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ORIGINAL ARTICLE

Enhancing the Durability of Connecting Rod of a Heavy-Duty Diesel Engine

I. Ismail[1*], S. M. Abdelrazek[2], M. Ismail[3] and A. Emara[3,4]

[1]Mechanical Power Engineering Department, Faculty of Engineering-Egyptian Russian University, 11829 Cairo-Egypt
Phone: +201226160979
[2]Mechanical Power Engineering Department, Faculty of Engineering-Misr University for Science & Technology, 12573 Cairo-Egypt
[3]Mechanical Power Engineering Department, Faculty of Engineering-Mataria-Helwan University, 11718 Cairo-Egypt
[4]Mechanical Power Engineering Department, Faculty of Engineering-Badr University in Cairo, 11829 Cairo-Egypt

ABSTRACT – This paper investigates the mechanical loads resulting from the combustion pressure and dynamic inertia and their effects on the connecting rod of a direct injection turbocharged diesel engine. The main purpose is to enhance the durability of the connecting rod in order to withstand more engine power increase. The distribution of the axial (compressive/tensile) stress, deformation, and safety factors are calculated in order to predict any possible mechanical failure. The finite element routine is used by ANSYS Workbench to analyse the loading on the connecting rod model. The current study is applied to the connecting rod of a 300 hp diesel engine in order to increase the engine power by 17%. The connecting rod operates safely and withstands the applied loads until the power increase reaches 72%. The most stressed points are at the connecting rod shank, while less stressed are experienced at the big end. Calculations show that introducing some changes to the connecting rod geometry may result in decreasing the excessive stresses. These changes include increasing the thickness of the shank cross-section, increasing the fillets radii and slightly reducing the dimensions of the big end in order to maintain the same mass. The new geometry could significantly reduce the maximum stress by 25.5% with an insignificant reduction in the total mass of the connecting rod.

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INTRODUCTION

The connecting rod of a diesel engine is subjected to high cyclic loads, changing from the extreme compression (due to the maximum pressure of the combustion gases) to the highest tension (due to the inertia of the reciprocating masses), all at high frequencies. These loads induce heavy axial and bending stresses on the connecting rod parts, which requires the use of high strength material in order to achieve adequate durability [1]. The shank buckling, small and big ends cracking, and bolts rupture are a few of the consequent failure patterns of the connecting rod. Thus, investigating the stress, deformation and safety factor of each connecting rod section is important in order to avoid unforeseen breakdowns. Besides, the current automobile industry is faced with the growing demands for more engine power, less weight, and higher efficiency. This necessitates extra design optimisation, using innovative materials and deeper stress-strain analysis of the main engine components. The connecting rod optimisation comprises weight reduction, strength increase, and less manufacturing cost [2]. Shenoy et al.. [3] performed quasi-dynamic finite element analysis for the connecting rod in order to analyse the changes of stresses along one operation cycle. The results showed that the bending load has a great effect on the connecting rod strength, in addition to the effects of typical axial compression and tension loads. The same authors [4] introduced a further study aimed at reducing the weight and production cost of the connecting rod. In this study, finite element techniques were used to study the effect of both dynamic tensile loads and extreme static compression loads. The published results showed that the connecting rod shank has the largest potential of weight reduction; its section modulus, however, must be high enough to resist bending.

Asadi et al. [5] analysed the loading on the connecting rod of a tractor engine under different service conditions. They concluded that the maximum stressed points due to compression were observed at the transition between the small end and the shank. The maximum stressed points due to the tension were obtained in the lower half of the small end. Pathade et al. [6] executed a similar finite element analysis and conducted an experimental study (using the photo elasticity technique) in order to validate the calculated results. They reported that both methods agreed that the stresses are higher at the fillet of the small end rather than at the big end. Rakic et al. [7] analysed the fractures that occurred to the connecting rod of a twelve-cylinder diesel engine using visual inspection, metallography, micro fractography, and finite element method. The results showed that the main reason for the connecting rod failure was the engine operation for a long time at the maximum load. They recommended increasing the fillets at the connecting rod at the points having highly concentrated stresses. Zhu et al. [8] investigated the failure problem for a connecting rod cap and bolts installed in a reciprocating compressor. The investigation was performed using a scanning electron microscope, an optical microscope, and finite element analysis technique. The results revealed that there was an initial crack in the connecting rod cap which led to stress concentration at this location, causing the consequent failure with high cyclic loading. Shaari et al. [9]

***CORRESPONDING AUTHOR** | I. Ismail | ✉ eng.islamismail91@gmail.com

analysed the stresses and strains of a connecting rod in order to decrease the rod weight. The maximum stresses were found along the connecting rod shank, while the connecting rod big end had an adequate potential for weight reduction. Khare et al. [10] discussed the problem of the cracks and pits at the internal surface of the connecting rod big end using a finite element model. The results showed that the contact pressures between the connecting rod big end, roller bearing, and crankpin are excessively high. The original connecting rod was then modified by increasing the web area of the connecting rod shank to reduce its stiffness and consequently decrease the contact pressure. Bari et al. [11] investigated the reason for the fracture that occurred to the connecting rod of a motorcycle engine. The scanning electron microscope, optical microscopy and visual inspection were used as well as a finite element analysis. The observations revealed that there were micro-cracks at the connecting rod big end due to sulphur inclusions, leading to the failure during fatigue loading. Alam et al. [12] presented a detailed investigation for a connecting rod failure during a fatigue test. They studied the different factors of failure such as the structural design, material strength, and machining processes by using finite element analysis, visual observation, dimensional inspection, metallurgical, fractography analysis, hardness testing, and residual stresses analysis. The finite element analysis results showed that the failure is not related to design issues. The other tests suggested that the connecting rod failure was due to the residual tensile stress at the oil holes of the small end. This may be related to improper drilling and machining process.

Son et al. [13] simulated the realistic behaviour of the connecting rod of a marine engine under the operation cycles. The used connecting rod was a marine head type that has three split surfaces. The Ruiz criterion was used to predict any fretting fatigue failure, which includes slip, tangential and shear stress analysis. The results concluded that the Ruiz criterion that is improved to predict the fretting damage is very practical at the contact surfaces of the connecting rod body. Chao [14] analysed the sudden failure of an oblique split connecting rod of a medium speed engine after 20000 hours of operation. A material characterisation analysis was performed at the fracture zone to determine the cause of the failure. The results indicated that micro-movements between the housing bore and bearing back was the cause of the fatigue failure at the inner surface of the connecting rod cap bearing. Witek et al. [15] performed a stress and failure analysis for the connecting rod of a diesel engine. Visual observation at the fracture zone indicated that fatigue failure has occurred. Microscopic examination revealed that no material defects or corrosion at the crack region. Finite element analysis was applied to investigate the problem of failure. The analysis results showed that the connecting rod failed near the hole of the bolt because of the high pretension torque of the bolts. Also, the small radius of the fillet near the bolts accelerated the failure. Rao [16] determined the fatigue life of the connecting rod of a diesel engine at a variable compression ratio and searched for alternative materials. MATLAB software was used to perform the kinematic and dynamic analysis of the connecting rod then ANSYS workbench for determining von Mises stresses, deformation and fatigue life. The results concluded that the titanium alloy is the best material, especially at high loads, forged steel considered a commercial aspect and aluminium alloy has a low cost. Parkash et al. [17] re-optimising the design of the connecting rod of a tractor to reduce the weight and increase life. The base design of the connecting rod was analysed under fatigue loading using ANSYS software to evaluate the critical regions. This analysis revealed that the maximum stress occurred at the shank near the piston pin end. These results were used to modify the connecting rod design, which led to reduce the weight, inertia force and cost of the material.

In this study, the finite element method is used to analyse the mechanical loadings on the forged steel connecting rod of a 300 hp turbocharged diesel engine. The analysis is performed under the conditions of engine maximum torque and maximum speed and repeated with increased turbocharging pressure. Axial and bending stresses and the factors of safety are evaluated in all sections. The study is conducted to predict any failure in the connecting rod with increased boost pressure in order to increase the engine power to 350 hp and further. Some modifications to the connecting rod design are then suggested in order to increase the connecting rod strength without increasing its mass.

ENGINE SPECIFICATION

The engine main characteristics are experimentally measured at different speeds and loads, especially at the conditions of the maximum torque and power output with and without increasing the boost pressure (basic and upgraded engine) [18-21]. The main features of the engine and the connecting rod are listed in Table 1.

Table 1. Base and upgraded engines specifications.

Parameters	Base engine	Upgraded engine
Type	4 stroke water cooled DI diesel engine	
No. of cylinders	6 Inline	The same engine with increased turbocharging pressure
Aspiration type	turbocharged	
Bore x Stroke (mm)	150 x 180	
Displacement (cm^3)	19100	
Crank shaft radius (mm)	90	
Connecting rod length (mm)	320	
Compression ratio	15:1	
Pressure ratio (boost)	1.35	1.5
Rated power (hp)	300 @ 1800 rpm	350 @ 1900 rpm
Max. torque (N.m)	1258.2 @ 1200 rpm	1488.9 @ 1300 rpm

NUMERICAL ANALYSIS

The connecting rod is subjected to mechanical loads due to combustion gas pressure and inertia forces of the reciprocating mass acceleration, with the former increasing with an increased boost, as shown in Figure 1. These loads, which largely vary in magnitude and direction during the engine cycle, affect all connecting rod parts, namely small end, shank, and big end. Thus, all sections of the connecting rod are under variable tension-compression loading at high frequency. The net force of the gas pressure and inertia forces acting on the connecting rod *Fc* is determined from the dynamic analysis for the crank mechanism, as shown in Figure 2.

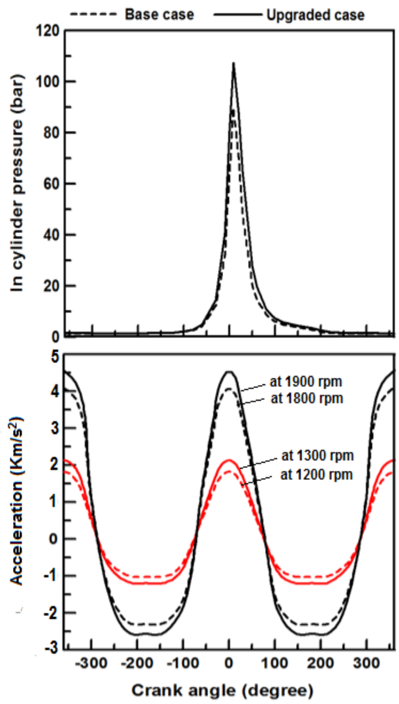


Figure 1. Gas pressure and the acceleration of the reciprocating mass versus crank angle degrees.

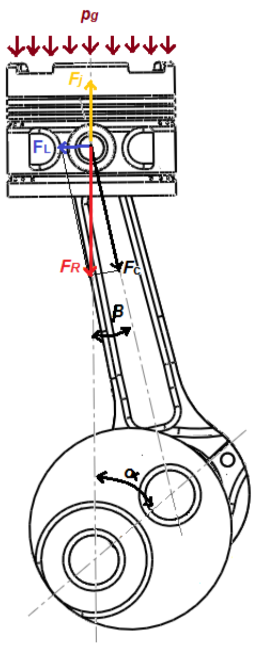


Figure 2. Dynamics of engine crank mechanism.

The net force, F_C acts on the connecting rod is calculated as Eq. (1).

$$F_C = F_R / \cos\beta \tag{1}$$

Where, $\beta = sin^{-1}(\lambda \sin \alpha)$, $F_R = pg.Ap + Fj$, $Fj = -mj.a$, and $a = R\omega^2(\cos \alpha + \lambda \cos 2\alpha)$. p_g is the instantaneous gas pressure (MPa), A_p is the piston crown area (m^2), F_j is the inertia force, m_j is the reciprocating mass of the piston assembly group and connecting rod small end (kg), a is the piston acceleration (m^2/s), R is the crank radius (m), ω is the angular velocity (rad/s), λ is the ratio of the crank radius and connecting rod length, β is the angle between the connecting rod and cylinder axis, α is the crank angle.

Figure 3 presents the variation of the force acting on the connecting rod with crank angle degree. This force reaches the maximum in tension because of inertia when the piston is at TDC. The maximum net compression force on the other hand, is caused by the gas pressure while opposed by the force of inertia. The maximum pressure is typically attained 8 crank degrees after TDC at the beginning of the expansion process. The connecting rod is usually designed to withstand the fatigue loads occurring at extreme operating conditions, namely at maximum power/speed (n = n_{max}) and maximum load/torque (T = T_{max}) [22].

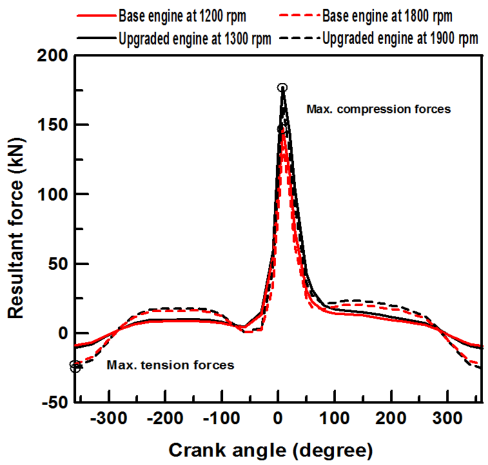


Figure 3. The force acting on the connecting rod.

Static analysis is conducted on the connecting rod to obtain the stress time history using the finite element ANSYS software. This technique can capture the actual connecting rod behaviour during the engine cycle [4]. At first, the prepared connecting rod geometry is imported, then the material property is defined, and a mesh study is performed. The studied connecting rod has a big oblique end splitting plane, as shown in Figure 4, and the used material is alloy steel with high ultimate strength and yield point (see Table 2). The grid mesh for the finite element analysis is generated using the tetrahedral type to ensure calculation accuracy. A comprehensive mesh study was done to check the best distribution and resolution of the results at different element sizes. The best convergence was achieved with a minimum element size of 2 mm, 460888 nodes, 291517 elements, 96.6% of elements with aspect ratio < 3, and 0.107% of elements with aspect ratio > 10, and 0% of distorted elements. The piston, piston pin, connecting rod and crankshaft are assembled together with a surface to surface contact between each other. The gas pressure is applied to the top surface of the piston crown, and the acceleration of the reciprocating mass at the centroid as the main mechanical loads [10, 23]. These loads are applied at the crank angles of the maximum compression and tension forces acting on the connecting rod as discussed above. This allows catching different types of loads that the connecting rod is subjected to and finding the critical zones [24].

Table 2. Properties of connecting rod material.

Properties	Values of the properties
Connecting rod material	Alloyed steel DIN 1.6747 (30NiCrMo16-6)
Density	7800 (Kg/m^3)
Yield strength	900 (MPa)
Young's modulus	210 (GPa)
Poisson ratio	0.3
Fatigue strength	460 (MPa)

RESULTS AND DISCUSSION

Figure 5 shows the distribution of the calculated stress along the connecting rod body at 8º crank angle during the expansion stroke (the instant of highest stress). It can be observed that the stresses concentrate at the fillet of the joint between the small end and the shank. This is related to the smallest section of the shank being at this point. The maximum stress is 299.9 and 359.8 MPa for the base and upgraded case, respectively, at the full engine load. The stress then decreases gradually along the shank down to the big end. The net high force acting on the piston downwards at this instant causes high surface pressure at the contact area of the lower half of connecting rod small end. The pressure is usually unevenly distributed along the middle of 120º of the lower surface, of the small end. The maximum value of this pressure is 154.0 MPa for the base engine and 184.5 MPa for the upgraded one, typically at the edges of the small end. The distribution of the resulted deformation from the maximum combustion pressure acts on the connecting rod body is shown in Figure 6. It can be observed that the maximum deformation occurs at the lower surface of the connecting rod small end, 0.254 mm for the base case and 0.305 mm for the upgraded one. The deformation distribution decreases gradually along the connecting rod shank. There are no concentrated deformations observed on the connecting rod shank sides; hence no chances for buckling the connecting rod shank.

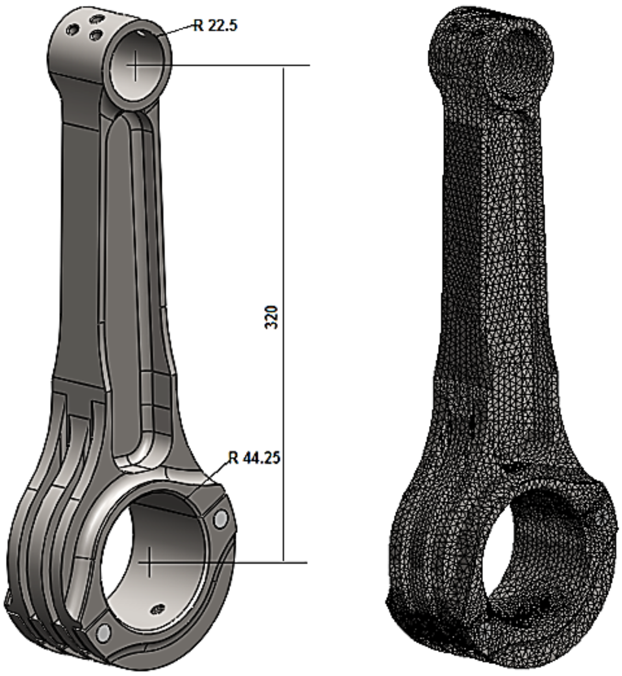
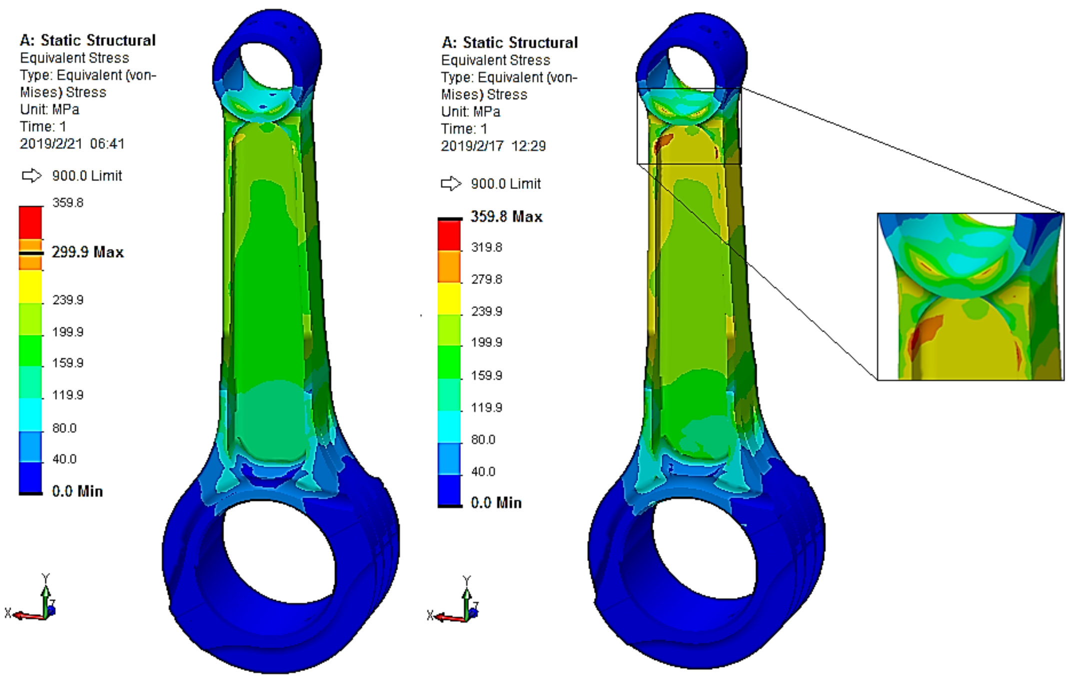
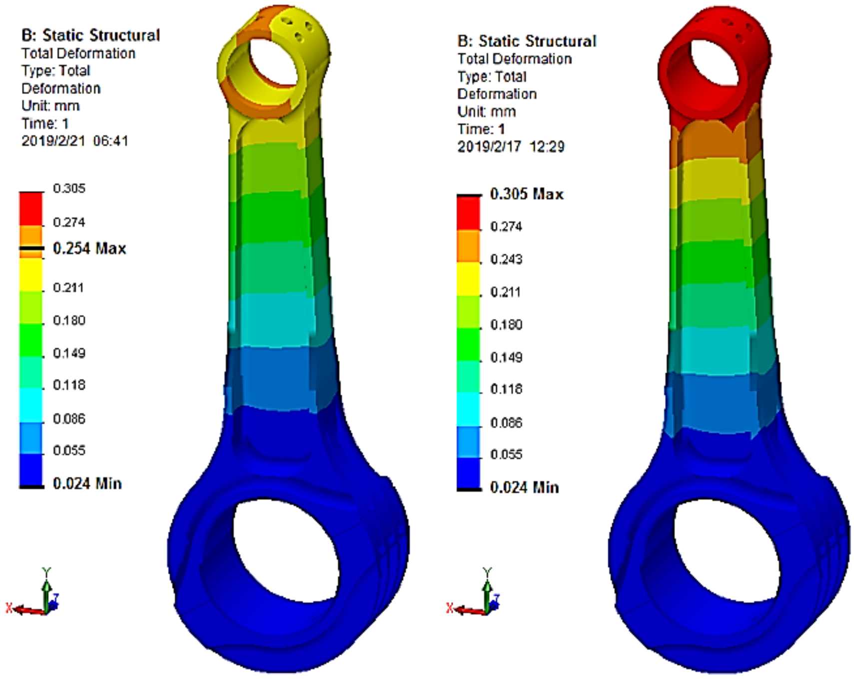


Figure 4. The geometry and meshing of the connecting rod model.



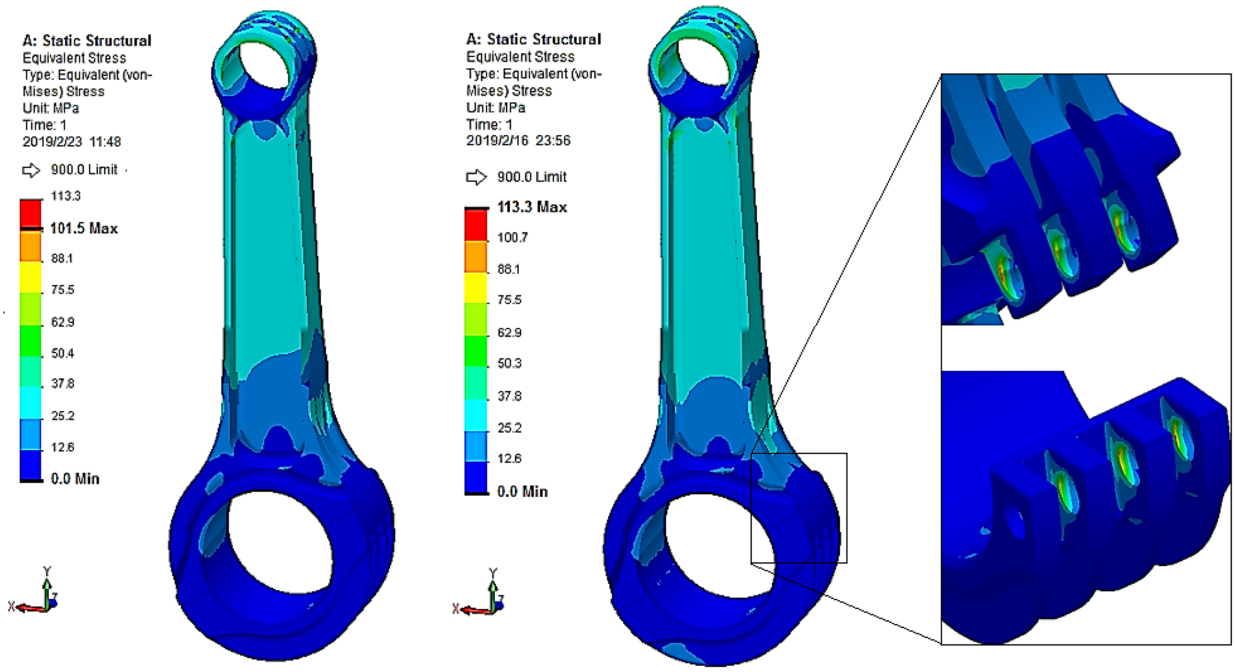
(a) base case (b) upgraded case

Figure 5. Stresses distribution over the connecting rod body due to the maximum compression load.



(a) base case (b) upgraded case

Figure 6. Deformation distribution over the connecting rod body due to the maximum compression load.



(a) base case (b) upgraded case

Figure 7. Stresses distribution over the connecting rod body due to the maximum tension load.

The distribution of the calculated stress at TDC during suction in Figure 7 shows that the highest stress occurs at the pinholes of the big end. This is due to the tension resulting from the highest inertia force upwards. At this region of the interference between the connecting rod and cap, the surface roughness of the pins and holes are neglected. The value of the concentrated stress is 101.5 MPa for the base case and 113.3 MPa for the upgraded one at maximum engine power. Besides, it can be observed that the pulling of piston pin up is due to the highest inertia results in surface pressure on the upper surface of the connecting rod small end hole. This surface pressure is typically distributed along 180° of the small end upper surface. The maximum surface pressure is 24.8 and 27.7 MPa for the base and upgraded engine, respectively, concentrated at the edges of the small end. Dynamic (fatigue) load safety factor is evaluated considering the alternating nature of tension/compression due to the combined effect of inertia and gas pressure forces. Goodman method [25] is used for calculating the dynamic safety factor, and the minimum to maximum load ratio is evaluated along one engine cycle. The connecting rod is analysed for 10^6 to 10^9 cycles (infinite life), as shown in Figure 8.

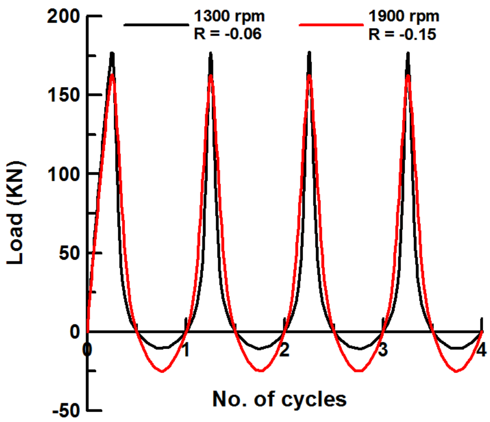


Figure 8. Connecting rod load ratio.

The minimum dynamic safety factor is 2.56 and 2.2 for the base case and upgraded one, respectively, found in the upper section of the connecting rod shank under the small end. The studied connecting rod can withstand loads with engine power increased up to 770 hp. Further increase in power requires modifications to the connecting rod design. Changing the rod material and/or dimensions are the major modifications that can be introduced in order to improve the strength and reduce the weight and cost at the same time. The most used materials for manufacturing the connecting rod are steel, aluminium and titanium alloys. Aluminium alloys are used in light-duty engines, while titanium alloys are used in high power engines, at extra cost, however. Steel alloys, commonly used, satisfy high strength requirements at a moderate cost [16]. The studied connecting rod is made of steel alloy, thus introducing design modifications is adopted in order to increase the rigidity. The lowest calculated safety factor is observed at the upper part of the connecting rod, just below the small end. The most and least stressed parts are the shank and the big end, respectively. Modifying the shank I-section and its transition with the small end would be most effective in improving the strength. Changing the thickness and diameter of the connecting rod big end, in order to decrease the mass, is also seen as a potential method for more improvement.

Several designs with different modifications are introduced, with the analytical calculations repeated with each design at the loading conditions of the upgraded engine. Some constraints are imposed in the connecting rod modification, include the main bearing diameter, the connecting rod length from the center of the small end to the center of the big end, type of the shank cross-section, and dimensions of the small end oil holes, and connecting rod material. Figure 9 shows a few of the tested, modified designs of the connecting rod.

The stresses calculated with all the introduced designs were compared to the basic design and summarised in Figure 10. It is found that with the increase in the thickness of the shank cross-section from 5 mm (initial design) to 7 mm (modified design 2), the maximum stress is reduced by 16.6%. Moreover, the weight of the connecting rod increased from 5.11 kg to 5.51 kg, nearly 7.7%. The increase in the inner diameter of the small end by 1 mm and the fillet between the shank and the small end by 2 mm (modified design 3) further reduces the stresses and mass by 9.4% and 0.6%, respectively. The reduction in stresses is significant; however, the increased mass leads to higher inertia and row material cost. Thus, successive attempts are made to decrease the connecting rod mass. Firstly, the thickness of the cap reduced by 1 mm (modified design 4), reducing the weight by 1.3%. Any further decrease in the thickness will lead to increased stresses. The second modification reduces the outer and inner diameter of the big end ribs (modified design 5 and 6). This modification decreases the total mass by 6.8%, with only a 0.5% increase in stresses (considered insignificant). Due to the modifications in the shank, fillets, and big end of the connecting rod (discussed above in details), the maximum induced stresses on the connecting rod reduced from 359.8 MPa to 267.8 MPa by 25.5%, and mass reduced from 5.11 kg to 5.03 kg by 1%. The stresses decreased along the connecting rod shank and concentrated only under the small end due to the increase in the shank cross-section, see Figure 11. The maximum deformation at the lower surface of the small end decreased from 0.305 mm to 0.264 mm by 13.4%. The results indicate that the basic connecting rod design may withstand the mechanical loading even with a 156.6% increase in engine power. With simple design modifications introduced that has no significant effect on the mass, the modified connecting rod can withstand the extra loads with engine power further increased up to 183.3%.

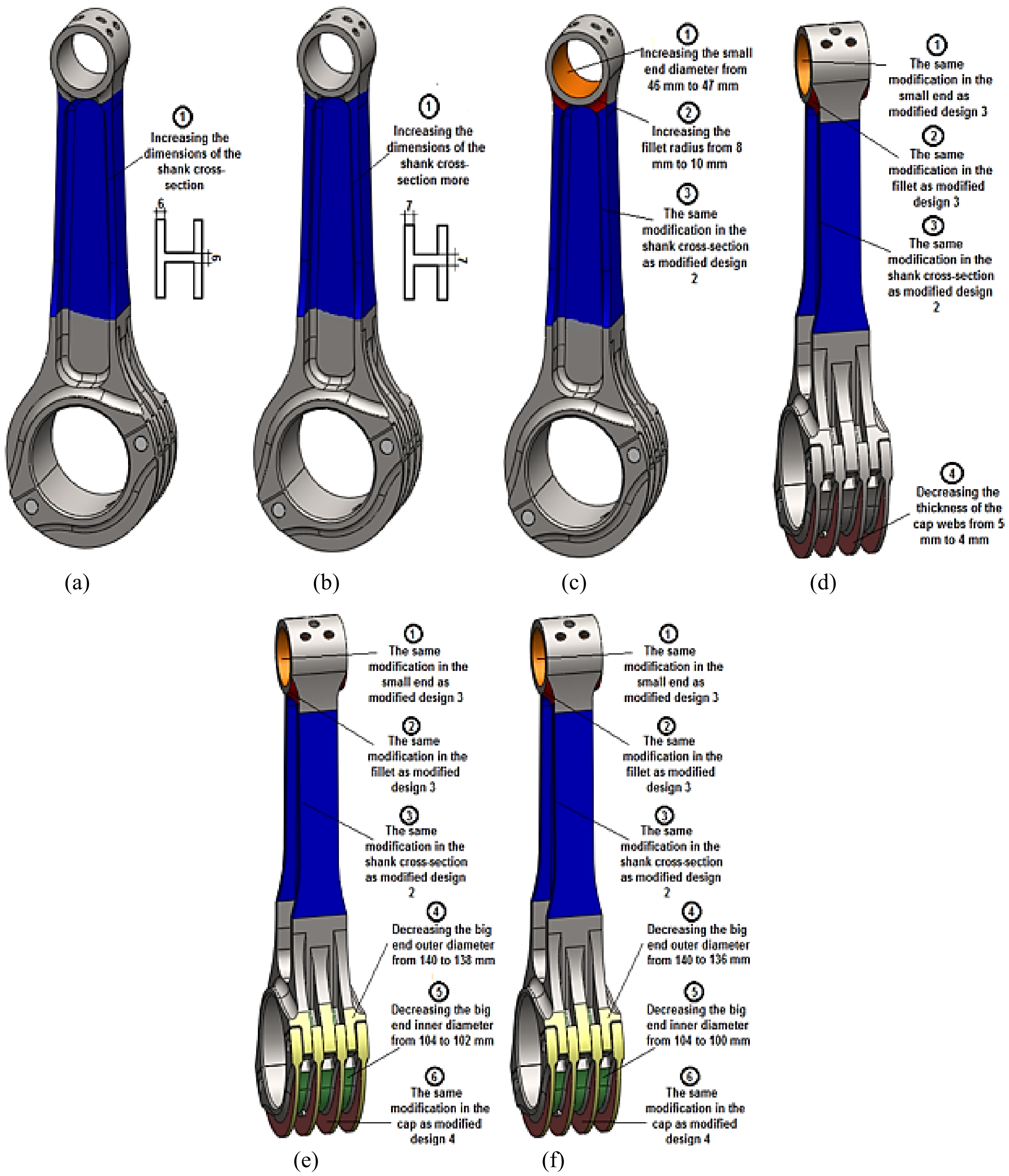


Figure 9. The modified designs of the connecting rod model; (a) modified design 1, (b) modified design 2, (c) modified design 3, (d) modified design 4, (e) modified design 5, and (f) modified design 6.

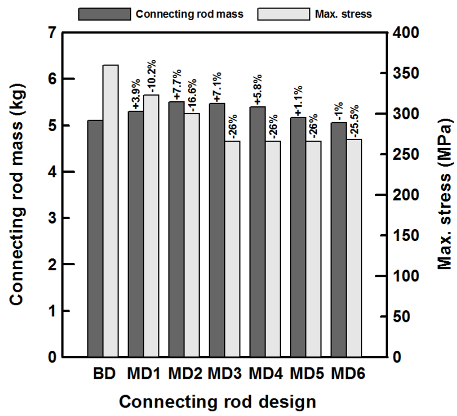
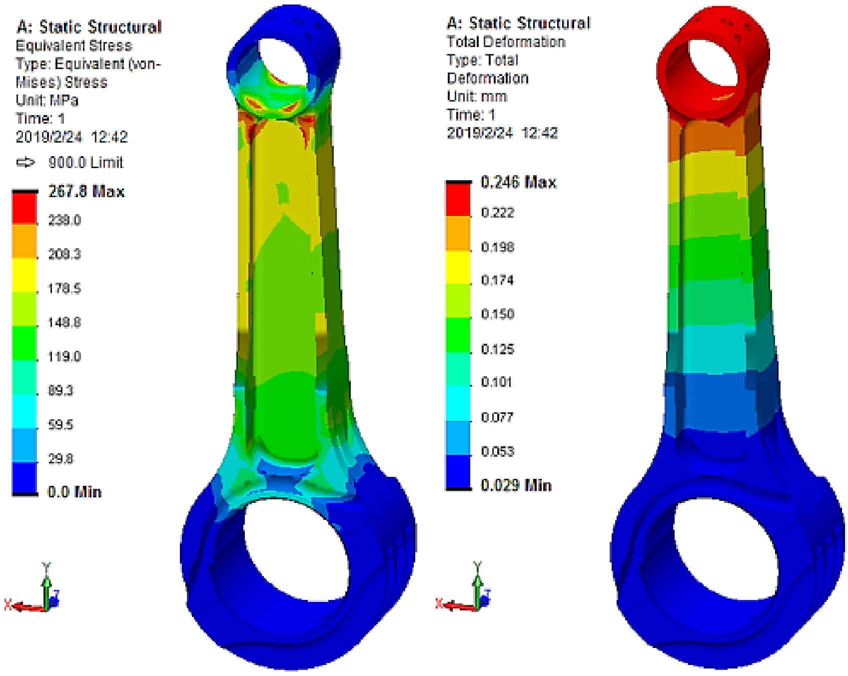


Figure 10. Percentage of stress and weigh change for the different connecting rod designs.



(a) (b)

Figure 11. (a) Stresses and (b) deformation distribution over the modified design of the connecting rod body due to the combustion pressure.

CONCLUSION

The present study addresses the analysis of stresses acting on the connecting rod body due to mechanical loading. The analysis is made on the connecting rod of a turbocharged engine, with the purpose of increasing the boost pressure in order to increase the engine power from 300 hp to 350 hp. Several design modifications are introduced, in order to increase the connecting rod strength with minimum cost penalties. The concluding remarks of this study may be summarised as follows:

i. Under the combined effect of the axial and bending loading, the highest stress is 299.9 and 359.8 MPa for the initial and increased power engine respectively. This stress occurs at the transition between the shank and small end.
ii. At these stresses, the safety factor is calculated as 2.56 for the initial engine, and decreases by 14% with engine power increased from 300 to 350 hp.
iii. The connecting rod has enough strength and can operate safely even with the engine power increased by 156.6%.
iv. The increase in the thickness of the shank cross section and the fillets by 40% reduces the maximum stresses by 25.5%.

The modified connecting rod with the bigger cross-section dimensions can withstand the stresses with engine power increased by 183.3%.

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