International Journal of Automotive Technology, Vol. 8, No. 3, pp. 319–325 (2007)

1229–9138/2007/034–08

PROCEDURE FOR COMPUTER-AIDED PRELOAD SELECTION OF ENGINE CONNECTING-ROD BOLTS

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(Received 28 August 2006; Revised 3 April 2007)

ABSTRACT–Preload of critical engine bolts affects the performance and durability of engines. In modern engines that pursue higher power outputs and which are of lighter weight, it becomes more difficult to select an optimal target preload in consideration of various factors such as the role and structural characteristics of joint members, joint load, and fatigue durability of bolts and joint members. A procedure to select the bolt preload using computer-aided engineering technology, especially the finite element method, has been developed. The procedure is illustrated with connecting-rod bolts for which an appropriate preload is known. The selection criteria of target preload and the finite element modeling technique for connecting-rod bolts are also explained.

KEY WORDS : Bolt preload, Computer-aided engineering, Finite element analysis, Engine connecting-rod, Joint stiffness, Fatigue durability

1. INTRODUCTION

Bolts are used to clamp joint members together with enough force to prevent them from separating. Bolts in internal combustion engines are also used as clamping means. However, the initial clamping force, i.e., the preload of some critical engine bolts affects the engine performance. For instances, the connecting-rod bolts and the main bearing cap bolts are involved in the construction of journal bearings through which the engine power is transmitted. The cylinder head bolts provide the cylinder head gasket with elastic energy which is used to generate the sealing force (Cho *et al.*, 2005). The crank pulley bolts and flywheel bolts serve as elements for power transmission. If the preload of these bolts are not sufficient, the bolted components cannot fulfill their roles successfully, and thus engine power and efficiency diminish.

The preload of critical engine bolts also affects the fatigue durability of engines. It is noted that the bolted joints are subject to varying loads during engine operation. When the bolt preload is too small to prevent the joint separation, a larger portion of the joint load is transmitted to the bolt, and thus the bolt load fluctuates more significantly. On the other hand, the excessive preload increases the mean load of the bolt (Bickford, 1995). Both the extreme cases may result in the fatigue fracture of the engine as well as the bolts.

The reliability of the bolted joints depends on various factors such as the reliability of target bolt preload, quality of the bolt, accurate determination and application of the tightening torque and/or angle to generate the target preload. Provided that the bolt quality is assured, the target bolt preload should be selected reliably and then applied accurately during the assembly. These days, the target bolt preload can be applied accurately with the aid of various techniques including the bolt stretch control technique that correlates the preload with the tightening torque and/or angle using depth micrometers, gage pin bolts, ultrasound, strain-gauged bolts, or force washers (Corbett, 1998; Shoberg, 1998). If the target bolt preload itself is inappropriate, the bolted joints may fail although the bolt is tightened accurately to the target preload. In this case the target bolt preload is usually sought with engine test runs at various bolt preloads. If an appropriate target bolt preload cannot be found with the test runs, the joint members as well as the bolts should be redesigned to increase the strength, stiffness and durability of the bolted joints, and then the same process is repeated. Since this process requires much time and cost, selecting the target bolt preload reliably is of the utmost concern in the design of bolted joints.

The target bolt preload should be selected in consideration of various factors such as role and structural

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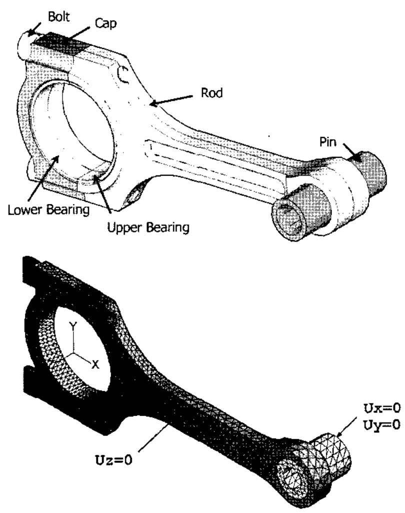

Figure 1. Schematic representation and finite element model of connecting-rod.

characteristics of joint members, joint load, and fatigue durability of bolts and joint members. Most of the automobile companies, to the best of the authors' knowledge, employ simple empirical equations to determine the target bolt preload at the early stage of engine development. Since the contributing factors are considered as a constant in the equations, the determined target bolt preload may be unreliable and, therefore, a large safety factor is adopted. It is noted that too large a preload leads to an overdesign of engine.

Since modern engines pursue higher power output and lighter weight, the automobile companies prefer smaller bolts, and have been substituting the yield tightening methods for the elastic tightening methods (Wallace, 1998). This means that the appropriate bolt preload exists in a narrower range so that more attention should be paid to the selection of the target bolt preload. This demand may be satisfied with computer-aided engineering (CAE) technology including the finite element method. It is a surprise that little literatures on the CAE approach for selection of the target bolt preload have been published. The present authors have developed a procedure to select the target preload of critical engine bolts systematically using finite element analysis with fatigue data. This paper illustrates the procedure with engine connecting-rod bolts for which an appropriate preload is known. The selection criteria of the target preload and the finite element modeling technique for the connecting-rod bolts are also explained.

2. FINITE ELEMENT ANALYSIS

2.1. Finite Element Model

The finite element analysis is conducted with a commercial finite element code, ABAQUS. Figure 1 shows shape and finite element model of an engine connecting-rod that consists of a rod, a bearing cap, a pair of bearing shells, and two bolts. The rod and bearing cap are assembled with bolts at the big end, and the bearing shells are interference-fitted into the big-end bore. The piston pin is interference-fitted into the bore of small end. A half finite element model is constructed because the shape and deformation of connecting-rod is symmetric with respect to the center plane. Each parts is modeled separately and then assembled as the real parts. However, the threaded parts of the bolt and rod are modeled as a cylinder with a cross-sectional area equal to the tensile stress area of the bolt. The bolt preload is generated with the reduction in the length of bolt shank. The contact surfaces between the bolt and rod are assumed to be tied whereas the other contact surfaces (cap/rod, bolt/cap, lower bearing/cap, upper bearing/rod, upper/lower bearings, and pin/rod) are allowed to be separated and/or slip with a friction coefficent of 0.2. All the nodes on the cut-off plane are constrained in the normal direction. The end surface of the piston pin is constrained to prevent the rigid body motion of the model.

The model has been developed for quasi-dynamic analysis in which inertia and reaction forces are treated as external loads. Figures 2(a) and 2(b) show the inertia force and moment at the center of mass by way of illustration when the crank shaft rotates at a constant angular velocity. The inertia loads vary with the crank angle during one engine cycle. In addition, the inertia force also varies with the location in the connecting rod. This means that the connecting rod is subject to non-uniform distributed body force and moment that vary with the crank angle. In order to mimic the nature of the inertia loads in the model, a subroutine has been developed that estimates the inertia loads of each elements at any given crank angle by the aid of dynamic analysis of the slider-crank mechanism (Wilson *et al.*, 1983).

Figures 2(c) shows the resultant reaction force at the big end. Since the big end is a dynamically-loaded journal bearing, the load should be inputed into the model as the oil film pressure. There are three methods which are most-widely used for the conversion. In the first method, which is based on an experimental observation (Webster *et al.*, 1983) that the pressure profile can be approximated as a cosine function along the circumference and as a parabolic function along the width, the resultant load is converted directly to the pressure of the presumed profile.

The others employ the hydrodynamic (Booker, 1965) and the elastohydrodynamic (EHD) journal bearing theory. Major distictions among the methods are as follows. The presumed pressure profile (PPP) method requires the least amount of informations about the bearing, and yields the pressure profile in a short time without employing a

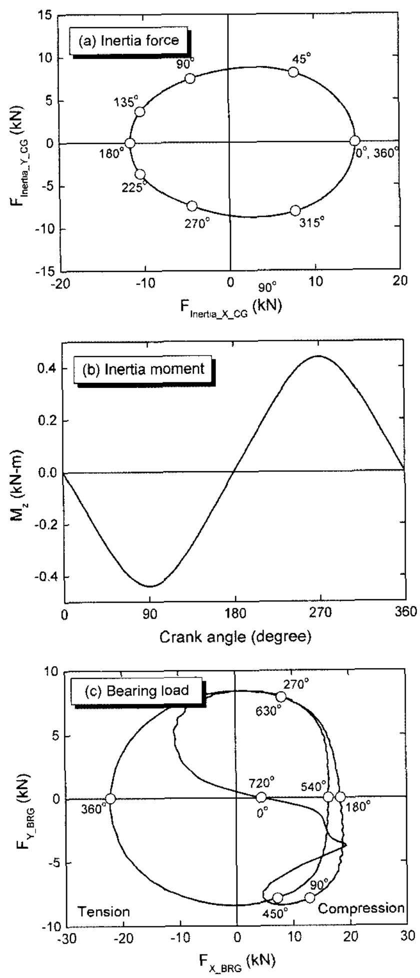

Figure 2. External forces of the connecting-rod in the quasi-dynamic analysis.

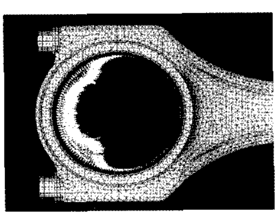

Figure 3. Nodal forces equivalent to the oil film pressure of the big end bearing at a crank angle of 360°.

numerical scheme, but the profile is the most inaccurate. The EHD method requires the most details of the bearing including the stiffness, and estimates the profile numerically, but the profile is the most realistic. By way of illustration, Figure 3 shows the nodal forces equivalent to the oil film pressure that was obtained with a commercial engine bearing EHD analysis code, AVL EXCITE. The HD method possesses the intermediate features. Moreover, the PPP method is readily used in the connecting-rod

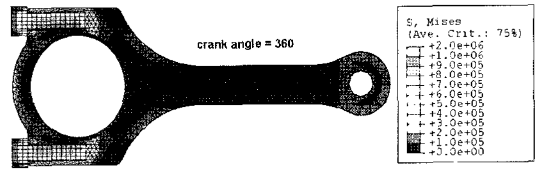

Figure 4. Finite element analysis result showing equivalent stress distribution.

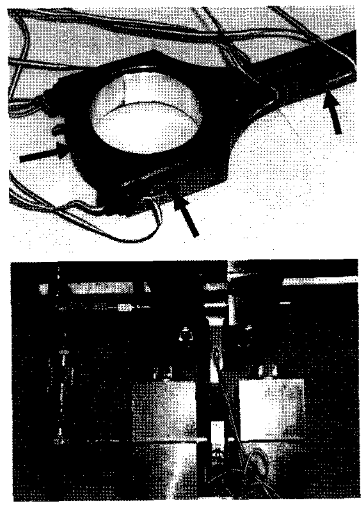

Figure 5. Experiment setup for verification of the finite element model.

analysis, whereas in the EHD method the pressure profile should be obtained with additional analyses and then transferred to the connecting rod analysis model, implying a significant amount of time and cost. A suitable method should be chosen with respect to the purpose of analysis. The PPP method may be the most suitable at the early stage of engine development where both the bolt preload and the connecting-rod design are determined concurrently. However, the EHD method is preferred for assessing the final design. In the present work the PPP method was adopted. Both the estimation of the big end load at various crank angles and the conversion to the pressure load are included in the subroutine mentioned above.

The small end load should be applied as a Hertzian pressure. Since the piston pin is interference-fitted into the bore to generate the initial Hertzian contact pressure and its end surface is constrained, the reaction pressure load is generated during the analysis. Thus, it is not inputed into the model.

The analysis starts with assembling the connecting-rod, during which stresses due to bolt-tightening and interference-fitting are generated. Then the quasi-dynamic analysis is conducted incrementally as a function of crank angle for one engine cycle. Figure 4 shows the equivalent

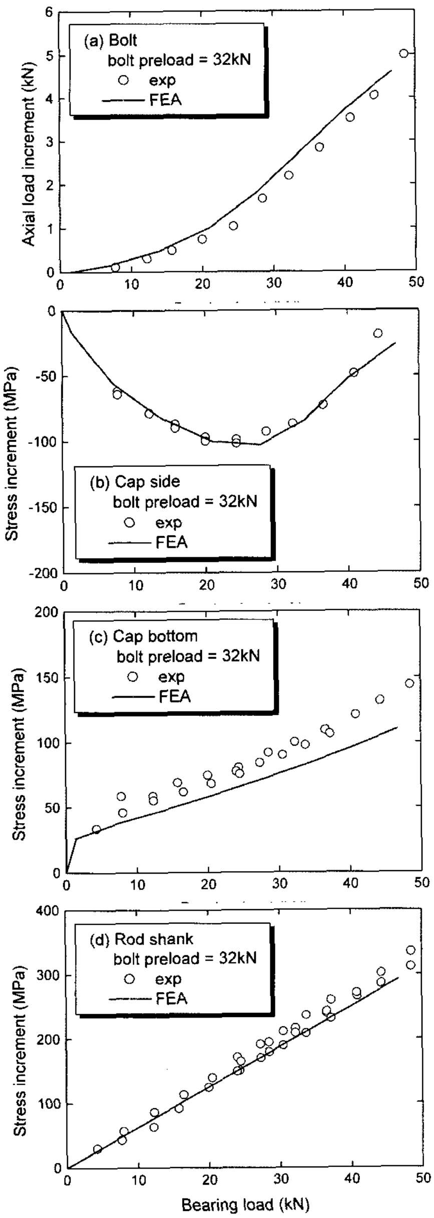

Figure 6. Verification of the finite element model.

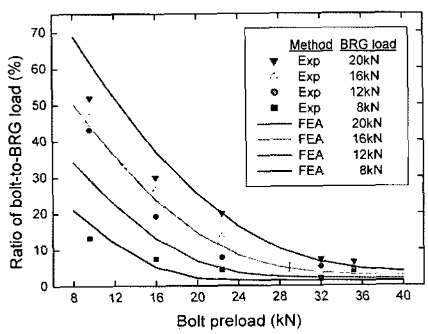

Figure 7. Dependence of the bolt-to-bearing load ratio on the bolt preload and the bearing load.

stresses at the crank angle of 360° where the maximum tensile bearing load is applied. The bolt stresses are relatively large and exhibit a gradient due to bending.

2.2. Model Verification

Experiments were conducted to verify the model. Figure 5 shows the experimental setup. Strain gauges are attached to the bottom and side of the cap, and the side of rod shank. The connecting-rod is assembled with two strain-gauged bolts at the preload of 32 kN, and pins are inserted into each bore of the big and small ends. Oil is supplied into the gap between the pin and the big end bearing. While the pin at the big end is fixed, a sinusoidal load is applied to the pin of the small end in order to generate an oil film pressure at the big end bearing. The axial load of the bolts and the stresses in the connecting-rod are measured at various amplitudes of the sinusoidal load.

Figure 6 shows the increments of the average axial bolt load and the stresses of the connecting-rod at various tensile bearing load. The bolt load increases in proportion to the bearing load, and the slope of the curve changes near the bearing load of 25 kN due to the partial separation of the rod/cap joint. The side of the cap is compressed due to the bending as well as tensioned due to the tensile bearing load itself. The result shows that the compression dominates the tension in a practical range of loads. The dominance of compression is the most pronounced near the bearing load of 25 kN. This means that a

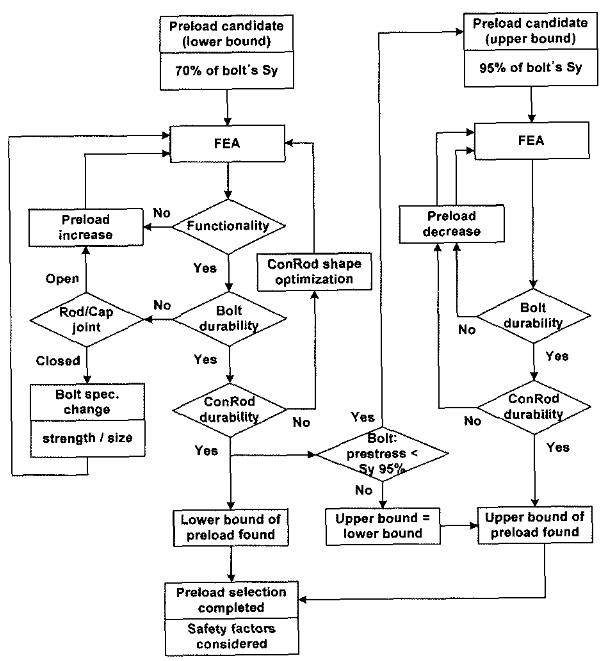

Figure 8. A procedure to select the target preload of connecting-rod bolts.

larger tensile bearing load compresses the cap side more before the rod/cap joint is separated. The stresses at the bottom of the cap and the rod shank increases linearly with the bearing load. Since the analysis results (line curves) agree well with the experiment results (symbols), it is argued that the model is reliable for the stress analysis.

Figure 7 shows the dependence of the bolt-to-bearing load ratio on both the bolt preload and the bearing load. The load ratio is defined as the ratio of the bolt load increment to the bearing load. As the bolt preload increases, the load ratio decreases to a constant value. The constant load ratio is achieved at the relatively high bolt preload as the bearing load increases. It is noted that the constant load ratio results from no separation at the rod/cap joint. Since both the experiment (symbols) and the analysis (line curves) yield similar results, it is argued again that the model is reliable.

3. BOLT PRELOAD SELECTION

3.1. Selection Criteria

The big end of the connecting-rod should be assembled rigidly to behave as a single body so that the bearing plays its role successfully, and the big end does not fail due to fatigue. Criteria to assure the functionality and durability have been estabilished as follows.

Functionality criterion: Separation and slip should not occur at the joints between the upper and lower bearing shells.

Durability criterion: Fatigue failure should not occur due to either insufficient or excessive bolt preload.

No separation and no slip conditions at the rod/cap joints are not included in the functionality criterion because the interference-fitted bearing can play its role successfully even when partial separation or slip occurs at the joints. This is verified in section 3.3.1.

Since appropriate bolt preload exists in a range, the lower bound is determined with both the functionality and the fatigue failure due to insufficient bolt preload, whereas the upper bound with the fatigue failure due to excessive preload.

3.2. Selection Procedure

Figure 8 shows a procedure to select the target preload of connecting-rod bolts. The procedure looks for the lower and upper bounds of the appropriate bolt preload range, and then the designer selects the smallest value in the range in consideration of safety factor. The procedure also helps the designer modify the design systematically in case an appropriate preload range does not exist for a given design. It is noted that the procedure can be applied to the other bolted joints with modification of the bolt preload selection criteria.

The procedure starts with the finite element analysis for a relatively low value of the bolt preload that is expected to be appropriate from experience. In Figure 8 the preload equivalent to 70% of the bolt yield strength is given as an example. The analysis result is used to examine whether separation or slip occurs at the joints of the upper/lower bearing shells. Unless the functionality criterion is satisfied, the analysis is conducted again for an increased preload value. Repetition of the loop eventually yields the minimum preload to guarantee the functionality.

The second step assesses the fatigue durability of the bolts by the aid of high-cycle fatigue theory with the analysis result. If the bolt fracture is expected and if the separation occurs at the rod/cap joint, the analysis is conducted again for an increased preload value. It is recalled that the separation at the rod/cap joints results in the increase in the bolt-to-bearing load ratio and thus significant fluctuation of the bolt load. This loop yields the minimum preload that guarantees the fatigue durability of the bolts. If the bolt fracture is anticipated without any separation at the rod/cap joints, the strength or size of bolts should be increased because Figure 7 implies that a higher bolt preload increases only the mean value of alternating bolt load without reducing the amplitude. Preferred is to increase the bolt strength since the design of the connecting-rod need not be changed. If the bolt strength cannot be increased, its size is increased.

The third step assesses the fatigue durability of the rod and the cap. If no failure is expected at the preload obtained in the previous steps, this preload value is the lower bound of appropriate preload range. Otherwise,

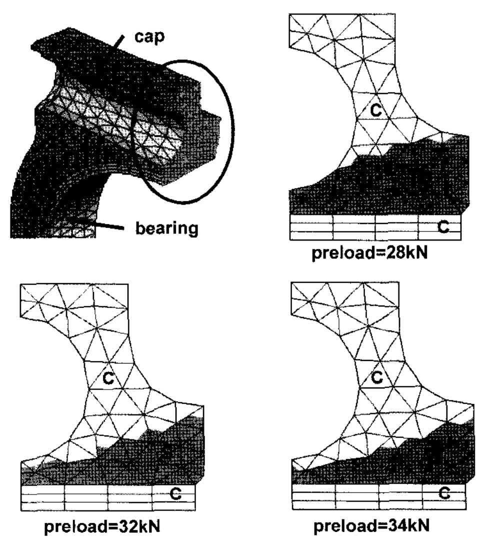

Figure 9. Contact status at the joints of rod/cap and upper/lower bearing shells. The letters C and S stand for closed and separated, respectively.

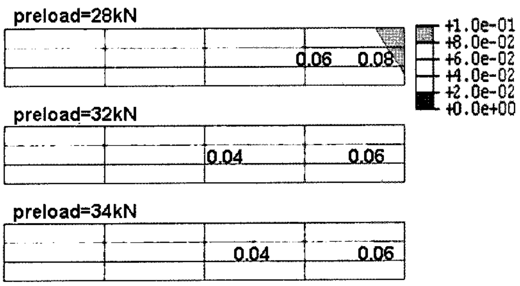

Figure 10. Shear-to-normal stress ratio in the contact surface between the upper and lower bearing shells.

weak points of the rod or cap should be reinforced.

The upper bound is determind with a process similar to that for the lower bound. The analysis starts with an anticipated upper bound value such as that equivalent to 95% of the bolt yield strength in Figure 8. If the joint members are expected to fail by fatigue, the analysis is repeated for a reduced preload value until the upper bound is found. It is noted that the upper bound determination procedure may be omitted in case that the lower bound is close to the yield strength of the bolt.

At the final step the designer selects the smallest preload value in the range in consideration of the safety factor.

3.3. Application and Verification

Application of the procedure is demonstrated and the reliability of both the preload selection criteria and the finite element model is verified for the connecting-rod bolts for which an appropriate preload is known from the engine tests.

3.3.1. Functionality

Figure 9 shows the contact status at the joints of the rod/cap (region of triangles) and upper/lower bearing shells (region of rectangles) at a crank angle of 360° for bolt preloads of 28, 32 and 34 kN. It is noted that the joint is subject to the strongest bearing tensile load at a crank angle of 360°. The rod/cap joint separates in the dark region (denoted by S) while no separation occurs at the joint of bearing shells. The separated region shrinks with the increase in the preload, and the bolt hole periphery is partially separated at the preload of 28 kN. It is demonstrated in section 3.3.2 that the bolt fractures by fatigue at a preload of 28 kN.

Figure 10 shows the shear-to-normal stress ratio at the joint of bearing shells. The ratio is greater for the higher preload. The largest ratio that occurs at the upper right corner is smaller than 0.1. Since the friction coefficient is 0.2, no slip occurs between the bearing shells for all the three preloads.

It is summarized that the functionality condition is

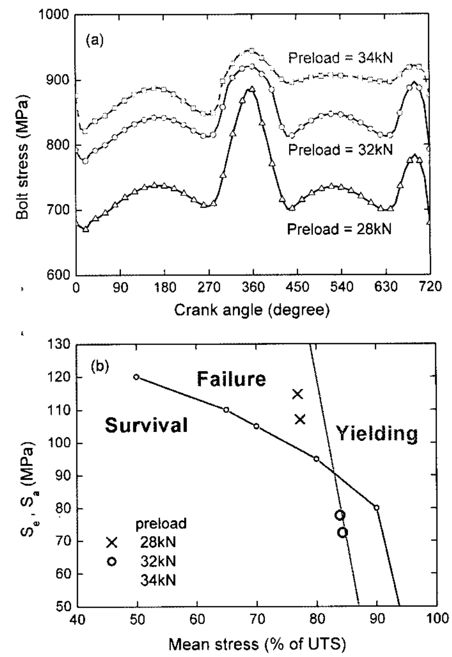

Figure 11. Variation of bolt tensile stress during one engine cycle and assessment of bolt fatigue durability.

satisfied for all the three preload values. It is noted that the preload 32 kN was proven to be appropriate in the engine tests.

3.3.2. Fatigue durability

Figure 11(a) shows the variation of axial bolt stress at the first bolt-nut engagement point during one engine cycle. It has been known that most of the bolt fatigue failure occurs at the first engagement point. The bolt is subject to a variable amplitude cyclic stress. The amplitude of the primary cycle diminishes with the increase in the bolt preload. Since infinite lives are required for the bolts, only the primary cycle is used to obtain the stress amplitude and mean stress for fatigue assessment.

Figure 11(b) shows the Haigh diagram of the bolt endurance limit with the analysis results obtained from Figure 11(a). The endurance limits at various mean stresses were obtained from the bolt fatigue tests under tensile load. It is noted that the Goodmann equation for compensation of the mean stress effect on the endurance limit cannot be applied to the bolt that is a kind of notched member. The bolts are expected to fracture at the preload of 28 kN, but not at the preloads of 32 and 34 kN. Meanwhile, plastic deformation of the bolts are anticipated at the preload of 34kN. Hence, only the preload of 32 kN can guarantee the fatigue durability of bolts without any plastic deformation.

Figure 12 shows the distribution of the fatigue safety factor of the rod and cap. The fatigue safety factor is

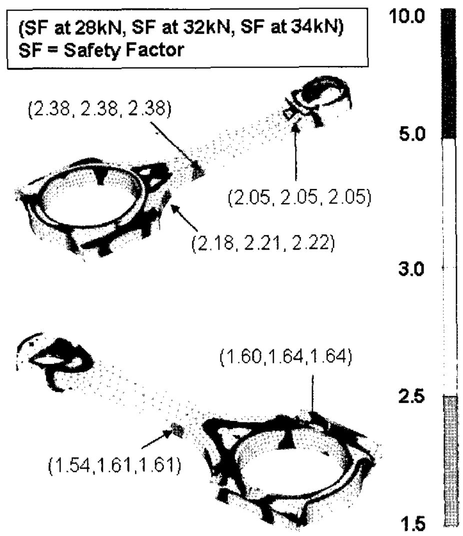

Figure 12. Fatigue durability safety factor of rod and cap.

defined as the ratio of the endurance limit to the equivalent stress amplitude. A commercial finite element fatigue code, FEMFAT was employed for the calculation with all the stress data during one engine cycle and the endurance limit 330 MPa at the stress ratio of zero. For the three preload values, the safety factor at all locations is greater than one. Hence, both the rod and the cap will not fracture by fatigue.

It is summarized that only the preload 32 kN can guarantee both the functionality and durability of the connecting-rod. This result agrees with the result of engine tests conducted in the Hyundai Motor Company. Hence, it is claimed that the procedure, criteria, and finite element modeling and analysis methods for the selection of bolt preload are reliable.

4. CONCLUSION

A procedure to select the bolt preload using computer-aided engineering technology, especially the finite element mothod, has been developed. The procedure also helps the designer modify the design systematically in case an appropriate preload range does not exist for a given design.

Application of the procedure was demonstrated and the reliability of both the preload selection criteria and the finite element model was verified for the connecting-rod bolts for which an appropriate preload is known. It was illustrated that the finite element analysis provides invaluable information on the deformation and contact behaviors of the bolted joint, bolt, and joint members which is difficult to obtain in experiments.

The procedure can be applied to the other bolted joint where the joint stiffness is of concern. An expansion of the procedure into the other types of bolted joints, such as gasketed joints, is in progress. The result will be reported in the near future.

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