an ultimate tensile strength of 950 MPa and an endurance limit of 350 MPa based on the Goodman criterion (Figure 6).

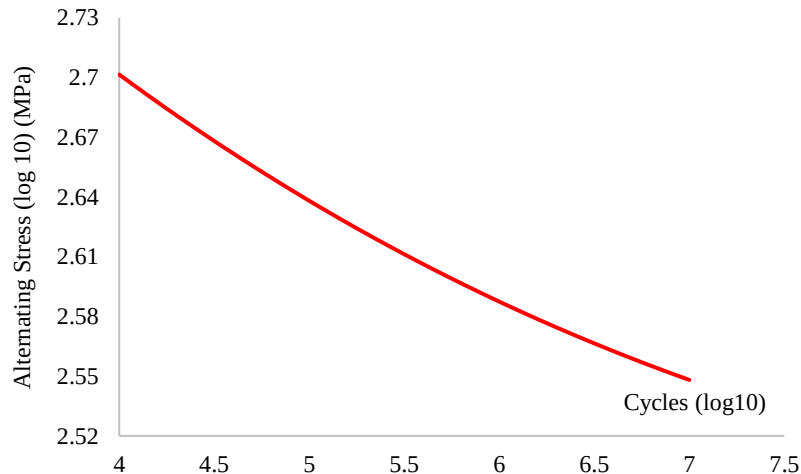


Figure 5. Connecting rod S–N curve

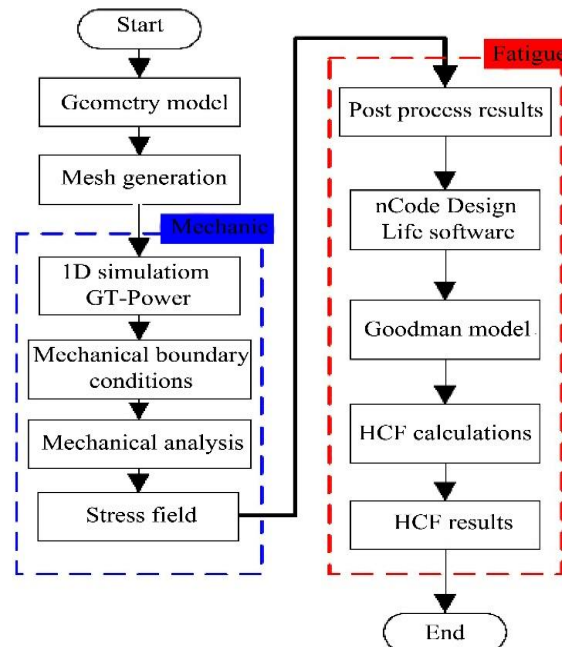


Figure 6. Connecting rod HCF analysis flowchart

3. Results and Discussion

3.1 In-Cylinder Pressure

The GT-Power was used for 1D simulation of the engine combustion process and to determine the boundary conditions of the connecting rod, which is crucial for mechanical analysis and fatigue life prediction. Figure 7 shows the pressure diagram inside the cylinder based on the crankshaft angle in gasoline and bioethanol conditions. The maximum pressure within the cylinder occurs shortly after combustion initiation, followed by a decrease in the slope of the pressure curve.

The maximum inner-cylinder pressure in the gasoline condition was 77.5 bar at a 376.5-degree crankshaft angle. In the bioethanol condition, on the other hand, the maximum pressure in the cylinder was 81.07 bar, occurred at a 375.4-degree crankshaft angle. Accordingly, the inner-cylinder pressure in bioethanol conditions would be increased by about 3.57 MPa or 4.6%. Gas pressure is a major load acting on the piston. The approximately 4.6% higher in-cylinder pressure under bioethanol operation indicates increased thermomechanical loading, which can consequently reduce the piston's HCF life.

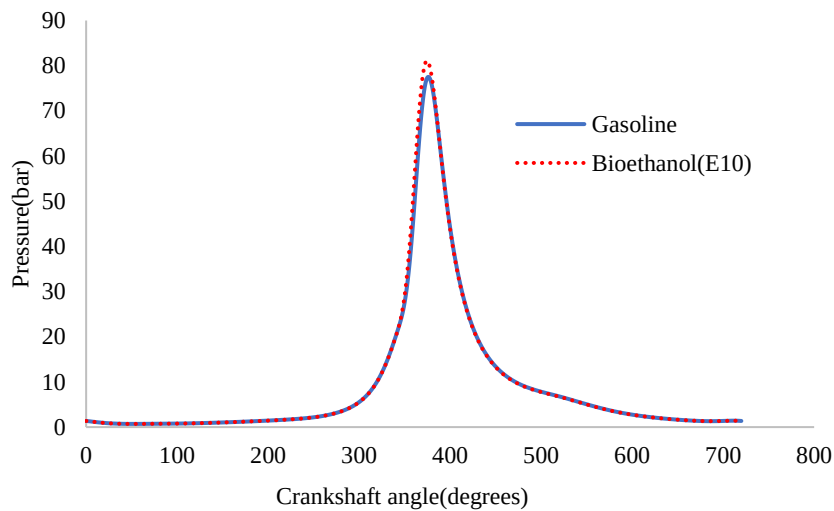


Figure 7. In-cylinder pressure versus crank angle

The higher in-cylinder pressure under bioethanol operation can be attributed to its oxygenated structure, which promotes more efficient combustion. Additionally, its higher latent heat of vaporization enhances charge cooling and combustion characteristics, while its higher octane number improves knock resistance, leading to higher peak cylinder pressures. The higher flame speed of bioethanol further contributes to increase in-cylinder pressure. Therefore, the connecting rod is subjected to greater forces and stresses under bioethanol operation [6, 7], [46], [47].

3.2 Mechanical Forces Applied on the Connecting Rod

Using the inner-cylinder pressure diagram and equations 1-16, a MATLAB-written program was used to analyze the forces applied to the connecting rod. Figures 8 and 9 depict the forces applied to the small and big ends in the axial and perpendicular directions to the connecting rod axis. Figures 8 and 9 show the inequality of the forces perpendicular to the connecting rod axis, which is due to the rotational acceleration of the connecting rod. On the other hand, the linear acceleration of the connecting rod center of gravity causes the difference between the axial forces at the small and big ends of the connecting rod. As seen in Figure 8, the maximum compressive forces applied to the connecting rod are 28321.1 and 29961 N in gasoline and bioethanol conditions, respectively, which are typically considered as the compressive load criterion in the design. The normal force diagrams, on the other hand, are alike at both conditions, with a maximum force of 1688.58 N. The maximum axial force in gasoline and bioethanol conditions is 19265.6 and 20874.9 N, respectively (Figure 9).

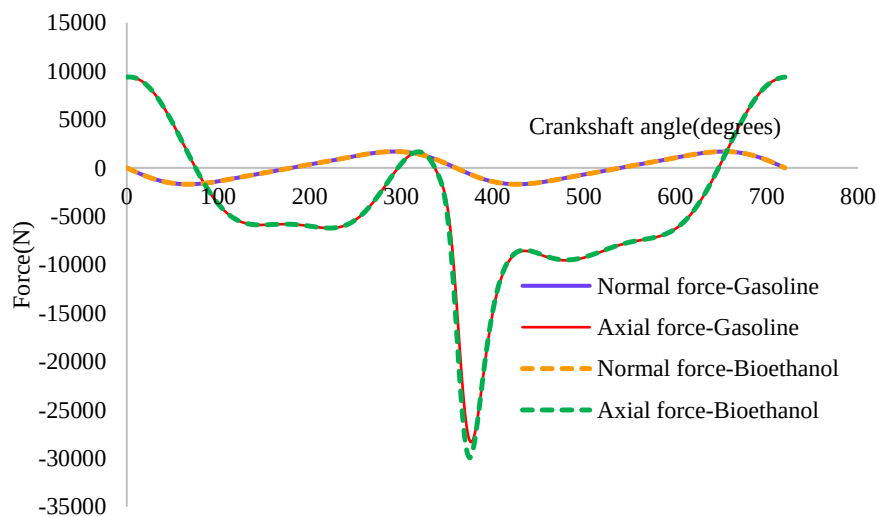


Figure 8. Diagram of the small end of the connecting rod forces calculated by MATLAB

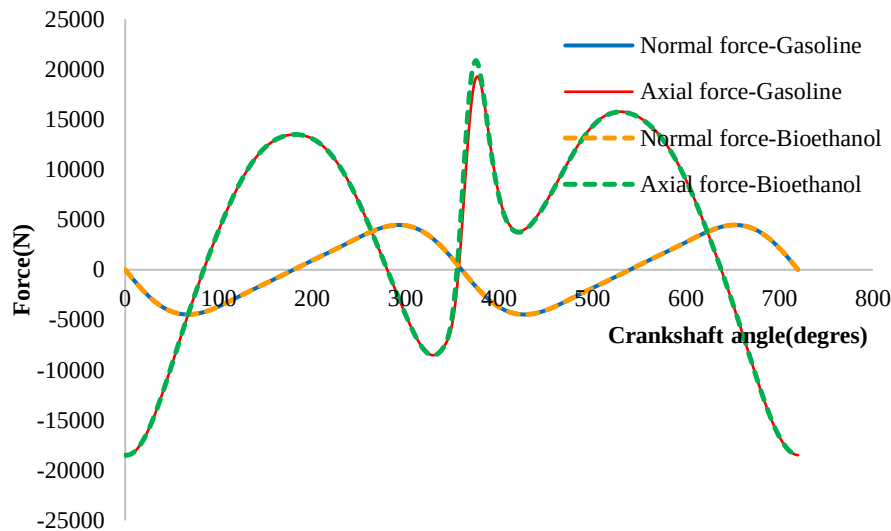


Figure 9. Diagram of the big end of the connecting rod forces calculated by MATLAB

3.3 The Mechanical Analysis using the Results of Dynamic Analysis of Forces Applied to the Connecting Rod

The applied stresses result from the bending moment and axial forces. The bending moment applied to the connecting rod results from angular acceleration, which is repeat every 360 degrees. Conversely, variations in gas pressure inside the cylinder cause a 720-degree alteration period for the axial forces. The dynamic forces applied to the connecting rod, which were calculated earlier, were used to analyze the stress. Due to the inability to directly apply dynamic forces in ANSYS (by continuous variation in their magnitude and direction), multiple analyses were performed at certain intervals using dynamic force inputs to determine the stress history at various points on the connecting rod. The results obtained from such a method are referred to as semi-dynamic analysis. The mechanical analysis was performed in two stages. First, the preload force of the connecting rod bolts was considered, and then the other forces were applied to the connecting rod. A 11900 kN preload force was assumed for the bolts.

The von-Mises stress distribution under gasoline conditions (Figure 10(a)) can be related to the angle at which the inner-cylinder gas pressure reached its maximum. At this moment, the connecting rod tolerates the highest axial force. The mechanical analysis under gasoline conditions suggests two critical areas in the region between the shank and the small and big ends of the connecting rod, with a maximum stress of 166.83 MPa in the area between the shank and the small end, aligning with [16] and [18]. The von-Mises stress distribution of the bioethanol condition is illustrated in Figure 10(b). The maximum von-Mises stress in the connecting rod in such a condition was 172.6 MPa, similar to the gasoline condition in the area between the shank and the small end. Based on the mechanical analysis approach, Figure 8 shows that the two areas between the shank and the small and big ends are critical, and the connecting rod tolerates about 5.77 MPa higher stress in the bioethanol condition. Therefore, using bioethanol instead of gasoline increases the connecting rod stress by about 3.46%. Due to the nonlinear relationship between stress and fatigue life, even a small increase in stress can substantially reduce fatigue life [15]. Mechanical analysis identified the regions between the shank and the small and big ends as the critical areas due to their high stress levels. Since bioethanol operation increases the stresses, the HCF of the connecting rod and its factor of safety were evaluated to assess its reliability under actual operating conditions.

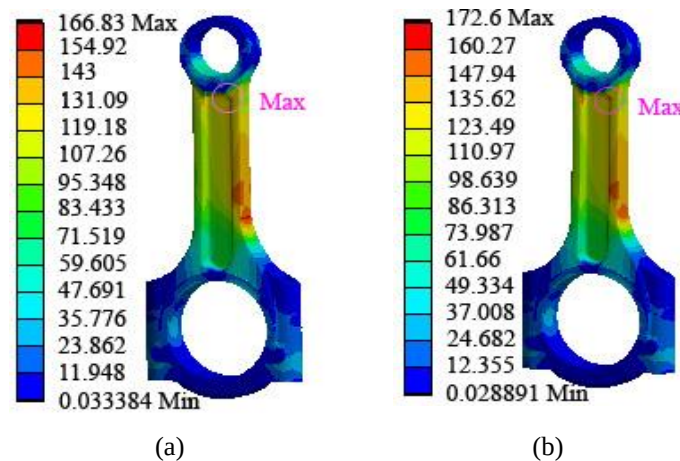


Figure 10. Stress contours in the connecting rod (a) gasoline and (b) bioethanol

Figure 11 presents the stress history at the critical node in gasoline and bioethanol conditions, calculated through the dynamic analysis of the applied forces. This figure indicates that the maximum stress in both mentioned conditions occur at the combustion moment.

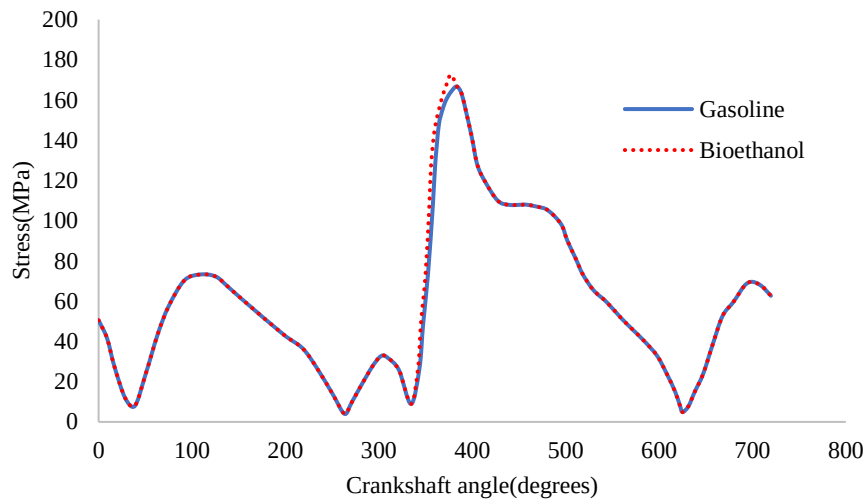


Figure 11. The variations of the von-Mises stress distribution of the critical node for the complete combustion cycle

The connecting rod static analysis results under the gasoline conditions are presented in Figure 12 to compare the static and dynamic methods. This figure suggests that the area between the shank and the small end is a critical one, with a maximum stress of 223.79 MPa. The static and dynamic analysis results in the gasoline condition reveal that using the static method in the mechanical analysis of the connecting rod leads to an estimation of approximately 56.96 MPa higher applied stresses, which is 34% higher than the permitted limit, confirming the considerable difference between the static and dynamic results.

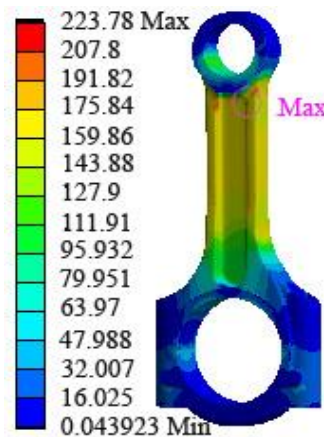


Figure 12. The von-Mises stress distribution in static analysis of the connecting rod in gasoline condition

The mechanical analysis under gasoline operation showed that the von Mises stress increased with the mesh density (Figure 13). However, beyond 74984 elements, the stress variation became negligible, indicating that 74984 elements provided sufficient mesh convergence and were therefore selected for the analysis.

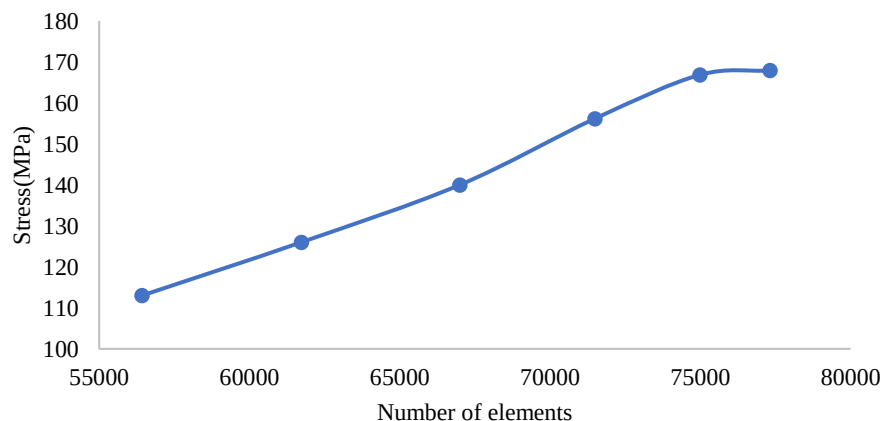


Figure 13. Stress versus element number

3.4 HCF Life Prediction

The connecting rod should be designed to be highly reliable and sufficiently strong to withstand the applied forces. Such a design necessitates careful analysis of the applied stresses to the connecting rod, and subsequently, the prediction of its fatigue life in the actual operational condition of the engine. The stress history was used in the nCode Design Life software to examine the connecting rod fatigue life in gasoline and bioethanol states. The results of the high-cycle fatigue analysis of the connecting rod in the gasoline state are presented in Figure 14(a), indicating that the two areas between the shank and the small and big ends are critical, with the 2.744×10^9 cycles as the minimum HCF life of the connecting rod, observable in the area between the shank and small end. The fatigue life analysis of the connecting rod in the bioethanol state is shown in Figure 14(b). According to the figure, the minimum high-cycle fatigue life of the connecting rod in this condition was 2.356×10^9 cycles, which occurred in the area between the shank and small end, similar to the gasoline state. The fatigue analysis results show an approximately 0.388×10^9 cycles higher fatigue life of the connecting rod in the bioethanol condition, indicating that employing bioethanol can decrease the connecting rod fatigue life by about 14.14%.

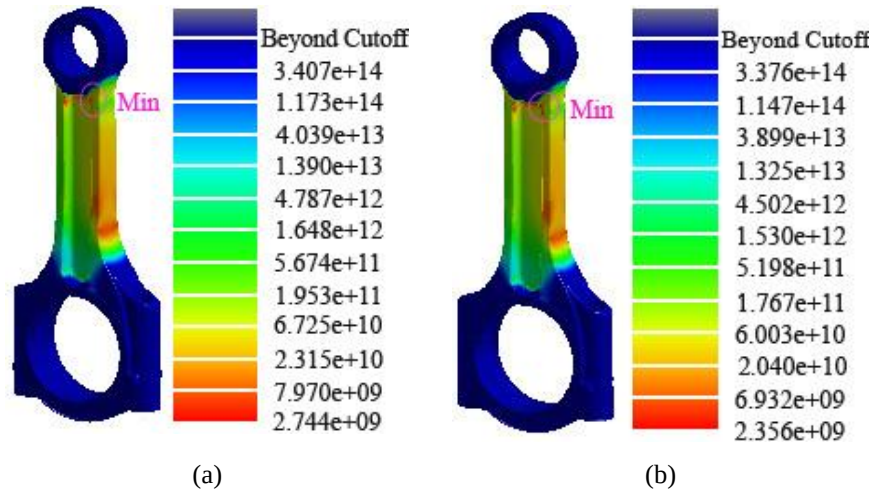


Figure 14. The HCF life prediction in the connecting rod (a) gasoline and (b) bioethanol

The safety factor is estimated based on the HCF method using the Goodman equation. The results revealed no critical areas in this regard (Figure 15). The connecting rod minimum safety factor in the gasoline and bioethanol conditions, resulting in the areas between the shank and small end, are 2.15 and 1.905, respectively. The results of mechanical analysis and fatigue life reveal that the two areas between the shank and the small and big ends are critical, and the maximum von-Mises stress and the minimum fatigue life can be captured in the area between the shank and the small end. The fatigue damage distribution is illustrated in Figure 16 to assess the various areas of the connecting rod that are potentially vulnerable to damage. Evaluation of fatigue damage distribution results suggests that the critical areas from a fatigue damage viewpoint correspond with the critical areas in the mechanical analysis and fatigue life approach. Therefore, in addition to the area between the shank and the small end, which is the most critical area with maximum and minimum von-Mises stress and HCF life, respectively, the area between the shank and the big end of the connecting rod is also prone to fatigue cracks.

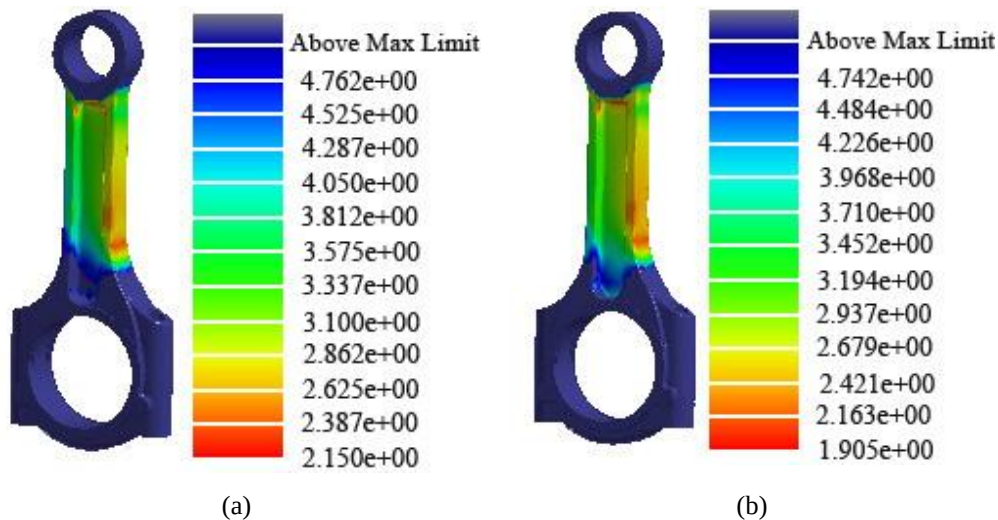


Figure 15. Calculated HCF safety factor in the connecting rod (a) gasoline and (b) bioethanol

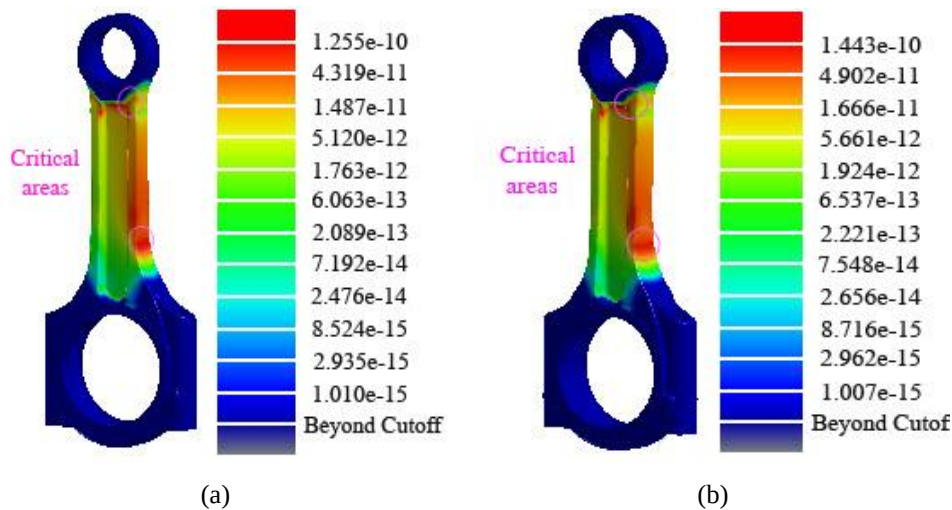


Figure 16. Fatigue damage contours in the connecting rod (a) gasoline and (b) bioethanol

The comparison of the finite element analysis results for gasoline and bioethanol operation is presented in Table 4, showing an approximately 3.46% increase in stress and a 14.14% reduction in the HCF life under bioethanol operation.

Table 4. Comparison of FEA results

Conditions	Stress (MPa)	HCF (cycles)
Gasoline	166.83	172.6
Bioethanol	2.744×10^9	2.356×10^9

3.5 Validation

The mechanical analysis results were evaluated based on [48] and [49]. The mechanical analysis in the latter was conducted using dynamic analysis results of the applied forces on the connecting rod, showing two areas between the shank and the small and big ends as critical, with a maximum stress of 175.25 MPa occurring between the shank and the small end. Likewise, the mechanical analysis of the present study showed that the two areas between the shank and the small and big ends are critical, with a maximum von Mises stress of 166.83 MPa in the gasoline state, observable in the area between the shank and the small end of the connecting rod. The 4.8% discrepancy between the mechanical analysis results of the present study and [49] indicates a good correlation between the two studies. On the other hand, [48] performed the mechanical analysis of the connecting rod through the static method, showing the maximum von-Mises stress in the gasoline state as 234.85 MPa at the area between the shank and the small end. The results of using the static method in the present research also showed that the maximum von-Mises stress of 223.78 MPa in the gasoline state occurred at the area between the shank and the small end. In other words, the difference between the mechanical analysis results and the static method in [48] and the present study is approximately 4.71%. Therefore, the critical areas in this research and [48] are the same, validating the accuracy of the mechanical analysis. However, as stated earlier and by [5] and [42], it would be improper to employ the static method in mechanical analysis due to the significant discrepancy between the static and dynamic outcomes. Thus, this study employed the dynamic method to elevate the accuracy of the mechanical analysis and fatigue life prediction of the connecting rod. According to the HCF safety factor results (Figure 15), no critical areas can be reported. Due to the scope of this study, the 800-hour endurance test was performed on the engine to simulate the real operating conditions and ensure the engine's capability of maintaining the desired function under stress and adverse conditions. During the test, various functional data were collected, including torque and output power, fuel consumption, and temperature. Upon completion, the engine is disassembled, and its components are inspected for wear, failure, damage, and cracking. The block program is presented in Table 5. The test was conducted in accordance with the established standards of Peugeot and FEV. The results of the 800-hour durability test reveal no damage in the critical areas of the connecting rod (Figure 17).

Table 5. Block program of 800-hour engine durability test [50]

Step	Time (minutes)	Engine speed (rpm)	Load specifications
1	5	6000	Rated power
2	4	2000	BMEP= 2bar
3	5	3500	Max torque
4	1	750	Low idle

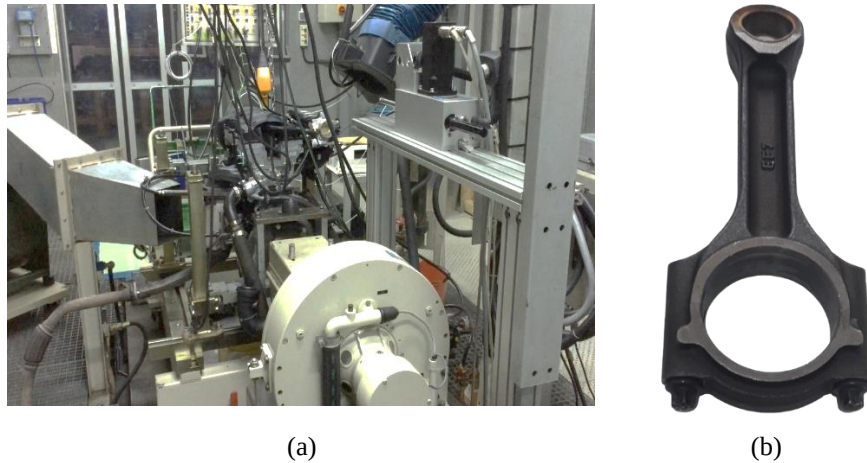


Figure 17. (a) HCF test device and (b) the connecting rod after HCF test

4. Conclusions

This study predicted the HCF life of the connecting rod by changing the fuel from gasoline to bioethanol using dynamic analysis results and employing FEA to provide a precise prediction of the stress distribution and fatigue life. As the mechanical analysis indicated, the maximum von-Mises stress of the connecting rod in gasoline and bioethanol states was 166.83 and 172.6 MPa, respectively, observable in the area between the shank and the small end. In other words, the connecting rod in the bioethanol state tolerates about 5.77 MPa higher stress. Therefore, replacing gasoline with bioethanol increases the connecting rod stress by about 3.46%. The FEA analysis results demonstrated the inaccuracy of the static method in the mechanical analysis of the connecting rod due to the significant discrepancy between the static and dynamic results. According to the high-cycle fatigue life analysis, the minimum fracture cycles of the connecting rod in gasoline and bioethanol states were 2.744×10^9 and 2.356×10^9 cycles, respectively, observable in the area between the shank and the small end. Therefore, the HCF life of the connecting rod in the bioethanol state would be reduced by about 0.388×10^9 cycles, or 14.14%. As the simulated results suggest, in addition to the area between the shank and the small end of the connecting rod, which is the most critical area with maximum von-Mises stress and minimum HCF life, the area between the shank and the big end is also prone to initiating a fatigue crack. The comparison of the simulated results with the mechanical analysis results of [46] validated the accuracy of the mechanical analysis with an excellent correspondence between the findings and the critical areas of the connecting rod. The HCF safety factor showed no critical area in the connecting rod. The FEA results were compared with the outcome from the engine durability test, which validated the results. The 800-hour test results, as an index of urban driving for the component's HCF evaluation, showed no specific damage in the connecting rod. Despite previous reported results on improved engine performance and reduced emissions with bioethanol, the present study indicates that its use in dual-fuel engines increases the stresses acting on the connecting rod. Therefore, mechanical analysis is required to determine the stress distribution in critical regions and ensure that the stresses remain within permissible limits. Furthermore, predicting the HCF life and factor of safety of the connecting rod is essential for assessing its structural reliability and preventing fatigue failure under actual engine operating conditions.

Funding

This study was not supported by any grants from funding bodies in the public, private, or not-for-profit sectors.

Declaration of Competing Interests

The author declares no conflicts of interest.

CRedit Author Contribution Statement

Hojjat Ashouri: Modeling, Simulation, Analysis, Validation, Writing, Review and Editing.

Availability of Data and Materials

The datasets generated and/or analyzed during the current study are available from the corresponding author.

Ethics Declarations

This study did not involve human participants or animals. Ethical approval was therefore not required.

Generative Artificial Intelligence Disclosure

No generative artificial intelligence tools were used in the preparation of this manuscript. All intellectual content is original and solely attributable to the author.

References

- [1] M. T. Alam, A. Thakur, V. Kumar PS, and S. Ghadei, "Fatigue failure analysis of diesel engine connecting rod," in *International Conference on Advances in Design, Materials, Manufacturing and Surface Engineering for Mobility*, Chennai, India, July 20, 2018, <https://doi.org/10.4271/2018-28-0067>.
- [2] T. Kaiho, S. Yamada, T. Tsuruda, M. Kubota, and S. Teramoto, "Weight reduction of connecting rod by using high strength steel," in *JSAE/SAE 2015 International Powertrains, Fuels & Lubricants Meeting*, Kyoto, Japan, September 1, 2015, <https://doi.org/10.4271/2015-01-1975>.
- [3] N. Thekke Kolayath, S. T, and R. Giles, "Connecting rod durability and big-end bore distortion study," in *WCX SAE World Congress Experience*, Detroit, Michigan, United States, April 21, 2020, <https://doi.org/10.4271/2020-01-0184>.
- [4] S. K. Kandreegula, S. Paroche, S. Mukherjee, D. Ayyar, and U. Gupta, "Fatigue analysis and validation of fracture split connecting rod for value engineering using 1D and 3D simulation tools," in *The 12th International Conference on Automotive Engineering*, Bangkok, Thailand, March 28, 2016, <https://doi.org/10.4271/2016-01-1735>.
- [5] L. Di Angelo, E. Mancini, and P. Di Stefano, "Numerical methodology for design and optimization of a connecting rod for very high speed engines," *International Journal on Interactive Design and Manufacturing*, vol. 16, pp. 109-134, 2022, <https://doi.org/10.1007/s12008-021-00812-x>.
- [6] W. Liu, M. S. Shadloo, I. Tlili, A. Maleki, and Q.-V. Bach, "The effect of alcohol–gasoline fuel blends on the engines' performances and emissions," *Fuel*, vol. 276, p. 117977, 2020, <https://doi.org/10.1016/j.fuel.2020.117977>.
- [7] İ. Örs, S. Yelbey, H. E. Gülcan, B. S. Kul, and M. Ciniviz, "Evaluation of detailed combustion, energy and exergy analysis on ethanol-gasoline and methanol-gasoline blends of a spark ignition engine," *Fuel*, vol. 354, p. 129340, 2023, <https://doi.org/10.1016/j.fuel.2023.129340>.
- [8] H. Hosseini, A. Hajialimohammadi, I. J. Gavzan, and M. A. Hajimousa, "Numerical and experimental investigation on the effect of using blended gasoline-ethanol fuel on the performance and the emissions of the bi-fuel Iranian national engine," *Fuel*, vol. 337, p. 127252, 2023, <https://doi.org/10.1016/j.fuel.2022.127252>.
- [9] M. Jamshidi, "The effect of plant and fossil fuels on piston life and reliability," M.Sc. thesis, Shahid Beheshti university, Tehran, Iran, 2022.
- [10] A. A. Elshenawy, S. M. Abdel Razik, and M. Gad, "Modeling of combustion and emissions behavior on the effect of ethanol–gasoline blends in a four stroke SI engine," *Advances in Mechanical Engineering*, vol. 15, p. 16878132231157178, 2023, <https://doi.org/10.1177/16878132231157178>.
- [11] S. Nuthalapati and D. Malkapuram, "Design and analysis of aluminium-flyash composite for connecting rod," in *International Conference on Advances in Design, Materials, Manufacturing and Surface Engineering for Mobility*, Chennai, India, October 11, 2019, <https://doi.org/10.4271/2019-28-0166>.
- [12] R. A. de Lima e Silva and M. L. Bittencourt, "Optimization methods applied to automotive connecting rod mechanical design," in *22nd SAE Brasil International Congress and Display*, Sao Paulo, Brazil, October 7, 2013, <https://doi.org/10.4271/2013-36-0633>.
- [13] T. Sathish, S. D. Kumar, and S. Karthick, "Modelling and analysis of different connecting rod material through finite element route," *Materials Today: Proceedings*, vol. 21, pp. 971-975, 2020, <https://doi.org/10.1016/j.matpr.2019.09.139>.
- [14] L. Witek and P. Zelek, "Stress and failure analysis of the connecting rod of diesel engine," *Engineering failure analysis*, vol. 97, pp. 374-382, 2019, <https://doi.org/10.1016/j.engfailanal.2019.01.004>.
- [15] S. S. Gandhul and A. Shriwastava, "Improved calculation methodology for design of connecting rod considering fatigue loads and stresses in IC engine," in *International Conference on Advances in Design, Materials, Manufacturing and Surface Engineering for Mobility*, Chennai, India, September 25, 2020, <https://doi.org/10.4271/2020-28-0335>.
- [16] A. M. Mohsin and H. Mohamed, "Analysis of fatigue stresses on connecting rod subjected to concentrated loads at the big end," *Materials Today: Proceedings*, vol. 2, pp. 2094-2103, 2015, <https://doi.org/10.1016/j.matpr.2015.07.205>.
- [17] S. Godara, V. Brenia, A. K. Soni, R. S. Shekhawat, and K. Saxena, "Design & analysis of connecting rod using ANSYS software," *Materials Today: Proceedings*, vol. 56, pp. 1896-1903, 2022, <https://doi.org/10.1016/j.matpr.2021.11.166>.
- [18] A. Bhamdare, A. Atulkar, and R. Porwal, "Design, static structural analysis and topology optimization of a four wheeler connecting rod," *Materials Today: Proceedings*, vol. 66, pp. 829-833, 2022, <https://doi.org/10.1016/j.matpr.2022.04.430>.
- [19] W. Xu, O. L. R. de Bretagne, M. Grasso, M. F. Harrison, O. Carless, and C. StLeger-Harris, "On a numerical methodology to assess the fatigue life of connecting rods," *Proceedings of the Institution of Mechanical Engineers, Part D: Journal of Automobile Engineering*, vol. 238, pp. 3232-3241, 2024, <https://doi.org/10.1177/09544070231180996>.
- [20] D. Jia, Y. Li, X. Deng, J. Lei, and W. Deng, "Design research on forging mark of connecting rod," *Engineering Failure Analysis*, vol. 113, p. 104537, 2020, <https://doi.org/10.1016/j.engfailanal.2020.104537>.
- [21] A. Muhammad, M. A. Ali, and I. H. Shanono, "Design optimization of a diesel connecting rod," *Materials Today: Proceedings*, vol. 22, pp. 1600-1609, 2020, <https://doi.org/10.1016/j.matpr.2020.02.122>.
- [22] S. Rakic, U. Bugaric, I. Radisavljevic, and Z. Bulatovic, "Failure analysis of a special vehicle engine connecting rod," *Engineering failure analysis*, vol. 79, pp. 98-109, 2017, <https://doi.org/10.1016/j.engfailanal.2017.04.014>.

- [23] A. Baldini, A. Strozzi, E. Bertocchi, and S. Mantovani, "Stresses in the cap of a connecting rod," *Engineering Failure Analysis*, vol. 129, p. 105693, 2021, <https://doi.org/10.1016/j.engfailanal.2021.105693>.
- [24] H. Savaripour, S. A. Hosseini, F. Farhani, and M. Akbari, "Finite element analysis of fatigue in the connecting rod of a spark ignition engine based on results of combustion tests," *Proceedings of the Institution of Mechanical Engineers, Part D: Journal of Automobile Engineering*, vol. 240, pp. 1328-1339, 2026, <https://doi.org/10.1177/09544070251320130>.
- [25] S. Seralathan, S. V. Mitnala, R. S. K. Reddy, I. G. Venkat, D. R. T. Reddy, V. Hariram, et al., "Stress analysis of the connecting rod of compression ignition engine," *Materials Today: Proceedings*, vol. 33, pp. 3722-3728, 2020, <https://doi.org/10.1016/j.matpr.2020.06.137>.
- [26] G. Shanmugasundar, M. Dharanidharan, D. Vishwa, and A. S. Kumar, "Design, analysis and topology optimization of connecting rod," *Materials Today: Proceedings*, vol. 46, pp. 3430-3438, 2021, <https://doi.org/10.1016/j.matpr.2020.11.778>.
- [27] S. U. M. Rao, T. H. Rao, K. Satyanarayana, and B. Nagaraju, "Fatigue analysis of sundry ic engine connecting rods," *Materials Today: Proceedings*, vol. 5, pp. 4958-4964, 2018, <https://doi.org/10.1016/j.matpr.2017.12.073>.
- [28] G. E. Honarvar, A. Babakhanian, and A. Haerian, "Study and Simulation of Shot peening Effect on Fatigue Life of a Powder Forged Connecting rod," *Automotive Science and Engineering*, vol. 4, no. 1, pp. 654-663, 2014, <http://ijae.iust.ac.ir/article-1-253-en.html>.
- [29] C. Juarez, F. Rumiche, A. Rozas, J. Cuisano, and d. P. Lean, "Failure analysis of a diesel generator connecting rod," *Case Studies in Engineering Failure Analysis*, vol. 7, pp. 24-31, 2016, <https://doi.org/10.1016/j.csefa.2016.06.001>.
- [30] X. Zhu, J. Xu, Y. Liu, B. Cen, X. Lu, and Z. Zeng, "Failure analysis of a failed connecting rod cap and connecting bolts of a reciprocating compressor," *Engineering Failure Analysis*, vol. 74, pp. 218-227, 2017, <https://doi.org/10.1016/j.engfailanal.2017.01.016>.
- [31] Z. Pan and Y. Zhang, "Numerical investigation into high cycle fatigue of aero kerosene piston engine connecting rod," *Engineering Failure Analysis*, vol. 120, p. 105028, 2021, <https://doi.org/10.1016/j.engfailanal.2020.105028>.
- [32] Y. Zheng, H. Wang, and Z. Wu, "A simulation method for predicting connecting rod buckling stress and fatigue safety factor of a four-cylinder small displacement engine," in *Society of Automotive Engineers (SAE)-China Congress*, 2017, pp. 797-818, https://doi.org/10.1007/978-981-10-8506-2_53.
- [33] I. Ismail, E. Abdelrazek, M. Ismail, and A. Emara, "Enhancing the durability of connecting rod of a heavy-duty diesel engine," *International Journal of Automotive and Mechanical Engineering*, vol. 18, pp. 8728-8737, 2021, <https://doi.org/10.15282/ijame.18.2.2021.10.0666>.
- [34] A. Strozzi, A. Baldini, M. Giacomini, E. Bertocchi, and S. Mantovani, "A repertoire of failures in connecting rods for internal combustion engines, and indications on traditional and advanced design methods," *Engineering failure analysis*, vol. 60, pp. 20-39, 2016, <https://doi.org/10.1016/j.engfailanal.2015.11.034>.
- [35] Y. Ketmuang and B. Wongthong, "Effect on fatigue behavior of connecting rod in gasoline engine," *Engineering, Technology & Applied Science Research*, vol. 14, pp. 14539-14545, 2024, <https://doi.org/10.48084/etasr.7239>.
- [36] S. Cecchel, N. Razavi, F. Mega, et al., "Fatigue testing and end of life investigation of a topology optimized connecting rod fabricated via selective laser melting," *International Journal of Fatigue*, vol. 164, p. 107134, 2022, <https://doi.org/10.1016/j.ijfatigue.2022.107134>.
- [37] B. Rahmatinejad, H. R. Asiabaraki, and F. A. Shishevan, "Static analysis, fatigue and buckling of connecting rod by finite element method," *The Journal of Engine Research*, vol. 71, pp. 32-44, 2024, <https://doi.org/10.22034/er.2024.2045296.1066>.
- [38] A. Elmaihi, M. A. El-latif, S. A. Mohamad, and M. Bennaya, "Influence of design parameters on stresses in the small end of the connecting rod in sailboat and passenger car engines," *Alexandria Engineering Journal*, vol. 133, pp. 117-136, 2025, <https://doi.org/10.1016/j.aej.2025.11.009>.
- [39] B. Kuldeep, K. Ravikumar, B. Guruprasad, et al., "A novel composite connecting rod: Study on mechanical and dynamic behaviour through experimental and finite element approach," *Composites Part C: Open Access*, vol. 12, p. 100413, 2023, <https://doi.org/10.1016/j.jcomc.2023.100413>.
- [40] R. D. Sulamet-Ariobimo, M. Fadhlan, Y. Oktaviano, T. Sukarnoto, Y. Mujalis, and A. Witonohadi, "The effects of I-Beam thickness to microstructure and compression load of thin wall ductile iron connecting rod," *International Journal of Lightweight Materials and Manufacture*, vol. 6, pp. 392-404, 2023, <https://doi.org/10.1016/j.ijlmm.2023.01.001>.
- [41] B. Zheng, Y. Cai, and K. Tang, "RETRACTED: Optimization design of titanium alloy connecting rod based on structural topology optimization," *Journal of Engineering, Design and Technology*, vol. 24, pp. 26-34, 2026, <https://doi.org/10.1108/JEDT-01-2021-0045>.
- [42] P. S. Shenoy and A. Fatemi, "Dynamic analysis of loads and stresses in connecting rods," *Proceedings of the Institution of Mechanical Engineers, Part C: Journal of Mechanical Engineering Science*, vol. 220, pp. 615-624, 2006, <https://doi.org/10.1243/09544062JMES105>.
- [43] C. E. Wilson and J. P. Sadler, *Kinematics and Dynamics of Machinery*, Pearson Higher Ed, 2013.
- [44] M. Eyidogan, A. N. Ozsezen, M. Canakci, and A. Turkcan, "Impact of alcohol-gasoline fuel blends on the performance and combustion characteristics of an SI engine," *Fuel*, vol. 89, pp. 2713-2720, 2010, <https://doi.org/10.1016/j.fuel.2010.01.032>.
- [45] R. I. Stephens, A. Fatemi, R. R. Stephens, and H. O. Fuchs, *Metal fatigue in engineering*, John Wiley & Sons, 2000.

- [46] M. K. Balki, C. Sayin, and M. Canakci, "The effect of different alcohol fuels on the performance, emission and combustion characteristics of a gasoline engine," *Fuel*, vol. 115, pp. 901-906, 2014, <https://doi.org/10.1016/j.fuel.2012.09.020>.
- [47] T. C. C. de Melo, G. B. Machado, C. R. Belchior, M. J. Colaço, J. E. Barros, E. J. de Oliveira, et al., "Hydrous ethanol-gasoline blends-Combustion and emission investigations on a Flex-Fuel engine," *Fuel*, vol. 97, pp. 796-804, 2012, <https://doi.org/10.1016/j.fuel.2012.03.018>.
- [48] A. Mirmohammadi and P. Rahmati, "Proposing a new geometry for the connecting rod under static loading with the function of aim of stress," presented at the 12th Conference on Internal Combustion Engines and Oil, Teharn, 2022.
- [49] S. Bab, "Static, quasistatic and dynamic analysis of conrod," M.Sc. thesis, Tarbiat Modares university, Tehran, Iran, 2009.
- [50] M. Shariyat, J. F. Sola, and S. Jazayeri, "Experimentally validated combustion and piston fatigue life evaluation procedures for the bi-fuel engines, using an integral-type fatigue criterion," *Latin American Journal of Solids and Structures*, vol. 13, pp. 1030-1053, 2016, <https://doi.org/10.1590/1679-78251937>.