

# Study of Optimization of Reciprocating Parts for General-purpose Engine with Aluminum-alloy Connecting Rods

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## ABSTRACT

The connecting rods employed in the reciprocating parts of general-purpose engines are manufactured from aluminum alloy, and therefore display lower material strength than steel connecting rods. In terms of operating conditions, general-purpose engines are frequently operated under high load while being held at a comparatively low engine speed, necessitating strength and durability in relation to combustion pressure. Realizing a balance between the rigidity of each part is an important factor in reconciling the achievement of weight reduction in the reciprocating parts while also maintaining strength. The research discussed in this paper developed a structure optimization system to examine shape parameters for reciprocating parts, and studied shapes that would balance the reduction of weight with the maintaining of strength. It was shown from calculation results that the annular stress of a connecting rod originates in piston pin displacement, and in order to achieve both weight reduction and strength retention, results confirmed that reducing the diameter of the piston pin and the formation of thick walls of connecting rod small-end annular sections are effective. This realizes a combination of shapes in which the stress generated on each part is equivalent to or lower than the stress generated on the mass-production shape, while reducing the mass of the piston pins by 20%. Tests were conducted on the reduced-weight shapes, and the effectiveness of this optimization system was verified.

## 1. Introduction

Low weight and compactness are important factors for small general-purpose engines. Portability and freedom of fitting are demanded of the engines for use in applications such as construction equipment, agricultural equipment, and industrial equipment. In addition, the engines are frequently used under high loads while being held at comparatively low engine speed, demanding strength and durability in relation to combustion pressure more than inertia force.

A variety of methods have been used in the past to realize lightness and compactness in general-purpose engines<sup>(1)-(3)</sup>.

Of the reciprocating parts of general-purpose engines in most standard spark-ignition gasoline engines, the pistons and connecting rods are manufactured from aluminum alloy, and the piston pins are manufactured from steel. The engines use a fully floating configuration for the piston pins.

Aluminum alloy is employed for the connecting rods in order to reduce weight. It also makes bearings unnecessary, reducing the number of parts and increasing productivity, which represents a significant advantage in terms of

reducing costs.

The big ends of aluminum-alloy connecting rods also employ a separate cap structure, which makes the replacement of parts an easy matter and offers excellent maintenance performance.

It is often the case that users in emerging nations replace parts themselves, and ease of maintenance therefore contributes significantly to increasing commercial appeal.

Because they satisfy the needs of general-purpose engines so effectively for all of these reasons, aluminum-alloy connecting rods are used in the main moving sections of the majority of general-purpose engines.

The aluminum alloys used in connecting rods are lower in strength than steel materials, making detailed consideration of strength essential in the design of weight reduction around the piston pins.

For example, reducing the weight of piston pins with high specific gravity would be effective in achieving weight reduction, but reducing the thickness of the pins would increase annular crushing deformation of the pins due to combustion pressure. As a result, stress around the piston pins and stress on the connecting rod small ends would

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increase in tandem with the deformation of the piston pins.

As phenomena of this type indicate, the realization of weight reduction and the maintaining of strength in the reciprocating parts are in a tradeoff relationship.

Because of this, structural analysis using FEM is employed during the design of the connecting rod small ends and the area around the piston bosses in order to judge strength and increase the accuracy of the blueprints prior to the prototype stage.

The conventional method of review is to create a 3D model and determine strength following a structural analysis.

However, this presents the issue of requiring an enormous amount of time for the calculation of large numbers of models of shape differences while studying the optimum shapes.

The linkage of CAE analysis and optimization tools is today attracting attention as an approach to analyses of this type, and has been employed in a variety of development processes<sup>(4), (5)</sup>. The research discussed here set out to determine by calculation the respective part shapes of the reciprocating parts of a general-purpose engine that would maintain strength while also reducing weight, and to verify the effectiveness of these shapes in tests.

## 2. Overview of Shape Optimization System

### 2.1. Conditions for Structural Analysis of Reciprocating Parts

Generally, evaluation of the strength of connecting rods as reciprocating parts includes the buckling and bending strength of the rod sections in areas other than the vicinity of the piston pin holes. When aluminum alloy is employed in the connecting rods, the cross-sectional area of the rod section is increased in order to reduce stress, compensating for the low strength of the alloy. Because of this, the rod sections of connecting rods manufactured from aluminum alloy tend to display adequate stiffness, and buckling and bending strength rarely represent issues. Taking this into consideration, the present study focused on the strength of the connecting rod small end in the vicinity of the piston pin holes, and set conditions for the analysis on this basis. Figure 1 shows the analysis conditions in detail.

Because the strength of the connecting rod was focused on only in relation to the small end around the piston pin holes, the center of the rod section between the big and small ends was cut out. The cut section was constrained only in the z direction; deformation was allowable in the directions of the x and y planes. One-half-cut models were used for each part in order to allow a greater focus on deformation and reduce calculation time. Gap elements were used to reproduce load conditions and the distances between parts, based on consideration of an analysis discussed in a previously published report<sup>(6)</sup>. Table 1 shows the material specifications used in the structural analysis. Figure 2 shows the areas for evaluation of stress. Areas

Table 1 Material specifications

| Part                                 | Piston              | Piston pin  | Connecting rod       |
|--------------------------------------|---------------------|-------------|----------------------|
| Material                             | JIS AC8B equivalent | SCM420      | JIS ADC12 equivalent |
| Heat treatment                       | T7                  | Carburizing | T6                   |
| Specific gravity                     | 2.76                | 7.83        | 2.70                 |
| Young's modulus                      | 75 GPa              | 204 GPa     | 72 GPa               |
| Fatigue strength at room temperature | 120 MPa             | 600 MPa     | 100 MPa              |
| Fatigue strength at 150°C            | 115 MPa             | 600 MPa     | 95 MPa               |

in which high levels of stress are generated were selected based on the results of previously conducted structural analyses and tests. Two areas were selected in the piston and two in the connecting rod small end, for a total of four areas. Piston pin deformation was also evaluated in order to study the relationship between stress in each of the parts and piston pin stiffness. Despite the fact that high levels of stress are generated in the piston pins, because the material fatigue strength of the pistons and the connecting rods is lower than that of the piston pins, the pins were not part of the focus for stress evaluation in this research.

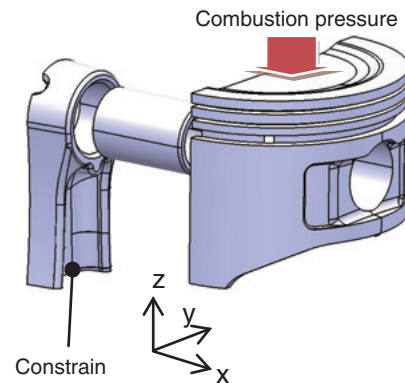


Fig. 1 Conditions for analysis

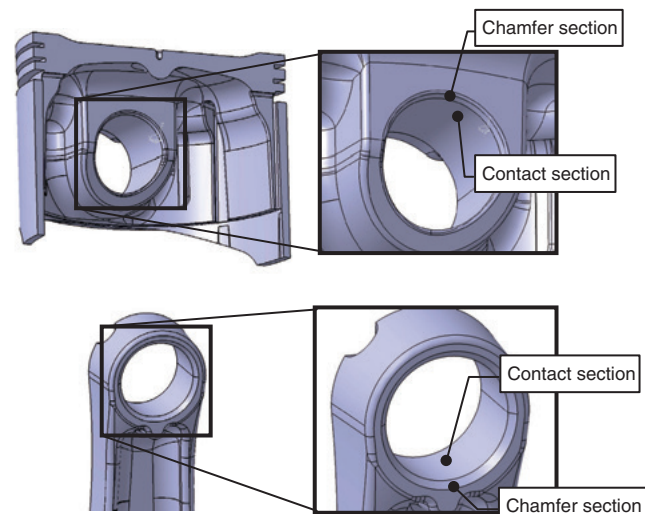


Fig. 2 Stress evaluation points

Figure 3 shows nine dimension parameters that have an effect on the four areas for stress evaluation. A parametric model was formulated, in which the three-dimensional shape was automatically updated to reflect changes in these dimensions. Dimensions were variable within appropriate limits, taking into consideration shape constraint conditions and mass producibility. In addition, of the nine dimension parameters, (e) was fixed at the initial stage of the study as a shape that reduced length by 8%. It was known from previous studies that the reduction of the length of (e) has the effect of reducing mass without reducing strength. Given this, calculation cost was reduced by reducing the degree of freedom of change in dimensions. This change reduced mass by 1.3% against the mass-production specification.

## 2.2. Optimization Algorithm

The aim of the present study was to minimize weight while attempting to help ensure that the stress generated in the four evaluation areas was equivalent to or less than figures for the mass-production shapes. The details of the optimization algorithm and the calculation conditions were as follows:

Test planning method: Uniform random number method  
 Number of initial generations: 36  
 Response curves: Kliging  
 Optimization algorithm: MOGA-II  
 Number of iterations: 30

Optimal solutions that produced the lightest models while also realizing levels of stress at the four evaluation points equivalent to or lower than the target values were selected from candidate solutions obtained from the calculations. Concrete details will be discussed in the following chapter.

With regard to software, CATIA V5 was used for structural analysis and for the formulation of the parametric model. The general-purpose optimization tool

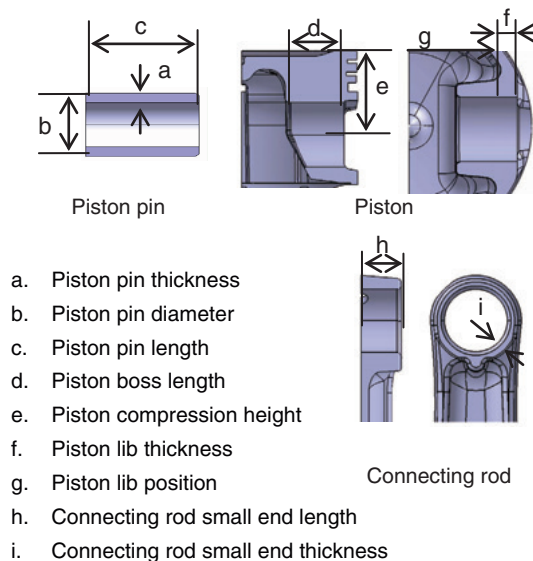


Fig. 3 Model parameters

Table 2 Engine specifications

| TYPE                                    | 4-stroke air-cooled single-cylinder gasoline engine (OHV) |
|-----------------------------------------|-----------------------------------------------------------|
| Number of cylinder                      | 1                                                         |
| Bore × Stroke                           | 68 × 54                                                   |
| Engine displacement (cm <sup>3</sup> )  | 196                                                       |
| Compression ratio                       | 8.5 : 1                                                   |
| Maximum power (kW/rpm)                  | 4.1 / 3,600                                               |
| Maximum torque (N·m/rpm)                | 12.4 / 2,500                                              |
| Rated power (kW/rpm)                    | 3.7 / 3,600                                               |
| Fuel category                           | Unleaded gasoline for cars                                |
| Dry mass (kg)                           | 17.1                                                      |
| Dimension: Length × Width × Height (mm) | 352 × 376 × 346                                           |

modeFRONTIER was employed for all processes from the implementation of the optimization algorithm to data processing.

## 3. Results of Analysis

Using the analytic method discussed in Chapter 2, shape optimization was conducted for the reciprocating parts of a general-purpose engine with a displacement of 200 cm<sup>3</sup>. Table 2 shows the specifications of the test engine.

### 3.1. Mechanism of Generation of Stress

Figure 4 shows the stress on each part when a combustion pressure load is acting on them. These results demonstrate that a high level of stress is generated on

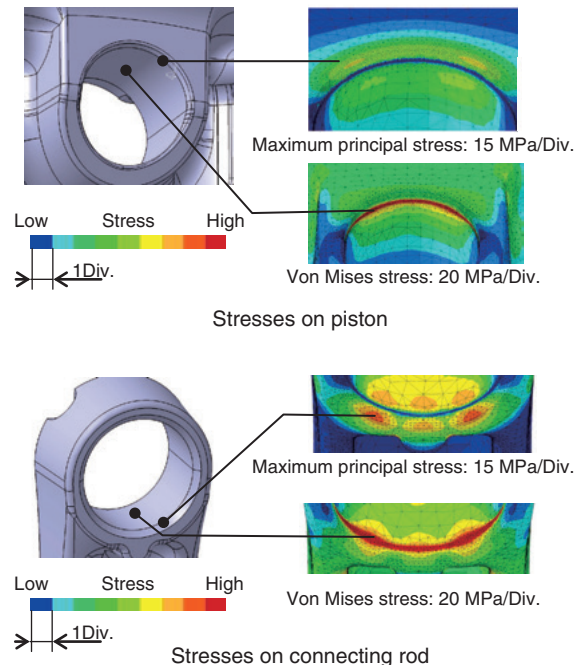


Fig. 4 Stress contour

the chamfer sections and the contact sections of both the pistons and the connecting rods, as discussed above. The method used to evaluate these two stress states was as follows. In the chamfer sections of both the pistons and the connecting rods, tensile stress in the circumferential direction is dominant, as shown in Fig. 5. Given this, maximum principal stress was used for the evaluation of these sections. The contact sections present biaxial stress fields in which compressive stress and tensile stress are respectively generated in the radial and the circumferential directions. Here, Von Mises stress was used as the indicator to make it possible to evaluate these two stress states based on a single value.

Using these stress indicators, stress on the pistons and the connecting rod small ends was compared. The connecting rods displayed 1.4x the stress on the pistons in the chamfer section, and 1.2x in the contact section. Because the level of material fatigue strength of the connecting rods is also somewhat lower than that of the pistons, the result is that the connecting rod small ends are the weakest of the reciprocating parts. This made it necessary to study specifications that would reduce weight while maintaining the strength of the connecting rod small ends. Next, the mechanism of generation of stress in the connecting rods was studied based on the state of deformation of the piston pins. The piston pins display two main deformation modes when a combustion pressure load is applied. The first of these shows bending deformation, as shown in Fig. 6(a). The second shows crushing deformation, in which the annulus is flattened, as shown in Fig. 6(b). Figure 7 shows the results of an analysis of the relationship of correlation between these two states of deformation and stress on the connecting rod small ends. The coefficients of correlation between each type of deformation and each type of stress were as follows: For the chamfer sections, 0.8 (a) for bending deformation and 0.5 (b) for crushing deformation;

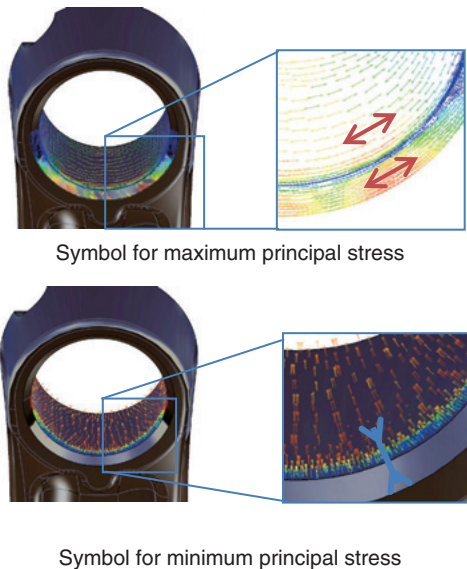


Fig. 5 Symbols for stress

and for stress in the contact sections, 0.9 (c) for bending deformation and 0.7 (d) for crushing deformation. Each type of stress therefore displayed a correlation with piston pin deformation. This indicated that the reduction of deformation in the piston pins would be effective in reducing stress on the four areas under study.

### 3.2. Identification of Optimum Shapes

Figure 8 shows the relationship between stress in each of the four areas for evaluation of stress and total mass. The solutions on the curved boundary line are Pareto solutions, and represent candidates for optimal solutions. The red lines show the mass of the reciprocating parts for the mass-production shapes, and the blue lines show stress on each part for the mass-production shapes. In the graphs, only solutions that satisfy the constraint conditions for mass and stress are plotted as black; solutions that do not satisfy the conditions are plotted as gray. The shapes that realized the

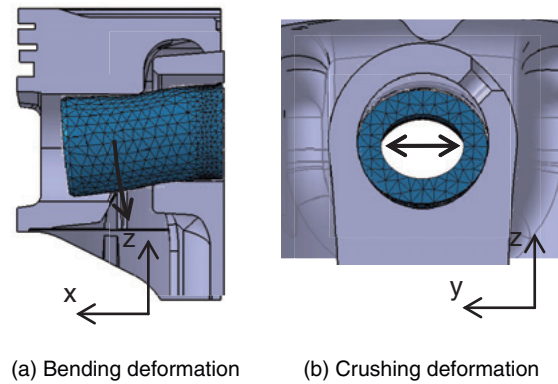


Fig. 6 Piston pin deformation

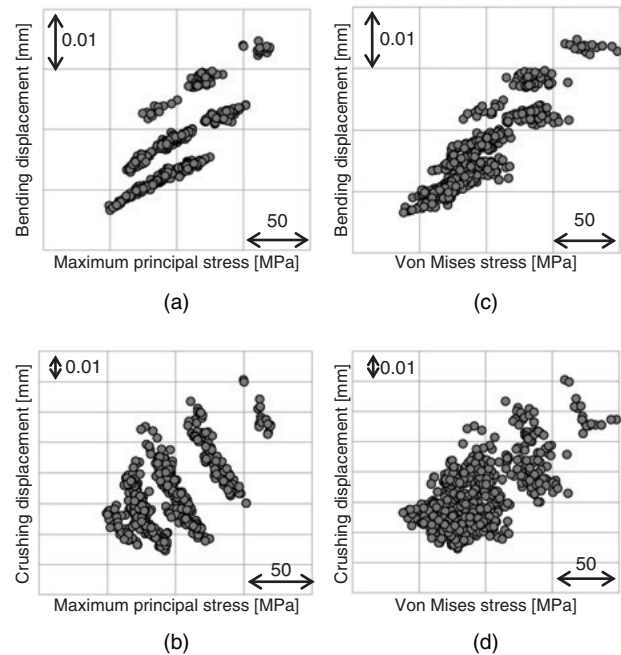


Fig. 7 Correlation matrix charts

lightest weight were selected from among the solutions satisfying stress conditions in each of the four evaluation areas as the optimal solutions. The shapes judged as optimal are shown as orange in the graphs. The results show that it is possible to reduce the weight of the optimal shapes by 10% against the mass-production shapes.

### 3.3. Characteristics of Optimal Shapes

Analysis of optimal shapes demonstrated the following characteristics. For the piston pins, diameter was reduced by 11% and length by 3%, while thickness remained the same as the conventional shape, for an overall reduction in mass of 20%. In the case of the connecting rod small ends, while a reduction in diameter increased the annular thickness of the pin holes by 25% against the mass-production shape, because the annular length of the boss was reduced by 8%, the mass of the part only increased by 2%.

These results show that despite the fact that there will be an increase in the mass of the connecting rod unit, it is possible to reduce the size of the piston pins.

### 3.4. Verification in Actual Engine

The optimal reciprocating part shapes selected in Section 3.3 were manufactured as prototypes, and their strength was tested in an actual engine. The prototype parts were fitted in a test engine, a static load corresponding to combustion pressure was applied to the combustion chamber, and stress was measured using a strain gauge. Figure 9 shows the conditions for measurement in the actual engine. Figure 10 shows the results of a comparison between stress measured in the chamfer sections of the piston pins and connecting rods and stress values predicted in the structural analysis. The measured stress values are within  $\pm 5\%$  of the predicted values. It is conjectured that this represents error due to

intersection of the clearance between the fully floating sections.

When using the developed system in product development and similar processes, it will be necessary to set a safety factor that considers factors such as dimensional variations of this type.

### 3.5. Analysis of Optimal Results

The orientation for weight reduction via the shapes discussed in the previous section was reduction of piston pin diameter and length and increase in the thickness of the connecting rods. The combination of these shapes can be considered an important element in reconciling strength and the reduction of weight in the reciprocating parts. Combination of three dimensions was studied. Using the results of optimization calculations, shapes that resembled each other were clustered in order to form groups for this study.

By means of grouping of shapes, the candidate optimal shapes were divided into three groups. The characteristics of each group were as follows.

Group A: Weight was reduced by reducing the diameter of the piston pins and stiffness was increased by increasing the annular thickness of the connecting rod small end.

Group B: Weight was reduced by reducing the diameter of the piston pins, and stiffness was increased by increasing the thickness of the piston pins.

Group C: Stiffness was increased by enlarging the diameter of the piston pins, and weight was reduced by reducing the thickness of the piston pins and reducing the annular thickness of the connecting rod small ends.

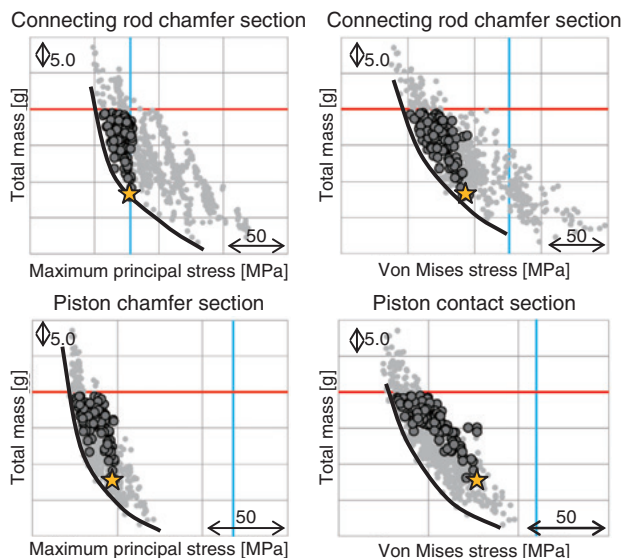


Fig. 8 Scatter diagrams

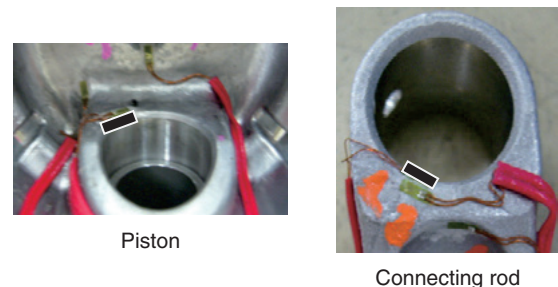


Fig. 9 Measurement of gauge stress

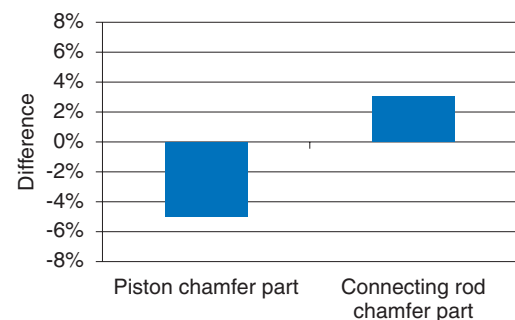


Fig. 10 Difference between measurement and calculation

Figure 11 shows a pattern diagram of the three relationships between dimensions and mass in these groups. In this research, the shapes in Group A were selected as offering the greatest reduction in weight.

The results discussed above indicate that in order to maintain strength and reduce weight in engine reciprocating parts using aluminum-alloy connecting rods, the appropriate design orientation is to reduce weight by reducing the diameter and length of the piston pins, while supplementing the reduced stiffness by increasing the annular thickness of the connecting rod small end.

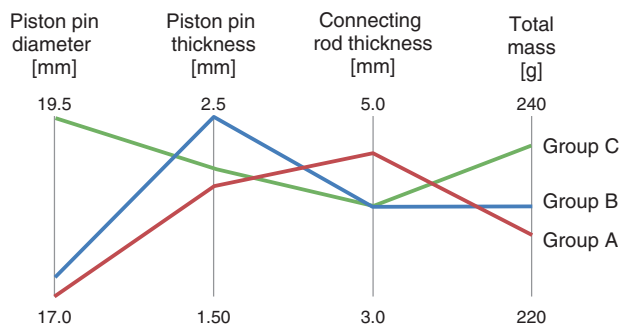


Fig. 11 Groups of optimal solutions

#### 4. Conclusion

The effectiveness of a structural optimization system for the optimization of the shape of reciprocating parts for general-purpose engines using aluminum-alloy connecting rods has been verified. The system has made it possible to introduce specifications that reduce the weight of the piston pins by 20%, maintaining strength while reducing the weight of the reciprocating parts. The lighter specifications reduce the diameter of the piston pins while maintaining the same thickness as the mass-production specifications, and increase the annular thickness of the connecting rod small end. The new specifications were tested in an actual vehicle, and satisfied strength criteria. This verified the effectiveness of the structural optimization system. The results of the research discussed in this paper were as follows:

- (1) The weakest section of the reciprocating parts is the chamfer section of the connecting rod small end, due to stress originating in bending and crushing deformation of the piston pins.
- (2) Three dimensions are related to stress on the connecting rod small end. By reducing the diameter of the piston pins, employing equivalent piston pin thickness to the mass-production specifications, and increasing the annular thickness of the connecting rod small end, it was possible to reduce the unit mass of the piston pins by 20% while maintaining the strength of the reciprocating parts.
- (3) When the strength of the weight-reducing specifications was tested in an actual engine, the results displayed an

error of around  $\pm 5\%$  with predicted values. Because of this, the application of the method to actual engine developments will necessitate the setting of safety factors that take into consideration the range of error.

- (4) The three dimensions discussed above were classified into three groups that would reduce weight while maintaining strength. The weight-reducing shapes selected in the present research belonged to the group that realized the greatest weight savings.

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