Optimal Design of Crankshaft for a Single-cylinder Diesel Engine based on ANSYS

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Abstract

The present paper discusses the critical prerequisites to improve lightweight design and structural reliability of the crankshaft of the single-cylinder diesel engines. Using ANSYS Workbench finite element analysis software, a three-dimensional solid model of the crankshaft is created and then static finite element analysis is done to determine the stress distribution when subjected to working conditions under real-life conditions. As shown by the findings, the highest value of the equivalent stress of the original crankshaft is 100.16 MPa and the least value of the safety factor is only 2.49. Moreover, there is a high level of stress concentration where the crankshaft meets both the main journal and crank pin, which can cause fatigue failure after many operating hours and negatively impact the lifespan of the diesel engine. These problems are solved with a specific optimization strategy, taking into account the features of stress distribution. Stress concentration regions are relieved using transition fillets to reduce local stress fluctuations. Also, the stress relief slots are provided at the crank arm to distribute the concentrated stress. At the same time, the diameter of the crank pin and main journal was optimized to make light weight at the cost of structural integrity. After optimization, the static finite element analysis was carried out to confirm the outcomes. It was found that the maximum equivalent stress of the crankshaft has been minimized to 87.713 MPa and the smallest safety factor was raised to 2.85, which greatly improved the reliability of structures. Simultaneously, the total mass of the crankshaft was decreased by 9.3% thus accomplishing the aim of light weighting. This study successfully resolves the trade-off between lightweight design and reliability of the crankshaft of the single-cylinder diesel engine by combining finite element analysis with structural optimization. The suggested optimization scheme is scientifically justified and feasible, and can be used as a dependable engineering guide to the best design of a crankshaft of a single-cylinder diesel engine, and it can be regarded as very valuable to improve the performance of diesel engines in general.

Keywords

Crankshaft; Finite Element; Optimal Design; Statics; Light Weighting.

1. Introduction

Diesel engines with their reputation of high efficiency and reliability are widely used in various fields like agricultural machinery, construction machines and small generator sets. The operation stability, power performance, and service life of such engines has direct impact on the efficiency of operation and maintenance expenses of the relevant equipment[1]. The crankshaft is the main carrying and transmitting part of the connecting rod assembly of a single-cylinder diesel engine. It plays a central role in converting the gas explosion pressure of the piston into rotational torque to be delivered to the outside world. At the same time, it also actuates the valve train and other auxiliary parts of the diesel engine, such as the fuel pump and the water pump. Therefore, it becomes an important factor in the transformation of thermal energy into mechanical energy in the diesel engine[2].

When a single-cylinder diesel engine is running, the crankshaft is subjected to a constant combined load consisting of cylinder burst pressure, reciprocating inertial force, rotational inertial force, and bearing reaction force. It is thus kept in the long-term multi-stress coupling condition of bending, torsion, and shearing. The situation increases the probability of stress concentration at the junctions of the crank pin and the crank arm, the main journal and the crank arm. These stress concentrations may cause fatigue cracks, plastic deformation and possibly fracture failure, directly compromising the safety and integrity of the whole machine[3,4].The traditional structure of the crankshaft is heavily based on engineering intuition and similarity procedures that make it challenging to precisely match the pattern of stress distribution with structural strength demands and lightweight goals. This dependence frequently leads to problems like superfluous materials, overabundance of structural weight, and significant stress concentration, thus drastically limiting the overall performance improvement of diesel engines[5].

The research paper centers on the crankshaft of a designated single-cylinder diesel engine. Three dimensional solid models are done through Solid Works software and static finite element analysis is conducted in ANSYS Workbench to determine the regions of stress concentration and the weak strength links of the crankshaft[6].The application of transition fillets, the implementation of unloading grooves, and other optimization methods of the structure, as well as the optimized design of the crank pin and main journal diameter, can help to lower the stress intensity of the crankshaft and its structural weight. The improvements provide a trustful simulation basis and can be used as an effective technical guide in the development of lightweight and highly reliable single-cylinder diesel engine crankshaft designs[7].

2. Building the Model

2.1 Crankshaft Finite Element Solid Model

Table 1. Crankshaft Basic Parameters

Parameters Name	Crankshaft length/mm	Main bearing diameter/mm	Crankshaft diameter/mm	Crankshaft weight/kg
numerical value	230	45	45	4.7082

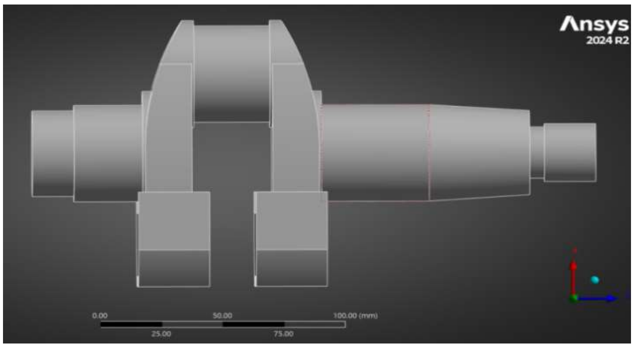

Figure 1. Three-dimensional model of the crankshaft

The 3D software that has been used in modeling the crankshaft is SolidWorks and the analysis by finite elements was done with Ansys Workbench [8]. The nature of diesel engine crankshafts is very complex. The mesh size was minimized to improve mesh quality and calculation complexity was minimized with suitable changes to the crankshaft model. The modified crankshaft model does not contain the features like chamfers, oil holes, and threads which have lower stress. Such a simplification increases the efficiency of finite element analysis applied to the crankshaft[9].Table 1

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presents the main parameters of the crankshaft, and Figure 1 illustrates the three-dimensional shape of the crankshaft.

2.2 Crankshaft Material and Attribute Settings

The material that forms the crankshaft of the single-cylinder diesel engine considered in this paper is structural steel and its material characteristic parameters are presented in Table 2.

Table 2. Properties of Crankshaft Materials

designation	materials	material density/kg/m3	yield strength/Mpa	tensile strength/Mpa	poisson ratio
numerical value	constructional steel	7850	250	460	0.3

2.3 Analysis of Crankshaft Forces

The crankshaft of a single-cylinder diesel engine primarily experiences gas pressure, reciprocating inertial forces, rotational inertial forces, and bearing support forces. These factors lead to the generation of loads, including bending, torsion, and shearing. A thorough understanding of the force conditions acting on the crankshaft is essential for analyzing its strength[10].

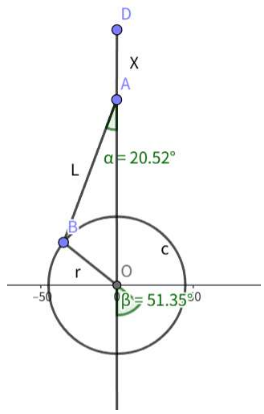

Figure 2. Crank and Connecting Rod Mechanism

Figure 2 illustrates the mechanism of the crankshaft connecting rod. Point O serves as the rotation center of the crankshaft, point A represents the contact point between the front end of the connecting rod and the piston pin, and point B denotes the contact point between the rear end of the connecting rod and the crankshaft.The crank OB rotates at a constant speed around point O, which serves as the center of rotation, while the piston A oscillates along the central line segment, exhibiting a stroke length of X.In the figure, α denotes the clockwise rotation angle of the connecting rod axis relative to the cylinder axis, β represents the crank rotation angle, and r indicates the rotation radius. To elucidate the magnitude of the pressure exerted by the cylinder on the crank, it is essential to determine the magnitudes of the piston displacement, velocity, and acceleration.The specific calculation is as follows:

piston displacement:

$$x = r\left(1 - \cos\alpha + \frac{\lambda}{2}\sin^2\alpha\right) \tag{1}$$

In the formula, λ represents the link ratio and its value ranges from 0.26 to 0.31.piston velocity:

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$$v = r\omega \sin\alpha + r\omega \frac{\lambda}{2} \sin 2\alpha \tag{2}$$

In the formula, ω represents the rotational acceleration of the crank:

$$a = r\omega^2 \cos\alpha + r\omega^2 \lambda \cos 2\alpha \tag{3}$$

The force exerted by the piston gas varies periodically with the angle of crankshaft rotation, reaching its peak near the top dead center, where the stress on the crankshaft is maximized[11]. At the starting position of the cylinder, the piston is at the top dead center, and the crankshaft experiences its maximum tensile stress:

$$F_A = \frac{\pi D^2}{4} P_z \tag{4}$$

In the formula,F_AThe formula represents the maximum pressure exerted on the crankshaft (in units of Newtons).P_zThe maximum explosion pressure of the diesel engine cylinder (in MPa); D is the diameter of the diesel engine cylinder (in mm).

The basic parameters of the single-cylinder oil extraction machine are shown in Table 3.

Table 3. Basic Parameters of Diesel Engines

designation	cylinder bore/ mm	route/ mm	displacement/ L	rated speed/ r/min	Maximum explosive pressure/Mpa
numerical value	95	115	0.815	3600	7.5

Based on the above basic parameters, the maximum compressive stress exerted on the crankshaft can be calculated:

$$F_A = \frac{\pi D^2}{4} P_z = 53161.6372(\mathrm{N})$$

2.4 Setting of Boundary Conditions at the Root

In this study, a detailed grid division was performed in regions experiencing higher stress, utilizing tetrahedral elements with a size of 4 mm and a resolution for size adjustment of 4. Following the solid mesh division, a total of 44,807 nodes and 26,813 elements were generated. The mesh division of the crankshaft is illustrated in Figure 3.

Cylindrical supports are incorporated into the cylindrical surfaces of the two primary journals of the crankshaft to stabilize radial movement while allowing for axial and tangential movements. Remote displacement constraints are applied to prevent self-rotation on the cylindrical surface of the crankshaft's output, with the displacement along the crankshaft's axial direction being fixed[12,13].

Displacement constraints should be imposed to restrict any movement at the end faces of the crankshaft. A downward bearing load, equal to the maximum compressive stress on the crankshaft, which is 53,161 N, will be applied to the crankpin. Additionally, a rotational speed of 1000 rad/s will be assigned to the crankshaft journal to simulate the crankshaft's rotation. The constraints and loads applied to the crankshaft are illustrated in Figure 4.

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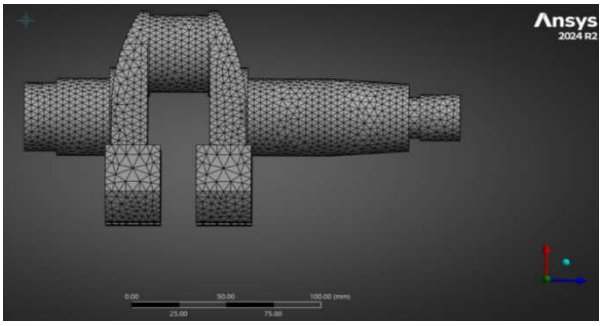

Figure 3. The crankshaft after the grid division

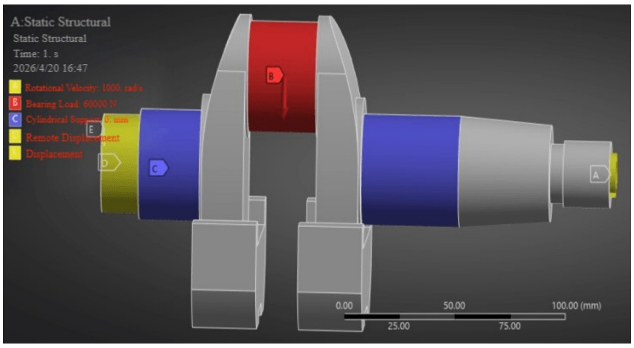

Figure 4. Crankshaft model after loading force application

3. Finite Element Results and Analysis

As can be seen in Figure 5, the equivalent stress in the crankshaft is analyzed. According to the data, the greatest stress is at the point where the connecting rod and the crankpin meet, and then the stress at the edge between the main journal and the crankshaft. The highest equivalent stress observed is 100.16 MPa which is considerably lower compared to maximum yield strength of the structural steel used in the crankshaft material which is 250 PMPa. As a result, the crankshaft has sufficient strength to meet the required standards.

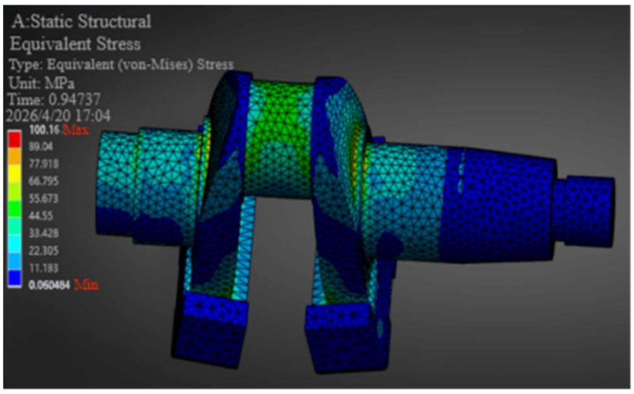

Figure 5. Stress distribution map of crankshaft equivalent stress

The equivalent elastic strain cloud diagram of the crankshaft is given in Figure 6. The diagram shows that the elastic strain is mainly located at the interface between the crankshaft and the main journal with the maximum equivalent elastic strain being 0.00077 mm. Such a small amount of elastic deformation meets the strength demands of the crankshaft.

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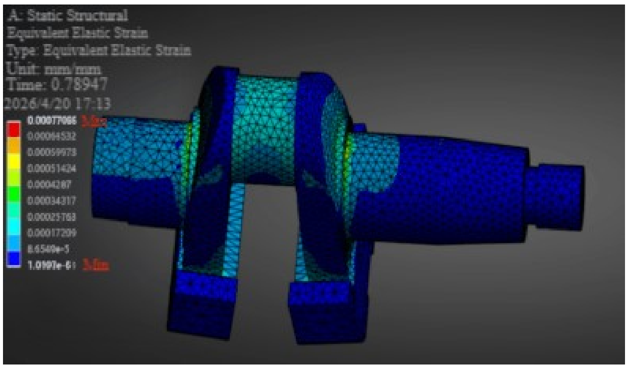

Figure 6. Equivalent elastic strain cloud map of the crankshaft

Figure 7 presents the cloud map of the crankshaft safety factor. The minimum safety factor, as indicated in the figure, is 2.4959, which exceeds the threshold of 1.5, thereby satisfying the requirements for the crankshaft safety factor.

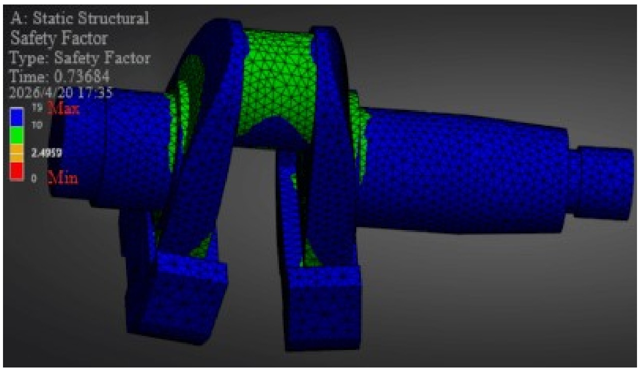

Figure 7. Cloud Chart of Crankshaft Safety Factor

4. Optimization Design

4.1 Establishment of the Objective Function

Take the main parameters of the crankshaft: the diameter of the crankshaft crank pin and the diameter of the crankshaft journal as the design variables. That is:

$$X = [B \quad D]^T = [x_1 \quad x_2]^T \tag{5}$$

B represents the diameter of the crank pin, denoted as X1, and D represents the diameter of the crankshaft journal, denoted as X2.

Reducing the crankshaft's mass can decrease the overall mass of the diesel engine, provided that the maximum equivalent stress the crankshaft can withstand is not exceeded[14]. This study considers the crankshaft's mass as the objective function, whereby:

$$m = \rho v \tag{6}$$

In the formula, ρ represents the density of the crankshaft, *v* represents the volume of the crankshaft, and *m* represents the mass of the crankshaft.

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4.2 Reduce Stress

4.2.1 Add Rounded Corners to the Crank Pins

The stress analysis indicates that the crankshaft experiences concentrated stress at the junction between the crank pin and the crank arm. This connection, which is initially formed at a right angle or a small angle, leads to a significant concentration of stress. Consequently, this region is particularly susceptible to fatigue fractures in the crankshaft[15]. In the optimization design presented in this paper, small-angle transition fillets are incorporated at the connection points, resulting in a significant reduction in peak stress. Figure 8 illustrates the equivalent stress distribution of the crankshaft following the addition of transition fillets, while Figure 9 depicts the safety factor distribution of the crankshaft after the same modification.

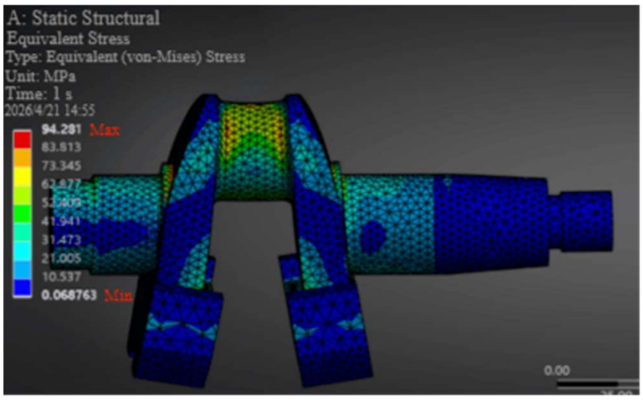

Figure 8. shows the equivalent stress cloud after adding transition fillets

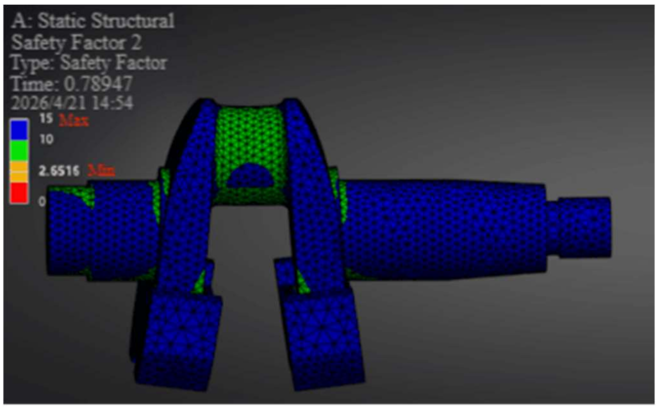

Figure 9. shows the safety factor cloud map of the crankshaft after adding transition fillet

The picture shown above shows that additional fillets may lead to a reduction in the stress concentration in the crankshaft. By this modification, there is an easier transmission of bending and torsional stresses that are spread out evenly, and thus avoid excessive local stresses and simultaneously improve the safety factor. After optimization, the peak stress of the crankshaft is 94.28 MPa which is 5 per cent lower than the original peak stress. Safety factor has been raised by 2.65 which is 6.4 per cent higher.

4.2.2 Add an Unloading Slot

In addition to alleviate the concentration of stress, unloading grooves are included in the crank pin. Through local excavation of such grooves, the high-stressed areas are reallocated and spread outwards, thus removing the extra stress that is caused by the interference fit. The figure 10 shows the equivalent stress distribution map of the crankshaft after the introduction of the unloading groove, whereas the

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figure 11 depicts the safety factor distribution map of the crankshaft when the unloading groove is introduced.

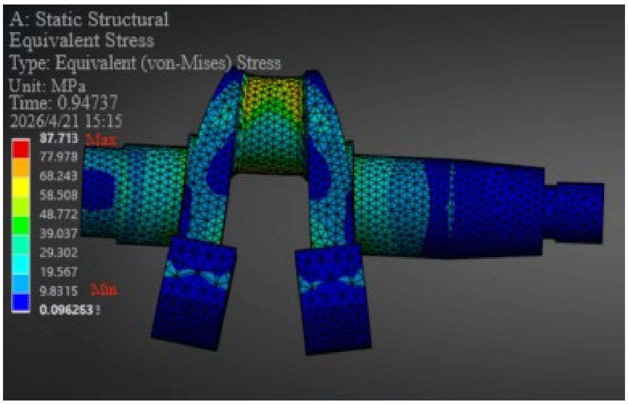

Figure 10. shows the equivalent stress cloud map after the unloading slot is opened

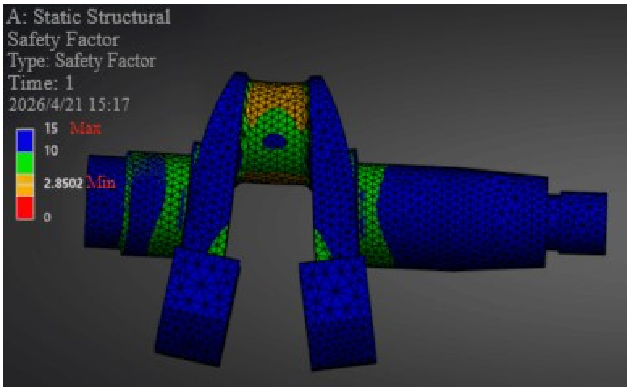

Figure 11. shows the cloud map of the safety factor after setting up the unloading slot

As shown in the figure above, the addition of an unloading slot is an effective way in reducing the maximum stress of the crankshaft. After optimization, the maximum stress level of the crankshaft is 87.713 MPa, which is a decrease of 12.4 percent of the original value. Also, safety factor has improved by 15 percent to 2.85.

The illustration in Table 4 shows that the optimized crankshaft has lower maximum stress. Also, the safety factor of 2.85 is very high compared to the minimum of 1.5, which means more safety of the crankshaft. The stress concentration regions were also intentionally loaded with unloading slots, which relieved stress, as well as reduced the weight of the crankshaft. This loss of mass then resulted in a smaller inertial force being acted upon by the crankshaft and led to a reduction in the material costs related to its manufacture.

Table 4. Comparison Table of Crankshaft Before and After Optimization

Design variable	Before optimization	After optimization
Stress (Mpa)	100.16	87.713
Safety factor	2.49	2.85

4.3 Reduce Quality

4.3.1 The Diameter of the Crank Pin X1 Varies

As indicated in Table 5, a reduction in the diameter of the crank pin leads to a decrease in the stiffness of the cylinder that contacts the connecting rod and crank pin. Consequently, the bearing load increases, resulting in higher maximum stress and a simultaneous decrease in the safety factor. The resulting safety factor is 2.06, which remains above the threshold of 1.5, thus falling within the acceptable range.

Table 5. Stress and Safety Factor Table after the diameter of the crank pin changes

Crank pin diameter/mm	45	43	40
Stress (Mpa)	87.713	108.95	121.16
Safety factor	2.85	2.294	2.06

4.3.2 The Variation of the Main Journal Diameter X2

Table 6. Stress and Safety Factor Table after the variation of the main journal diameter

Diameter of the main journal(mm)	45	43	40
Stress (Mpa)	121.16	126.56	128.4
Safety factor	2.06	2.007	1.947

As indicated in Table 6, the data presented in Table 5 demonstrate that a reduction in the diameter of the crankshaft journal leads to an increase in the maximum stress experienced, accompanied by a decrease in the safety factor. The resulting safety factor is 1.947, which remains above the threshold of 1.5 and is thus considered to be within a safe range.

The mass of the optimized crankshaft has been reduced from the original 4.7 kg to the current 4.3 kg, representing a weight reduction of 9.3%. Additionally, its volume has also decreased correspondingly.

5. Conclusion

(1) During the static analysis of crankshafts, three-dimensional solid modeling is conducted with Solid Works first and then the generated model is imported into ANSYS finite element software where modal analysis is performed. The results of the analysis show that there is a significant concentration of stress on the crankshaft at the transitional connection points between the crank pin and the crank arm, as well as between the main journal and the crank arm. To solve this problem, transition fillets are first added to the structural connection points to ensure that the local force distribution is more uniform and stress concentration is relieved. Based on this, an unloading groove is introduced to not only reduce the load concentration on the cylindrical surface of the crank pin but also to disperse and transfer the areas of high stress. It is a method that greatly alleviates the maximal equivalent stress of the crankshaft, alleviates the stress concentration effect, and minimizes the hazard of crankshaft fatigue fracture.

(2) This paper will use the radius of the crank pin and the main journal as the objective function variables in the static analysis of quality optimization. Reducing the volume of the crankshaft can reduce the mass of the crankshaft and consequently lower the total mass of the diesel engine and greatly reduce the manufacturing expense related to the crankshaft.

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