Connecting Rod Optimization for Weight and Cost Reduction

Pravardhan S. Shenoy and Ali Fatemi
The University of Toledo

ABSTRACT

An optimization study was performed on a steel forged connecting rod with a consideration for improvement in weight and production cost. Since the weight of the connecting rod has little influence on its total production cost, the cost and the weight were dealt with separately. Reduction in machining operations, achieved by change in material, was a significant factor in manufacturing cost reduction. Weight reduction was achieved by using an iterative procedure. Literature survey suggests cyclic loads comprised of static tensile and compressive loads are often used for design and optimization of connecting rods. However, in this study weight optimization is performed under a cyclic load comprising dynamic tensile load and static compressive load as the two extreme loads. Constraints of fatigue strength, static strength, buckling resistance and manufacturability were also imposed. The fatigue strength was the most significant factor in the optimization of the connecting rod. An estimate of the cost savings is also made. The study results in an optimized connecting rod that is 10% lighter and 25% less expensive, as compared to the existing connecting rod.

INTRODUCTION

The objective of this study was to optimize a forged steel connecting rod for its weight and manufacturing cost, taking into account recent developments. Typically, an optimum solution is the minimum or maximum possible value the objective function could achieve under a defined set of constraints. The optimization carried out here, however, is not in the true mathematical sense, since while reducing mass, manufacturing feasibility and cost reduction are integral parts of the optimization. In addition, software used in this work imposed restrictions in performing optimization under fatigue life constraint. Therefore, rather than using numerical optimization techniques for weight reduction, quantitative results were examined qualitatively, and the structure modified.

Yoo et al. [1] performed shape optimization of an engine connecting rod using variational equations of elasticity, material derivative idea of continuum mechanics, and an adjoint variable technique to calculate shape design sensitivities of stress. The results were then used in an iterative optimization algorithm to numerically solve for an optimal design solution. Serag et al. [2] developed approximate mathematical formulae to define connecting rod weight and cost as objective functions as well as constraints. The optimization was achieved using a geometric programming technique. Saruhan and Song [3] optimized the wrist pin end of an engine connecting rod with an interference fit. They generated an approximate design surface and performed optimization of this design surface. The objective and constraint functions were updated in an iterative process until convergence was achieved. The load cycle that was used consisted of compressive gas load corresponding to a maximum torque and a tensile load corresponding to maximum inertia load. The modified Goodman equation with alternating and mean octahedral shear stress was used for fatigue analysis. Pai [4] presented an approach to optimize the shape of a connecting rod subjected to a load cycle which consisted of the inertia load deducted from gas load as one extreme and peak inertia load exerted by the piston assembly mass as the other extreme. A finite element routine was first used to calculate the displacements and stresses in the rod, which were then used in another routine to calculate the total life. Fatigue life was defined as the sum of crack initiation and crack growth lives, with crack growth life obtained using fracture mechanics.

For this optimization problem, the weight of the connecting rod has little influence on the cost of the final component. Change in the material, resulting in a significant reduction in machining cost, was the key factor in cost reduction. As a result, in this optimization problem the cost and the weight were dealt with separately. The structural factors considered for weight reduction during the optimization include the buckling load factor, stresses under the loads, bending stiffness, and axial stiffness. Cost reduction is achieved by using C-70 steel, which is fracture crackable. It eliminates sawing and machining of the rod and cap mating faces and is believed to reduce the production cost by 25% [5].
Figure 1 shows the actual and the digitized connecting rods. The weight difference between the two when corrected for bolt head weight was less than 1%. This is an indication of the accuracy of the solid model. The engine configuration considered is tabulated in Table 1.

![Figure 1: The actual and the digitized connecting rods.](image1)

Table 1: Configuration of the engine to which the connecting rod belongs.

<p>| | |</p>
<table>
<thead>
<tr>
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</thead>
<tbody>
<tr>
<td>Crankshaft radius</td>
<td>48.5 mm</td>
</tr>
<tr>
<td>Piston diameter</td>
<td>86 mm</td>
</tr>
<tr>
<td>Mass of the connecting rod</td>
<td>0.439 kg</td>
</tr>
<tr>
<td>Mass of the piston assembly</td>
<td>0.434 kg</td>
</tr>
<tr>
<td>Connecting rod length</td>
<td>141 mm</td>
</tr>
<tr>
<td>(I_{zz}) about the center of gravity</td>
<td>0.00144 kg m^2</td>
</tr>
<tr>
<td>Distance of C.G. from crank end center</td>
<td>36.4 mm</td>
</tr>
<tr>
<td>Maximum gas pressure</td>
<td>37.3 Bar</td>
</tr>
</tbody>
</table>

LOAD ANALYSIS

The pressure crank angle diagram used is shown in Figure 2. The optimization was performed under a cyclic load comprising dynamic tensile load and static compressive load as the two extreme loads. Results consisting of the angular velocity and angular acceleration of the connecting rod, linear acceleration of the connecting rod crank end center and of the center of gravity, and forces at the ends were generated for a few engine speeds. Crank end and piston end forces as a function of crank angle for this connecting rod at the maximum engine speed of 5700 rev/min are shown in Figures 3(a) and 3(b), respectively. Two components of the force are plotted, one along the length of the connecting rod and the other normal to it. The resultant force is also shown. At any point in time the forces calculated at the ends form the external loads, while the inertia load forms the internal load acting on the connecting rod. These result in a set of completely equilibrated external and internal loads. As the speed increases, the maximum tensile load increases whereas the maximum compressive load at the crank end decreases. As can be seen, the maximum dynamic tensile load corresponds to 360° crank angle. Note that under the effect of dynamic load, forces at the two ends are different at a given instant of time.

![Figure 2: Pressure crank angle diagram used to calculate forces at the connecting rod ends.](image2)

![Figure 3: Axial, normal, and the resultant force at the crank end (a) and at the piston pin end (b) at crank speed of 5700 rev/min.](image3)
STRESS-TIME HISTORY

Quasi-dynamic FEA was performed to obtain the stress-time history. Quasi-dynamic FEA rather than static FEA can capture the actual conrod structural behavior. While performing quasi-dynamic FEA of the connecting rod, the external loads consisting of reactions or the loads computed at the connecting rod ends at the crank angle of interest, were applied to the crank and the piston pin ends of the connecting rod. The angular velocity, angular acceleration, and linear acceleration at the same crank angle were specified in both magnitude and direction for the connecting rod. The inertia and dynamic loads were calculated and applied internally based on these inputs. This ensures that the applied loads form a set of completely equilibrated internal and external loads.

The applied load distributions were based on research by Webster et al. [6]. The tensile load was applied over 180° of crank contact surface with cosine distribution, whereas compressive load was applied as a uniformly distributed load over 120° of crank contact surface. The details have been discussed in [6]. Of the fifteen locations shown in Figure 4, the stress-time history for locations 9, 12 and 13 are shown in Figure 5. From this figure it is clear that the maximum stress at location 9 occurs at 360° crank angle, just as it does for the maximum load. At location 13, however, the maximum stress occurs at 348° crank angle, due to the influence of bending stresses. This highlights the significance of the bending stresses, which will not be considered if designing/optimizing connecting rod based on axial loads alone. The bending stresses were found to be about 20% of the overall stress amplitude at the shank center [7].

Other important observations from stress-time histories of the locations in Figure 4 were that locations near the oil hole and at the crank end transition have significant multiaxiality requiring use of equivalent stress (i.e. von Mises). Also, the R ratio (i.e. minimum to maximum stress ratio) varied with location and engine speed. For example, at the middle of the shank (i.e. point 12 in Figure 4), the R ratio varies from – 18.8 at 2000 rev/min to – 0.86 at 5700 rev/min. These are discussed in [7].

OPTIMIZATION STATEMENT AND CONSTRAINTS

The objective of the optimization was to minimize the mass of the connecting rod under the effect of a load range comprising the peak compressive gas load and the peak dynamic tensile load at 5700 rev/min (at 360° crank angle), such that the maximum, minimum, and the equivalent stress amplitude are within the limits of the allowable stresses. The production cost of the connecting rod was also to be minimized. Furthermore, the buckling load factor under the peak gas load has to be permissible. As far as the optimized geometry, the connecting rod has to be interchangeable with the existing one in the current engine. Each of these requirements or constraints, are now briefly discussed.

APPLIED LOADS - The tensile load used was the dynamic tensile load at 360° crank angle at 5700 rev/min. This is because the maximum tensile stress at all locations in the connecting rod occurs at or near 360° crank angle at 5700 rev/min engine speed. The compressive gas load used was 21.8 kN, corresponding to peak cylinder pressure in the pressure crank angle diagram (see Figure 2). The compressive load of 21.8 kN is independent of the geometry of the connecting rod.

ALLOWABLE STRESS - Allowable stress is the ratio of material strength to the factor of safety. The material chosen for the optimized connecting rod is C-70 steel due to its fracture crackability. C-70 steel has a percent elongation of 27%, and the monotonic stress-strain curve [8] shows the behavior of a relatively ductile material. As a result, the factor of safety for static loading was defined with respect to the yield strength, rather than the ultimate tensile strength.

With regards to cyclic loading, the connecting rod is expected to survive between $10^5$ to $10^8$ cycles. The
endurance limit for the C-70 steel is 339 MPa, whereas for the existing forged steel material it is 423 MPa. The surface finish factor has not been taken into account, since the shot peening process negates the negative impact that the forged surface finish has on the fatigue life, by inducing compressive residual stress on the surface. This is confirmed by subsurface failures during axial fatigue tests of the connecting rods [8].

FAILURE INDEX - A concept similar to factor of safety, the failure index (FI), was used in this work to quantify severity of the applied stresses with respect to the available strength. Failure index (FI) is the inverse of the safety factor, and can be defined as the ratio of the von Mises stress to the strength (either yield strength or fatigue strength, depending on the static or cyclic nature of the stress, respectively) of the material. The closer the FI to one, the higher the possibility of failure.

The factor of safety applied to the yield strength in this work was 2.1, corresponding to a FI of 0.48, based on guidelines recommended by Norton [9], as well as other studies on design of connecting rods, such as that by Folgar et al. [10]. Either this assumed FI of 0.48 or the FI in the existing component, whichever was higher, was used for obtaining the allowable stress at a given location or region of the connecting rod.

For cyclic loading, Sonsino and Esper [11] used a safety factor of 1.66 on the endurable load amplitude for a PM connecting rod. The same factor was used for the allowable stress amplitude here, and corresponds to a FI of 0.60. Similar to the case for axial loading, this assumed FI of 0.60 or the FI in the existing component, whichever was higher, was used for obtaining the allowable stress amplitude at a given location or region of the connecting rod. Figure 7 shows the FI distribution for the existing geometry with respect to the endurance limit of the existing material under cyclic load.

![Figure 7: Failure Index (FI), defined as the ratio of equivalent stress amplitude at R = -1 to the endurance limit of 423 MPa, for the existing connecting rod and material.](image)

GEOMETRY CONSTRAINTS - The optimized connecting rod was assumed to be interchangeable with the existing one. Therefore, the diameters of the crank pin and the piston pin holes, the overall thickness of the connecting rod, and the crank pin center to piston pin center distance could not be changed. The piston pin end fits under the piston and is supposed to clear off the piston skirt and the piston bottom, when in operation. The dimensions of the bolts and their holes were also retained. This is because modeling the bolt-connecting rod interface is a complex problem and beyond the scope of this work. All other dimensions of the connecting rod could be varied, within practical limits.

OPTIMIZATION PROCEDURE AND RESULTS

Comparison of FEA results for the existing connecting rod against the allowable stresses indicate that the shank region of the connecting rod offers the greatest potential for weight reduction. Regions near locations 3, 4, 9, 10, and 11 (Figure 4) are already highly stressed. Even though regions near locations 5, 6, 7, and 8 may offer potential for weight reduction, these regions involve complex interface between the bolt and the connecting rod. As a result, this region was not the focus in the study.

In the shank region, the rib and the web thicknesses were reduced, however, only to a certain limit to maintain forgeability. The section modulus of the optimized connecting rod should be high enough to prevent high bending stresses. Bending stresses exist due to inertia forces, and can also occur due to eccentricities as well as crankshaft and case wall deformations. In order for the section modulus to be as high as possible, the width of the rib was increased.

The volume of the pin end region was increased to increase the strength in this region due to the lower yield strength and endurance limit of C-70 steel, as compared to the existing forged steel. Based on the guideline outlined by Repgen [5], the jig spot was relocated from the existing location to be entirely on the cap. The radii at the transitions to the crank end and the pin end were increased, since relaxing the radii has the effect of reducing the stress concentration effect. However, the overall width, diameters of the pin end and crank end bores, and the pin end center-to-crank end center length of the connecting rod were not changed, to maintain interchangeability with the existing connecting rod. It must be mentioned that in addition to the dimensions just discussed, dimensions of chamfers in the web and the cap of the connecting rod were also varied during the optimization process.

After several iterations, which involved determining the loads and performing FEA for the resulting geometry of each iteration step, an optimized geometry was obtained, shown in Figure 8. Mass of the optimized connecting rod is 396 grams, which is lower than the mass of the original connecting rod by 10%. This
geometry was found to satisfy the aforementioned design constraints.

Figure 8: The geometry of the optimized connecting rod.

Failure indexes (FI) with respect to the yield strength of C-70 steel under the action of tensile and compressive loads were obtained. The highest FI under tensile load was found to be on the surface of the pin end bore. However, in an analysis which involved FEA of the connecting rod with the piston pin and the bushing, it was observed that the stresses in this region (bore of the pin end) were significantly lower than the stresses predicted with cosine loading.

For cyclic loading, failure index (FI) was obtained with respect to the endurance limit of the C-70 steel. The equivalent stress amplitude $S_{qa}$ was calculated based on von Mises criterion, and the equivalent mean stress $S_{qm}$ was calculated as:

$$S_{qm} = S_{mx} + S_{my} + S_{mz}$$  \hspace{1cm} (1)

After obtaining the equivalent mean stress and stress amplitude, the equivalent stress amplitude at $R = -1$ (corresponds to $S_{qa}$) was obtained by using the commonly used modified Goodman equation:

$$\frac{S_{qa}}{S_{Nf}} + \frac{S_{qm}}{S_u} = 1$$  \hspace{1cm} (2)

Figure 9 shows the FI distribution for the optimized connecting rod, for cyclic loading. This figure indicates that the maximum FI is at the oil hole. The stresses at this location are sensitive to the boundary conditions. Similar boundary conditions were used for both the existing and the optimized connecting rods. The value of FI for the optimized connecting rod at the oil hole was found to be less than that for the original connecting rod.

Linear buckling analysis was also performed on the connecting rod with the compressive load of 21.8 kN, which is the maximum compressive load in the loading cycle. The buckling load factor for the original connecting rod is 7.8, while for the optimized connecting rod the buckling load factor is 9.6, which is more than adequate.

Figure 9: Failure Index (FI) distribution for the optimized connecting rod.

Table 2 lists various material and geometric properties for the two connecting rods. Note that in spite of weight reduction, the axial stiffness has slightly increased. The mass moment of inertia of the connecting rod about the axis normal to its plane of motion (z axis) and passing through its C.G., $I_{zz}$, and the weight are also listed for the two connecting rods. The weight does not include the weight of the bolt heads and is the weight of the geometry generated for FEA. It should be noted that by using other fracture crackable materials such as micro-alloyed steels having higher yield strength and endurance limit, the weight at the piston pin end and the crank end can be further reduced. Weight reduction in the shank region is, however, limited by manufacturing constraints.

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<th>Material Properties</th>
<th>Optimized</th>
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<th>% Change</th>
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<tbody>
<tr>
<td>E (GPa)</td>
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<td>207</td>
<td>2</td>
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<tr>
<td>Yield Strength (MPa)</td>
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<tr>
<td>Percent Reduction in Area</td>
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<tr>
<td>Tensile Strength (MPa)</td>
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<td>Endurance Limit (MPa)</td>
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<td>423</td>
<td>-20</td>
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<table>
<thead>
<tr>
<th>Other Factors</th>
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<th>Original</th>
<th>% Change</th>
</tr>
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<tbody>
<tr>
<td>Axial Displacement$^1$ (mm)</td>
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<td>0.206</td>
<td>-1</td>
</tr>
<tr>
<td>Weight (gms)</td>
<td>396</td>
<td>440</td>
<td>-10</td>
</tr>
<tr>
<td>$I_{zz}$ (kg m$^2$)</td>
<td>0.00139</td>
<td>0.00144</td>
<td>-4</td>
</tr>
<tr>
<td>Buckling Load Factor</td>
<td>9.6</td>
<td>7.8</td>
<td>23</td>
</tr>
</tbody>
</table>

$^1$Overall displacement along connecting rod length in tension.

VALIDATION OF THE CRANK END DESIGN

As mentioned earlier, the jig spot was relocated so that it would not be in the path of the fracture crack during the fracture cracking of the cap and rod. In order to validate the adequacy of the design, a more detailed FEA was performed for the crank end. For this FE analysis the
bolt holes were modeled, and the cap was split from the rod and connected with two bolts. The connection between the bolt and the connecting rod at the threads was modeled using spring elements with high stiffness. The connection region was not modeled with great details, since the behavior at the bolt–rod interface was not of interest, as the original design of the bolt was being retained. The behaviors at the cap-rod interface and of the rod and the cap under more realistic loading conditions were of primary interest. The bolt pretension was also modeled. The pretension was estimated to be 12.8 kN per bolt, based on a specified bolt torque of 20 Nm for the connecting rod. Contact elements were defined between the rod and the cap at their mating surface. Contact elements were also defined between the bolt head and the bolt seat on the cap. The pin end was restrained, and a tensile load corresponding to 360° crank angle at 5700 rev/min and a compressive load of 21.8 kN were applied to the connecting rod. Thus two FEA models were solved, one with a tensile load and the other with a compressive load.

Figure 10 shows the von Mises stress variation and the displacements of the model just discussed under the tensile load. The displacement shape of the cap and the rod, especially at the rod-cap interface indicates that FEM was modeled appropriately.

The maximum von Mises stress is at the outer corners of rod-cap interface in Figure 10. Of the many nodes on the inner cap edge (at the rod-cap interface), the node with minimum radial displacement had a radial displacement value of 0.077 mm. This displacement is towards the center of the connecting rod bore. However, the clearance between the crank end bearing and the crankshaft is of the order of 0.026 mm for connecting rods in this size range [12, 13].

A few additional cases were investigated to compare the rigidity of the connecting rod assembly (with the crank pin and bearing) with the rigidity of the connecting rod alone. These cases included connecting rod with a bearing, connecting rod with bearing in which the stiffness of the bearing was reduced to a very low value, and a connecting rod with a pin (representing the crankshaft) having radial clearance of 0.026 mm. The connecting rod used in these analysis cases had the cap integral to the rod. The only factor not accounted for in the analysis is the presence of the oil film. The oil film, however, is not expected to affect the rigidity.

Comparison of these results indicated that the rigidity of the connecting rod crank end increased with the bearing assembled in the crank end bore. Moreover, the stresses and displacements were significantly lower at the critical location in the presence of the pin, indicating that presence of the pin increases the rigidity of the crank end of connecting rod assembly (when compared with its behavior under cosine loading used for the model in Figure 10). The displacements at the region corresponding to the inner edge (at the rod-cap interface) reduced significantly to lower than the clearance value. The fracture surface at the cap-rod interface, is also known to increase stiffness due to a firm contact between the cap and rod [14].

From these analyses it can be concluded that rigidity of the crank end is increased in the presence of the crankshaft and the bearing. As a result, springs were added across the opposite inner edges of the rod and cap to model the above-mentioned stiffness. Following this addition, the maximum stress at the outer edge of rod-cap interface drops significantly, as can be seen from Figure 11.

Figure 11. von Mises stress variation and displacements (magnified 20 times) of the connecting rod and cap under tensile load. The FE model with springs added across the edges of the cap and rod is shown on the right.

MANUFACTURING AND ECONOMIC COST ASPECTS

The following steps in the manufacturing of the existing forged steel connecting rod can be eliminated by introducing C-70 crackable steel: the heat treatment, the machining of the mating faces of the crank end, and drilling for the sleeve. The fracture splitting process eliminates the need to separately forge the cap and the body of the connecting rod or the need to saw or
machine a one-piece forged connecting rod into two. In addition, the two fracture split parts share a unique surface structure at the fractured surface that prevents the rod and the cap from relative movement [5]. This provides a firm contact and increases the stiffness in this region [14].

Maintaining the forgeability of the connecting rod was taken into account during the optimization process. While reducing the dimensions of the shank, the web and the rib dimensions were reduced to a certain limit. The web was retained in the shank for the same reason. Making a cut out in the shank would have resulted in more efficient utilization of the material, but the shape would not be forgeable without distortion. Another aspect addressed to maintain the forgeability is the draft angle provided on the connecting rod surface.

Cost is often a proprietary issue and not easily available. A study published by Clark et al. [15] in 1989 compared the manufacturing costs of powder forged and steel forged connecting rods. It was shown that machining cost comprises 62% of the total cost for the conventional forged steel (FS) connecting rod, and 42% of the total cost for the powder metal (PM) connecting rod. The total cost of the PM connecting rod was 6% lower than the conventional FS connecting rod. This indicates that the machining cost reduction from utilization of the fracture splitting process, which is used for PM connecting rods, results in 23% total cost savings for the FS connecting rod. Similarly, in a more recent study by Repgen in 1998 [5], with reference to forged steel connecting rods, he states that: “The development of fracture splitting the connecting rods achieves a total cost reduction up to 25% compared to conventionally designed connecting rods and is widely accepted in Europe”.

Paek et al. [16] introduced a powder metal connecting rod, for their Hyundai Motor Co. Engine. They note that by adapting powder material for connecting rods, without loss of stiffness, they saved 10.5% on product cost in comparison with hot steel forged connecting rod which required machining at the rod–cap joint face. Elimination of this machining for the hot steel forged connecting rod through the fracture splitting process would result in a 13% cost savings, as compared with the PM connecting rod, based on the Clark et al. cost analysis. Repgen [5] also makes a similar note: “In principle, a forged rough part can run on machining lines originally designed for powder metal connecting rods. An automotive manufacturer analyzed the costs and proved a cost reduction of 15%”.

**SUMMARY AND CONCLUSIONS**

Optimization was performed to reduce weight and manufacturing cost of a forged steel connecting rod subjected to cyclic load comprising the peak compressive gas load and the peak dynamic tensile load at 5700 rev/min, corresponding to 360° crank angle. The structural factors considered for weight reduction during the optimization process included fatigue strength, static strength, buckling resistance, bending stiffness, and axial stiffness. Additional constraints imposed during the optimization process included maintaining the forgeability as well as interchangeability of the optimized connecting rod with the existing one.

Cost was reduced by changing the material of the existing forged steel connecting rod to crackable forged steel (C-70). The fracture splitting process eliminates the need to separately forge the cap and the body of the connecting rod or the need to saw or machine a one-piece forged connecting rod into two. Heat treatment, machining of the mating faces of the crank end, and drilling for the sleeve are also eliminated.

The following conclusions can be drawn from the results of this study:

1) Fatigue strength was the most significant factor (i.e. design driving factor) in the design and optimization of the connecting rod.

2) Stresses and displacements were observed to be significantly lower under conditions of assembly (with bearings, crankshaft and piston pin and bushing), when compared to stresses obtained from unassembled connecting rod subjected to cosine loading.

3) The section modulus of the connecting rod should be high enough to prevent high bending stresses due to inertia forces, eccentricities, as well as crankshaft and case wall deformations.

4) The shank region of the connecting rod offered the greatest potential for weight reduction. The rib and the web thicknesses were reduced, while maintaining forgeability.

5) With using fracture splitting process, the two fracture split parts share a unique surface structure at the fractured surface that prevents the rod and the cap from relative movement, resulting in a firm contact. This increases the stiffness and reduces stress at critical locations in the crank end of the connecting rod.

6) The optimized geometry is 10% lighter than the current connecting rod for the same fatigue strength, in spite of lower yield strength and endurance limit of C-70 steel compared to the existing forged steel.

7) Reduction in machining operations achieved by using C-70 steel and utilization of the fracture splitting process reduces the production cost by about 25%. As compared with a PM connecting rod, the cost saving is estimated to be about 15%.
8) By using other facture crackable materials such as micro-alloyed steels having higher yield strength and endurance limit, it may be possible to further reduce the weight at the piston pin end and the crank end. Weight reduction in the shank region is, however, limited by manufacturing constraints.

REFERENCES


