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AUTOMOTIVE CONNECTING ROD

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ABSTRACT

The connecting rod links the piston to the crankshaft and converts the reciprocating motion of the piston into the rotary motion of the crank. In every engine cycle it is compressed by the gas force during combustion and pulled in tension by the inertia of the piston assembly near the top of the exhaust stroke, so it works under a fully alternating load at high frequency. A connecting rod failure usually destroys the engine block and crankshaft, and therefore the rod must be strong, stiff against buckling and resistant to fatigue. Because it is a reciprocating part, its mass also directly affects the inertia forces, the bearing loads and the balance of the engine.

This paper presents the design and finite element analysis of a connecting rod for a single-cylinder four-stroke petrol engine of 150 cc displacement with a bore of 57 mm and a stroke of 58.6 mm. The rod was sized by the standard I-section design procedure and modelled in SolidWorks with a centre distance of 112 mm. The model was analysed in ANSYS Workbench for a compressive gas load of 8.93 kN, corresponding to a peak cylinder pressure of 3.5 MPa, and a tensile inertia load of 4.2 kN at a maximum speed of 9,000 rpm. Static structural, linear buckling, modal and fatigue analyses were carried out.

Four materials were compared using the same geometry, mesh and loads: structural steel, forged alloy steel AISI 4340, aluminium alloy 7075-T6 and titanium alloy Ti-6Al-4V. Under the compressive load the maximum von Mises stress was between 166.4 and 168.9 MPa for all materials and occurred at the junction of the shank and the small end. The factor of safety on yield was 1.49 for structural steel, 4.20 for AISI 4340, 3.02 for aluminium 7075-T6 and 5.25 for Ti-6Al-4V. The fatigue analysis showed that structural steel and aluminium 7075-T6 could not reach the target life of one million cycles under the alternating load, while AISI 4340 and Ti-6Al-4V could.

Considering strength, fatigue life, buckling margin, manufacturability and cost, forged AISI 4340 was selected, and its I-section was then refined by thinning the web from 4.0 mm to 3.2 mm and tapering the flanges towards the small end. The refined rod has a mass of 0.282 kg against 0.312 kg for the original section, a reduction of 9.6 percent, with a maximum stress of 186.4 MPa, a factor of safety of 3.81, a buckling load factor of 24.1 and a fatigue safety factor of 1.68. Its first natural frequency is well above the crankshaft speed and its second harmonic.

The work demonstrates that a computer-aided design and finite element approach can identify the critical regions of a connecting rod, compare candidate materials on a common basis and guide a weight reduction without compromising safety. The approach shortens the design cycle compared with repeated prototype forging and

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fatigue testing, and it provides a clear record of the reasoning behind the final material and section choice.

I. INTRODUCTION

The internal combustion engine remains the main power source for two-wheelers and a large share of passenger cars, and its reliability depends heavily on the moving parts of the crank mechanism. Among these, the connecting rod carries the full force of combustion from the piston to the crankshaft. It consists of a small end that holds the piston pin, a shank usually made in an I-section, and a big end that is split and bolted around the crank pin. The rod moves with a combination of reciprocating and rotating motion, and its loads change in magnitude and direction many times every second.

During the power stroke the gas pressure acting on the piston produces a large compressive force along the rod, which can cause yielding at the transitions or buckling of the slender shank. Near top dead centre of the exhaust stroke, where there is little gas pressure, the inertia of the piston and the upper part of the rod pulls the rod in tension. The rod therefore experiences a cyclic load ranging from tension to compression, at a frequency equal to half the engine speed for the gas load and equal to the engine speed for the inertia load. At high speeds the inertia load becomes significant, and fatigue becomes the main design concern.

Connecting rods for mass-produced engines are commonly made from forged carbon or alloy steel, while sintered powder metal, cast iron and aluminium are used in some engines. High-performance and racing engines use titanium alloys or forged aluminium to reduce reciprocating mass and allow higher speeds. Each material offers a different balance of strength, stiffness, fatigue resistance, density and cost, and the choice must be supported by analysis rather than by habit. A lighter rod reduces inertia forces and bearing loads, but it must not sacrifice the fatigue strength that the component depends on.

In this work a connecting rod for a 150 cc single-cylinder engine is designed from basic engine data, modelled in three dimensions and analysed in ANSYS Workbench. The aims are to determine the stress and deformation under the compressive gas load and the tensile inertia load, to check the buckling margin and natural frequencies, to estimate fatigue safety under the alternating load, to compare four candidate materials and to reduce the mass of the selected design by refining the I-section while keeping all safety targets satisfied. The work is limited to static, buckling, modal and fatigue analysis of the rod alone, with the piston pin and crank pin represented by their bearing loads, which is the level of detail normally used at the concept design stage before detailed dynamic and contact studies are carried out.

II. LITERATURE SURVEY

Studies of connecting rod failures in service have shown that most failures are caused by fatigue rather than by a single overload. Fatigue cracks typically start at the transition between the shank and the small end, at the transition between the shank and the big end, and at the bolt holes of the big end cap. These regions combine a change of section with high cyclic stress, and any surface defect or forging flash line

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left there can become an initiation site. Investigations of failed rods have also pointed out the effect of bearing seizure, which adds bending and heat to the rod. Researchers have therefore recommended smooth fillets at the transitions, careful removal of flash, shot peening of critical surfaces and a design that is checked for fatigue under the full alternating load rather than for static strength alone.

The classical design procedure for connecting rods sizes the I-section of the shank so that it resists buckling under the maximum gas force, using the Rankine formula with different end-fixity assumptions in the plane of motion and perpendicular to it. Textbook procedures recommend an I-section with proportions that give nearly equal buckling strength about both axes, and they size the small and big ends from bearing pressure limits. Several studies have compared these hand calculations with finite element results and reported that the hand method gives reasonable shank dimensions but cannot predict the peak stresses at the transitions or around the cap bolts, which are the actual failure locations. Finite element analysis has therefore become a standard complement to the classical method.

Finite element studies of connecting rods usually model the rod in SolidWorks, CATIA or a similar package and analyse it in ANSYS with second-order tetrahedral elements. The load at the pin bores is often applied as a distributed bearing load over half of the bore rather than as a point force, since a point force produces unrealistic local stresses. Researchers have shown that the choice of boundary condition at the big end, whether a fixed bore, a cylindrical support or a modelled crank pin with contact, has a noticeable effect on the stress near the big end. Many studies analyse both the compressive gas load and the tensile inertia load as separate static cases and then combine them in a fatigue analysis with a mean stress correction such as Goodman or Gerber.

Material comparison has been a major theme in connecting rod research. Forged steel, powder-metal steel, cast iron, aluminium alloys, titanium alloys and metal matrix composites have all been compared under identical geometry. A recurring observation is that the stress distribution changes little between materials because the rod is loaded mainly through its axis, while deformation varies inversely with the elastic modulus. Forged steels offer high fatigue strength at low cost. Aluminium alloys give large weight savings, but their lower fatigue strength limits them to engines where a finite life is accepted. Titanium alloys combine low density with high fatigue strength but are expensive to forge and machine, and they are therefore used mainly in performance engines.

Optimisation of the connecting rod for reduced mass has been studied through both shape and topology methods. Reported work has reduced mass by thinning the web of the I-section, tapering the flanges, reducing material around the small end and introducing pockets on the big end, while keeping the stress, buckling and fatigue limits unchanged. Typical reported savings for steel rods are in the range of about five to fifteen percent, which is significant for a reciprocating part because it also reduces inertia forces at high speed. Authors have emphasised that after every change the rod must be re-checked for buckling and fatigue, and that manufacturing constraints such as forging draft angles and minimum web thickness must be respected. The present work applies this approach to a forged steel rod after a material comparison.

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III. SYSTEM ANALYSIS

The problem is to design a connecting rod for a 150 cc single-cylinder four-stroke petrol engine that withstands the compressive gas force and the tensile inertia force, resists buckling of the shank and survives at least one million load cycles, while keeping its mass as low as practical. The input data are a bore of 57 mm, a stroke of 58.6 mm, a peak cylinder pressure of 3.5 MPa and a maximum speed of 9,000 rpm, which give a compressive gas load of 8.93 kN and a tensile inertia load of about 4.2 kN. The rod has a centre distance of 112 mm, a small end bore of 15 mm for the piston pin and a big end bore of 32 mm around the crank pin. The outputs required are the equivalent von Mises stress, total deformation, factor of safety on yield, buckling load factor, first natural frequency, fatigue safety factor and mass. The design targets are a static factor of safety of at least 3.0, a buckling load factor of at least 10 and a fatigue safety factor above 1.5.

Existing System

In conventional practice the connecting rod is designed by the textbook procedure in which the shank is treated as a column and its I-section is sized by the Rankine formula, while the small and big ends are sized from permissible bearing pressures. These calculations give the main dimensions but treat the rod as a set of simple shapes, so they cannot predict the stress concentration at the shank transitions, the bolt bosses or the oil hole. Large empirical factors of safety are used to cover this uncertainty, and the rod usually ends up heavier than necessary. The design is then validated by forging prototypes and testing them on a fatigue rig for millions of cycles, which takes several weeks for each variant. If cracks appear, the die must be modified and new forgings produced. Comparing several materials in this way is costly, so the material is often chosen from previous practice, and any opportunity to remove mass from lightly loaded regions is rarely explored.

Disadvantages of Existing System

- The shank is treated as a simple column, ignoring the real three-dimensional shape.
- Stress concentration at the shank transitions and bolt bosses is not predicted.
- Large empirical safety factors make the rod heavier than it needs to be.
- Each design variant requires new forging dies and prototype parts.
- Fatigue rig testing of every variant takes several weeks.
- Only a limited number of materials can be compared within the budget.
- Material choice is often based on previous practice rather than on analysis.
- The deformation pattern of the big end and small end is not visible.
- Lightly loaded regions where mass could be saved are not identified.
- Design decisions are poorly documented for later revisions.

Proposed System

The proposed approach combines the classical sizing procedure with three-dimensional modelling and finite element analysis. The main dimensions obtained from the Rankine and bearing pressure calculations are used to build a parametric

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model of the rod in SolidWorks, including the I-section shank, the small end, the split big end with its cap and bolt bosses, and fillets at all transitions. The model is imported into ANSYS Workbench, where structural steel, forged AISI 4340, aluminium 7075-T6 and Ti-6Al-4V are assigned in turn with the same mesh and loads. Static structural analysis is carried out separately for the compressive gas load and the tensile inertia load, both applied as bearing loads over half of the pin bore, with the opposite bore supported. Linear buckling and modal analyses check stability and resonance, and the fatigue tool combines the two static cases into a fully alternating load with Goodman mean stress correction. The selected material is then used in a refined I-section, which is re-analysed for all criteria.

Advantages of Proposed System

- The real geometry of the shank, ends and bolt bosses is analysed in detail.
- Peak stresses at the shank transitions are located before any forging is made.
- Compressive and tensile load cases are evaluated separately and then combined.
- Four materials are compared under identical geometry, mesh and loading.
- Buckling margin is obtained directly from a linear buckling analysis.
- Natural frequencies are checked against crankshaft speed and its harmonics.
- Fatigue safety under the alternating load is estimated with mean stress correction.
- The I-section is refined to reduce reciprocating mass by 9.6 percent.
- Prototype forging and rig testing are reserved for the final design only.
- The analysis record explains clearly why the final material was chosen.

IV. METHODOLOGY

The methodology started with the calculation of design loads from the engine data, giving 8.93 kN compressive and 4.2 kN tensile force. The shank was sized as an I-section of overall depth 20 mm and width 14 mm using the Rankine formula, and the small and big ends were sized from bearing pressure. The rod was modelled in SolidWorks and imported into ANSYS Workbench, and properties for the four materials were taken from the engineering data library. The model was meshed with ten-node tetrahedral elements with local refinement at the transitions; a convergence study at 2, 1.5, 1 and 0.75 mm element size showed a change of 1.1 percent in peak stress between the last two sizes, so a 1 mm mesh with 98,436 nodes and 63,207 elements was used. For the compressive case the big end bore was fixed and a bearing load was applied on the small end bore, and the direction was reversed for the tensile case. Static, buckling, modal and fatigue solutions were then solved and post-processed.

SYSTEM ARCHITECTURE

The design workflow is organised in four parallel lanes that pass information between the tools used. The engine data lane holds the bore, stroke, peak pressure and maximum speed and produces the gas and inertia loads. The CAD lane in SolidWorks converts the Rankine and bearing pressure dimensions into a parametric solid model of the rod and later applies section changes during refinement. The ANSYS

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Workbench lane contains the engineering data library with the four materials, the meshing step with convergence check, the bearing loads and supports, and the four solvers for static structural, linear buckling, modal and fatigue analysis. The evaluation lane collects von Mises stress, deformation, safety factor, buckling load factor, natural frequency, fatigue safety factor and mass in a comparison table, checks them against the targets and selects the material. When a margin is larger than required, the evaluation lane sends a section change back to the CAD lane, and the loop continues until the refined rod satisfies every target.

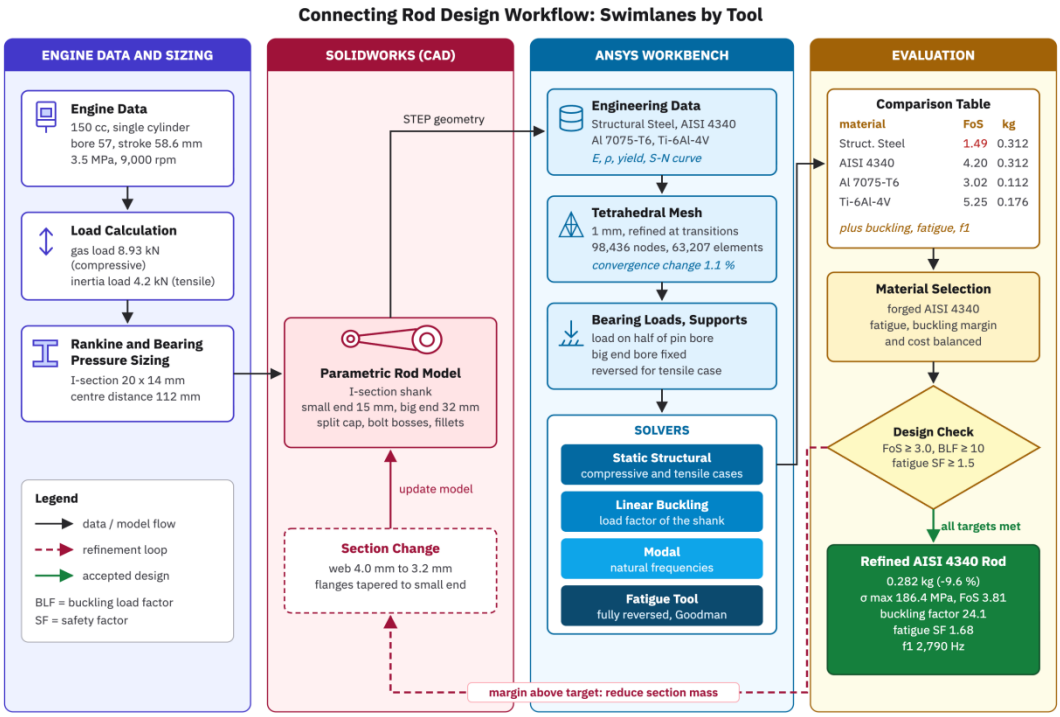


Fig. 1 System Architecture

V. RESULT AND OUTPUT

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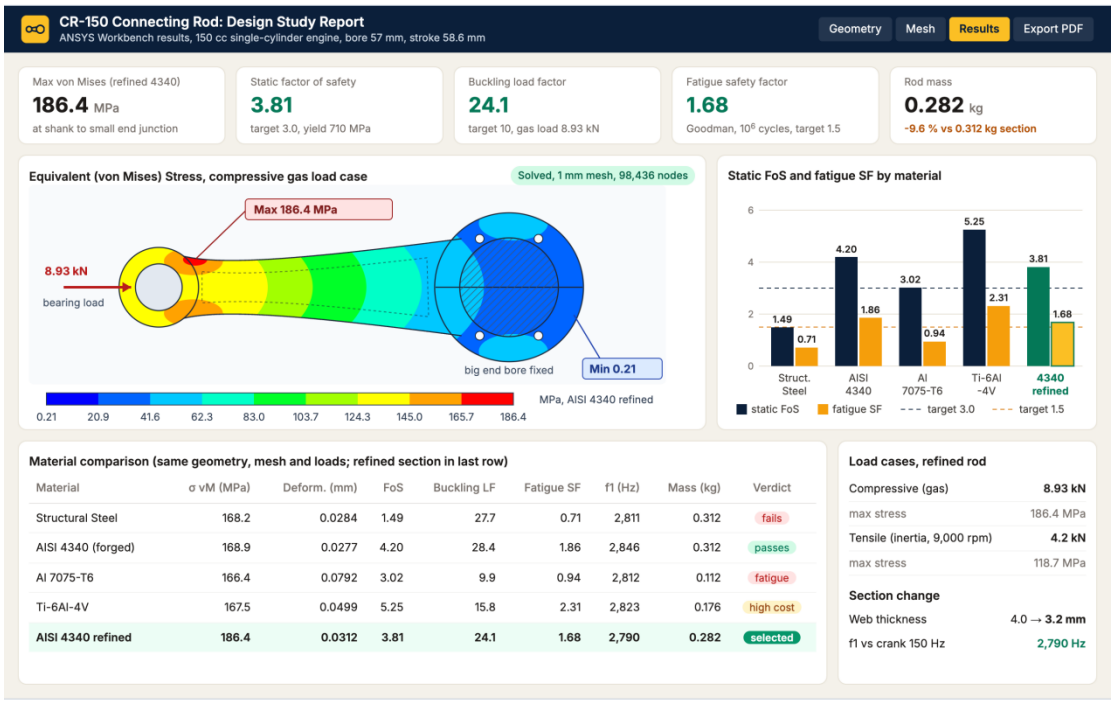


Fig. 2 Result and Output Screen

VI. CONCLUSION

A connecting rod for a 150 cc single-cylinder four-stroke petrol engine was designed from basic engine data, modelled in SolidWorks and analysed in ANSYS Workbench under a compressive gas load of 8.93 kN and a tensile inertia load of 4.2 kN. In all materials the highest stress occurred at the junction of the shank and the small end, which matches the crack initiation sites reported for rods in service. The mesh convergence study showed that the peak values were stable, and the bearing-load boundary conditions avoided artificial stress peaks at the pin bores.

The material comparison showed nearly equal stress levels for all four materials, while their safety factors, deflections and masses differed considerably. Structural steel gave a static factor of safety of only 1.49 and failed the fatigue target. Aluminium 7075-T6 gave a static factor of 3.02 and a mass of only 0.112 kg, but its fatigue safety factor of 0.94 and buckling load factor of 9.9 fall below the targets for an engine expected to run for a long service life. Ti-6Al-4V gave the best margins at 0.176 kg but at a high cost, so forged AISI 4340 was selected as the most practical material.

The refined AISI 4340 rod, with the web thinned to 3.2 mm and tapered flanges, weighs 0.282 kg, which is 9.6 percent lighter than the original section. It has a maximum stress of 186.4 MPa, a factor of safety of 3.81, a buckling load factor of 24.1, a fatigue safety factor of 1.68 and a first natural frequency of 2,790 Hz, which is far above the crankshaft frequency of 150 Hz at 9,000 rpm and its second harmonic. All targets are therefore satisfied with a lighter reciprocating part. The lower mass also reduces the reciprocating inertia force at 9,000 rpm in proportion, which lightens the load on the big end bearing and the crank pin and slightly improves the balance of the engine.

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Future work can apply the loads as a time history over the complete engine cycle obtained from a dynamic model of the crank mechanism, include the pretension of the big end cap bolts and the contact between cap and rod, and study the effect of the oil hole on local stress. Physical validation through strain measurement and fatigue testing of forged prototypes would confirm the predicted margins before the design is released for production.

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