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DEPENDENCE OF FATIGUE LIMIT OF HIGH-TENSION BOLTS ON MEAN STRESS AND ULTIMATE TENSILE STRENGTH

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ABSTRACT−High tension bolts in critical joints in internal combustion engines are susceptible to fatigue failure. Computer-aided bolted joint design procedures require knowledge of the dependence of bolt fatigue limit on the mean stress and ultimate tensile strength. This dependence is investigated with staircase fatigue limit tests. The test results show that when the bolt fatigue limit is estimated with the nominal stress of the bolt, it decreases with increasing tensile strength and nominal mean stress. However, there is a range of the nominal mean stress where the bolt fatigue limit is almost constant. The test results are interpreted with finite element analysis.

KEY WORDS : Fatigue limit, High tension bolt, Mean stress, Ultimate tensile strength, Staircase fatigue test, Finite element analysis

1. INTRODUCTION

Critical bolted joints in internal combustion engines, including connecting-rod big end joints, cylinder head-block joints, and main bearing cap-block joints, require great care in determining the size, strength and preload of the bolt in order to assure the fatigue durability of engines as well as the performance (Cioto and Collarse, 2003). For example, if the bolted joint is partially separated due to the lack of bolt preload, the bolt load fluctuates significantly during the engine run. On the other hand, excessive bolt preload generates too large a mean stress in the bolt. Both of these two extreme cases may result in the fatigue failure of bolts (Bickford, 1995) and thus, engine failure. It is common in the automobile industry to tighten the bolts to the yield point (Wallace, 1998) to reduce the engine weight via a reduction in the bolt size. This implies that the critical bolts in modern engines are more likely to fail due to fatigue.

The fatigue failure of critical engine bolts usually occurs at the thread root of the first bolt-nut engagement point. Since the bolt threads are cold-rolled and heat-treated, their fatigue limit differs substantially from that of the raw material. It is almost impossible to theoretically deduce the fatigue limit of a bolt from that of the raw material (Wallace, 1998), so the bolt fatigue limit must be obtained directly with bolt fatigue testing. Since the bolt fatigue limit is estimated with the nominal stress of the bolt, the nominal alternating stress is compared with the bolt fatigue limit for the fatigue assessment.

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The fatigue failure of a bolt in the joint is associated with various factors such as external loads, the joint stiffness and the structural characteristics of the whole assembly. Only finite element analysis considers the effects of these factors realistically. In the finite element model of bolted joints, the threaded portion of bolt is usually modeled as a solid cylinder for computational efficiency, and the relative movement at the interface of the bolt-nut engagement is constrained. Since the constraint acts as a stress riser, the stress at the first engagement point is greater than the nominal bolt stress and thus cannot be compared directly with the bolt fatigue limit for the fatigue assessment. This difficulty is overcome using the extrapolation technique (Cho *et al*., 2008). Therefore, the fatigue durability of bolt can be assessed using the finite element analysis of a bolted joint and bolt fatigue limit data.

The bolted joint design using the finite element analysis requires selection of the type and preload of the bolt and then assessment of fatigue durability of the bolt as well as performance of the bolted joint (Cho *et al*., 2007). If either fatigue failure or malfunction of the bolted joint results from the lack of bolt preload, the first remedy is to increase the bolt preload. If the bolt failure continues to occur until its prestress is close to the yield stress, the second remedy is to strengthen the bolt for application of higher preload. The strengthening is usually carried out using heat treatment because the bolt size is not changed. If even the strengthened bolt fails due to fatigue, a larger bolt should be employed, and thus, the members of bolted joint should be designed again. This brief description of the bolted joint design procedure reveals the necessity of knowledge of the dependence of the bolt fatigue limit on both the mean stress and the ultimate tensile strength.

It was reported previously (Burguete and Patterson, 1995) that the mean stress effect cannot be predicted with commonly used empirical rules, such as the Goodman line or Gerber parabola, but can be predicted in limited ranges with the modified rules of Gunn and Cook. However, their test data show that the bolt fatigue limit decreases slowly with relatively large fluctuations as the mean stress increases, and no explanation for the fluctuation was provided. Unfortunately, no other literature on the effects of the mean stress and tensile strength can be found. Therefore, this paper aims to enhance our understanding of the dependence of the bolt fatigue limit on the mean stress and tensile strength.

2. FATIGUE TESTS AND RESULTS

2.1. Test

The fatigue limits of M12×P1.5 high tension bolts with a phosphate conversion coating were tested. The bolts were made of steel with a major alloying composition of C 0.33~0.38%, Mn 0.50~0.85%, Cr 0.90~1.20%, Mo 0.15~0.30% (JIS G4105: SCM435). The bolt threads were cold-rolled before quenching and tempering. The tempering temperature was controlled to produce the two kinds of bolts whose ultimate tensile strengths were 1058 MPa and 1245 MPa. Figure 1 shows the stress-strain curves of the bolts, in which the nominal bolt stress that was estimated with the tensile stress area of bolt was used. The elastic modulus is 207 GPa for both curves. The yield strengths are 920 MPa and 1083 MPa for tensile strengths of 1058 MPa and 1245 MPa, respectively.

The staircase test method in the infinite life region (ISO 3800:1993) was employed. The step size of the stress amplitude was set to 5 MPa. The mean stress was selected in the range from 60 to 90 percent of the bolt tensile strength, which corresponds to the preload range in the elastic and yield-point tightening methods. Thirty-one bolts were tested for each test condition. The number of loading cycles was set to 5×10^6 cycles. It is noted that when SCM435 steel is tempered to a tensile strength below 1300 MPa, its fatigue limit is observed below 10^6 cycles and is maintained up to 10^9 cycles (Sakai *et al*., 2006). The cyclic load was applied at a frequency around 100 Hz on a

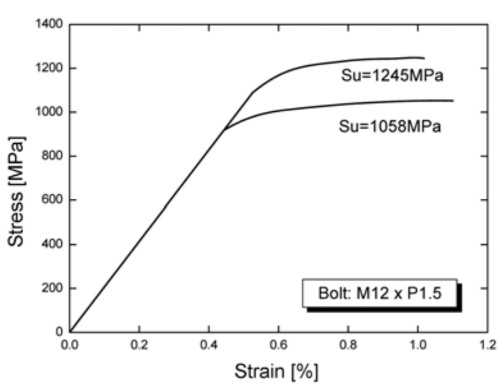

Figure 1. Stress-strain curves of bolts.

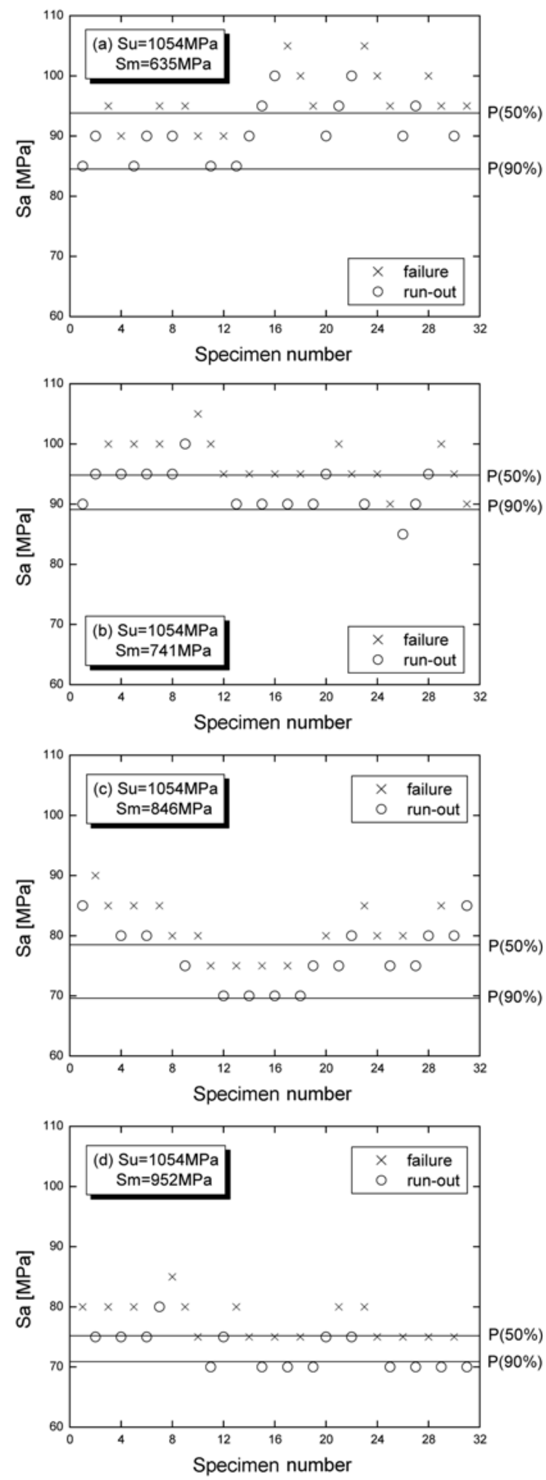

Figure 2. Staircase fatigue limit test results for bolts with a tensile strength of 1054 MPa at the mean stress levels of (a) 60%, (b) 70%, (c) 80% and (d) 90% of the ultimate tensile strength.

resonance fatigue test machine. The tests were conducted at room temperature. It is noted that temperature of engine bolts usually does not exceed 250ºC under normal running condtions. Meanwhile, the fatigue limit of most steels does not decrease with temperature when the temperature is below 400ºC (Stephens *et al*., 2001). This implies that the room-temperature fatigue limit can be used as a conservative criterion in the fatigue design and assessment of engine bolts.

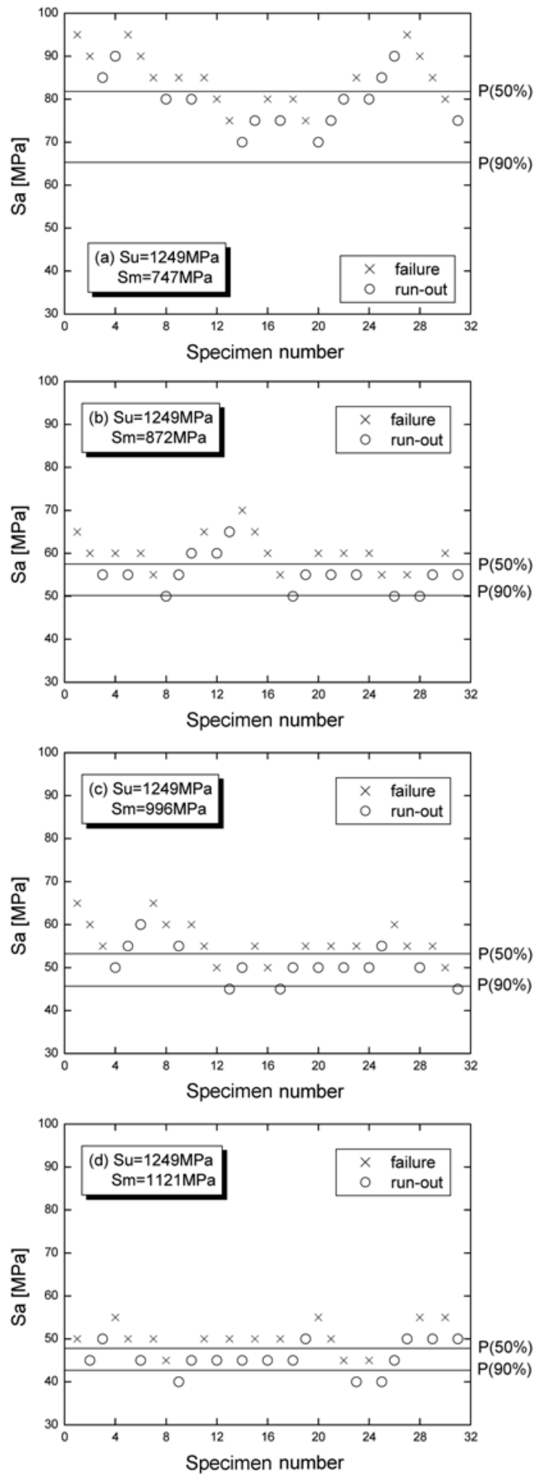

Figure 3. Staircase fatigue limit test results for bolts with a tensile strength of 1249 MPa at the mean stress levels of (a) 60%, (b) 70%, (c) 80% and (d) 90% of the ultimate tensile strength.

2.2. Test Results

Figures 2 and 3 show the staircase fatigue limit test results at various mean stress levels for bolts with tensile strengths of 1054 MPa and 1249 MPa, respectively. The bolt fatigue limit values at the non-failure probabilities of 50% and 90% are also shown as horizontal lines. The statistical analysis was conducted with the method described in the standard ISO 3800:1993. The difference between the bolt fatigue limit values at the two different probabilities decreases with increasing mean stress. This implies that the bolt fatigue limit exhibits smaller scattering at higher mean stress levels.

Figure 4 shows the effect of the nominal mean stress and tensile strength on the nominal bolt fatigue limit at a non-failure probability of 50%. The nominal bolt fatigue limit and the nominal mean stress imply that they are estimated with the nominal stress of bolt. The bolt fatigue limit decreases overall with the mean stress although those at mean stresses of 635 MPa and 741 MPa for the bolts with tensile strength 1054 MPa are almost the same. A linear or parabolic proportionality between the bolt fatigue limit and the mean stress is not observed. The bolts with a higher tensile strength exhibits a lower bolt fatigue limit.

3. ANALYSIS AND DISCUSSIONS

3.1. Finite Element Analysis

It is known that the fatigue limit of metal increases with the tensile strength. However, the test results show that the nominal fatigue limit of the bolts decreases with the tensile strength. It is inferred that the opposite dependence is

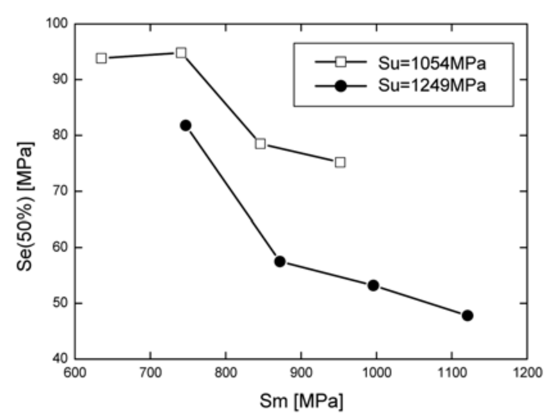

Figure 4. Variation of the nominal bolt fatigue limit with the nominal mean stress and tensile strength of the bolts. The nominal bolt fatigue limit means that it is estimated with the nominal stress of the bolts.

associated with the thread root geometry of bolt. More specifically, the fatigue crack in the bolt initiates at the thread root where both a local stress increase and plastic deformation occur, whereas the nominal bolt fatigue limit in Figure 4 is estimated with a nominal bolt stress that does not reflect the local stress rise at the thread root. Thus, it is necessary to examine the tensile strength effect with the bolt fatigue limit that is estimated with the stress at the thread root where the fatigue crack initiates, instead of the nominal stress of the bolt. Hence, finite element analysis was conducted.

The pairing of an engaged bolt and nut was modeled with axisymmetric elements, as shown in Figure 5. The stress-strain curve in Figure 1 was used for the bolt because the preliminary analysis showed the negligible effect of threads on the overall deformation behavior of the bolt. The nut, made of SNCM8 steel, was modeled with a stress-strain curve with an elastic modulus of 207 MPa, a yield strength of 880 MPa and a tensile strength of 980 MPa. The friction coefficient at the bolt-nut interface was set to 0.19. The bottom surface of the nut was fixed, and the bottom surface of the bolt head was subjected to an alternating load that resulted in the mean stress level and the stress amplitude in Figure 4. The alternating load was repeated six times to examine any transient variation in stress at the thread root during the beginning of the cycle. No transient variation was observed, so the axial stress at the thread root during the first cycle was used to estimate the bolt fatigue limit and the mean stress.

3.2. Analysis Results and Discussions

Figure 5 shows the axial stress distribution around the first bolt-nut engagement point. A stress rise is observed at the thread roots, and it is most significant at the first thread root where the fatigue crack initiates.

Figure 6 shows the effect of the actual mean stress (σ_m) on the actual bolt fatigue limit (σ_e) at a non-failure probability of 50%. The actual mean stress and the actual bolt fatigue limit are estimated with the stress at the thread root where the fatigue crack is initiated. The actual bolt fatigue limit decreases linearly with the actual mean stress. It is noted that such a clear linearity is not observed in Figure 4. In the case of bolts with a tensile strength of 1054 MPa, the nominal mean stress levels of 635 MPa and 741 MPa result in almost the same actual mean stress and thus, almost the same values of the corresponding nominal bolt fatigue limit. This explanation may be applied to the test results mentioned in the introduction (Burguete and Patterson, 1995).

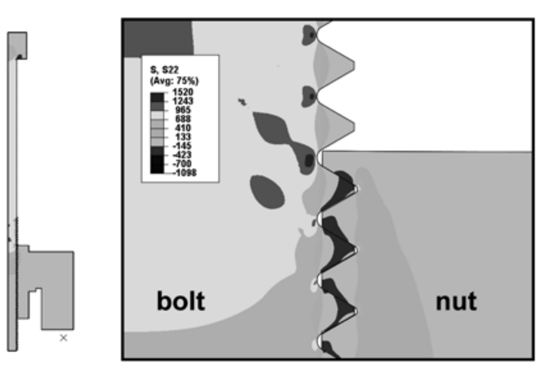

Figure 5. Finite element analysis result for the axial stress distribution around the first bolt-nut engagement point.

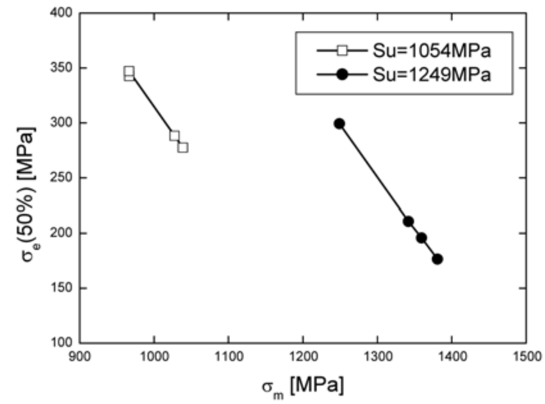

Figure 6. Variation of the actual bolt fatigue limit with the actual mean stress and tensile strength of the bolts. The actual bolt fatigue limit and the actual mean stress were estimated using the axial stress at the thread root where the fatigue crack initiates.

Figure 6 also shows the effect of tensile strength. Although the actual mean stress ranges of the two curves do not overlap, it can be inferred that the bolt with a higher tensile strength possesses a greater actual bolt fatigue limit. This result is consistent with the fact that stronger metals possess greater fatigue limits. Hence, it is claimed that lower nominal fatigue limit of stronger bolt shown in Figure 4 results from the use of nominal bolt stress, which does not reflect the stress state at the thread root where the fatigue crack is initiated.

The nominal stress in the bolts is used in industry to estimate the mechanical properties of bolts, such as the fatigue limit, and the load levels, such as the mean stress and the stress amplitude, because of its convenience. Hence, it should be kept in mind in designing bolted joints that when the bolt is strengthened using heat treatment, the nominal fatigue limit of bolt is reduced, although the fatigue limit of the bolt material is increased.

4. CONCLUSION

The staircase fatigue limit tests in the infinite life region were conducted at various mean stress levels for high tension bolts of different tensile strengths, and the test results were interpreted with finite element analysis.

When the bolt fatigue limit and mean stress are estimated with the axial stress at the thread root where the fatigue crack is initiated, the bolt fatigue limit increases with the tensile strength and decreases linearly with increasing mean stress at the thread root. This result is consistent with the well-known fact that the fatigue limit of metals increases with tensile strength. However, when the bolt fatigue limit and mean stress are estimated with the nominal stress of the bolt, which does not reflect the local stress rise at the thread root, the bolt fatigue limit decreases with increasing tensile strength and the nominal mean stress, and there is a range of the nominal mean stress where the bolt fatigue limit is almost constant.

Since the nominal stress of bolts is used in industry to estimate the mechanical properties of bolts, such as the fatigue limit, and the load levels, such as the mean stress and the stress amplitude, because of its convenience, it should be kept in mind in designing bolted joints that when the bolt is strengthened using the heat treatment, the nominal fatigue limit of the bolt is reduced though the fatigue limit of the bolt material is increased.

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REFERENCES

Bickford, J. H. (1995). *An Introduction to the Design and Behavior of Bolted Joints*. Marcel Decker. New York. 565−601.

Burguete, R. L. and Patterson, E. A. (1995). The effect of mean stress on the fatigue limit of high tension bolts. *Proc. Institution of Mechanical Engineers*, **209**, 257–262.

Cho, S.-S., Chang, H. and Lee, K. W. (2007). Procedure for conputer-aided preload selection of engine connecting-rod bolts. *Int. J. Automotive Technology* **8**, **3**, 319−325.

Cho, S.-S., Chang, H. and Lee, K. W. (2008). Finite element-based fatigue assessment of engine connecting-rod bolts. *Trans. Korean Society of Automotive Engineers* **16**, **4**, 14–20.

Cioto, R. and Collarse, A. R. (2003). Reduction of fatigue resistance on bolts with thread rolled after heat treatment, as a result of superficial treatment. *SAE Paper No*. 2003-01-3657.

Sakai, T., Sakai, T., Okada, K., Furuichi, M., Nishikawa, I. and Sugeta, A. (2006). Statistical fatigue properties of SCM435 steel in ultra-long-life regime based on JSME database on fatigue strength of metallic materials. *Int. J. Fatigue*, **28**, 1486−1492.

Stephens, R. I., Fatemi, A., Stephens, R. R. and Fuchs, H. O. (2001). *Metal Fatigue in Engineering*. John Wiley & Sons. New York. 373−381.

Wallace, P. W. (1998). Increasing joint performance by tightening bolts to yield. *Handbook of Bolts and Bolted Joints*. Editted by Bickford and Nassar. Marcel Dekker. New York. 591−602.