

Application of the Finite Element Method for Stress Evaluation of a Gasoline Engine Crankshaft under Turbocharged Conditions

Nguyen Xuan Khoa

School of Mechanical and Automotive Engineering,
Hanoi University of Industry, No. 298, Cau Dien Street, Tay Tuu Ward, Hanoi, 100000, Vietnam.
E-mail: khoanx@hau.edu.vn
ORCID: <https://orcid.org/0000-0003-2869-465X>

Nguyen Van Tuan

Mechanical Power Engineering and Automation Research Group,
University of Transport Technology, Hanoi, 100000, Vietnam.
Corresponding author: nguyenvantuan@utt.edu.vn
ORCID: <https://orcid.org/0000-0002-8340-8532>

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Abstract

Turbocharging is applied to internal combustion engines to improve power density and fuel economy, and is used broadly. This can be especially beneficial if used in conjunction with engine downsizing to enable efficient city driving while maintaining the necessary power output for highway speeds. These advantages come at a price, however; the mechanical loading imposed on the crankshaft-connecting rod assembly, which consequently impacts the durability of the crankshaft. The authors compared the stress state of the crankshaft of a turbocharged gasoline engine under two conditions, namely, naturally aspirated and turbocharged, to examine the effect of mechanical load on the operating conditions of the crankshaft. The authors used the engine model based on AVL-Boost software to generate pressures and dynamic loads to be used as boundary conditions for the 3D finite element model of the crankshaft. The study was performed at two operating modes, one for the maximum torque and the other for the maximum power with the boost pressure ratio of 1.5. The outcomes show that with turbocharging the equivalent stress of the crankshaft is higher and it reaches a maximum of 301.79 MPa at 4200 rpm. The high-stress areas were primarily concentrated at the transition zones between the crankpin and the crank web. The results of calculations indicated that the loads applied on the crankshaft of the turbocharged gasoline engine were still below the allowable loads for 40XHMA steel. These findings demonstrate that the original crankshaft design maintains sufficient durability under the investigated turbocharging conditions, while also providing a basis for future research into structural optimization and fatigue strength assessment.

Keywords- Turbocharged gasoline engine, Crankshaft, Strength verification, Finite element analysis.

1. Introduction

Turbocharging is now widely used in gasoline engines because it allows a smaller engine to deliver the power output of a larger naturally aspirated unit. This approach supports engine downsizing and can improve fuel economy, particularly during urban operation, while maintaining adequate performance at high engine speeds. For these reasons, turbocharged configurations have become increasingly common in modern fuel-efficient gasoline engines. The improvement in engine performance, however, is accompanied by higher intake pressure, increased in-cylinder pressure, and greater thermal loading. These changes directly increase the mechanical forces acting on the piston-connecting rod-crankshaft mechanism and may reduce the durability of its components if the original design is retained without further assessment (Almoslehy and Hadi, 2024; Elmoselhy et al., 2022).

The crankshaft is one of the most highly loaded components in an internal combustion engine. It receives the gas force transmitted through the piston and connecting rod and converts the reciprocating motion of

the piston into rotational motion. At the same time, it is subjected to inertia forces arising from the reciprocating and rotating masses. The combined action of these forces produces bending and torsional loads that vary continuously with crank angle and engine speed.

During operation, the crankshaft is subjected to cyclic loads; consequently, fatigue is the most common failure mode. Stress concentrations are the main points of interest at the geometric change regions between the crankpin and the crank web, and between the crank web and the main journal. These are the areas where cracks can form and grow under repeated loading, compromising the crankshaft's structural integrity. When the crankshaft fails, it can result in the destruction of the crankshaft, and other components of the engine can also be seriously damaged, which may cause the entire engine to fail (Aliakbari, 2019; Aliakbari et al., 2022; Jiao et al., 2020; Xu and Yu, 2018). So, studying and assessing crankshaft durability plays a vital role in improving engine reliability, lifespan, and performance efficiency (Fonseca and de Faria, 2018; Fonte et al., 2019; Nozdrzykowski et al., 2022; Witek et al., 2017). Based on the literature review, one can observe that the research on crankshafts has been mostly related to material selection, structural design, and manufacturing technology. In fact, in practice forged or cast crankshafts are of common use because they can meet the requirement of strength and fatigue resistance.

In addition, many studies have been conducted to enhance the surface hardness and fatigue resistance of the crankshaft using surface treatment techniques like carburizing, nitriding, and induction hardening (Que et al., 2023; Songsong et al., 2021; Sun et al., 2022). In addition, there are studies on optimizing the structural geometry or manufacturing process to improve the durability and reduce the mass of the crankshaft (Bulut et al., 2021; Luo et al., 2026; Yang et al., 2025). With turbocharged gasoline engines, however, a rise in cylinder combustion pressure places greater stresses on the crankshaft, both in terms of their magnitude and how they are distributed. Therefore, it is a matter requiring attention and further study to determine the durability of crankshafts under the operating conditions of turbocharged engines. There are two ways to analyse crankshaft durability: analytical calculations and numerical simulations. Analytical methods have the benefit of being simple and quick to compute and can be used for the preliminary design stage or for initial durability checks. But in the case of complex geometry and changing loading conditions, these methods are not able to describe the real stress situation completely. The Finite Element Method (FEM) or Finite Element Analysis (FEA) is often employed in order to obtain more precise results of the analysis. This method enables the creation of a three-dimensional crankshaft model to determine stress distribution in the high-load areas, such as fillets and areas where there are abrupt geometric changes (Ang and Ku, 2021; Liu et al., 2020). FEA is widely used in structural analysis of high-load engine components like pistons, connecting rods, crankshafts, and other significant engine components (Ang and Ku, 2021; Tien and Tuan, 2026). Nevertheless, the accuracy of the analysis results not only relies on the finite element model, but is also very much dependent on the applied loads and the boundary conditions set during the simulation process.

In a number of published studies, crankshaft loads are represented by a single peak pressure, a characteristic gas force, or a limited number of operating points. In other cases, the structural model is developed separately from the engine-cycle simulation. These simplifications reduce the computational effort but may not fully represent the variation of cylinder pressure and inertia force over the engine cycle, especially under turbocharged operation (Almoslehy and Hadi, 2024; Elmoselhy et al., 2022). The tangential and normal forces acting on the crankpin also vary with crank angle, and their maximum values do not necessarily occur at the same angular position. Neglecting this variation may affect the calculated stress distribution and the identification of the most critical crankshaft regions (Xu et al., 2025; Zhang and Zheng, 2023).

In the present study, the engine-cycle and structural calculations are linked to provide a more representative description of crankshaft loading. A one-dimensional gasoline-engine model is first developed in AVL-Boost to determine the in-cylinder pressure and the corresponding forces under naturally aspirated and turbocharged conditions. These calculated loads are then applied to a 3D finite element model of the crankshaft in ANSYS. The study was carried out with a boost pressure ratio of 1.5, where two operating modes were considered: a maximum torque operating mode and a maximum power operating mode. The study aimed at assessing the variation of the stress state on the crankshaft when operating under turbocharged conditions and determining the regions where the stress concentration is maximum. The results of the analysis are used to evaluate the crankshaft's operational integrity, and to suggest remedial activities or changes in the crankshaft's structure for areas determined to be vulnerable to structural failure.

2. Methodology

2.1 Loads Acting on the Crankshaft

Diagrams of the acting forces and moments are used in the analysis of crankshaft durability for determining the operating loads of the crankshaft (Kirkpatrick, 2020; Viteri Barrionuevo et al., 2025). These loads include tangential and normal forces, as well as inertial forces from rotating and reciprocating masses and reaction forces from the main bearings. Under the combined effect of these load components, the crankshaft is subjected to simultaneous bending and torsion, resulting in a complex stress state during operation. This forms the basis for developing a finite element model to analyze stresses and verify the crankshaft's structural integrity. The forces acting on the crankshaft are defined as follows in **Figure 1**.

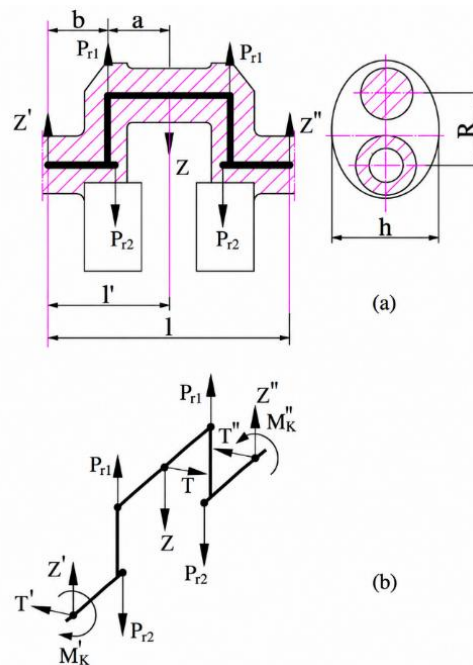


Figure 1. Schematic diagram of forces and moments acting on the crankshaft: (a) in the plane; (b) in space.

T and Z : Tangential force and normal force acting on the crankpin;

P_{r1} : Centrifugal inertia force of the crank web;

C_1 : Centrifugal inertia force of the crankpin;

C_2 : Centrifugal inertia force of the equivalent mass referred to the center of the big end of the connecting rod;

P_{r2} : Centrifugal inertia force of the counterweight;

Z', Z'' : Normal reaction forces at the left and right main bearings, respectively;

T', T'' : Tangential reaction forces at the left and right main bearings, respectively.

The torque acting on the left main journal, denoted as M'_K , is generated by the tangential forces of the crank throws located ahead of the crank under consideration. If the analyzed crank corresponds to index i , the torque can be expressed as:

$$M'_K = \sum T_{i-1} \cdot R \quad (MN.m) \quad (1)$$

where, $\sum T_{i-1}$ is the algebraic sum of the tangential forces of the crank throws located ahead of the crank with index i , and R represents the crank radius.

The torque acting on the right main journal (rear journal), denoted as M''_K , is generated by the tangential forces from the first crank to the crank with index i . Accordingly, it is given by:

$$M''_K = \sum T_i \cdot R = M'_K + T \cdot R \quad (MN.m) \quad (2)$$

where, T is the tangential force acting on the crank with index i .

The bending moment acting on the crankpin is calculated as:

$$M_u = Z'l' \quad (MN.m) \quad (3)$$

2.2 Engine Model and Turbocharging Configuration

To simulate the operating characteristics of the gasoline engine, a one-dimensional engine model was established using AVL-Boost software (AVL List GmbH, 2021). The model corresponds to a port fuel injection gasoline engine configuration and was developed to predict the engine performance parameters under different operating conditions, including turbocharged operation.

The fundamental engine input parameters required for the AVL-Boost model are summarized in **Table 1**. These parameters describe the main geometric dimensions and rated performance characteristics of the engine and serve as the basis for subsequent combustion, gas exchange, and load simulations.

Table 1. Basic engine specifications.

Parameter	Symbol	Value	Unit
Piston stroke	S	84.7	mm
Cylinder bore	D	75	mm
Number of cylinders	i	4	–
Rated power at $n = 6000$ rpm	$N_{e,max}$	80	kW
Maximum torque at $n = 4200$ rpm	$M_{e,max}$	140	Nm
Minimum specific fuel consumption	g_e	244	g/kW·h
Compression ratio	ε	10.5:1	–

The developed AVL-Boost model has been used to generate in-cylinder pressure histories and inertial load data which were needed as input for the subsequent finite element analysis of the crankshaft under naturally aspirated and turbocharged conditions. The simulation results for the power and Brake Specific Fuel Consumption (BSFC) of the naturally aspirated engine at the full load are shown in **Figure 2**, and compared with the manufacturer's published specifications. The results show the difference between the power

simulated and the reference value to be about 3.5% on average, throughout the engine speed range. As for BSFC, the average deviation between simulation results and manufacturer's data is 3.26% for the whole operating range. Both deviation values are below 5%, demonstrating the high accuracy and reliability of the developed engine model. Consequently, this model meets the requirements for determining the input load parameters needed to construct and analyze the finite element model of the crankshaft.

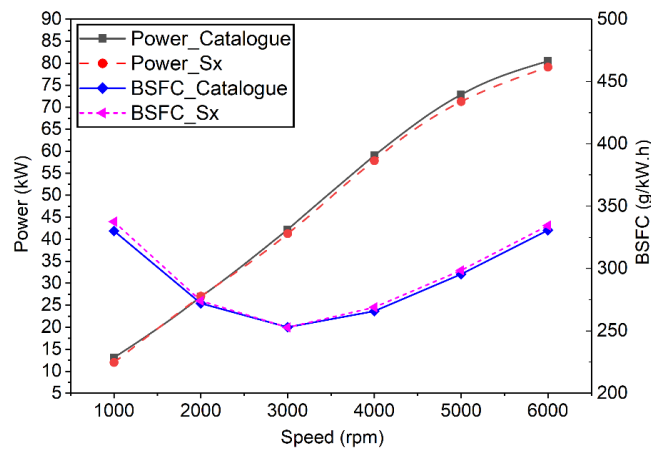


Figure 2. Simulated engine power and specific fuel consumption.

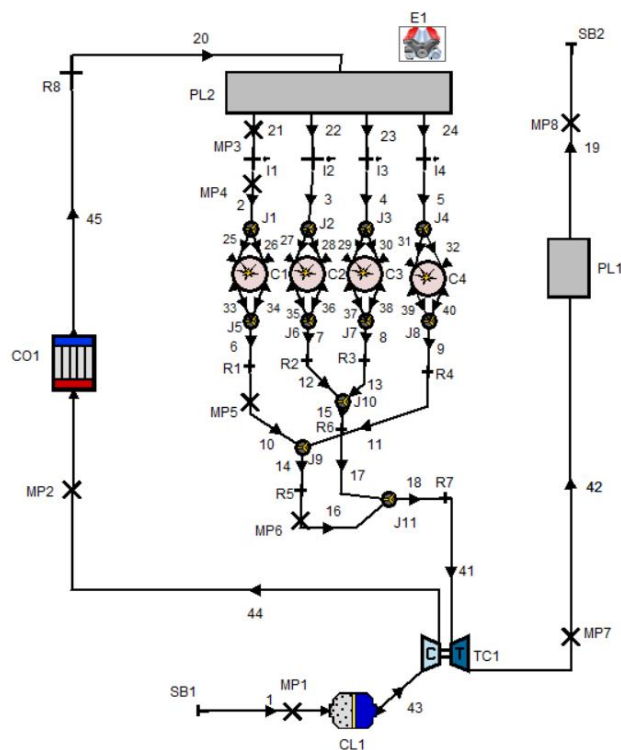


Figure 3. Turbocharged engine model in AVL-Boost.

Based on this validated naturally aspirated configuration, a corresponding turbocharged model was subsequently developed, as shown in **Figure 3**. A boost pressure ratio of 1.5 was selected as a representative moderate-boost condition for evaluating the structural response of the original crankshaft without geometric modification. This operating condition is considered representative and suitable for evaluating the structural performance of the crankshaft under turbocharged conditions.

2.3 Load Extraction and Boundary Conditions

The tangential (T) and normal (Z) forces acting on the crankpin are evaluated from the corresponding force diagrams or calculated using the analytical formulations presented in the engine dynamic analysis (Kirkpatrick, 2020; Viteri Barrionuevo et al., 2025).

$$Z = P_{\Sigma} \frac{\cos(\alpha+\beta)}{\cos\beta} \cdot F_p \quad (4)$$

$$T = P_{\Sigma} \frac{\sin(\alpha+\beta)}{\cos\beta} \cdot F_p \quad (5)$$

F_p : Piston crown area.

P_{Σ} : Centrifugal inertial force of the crank web and the counterweight.

α : Crankshaft rotation angle.

β : Connecting rod oscillation angle.

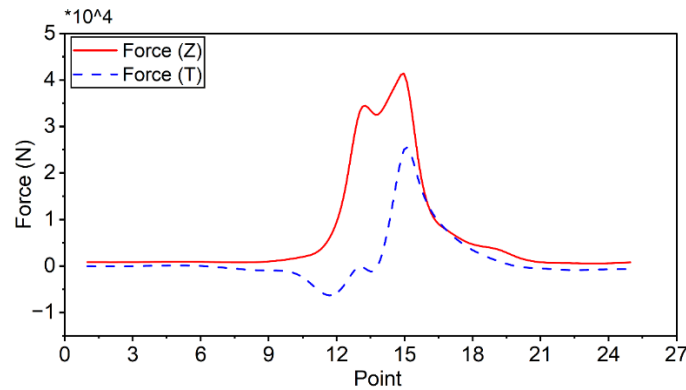


Figure 4. Tangential force (T) and normal force (Z) at an engine speed of 4200 rpm.

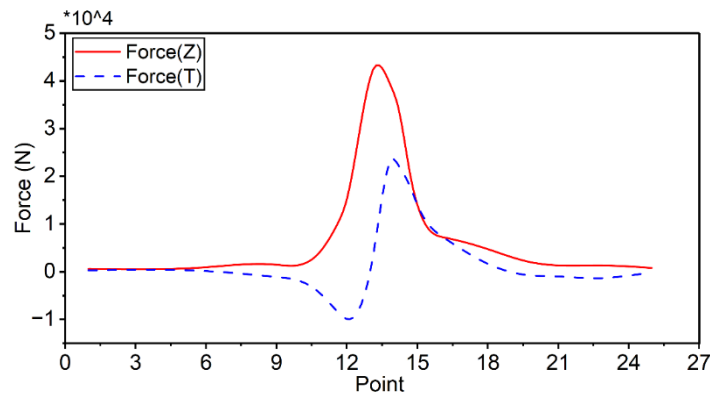


Figure 5. Tangential force (T) and normal force (Z) at an engine speed of 6000 rpm.

Figures 4 and 5 present the variation of tangential (T) and normal (Z) forces with crank angle at engine speeds of 4200 rpm and 6000 rpm, respectively. These forces exhibit cyclic behavior due to the combined effects of combustion and inertia. The tangential force is mainly responsible for torque generation, whereas the normal force contributes to bending loads on the crankshaft. Variations in these forces with engine speed directly affect the loading conditions and provide a basis for identifying critical cases in the finite element analysis.

2.4 Finite Element Model of the Crankshaft

2.4.1 Geometric Design and Material of the Crankshaft

The crankshaft of the four-cylinder gasoline engine is a single piece forged part that includes the following key components: the front end, eight crank webs, four counterweights, four main journals, and the rear end. The front end of the crankshaft has a keyway to mount the drive pulley that drives the cooling fan, alternator and water pump. It also has a crankshaft gear to drive the camshaft and other auxiliaries. Further, the end of the crankshaft is provided with an axial displacement control mechanism and an oil seal to minimize oil leakage at that location. The crank webs have an oval shape and provide the necessary rigidity and load-bearing capacity of the crankshaft during operation. This shape also allows the stress to be distributed more optimally, so that the stress concentration can be reduced if the crankshaft is subjected to bending and torsional moment simultaneously. To balance the inertial forces of rotating masses, counterweights are installed on the crank webs, which helps to reduce the load on the main bearings and prolong the life of the crankshaft.

To minimise the friction of working surfaces, lubricating oil is fed to the main journals and crankpins via a mixture of splash lubrication and forced lubrication, which is provided by an oil pump through internal oil passages in the crankshaft. The lubrication system provides an uninterrupted flow of oil to the friction surfaces, reducing wear, heat generation and improving the life span of the crankshaft during use.

Table 2. Geometric parameters of the engine crankshaft.

Parameter	Symbol	Value	Unit
Main journal diameter	d_{ct}	46	mm
Main journal length	l_{ct}	22	mm
Crank pin diameter	d_{ck}	39.5	mm
Crank pin length	l_{ck}	18.5	mm
Crank web width	h_{mk}	112	mm
Crank web thickness	b_{mk}	18	mm
Fillet radius of crank web	r_{mk}	3	mm
Crank radius	ρ_{mk}	26	mm
Counterweight radius	ρ_{dt}	60	mm
Overlap	ϕ	4	mm
Distance from journal center to critical section	r	22	mm

Geometric parameters of the crankshaft are shown in **Table 2**. These parameters were used to create a 3D model of the crankshaft, which is displayed in **Figure 6**. With this model, the authors will carry out calculations and a strength analysis of the crankshaft by the finite element method.

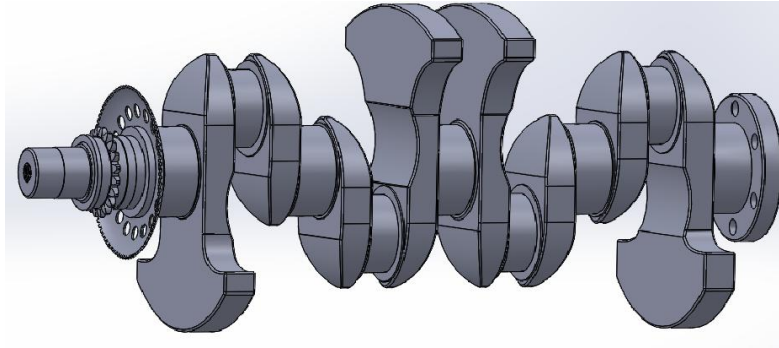


Figure 6. Final crankshaft model.

2.4.2 Material Properties and Failure Criterion

The alloy steel used for the manufacturing of the crankshaft in this study is 40XHMA alloy steel. This is a medium-carbon steel alloyed with chromium (Cr), nickel (Ni), and molybdenum (Mo) to improve its mechanical properties. Following heat treatment, the material exhibits the necessary strength, toughness, and fatigue resistance to meet the operational requirements of a crankshaft subjected to cyclic bending and torsional loads. Therefore, 40XHMA steel is widely used to produce crankshafts of internal combustion engines.

The mechanical properties of steel are as much a function of the manufacturing and heat treatment process as of the chemistry of the steel itself. The 40XHMA steel after quenching and tempering has a tempered martensite structure, which can improve the tensile strength and yield strength of the steel and ensure the necessary ductility (Songsong et al., 2021). The material has a medium carbon content and can be surface strengthened, e.g. by induction hardening and nitriding. These techniques are used to increase the hardness, wear resistance and fatigue strength of the high-stress surfaces. In addition, molybdenum (Mo) can reduce the susceptibility to temper brittleness and maintain the mechanical properties at high operating temperature, and chromium (Cr) increases wear resistance of contact surfaces especially the journals and bearings (Que et al., 2023).

In accordance with material published in the existing literature (Kirkpatrick, 2020; Viteri Barrionuevo et al., 2025), this study uses a permissible stress of 600 MPa for the 40XHMA alloy steel in order to assess the material's durability. Based on this value, the von Mises equivalent stress values from the simulation are compared to assess the load-bearing capacity of the crankshaft under the operating conditions studied.

2.5 Boundary Conditions and Mesh Generation

2.5.1 Boundary Conditions

In operation, the crankshaft is under gas forces caused by the combustion process and inertial forces from the reciprocating and rotating masses. Gas forces are transmitted from the piston through the connecting rod to the crankpin and the inertial forces depend on the angle of the crankshaft and its speed of rotation. These force components combine to cause bending and torsional moments in the crankshaft.

The tangential force (T) and normal force (Z) of the working cycle of the engine are calculated as a function of the crankshaft angle for each of the operating modes. The T component primarily generates torsional moment, whereas the Z component generates bending moment. The two forces are applied on the crankpin surface of the finite element model as per the scheme shown in **Figure 7**.

The main journals are used to define boundary conditions for the model to simulate the support from the engine bearings. In addition, the rotational speed corresponding to each operating mode and the transmitted torque are specified at the crankshaft output end. These boundary conditions are very similar to the actual working state of the component during power transmission (**Figure 8**).

Analysis is made in two engine operating modes of high load: at maximum torque (4200 rpm) and at maximum power (6000 rpm). For each of the speeds, the model is run for the naturally aspirated case and then for the turbocharged case. The tangential and normal force values are calculated according to the crankshaft angle to be able to determine the critical loading state for analysis. This allows the comparison of the differences in peak stress between operating modes and the evaluation of the impact of increased pressure on crankshaft durability. The boundary conditions and loads used in the model are shown in **Figures 7 and 8**. The above are the foundation for the analysis of von Mises equivalent stress distribution and the determination of high-stress concentration areas of crankshaft.

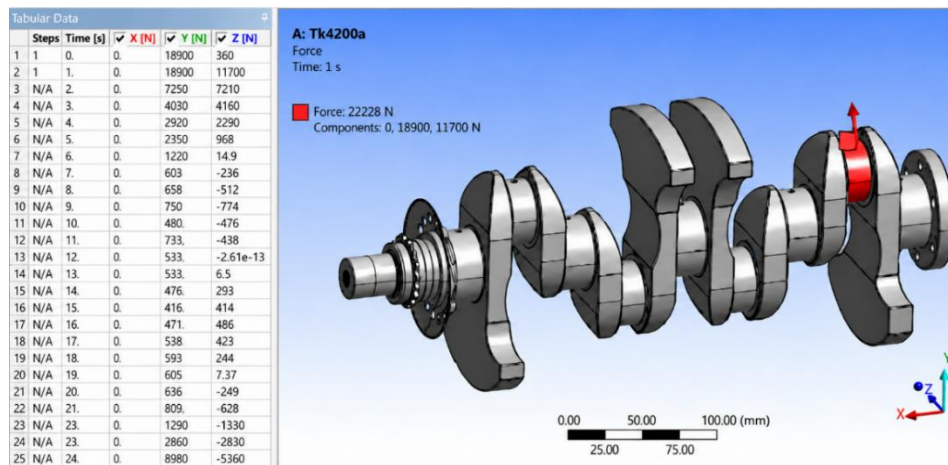


Figure 7. Definition of the forces acting on the crankshaft crankpin.

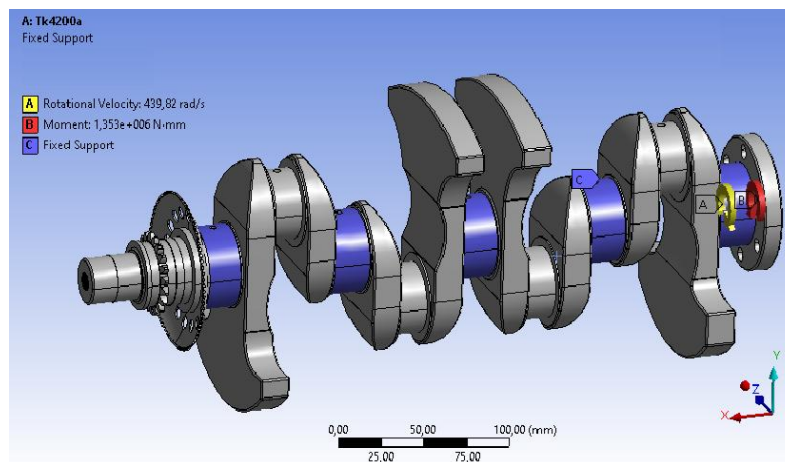


Figure 8. Definition of the forces acting on the main journal of the crankshaft.

2.5.2 Mesh Generation

The crankshaft geometry is quite complicated, consisting of main journals, crankpins, crank webs, counterweights and transition zones. Due to the different geometric features of the areas, the respective meshing techniques were used for each location. The model was meshed with solid elements in 3D. Swept hexahedrals were used for areas with relatively simple geometry like the main journals and crankpins. On the other hand, other more complex regions, such as crank webs, counterweights and transition zones were meshed with tetrahedrons. The combination of these two types of elements provides an accurate geometric representation with a relatively small number of elements, which can improve the computational efficiency and accuracy.

The meshing was done separately at the transition zones between the crankpins and crank webs, and between the main journals and crank webs. When the crankshaft is subjected to the combined influence of bending and torsional loads, these areas have a tendency to develop stress concentrations. For this reason, the element size used in these areas is smaller than in other areas. In this way, the mesh is refined to provide a better representation of stress gradients, and the reliability of the calculated maximum von Mises equivalent stress is increased.

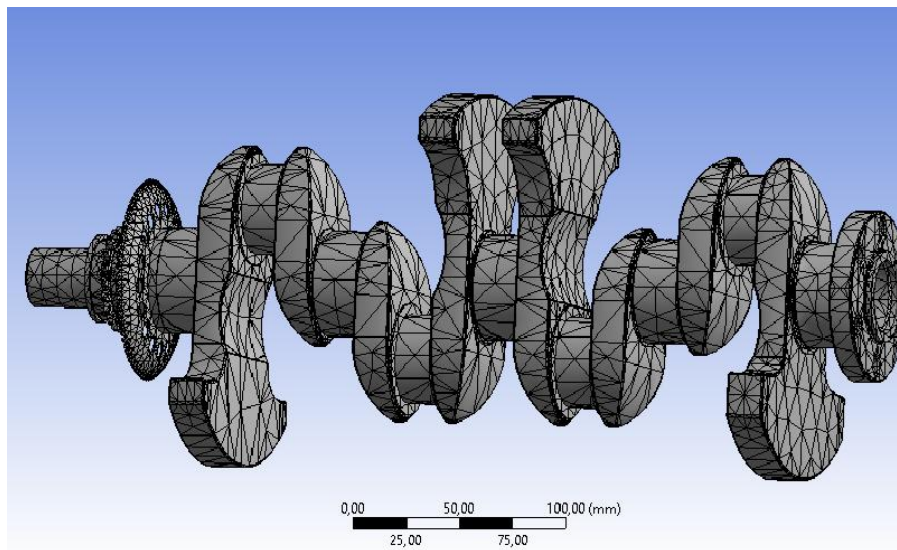


Figure 9. Meshing of the crankshaft model.

The completed finite element model, shown in **Figure 9**, contained approximately 480,000 elements and 720,000 nodes. The element size in the main body of the crankshaft ranged from 2.5 to 3.0 mm, whereas a smaller size of 0.8–1.2 mm was used at the critical fillet regions. This mesh arrangement was selected to provide sufficient resolution in the areas of stress concentration while maintaining a reasonable computational cost (Liu et al., 2020).

3. Results and Discussion

3.1 Results

During engine operation, the two most extreme operating scenarios are observed at engine speeds of 4200 rpm and 6000 rpm under 100% loading condition based on **Figures 10-13**, with more localized stress

amplification in the critical regions shown in **Figure 14**. These two cases correspond to conditions of maximum torque and maximum power respectively (Kirkpatrick, 2020; Tsitsilonis and Theotokatos, 2021; Zhang and Zheng, 2023). Therefore, in evaluating the structural strength of the crankshaft for a turbocharged gasoline engine, the authors focus on these two operating conditions.

The stress distribution along the crankshaft for various engine speeds under turbocharged condition is shown in **Table 3**, which shows that there is an overall increase in stress values under turbocharged condition. This result is consistent with previous investigations, where turbocharging enhances the in-cylinder pressure results in increased mechanical loadings experienced by the crankshaft (Almoslehy and Hadi, 2024; Elmoselhy et al., 2022).

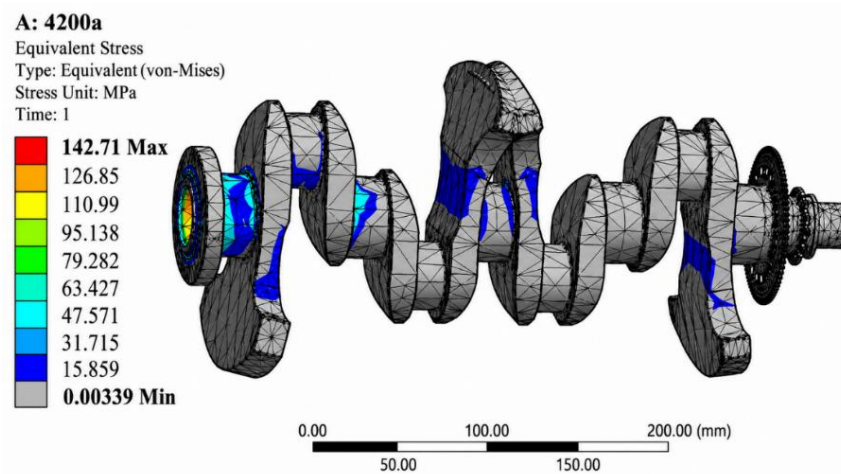


Figure 10. Stress diagram at 4200 rpm without turbocharging.

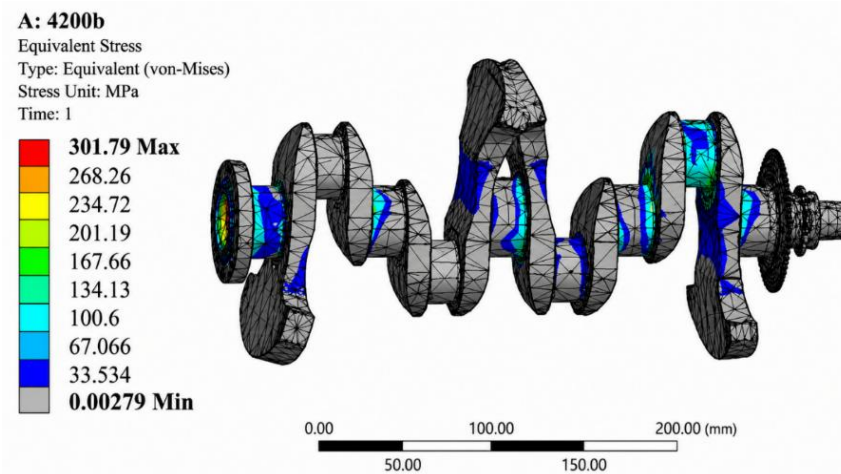


Figure 11. Stress diagram at 4200 rpm with turbocharging.

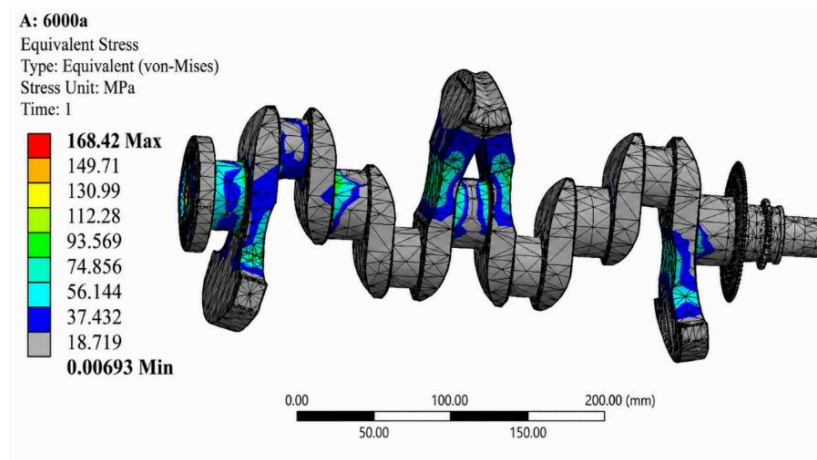


Figure 12. Stress diagram at 6000 rpm without turbocharging.

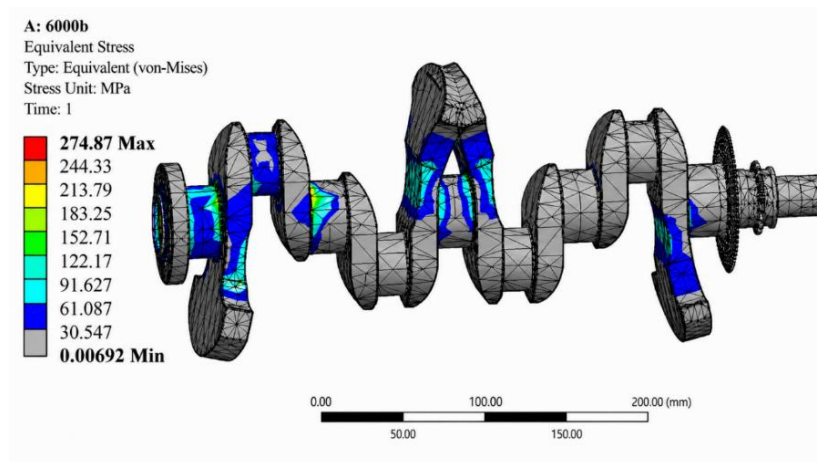


Figure 13. Stress diagram at 6000 rpm with turbocharging.

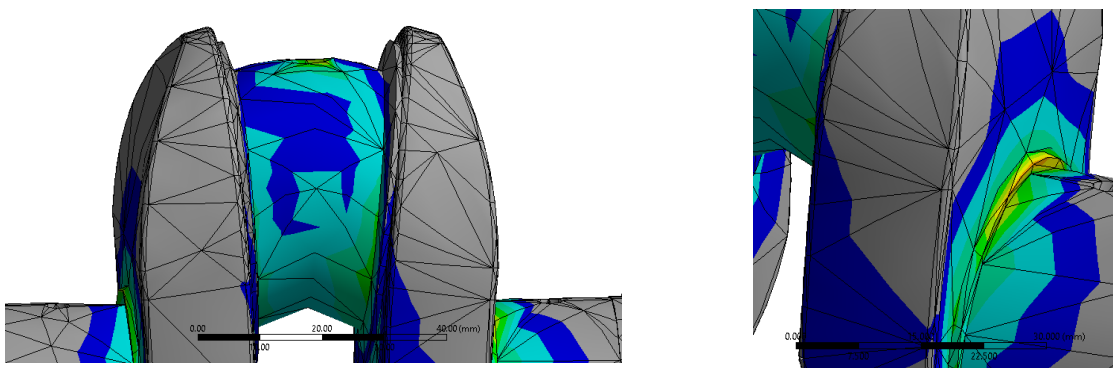


Figure 14. Stress distribution at the most critical location on the crankshaft at 4200 rpm with turbocharging.

The locations of the maximum stress are predominantly concentrated on the contact points of the crank web and crank pin, and crank web and main journal respectively. These regions show abrupt geometric changes, especially at the fillet transition points which amplify the stress concentration when subjected to combined bending and torsional loading condition. Similar behavior has been reported extensively in previous literature investigating crankshaft strength and fatigue where the fillet regions have been identified as the most critical locations for crack initiation due to local stress amplification (Que et al., 2023; Songsong et al., 2021; Tian et al., 2023; Xu et al., 2025).

Under all investigated cases, the stress value under different locations of crankshaft journal is different depending on the operating conditions. This difference can be clearly observed by examining the stress contour legend where the red-colored regions correspond to maximum stress levels representing the most critical and hazardous locations of the crankshaft (Ang and Ku, 2021; Liu et al., 2020).

Table 3. Comparison of crankshaft stress values at different engine speeds.

Engine speed (rpm)	Condition	Max stress (MPa)	Stress increase (MPa)	Increase (%)
4200	Naturally aspirated	142.71	–	–
4200	Turbocharged	301.79	159.08	111.5
6000	Naturally aspirated	168.42	–	–
6000	Turbocharged	274.87	106.45	63.2

Based on the results of the stress results shown in **Table 3**, the crankshaft stress under dynamic loading condition is evaluated using the safety factor approach. In this study, based on the extreme operating environment of the crankshaft, the safety factor of 1.7 has been adopted (Kirkpatrick, 2020; Songsong et al., 2021; Viteri Barrionuevo et al., 2025). As such, the corresponding stress under dynamic loading acting on the crankshaft was calculated as:

$$\sigma_{dynamic} = 1.7 * 301.79 = 513.04 \text{ MPa}.$$

3.2 Discussion

Analysis results indicate that when the engine operates with a boost pressure ratio of 1.5, the von Mises equivalent stress on the crankshaft increases under both investigated operating conditions compared to the naturally aspirated engine. The primary cause is the rise in in-cylinder combustion pressure, which increases the gas force acting on the piston. This force is transmitted via the connecting rod to the crankpin, simultaneously increasing the bending and torsional moments acting on the crankshaft (Almoslehy and Hadi, 2024; Jia et al., 2018; Zhang and Zheng, 2023; Zhu et al., 2023a).

From the stress distributions in **Figures 10-14**, it is observed that the maximum stresses are always at the fillet junction of the crankpin and the crank web and at the fillet junction of the main journal and the crank web. These locations feature abrupt changes in cross-section, making them prone to stress concentration when the component is subjected to simultaneous bending and torsional loads. The results are in agreement with previous simulation and experimental works (Que et al., 2023; Songsong et al., 2021; Xu et al., 2025). Thus, it was decided to refine the finite element mesh in these transition zones to more accurately reflect the stress changes and to improve the confidence of the results obtained.

The maximum equivalent stress increased by 111.5% from 142.71 MPa to 301.79 MPa after the boost was applied at maximum torque condition (i.e. engine speed 4200 rpm). This rise shows that the crankshaft stresses are largely dependent on gas loads at the mid-range of the engine's operating speed, when it is operating under high loads (Dagna et al., 2021; Tsitsilonis and Theotokatos, 2021).

The maximum power operating point was 6000 rpm, at which the maximum stress increased from 168.42 to 274.87 MPa, or 63.2%. Stress increased considerably after supercharging, but the rate of increase was lower than that observed at 4200 rpm. This means that as the engine speed increases, the effects of the inertial forces created by reciprocating and rotating masses increase. Thus the forces applied on the crankshaft are not only due to the combustion pressure in the cylinders but also to the inertial forces. This trend has also been observed in studies regarding crankshaft dynamics and vibration at high speeds (Sun et al., 2022; Xiang et al., 2026; Zhu et al., 2023a).

The design safety factor of 1.7 was used in the calculations to assess the structural integrity of the crankshaft. The maximum equivalent stress is 301.79 MPa, and the design stress is 513.04 MPa. This value is still less than the permissible stress of 600 MPa for 40XHMA alloy steel material. Thus, the current crankshaft design meets static strength requirements under the investigated supercharging conditions. But, these are only results of static load-bearing capacity and do not consider the effects of cyclic loading on fatigue life. In practice, the increase in stress amplitude resulting from supercharging can reduce component service life, even though peak stresses remain within permissible limits.

Many investigations have shown that fatigue cracks usually start in transition fillets, where there is stress concentration (Cho et al., 2025; Infante et al., 2019). For this reason, apart from providing structural strength, it is necessary to pay attention to enhancing the surface quality in these zones. Surface hardening methods like induction hardening or nitriding can be employed to increase the surface hardness, create compressive residual stresses, and boost the crankshaft's fatigue resistance (Que et al., 2023; Sun et al., 2022). Wear and surface degradation at the crankshaft–bearing interfaces should also be taken into account as they might affect the loading and support conditions during long engine operation (Elmoselhy et al., 2022; Zhu et al., 2023b).

For a higher boost pressure ratio, the present crankshaft may require further assessment rather than direct use without modification. Possible measures include increasing the fillet radius, improving the surface-hardening treatment, or applying fillet rolling to reduce the effective stress concentration and improve fatigue resistance (Bulut et al., 2021; Luo et al., 2026; Wang et al., 2023). These measures should be evaluated together with changes in crankshaft mass, bearing load, manufacturability, and engine balance.

Overall, linking the AVL-Boost engine model with the ANSYS structural model provides a more representative definition of crankshaft loading than the use of a single assumed cylinder-pressure value. The proposed approach makes it possible to relate changes in engine operating conditions directly to the resulting crankshaft stress. It can therefore support the preliminary evaluation of an existing crankshaft when a naturally aspirated gasoline engine is converted to turbocharged operation and provide a basis for subsequent fatigue analysis and structural optimization (Yang et al., 2025; Zheng et al., 2022).

4. Conclusion

This study evaluated the load-bearing capacity of a gasoline engine crankshaft following turbocharging, utilizing the 3D finite element method. Loads on crankshaft were calculated by using a boost pressure ratio of 1.5 and two characteristic operating modes: maximum torque and maximum power.

Turbocharging caused a substantial rise in the von Mises equivalent stress of the crankshaft as compared to a naturally aspirated engine. At the maximum torque mode (4200 rpm), the peak stress increased from 142.71 MPa to 301.79 MPa, representing an increase of 111.5%. At the maximum power mode (6000 rpm), this value increased from 168.42 MPa to 274.87 MPa, an increase of 63.2%. It was found that the rise at

4200 rpm is higher, showing that combustion pressure is the major influence on the crankshaft load when the engine is run in a maximum torque mode.

In either operating mode, the stress concentrations were at the fillet areas between the crankpins and the crank webs and between the main journals and the crank webs. Fatigue failure is possible at these locations and therefore needs to be carefully considered at the time of design, manufacture and surface treatment. A design safety factor of 1.7 was applied, the calculated stress was 513.04 MPa, which is less than 600 MPa, the permissible stress limit for 40XHMA alloy steel. This shows that the existing crankshaft design is adequate in terms of strength under the turbocharged condition of a boost pressure ratio of 1.5. The AVL-Boost engine model can be coupled with a finite element model, to allow a relatively accurate calculation of the loads on the crankshaft and an evaluation of the effects of supercharging on the stress state of the crankshaft. The results serve as a basis for a preliminary assessment of the crankshaft's operational capability prior to supercharging and to support further studies on fatigue strength, dynamic loading, and structural optimization aimed at enhancing the crankshaft's durability and service life.

Conflicts of Interest

The authors confirm that there is no conflict of interest to declare for this publication.

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AI Disclosure

The author(s) confirm that generative AI tools were not used during the preparation of this article.

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