Stress Analysis Method for Bolt Thread Roots of Automotive Engine under Operating Condition

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Abstract*: In order to analyze the stress at the thread roots of a bolt of an automotive engine under an operating condition, we developed a stress analysis method for it, which consists of a multi-body dynamics analysis, a linear transient analysis and nonlinear static analyses. It is able to consider the motion of bolted structures, the contact behavior and the plastic deformations with practical calculation costs. In this paper, we showed that the developed method is applied to in-line 4 engine systems as an example.*

Keywords*: Stress Analysis, Stress Amplitude, Fatigue, Thread roots, Bolt, Engine Vibration, Operating Condition, Finite Element Analysis, submodel Analysis, Zooming Analysis, Contact, Friction, Plasticity, Multi-Body Dynamics, Dynamic.*

1. Introduction

Bolted joints are widely used in mechanical structures. The thread roots of bolt are, because of their V-shapes, well known as sites where stress concentration occurs. Therefore, in designing bolted structures with specified fatigue strength, it is important to accurately predict the stress amplitude occurring at the thread roots of a bolt.

Since bolted structures are usually operated under complex loading, the motion of the structures under the operating condition must be taken into account for accurate prediction of the stress. Moreover, since the boundaries between bolted structures are in contact/friction conditions and a part of the thread roots of a bolt are thought to be yielding, nonlinearities of both boundary and material must also be taken into account. Nonlinear problems are, however, generally more expensive in terms of calculation cost than linear problems, and therefore it is practically difficult to take into account in analyses such nonlinearities for the whole bolted structures.

In this paper, we propose an analysis method using the finite element analysis (FEA), which is capable of predicting the stress of the thread roots of a bolt-hole with a motion and nonlinearities of boundary and material. We show an application of the proposed analysis method to thread roots of bolt-hole of an in-line 4 automotive engine as an example.

2. Method Overview

Figure 1 shows the flow of the proposed analysis method, which is applied to calculate the stress of the thread roots of an automotive engine under an operating condition. It consists of four tasks some of which uses submodel techniques. Task1 is a flexible multi-body dynamics analysis to calculate the loads on the bearings and the cylinder walls. Task2 is a linear transient finite element analysis with the operating loads obtained by the previous dynamics analysis. Task3 is a nonlinear static finite element analysis to obtain the displacements around the subject bolts. Task4 is another nonlinear static finite element analysis to calculate the stress at the thread roots of a bolt.

The proposed analysis method has two features below:

- Transition from the linear transient analysis (Task2) to nonlinear static analysis (Task3) considering nonlinearities such as contact and plasticity.

- Use of nodal forces, instead of nodal displacement, as the boundary conditions for the submodel analysis Task3.

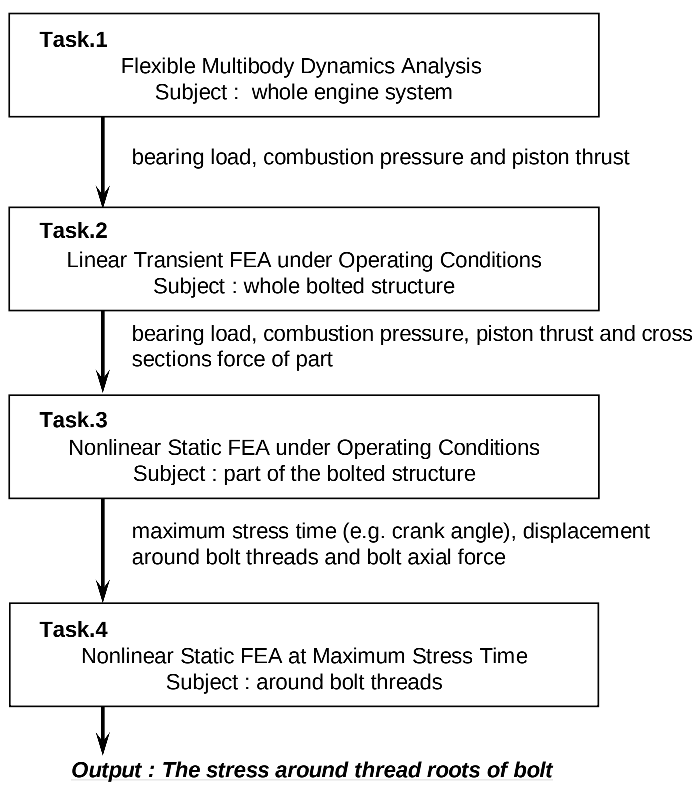


Figure 1. Flow of the proposed analysis method.

3. Multi-body dynamics analysis with flexible body under operating condition (Task1)

The purpose of this task is to obtain the time response of loads on the bearings, the combustion chambers and the cylinder walls. These loads are calculated by multi-body dynamics analysis with flexible bodies such as pistons, crankshaft and engine blocks and are used in the subsequent analysis (Task2). An in-line 4cylinder gasoline engine is taken for the following case study.

4. Linear transient finite element analysis of whole engine system under operating condition (Task2)

The purpose of this task is to obtain the vibration responses of the whole engine model by a linear transient analysis, which are used as the boundary conditions for the subsequent nonlinear analysis. Figure 2 shows the finite element model of the whole engine system ("the whole model" in the following) used in the linear transient analysis. Because the surface elements between each contact part, such as the cylinder block, the cylinder head and the ladder frame are modeled by using common nodes, nonlinear deformations such as slipping and opening/closing are not allowed during this analysis.

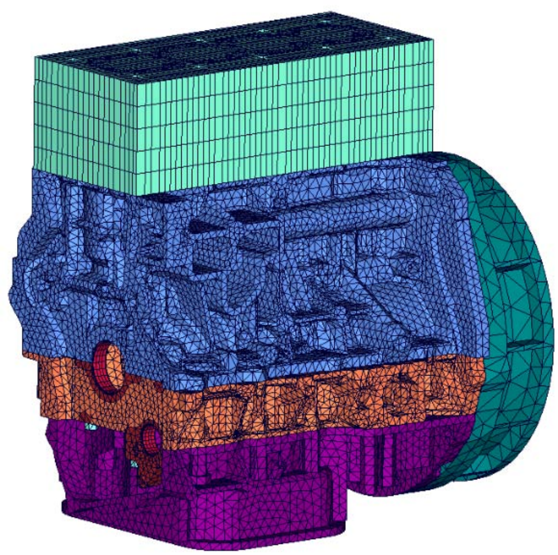


Figure 2. The whole model used in Task2.

The loads obtained from the previous dynamics analysis are applied to the bearings, the combustion chambers and the cylinder walls. As an example, Figure 3 shows the load characteristic on the main bearing No. 2 corresponding to two rotations of the crankshaft. The loads at the 20 and 550 degrees of the crank angle are much greater due to the first and second cylinder combustions than it is at the other crank angle.

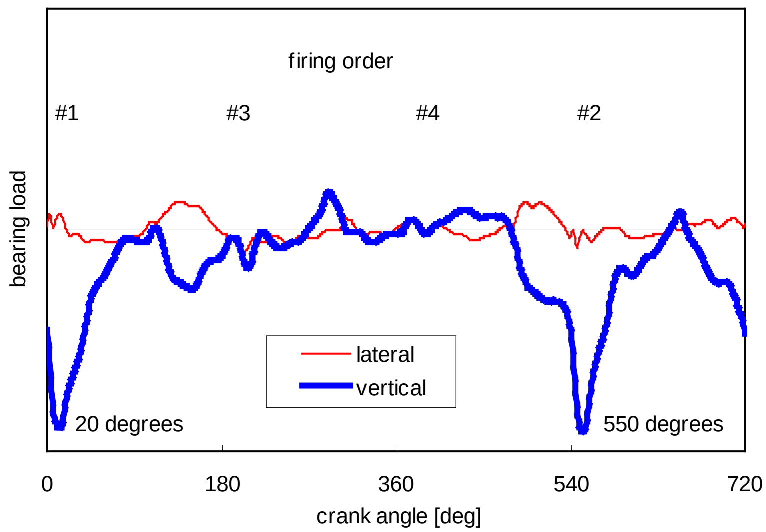


Figure 3. Bearing loads on the main bearing No.2.

5. Nonlinear static finite element analysis under operating condition (Task3) with Abaqus/Standard ver.6.6

The purpose of this task is to obtain the displacements around the subject bolt. They are used as the boundary conditions for the subsequent nonlinear static analysis with a fine mesh model of the subject bolt. In order to accurately calculate the displacements in this analysis, nonlinear analysis is necessary, which takes into account the axial forces of the bolts and the contact between bolted structures. However, if the whole engine finite element model is used for this nonlinear analysis, the calculation costs can be impracticable high.

To avoid this issue, the submodel analysis is employed, where an arbitrary part including the subject bolt is extracted from the whole model to decrease the number of degrees of freedom. In our case study, the bulk No.2 shown in Figure 4 is extracted ("the bulk model" in the following). Where, bulk is the part between centers of the adjoined two pistons.

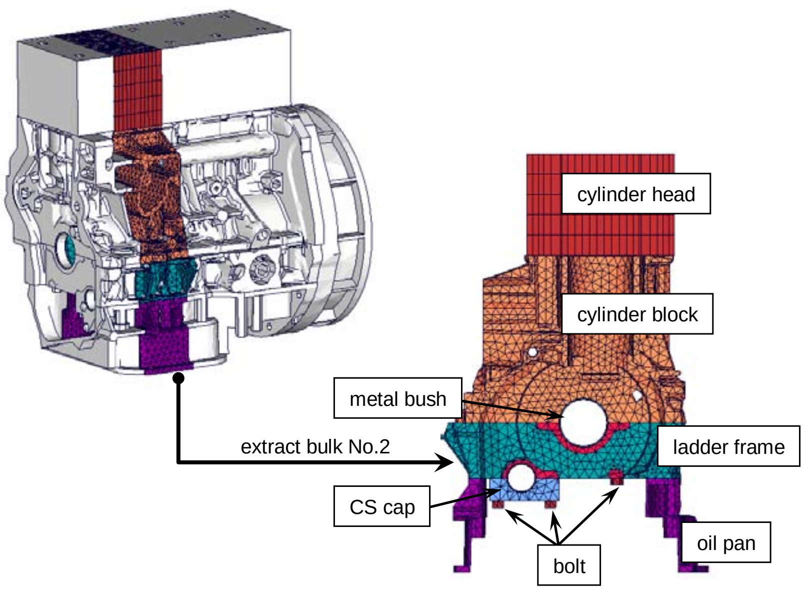


Figure 4. The bulk model used in task 3.

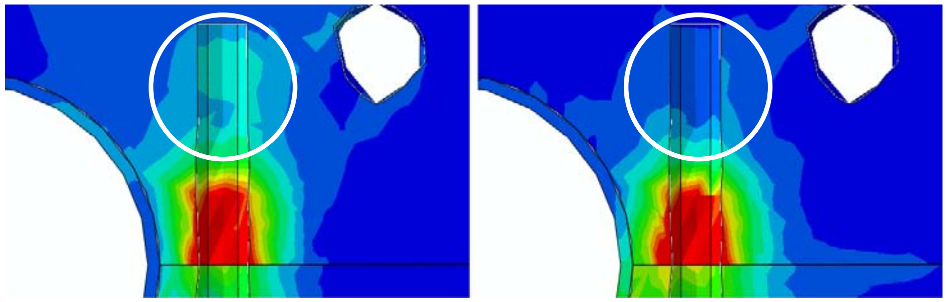
Two parts are changed to convert the whole model to the bulk model. The first part is between the cylinder block and the ladder frame, and the other part is between the ladder frame and the control shaft cap. Each part is changed from a common node joint model to a contact pair model.

The loads on the bearings, the combustion chambers and the cylinder wall, used also in the previous linear transient analysis are applied as well as the bolt axial force. Since deformations such as slipping and opening/closing are not allowed if nodal displacements are used as boundary conditions of the submodel, the nodal forces, instead of the nodal displacements, obtained by the previous analysis are used in order to accurately describe deformations around the contact surfaces.

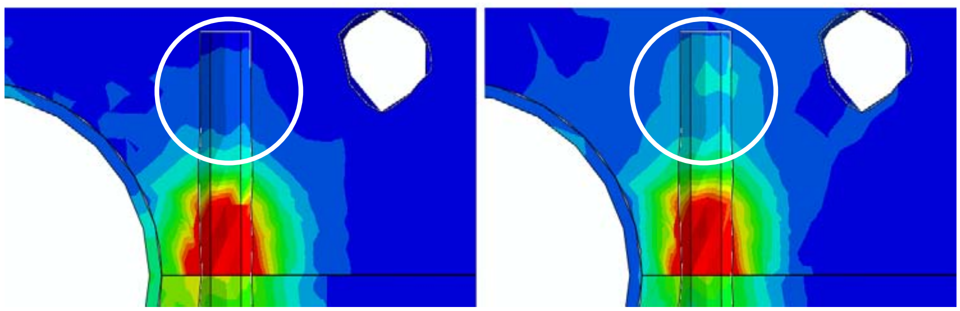
Figure 5 compares the distribution of the Mises stress contours around the subject bolt-hole for the crank angle of 20, 110, 150 and 550 degrees respectively and shows that the changes around the tip of bolt-hole are remarkable than other places.

The nodal Mises stresses at the tip of the bolt-hole are then evaluated along full cycle 720 degrees as shown in Figure 6. The nodal Mises stresses at the crank angle 20 and 550 degrees are higher than the other angles, where these angles correspond to the first and second cylinder combustions as shown in Figure 3.

The deformation calculated at the crank angle 20 degrees is shown in Figure 7. Since the downward appreciable load due to the cylinder combustion is applied to the main bearing No.2, the cylinder block and the ladder frame are V-shaped as shown in Figure 7(a). The contact surface between the cylinder block and the ladder frame slips as shown in Figure 7(b). These are some examples of nonlinear deformations that cannot be obtained by a linear domain analysis.



(a) at 20 degrees (b) at 150 degrees



(a) at 240 degrees (b) at 550 degrees

Figure 5. Distribution of the Mises stress.

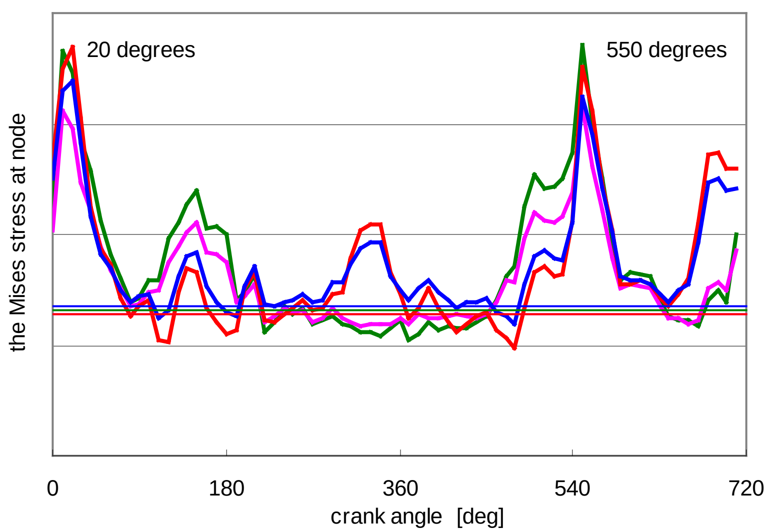
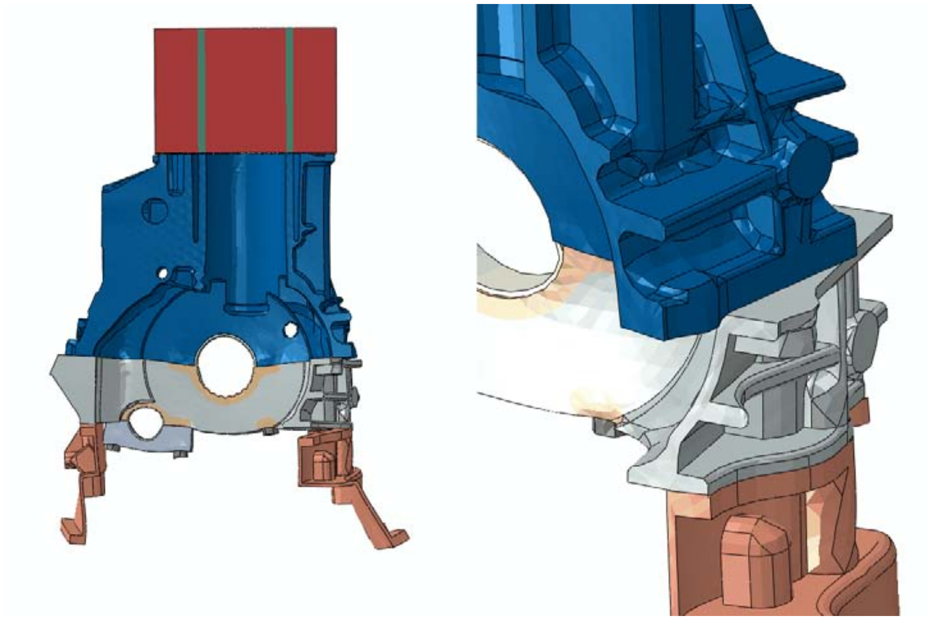


Figure 6. The Mises stress at the tip of the bolt-hole during two cycles of crankshaft.



(a) V-shaped deformation (b) slipped deformation

Figure 7. Deformations of the bulk model

6. Nonlinear static finite element analysis at maximum stress angle (Task4) with Abaqus/Standard ver.6.6

Since the purpose of this task is to obtain the maximum stress amplitude occurring at the thread roots of a bolt-hole using fine mesh models of the bolt and the bolt-hole, two conditions are considered: the just after the tightening process and the crank angle of 20 degrees. The common submodel analysis of Abaqus ver6.6, which uses the displacements as the boundary conditions, is conducted.

Figure 8 shows the fine-mesh bolt model as a submodel ("the bolt model" in the following) and the bulk model as a global model. The bolt model consists of a bolt and a cylinder block (bolt-hole side) with all elements being rectangular brick elements C3D8 as shown in Figure 9. The shapes of the bolt and the bolt-hole are modeled in detail. Although the actual threads are helical, they are approximated by an axisymmetric geometry. Since the bolt is made of steel and is much more rigid than the bolt-hole which are made of aluminum, bolt is characterized by an elasticity material, and the bolt-hole is characterized by a von Mises plasticity material with isotropic hardening. In order to obtain the accurate stress at the thread roots of bolt-hole in the cylinder block, sufficiently thin shell elements S4 are placed on the thread roots of the bolt-hole and the stress at the integration points of these thin shell elements is evaluated.

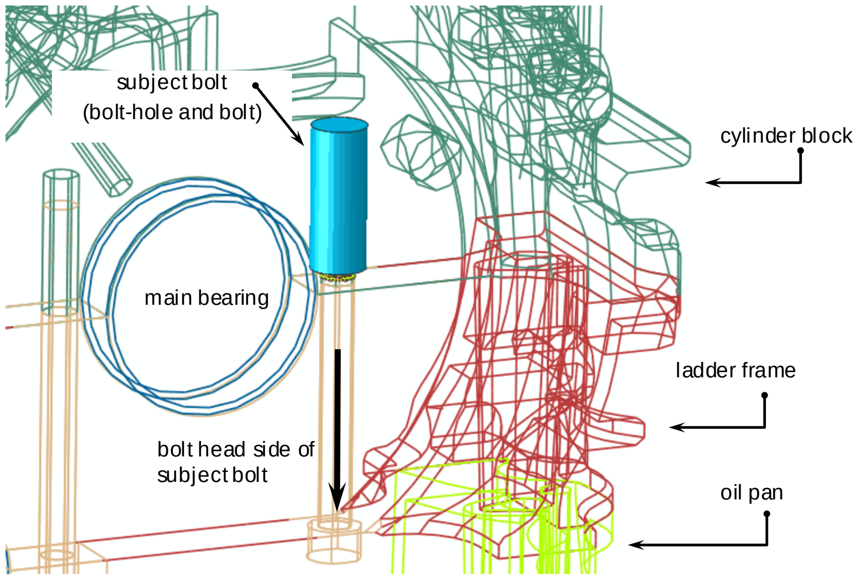
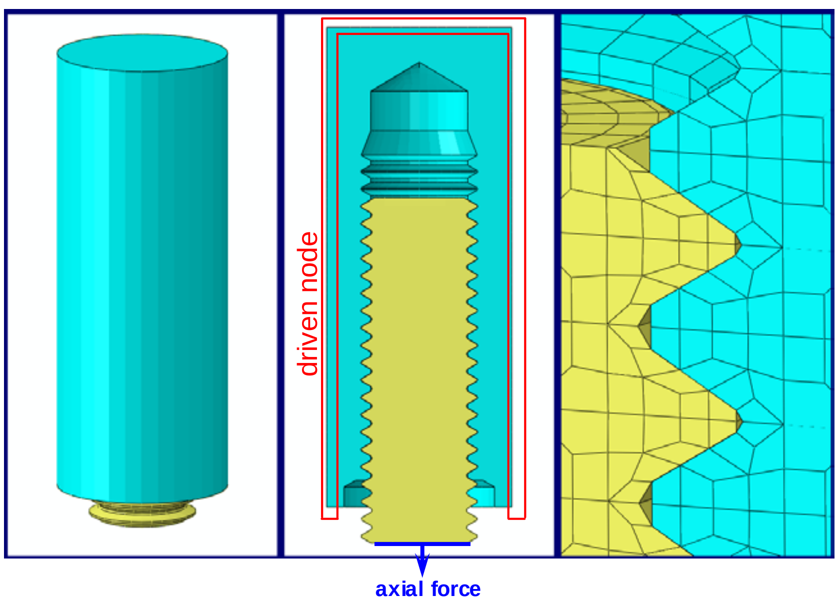


Figure 8. The bolt model (shaded, submodel) and the bulk model (lined, global model).



(a) isotropic view (b) cross section view (c) magnified view

Figure 9. The bolt model in detail

The displacements obtained in Task3 drive the boundary nodes of the submodel as shown in Figure 9(b). Practically, since the nodal displacements of the bolt of the boundary surface between global and submodel obtained in Task3 are not as accurate as the axial forces obtained also in Task3, the bolt axial forces are instead applied to the boundary nodes of the bolt as shown also in Figure 9. The inaccuracy stems from the fact that the detail shapes of bolt and nonlinearities such as contact and plasticity are not taken into account in the global model. The magnitude of the bolt axial forces applied are as follows:

- in tightening process, the standard bolt axial force is applied

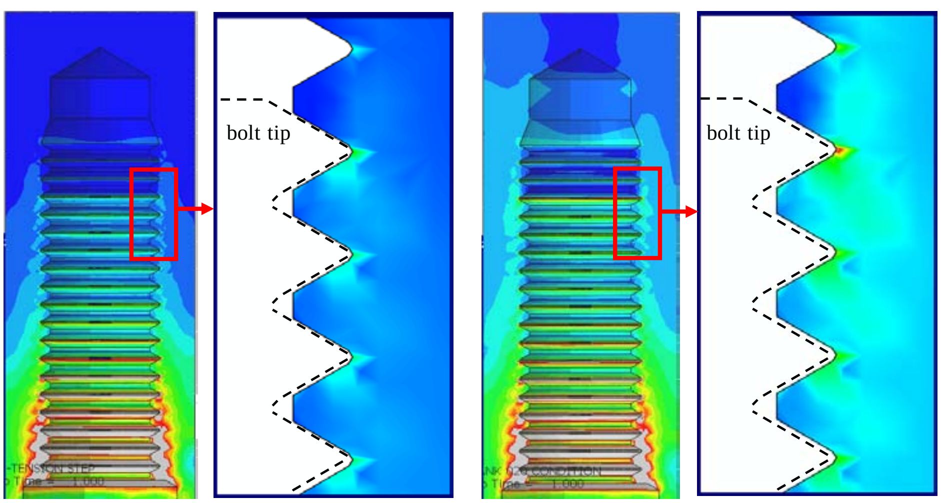
- at the crank angle 20 degrees, the bolt axial force obtained in Task3 (including changes of bolt axial force under an operating condition) is applied.

The Mises stress contours of the bolt-hole just after tightening and at the crank angle of 20 degrees are compared in Figure 10. Relatively large difference in the Mises stress is observed at the tip of the bolt-hole.

In Figure 11, the Mises stresses at the integration points of the monitor thin shell elements are compared along the thread roots of the bolt-hole at the above two conditions. The Mises stress amplitude, which is the difference between above two conditions, is also shown in Figure 11.

The stress occurring at the bolt head side tends to be larger than the stress occurring at the bolt tip side. The stress amplitude, on the other hand, is lower at the bolt head side than at the bolt tip side.

Next, the stress amplitude at the tip of the bolt-hole is evaluated along the circumference as shown in Figure 11. It is observed that the values of the stress amplitude vary along the circumference which is the result of considering deformations under operation conditions.



a) just after tightening process b) at the crank angle of 20 degrees

Figure 10. Distribution of the Mises stress of the bolt-hole

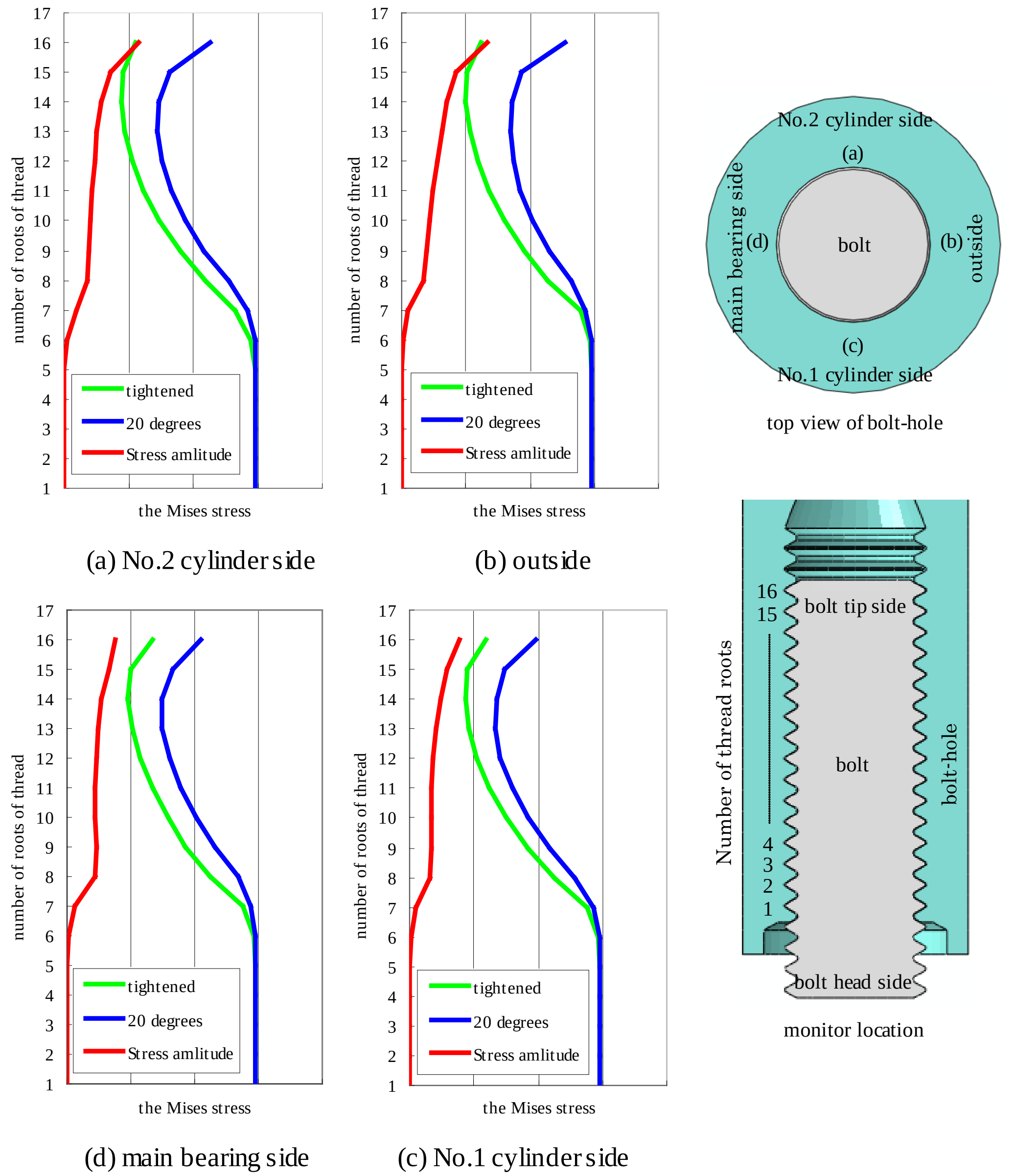


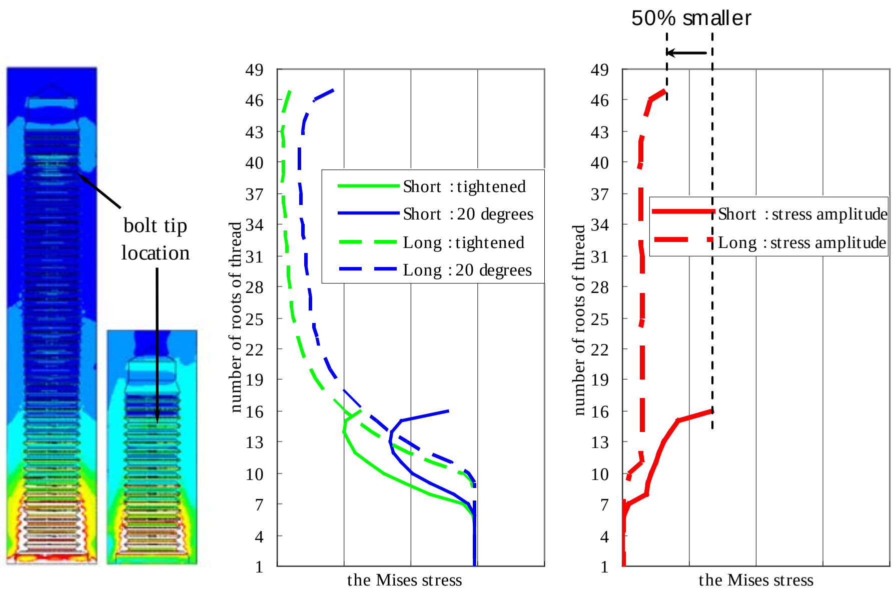
Figure 11. Distribution of the Mises stress of the bolt-hole side.

7. Effect of bolt length; Application to another cylinder block with different bolt length

The proposed analysis method is applied to a modified cylinder block with longer bolt length for comparison. Although only the content corresponding to Task 4 is mentioned here, all the tasks are actually applied to the modified engine system. The crank angle with the highest stress at the tip of the bolt-hole is also the crank angle of 20 degrees, which is the same as the original engine system.

The stress contours of the modified and the original cylinder blocks at the crank angle of 20 degrees are compared in Figure 12(a). The comparisons of the stress and the stress amplitude, corresponding to the *(b) outside* part of the short bolt-hole(in Figure 11), are also shown in Figure 12(b) and (c).

The stress tends to be lower toward the tip for the longer bolt. The tendency of the stress amplitude, which becomes the highest at the tip of the bolt, is not changed in comparison. On the other hand, the value of stress amplitude of the longer bolt is smaller by about fifty percent than that of the short bolt. It displays a tendency that the longer the bolt, the smaller its stress amplitude.



(a) the Mises stress contours (b) the Mises stress (c) the Mises stress amplitude

Figure 12. Comparison between the shorter and longer bolt.

8. Conclusions

We developed a new method to accurately predict the stress at the thread roots of a bolt of an automotive engine under an operating condition. It has been applied to in-line 4 engine systems.

- The stress occurring at the bolt head side tends to be larger than the stress occurring at the bolt tip side. The stress amplitude, on the other hand, is lower at the bolt head side than at the bolt tip side.

- It is observed that the values of the stress amplitude vary along the circumference which is the result of considering deformations under an operation condition.

- In comparison between longer bolt and shorter bolt, It displays a tendency that the longer the bolt, the smaller its stress amplitude.

9. References

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