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Effect of initial clearance on the maximum stress in pin-lug joints: A finite element investigation

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ABSTRACT

This study investigates the effect of initial clearance on the maximum stress distribution in pin-loaded lug connections, a critical factor influencing joint strength, durability, and design reliability. To this end, finite element analysis (FEA) was employed to model round-ended lug-pin joints with varying pin-hole clearances under different applied loads, and parametric simulations were conducted to identify the transition zone where tensile and compressive stresses become comparable. The results show that clearance has a profound influence on stress behavior: small clearances maintain relatively uniform stress distributions, whereas modest increases trigger a dramatic shift from tensile- to compressive-dominated regions. This transition is highly sensitive to applied load, producing nonlinear changes in the location of maximum stress. Larger clearances also lead to highly localized contact stresses, sharply amplifying compressive stresses and highlighting potential hotspots for surface damage, plastic deformation, and fatigue initiation. These findings demonstrate that critical stress behavior in lug-pin joints is dynamically dependent on both geometry and loading conditions, challenging traditional assumptions in joint design. The study provides a quantitative assessment of the interplay between clearance and applied load, offering practical guidance for achieving balanced stress distribution and improving the durability and reliability of lug-pin connections.

Pim-kulak bağlantılarında başlangıç boşluğunun maksimum gerilme üzerindeki etkisi: Sonlu elemanlar incelemesi

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ÖZET

Bu çalışma, pim yüklü kulak bağlantılarında başlangıç boşluğunun maksimum gerilme dağılımı üzerindeki etkisini incelemektedir. Bu etki, bağlantı dayanımı, dayanıklılığı ve tasarım güvenilirliği açısından kritik bir faktördür. Bu amaçla, farklı pim-delik boşluklarına sahip yuvarlatılmış uçlu kulak-pim bağlantıları sonlu elemanlar analizi (FEA) kullanılarak modellenmiş ve uygulanan farklı yükler altında parametrik simülasyonlar gerçekleştirilerek çekme ve basma gerilmelerinin karşılaştırılabilir hale geldiği geçiş bölgesi belirlenmiştir. Elde edilen sonuçlar, boşluğun gerilme davranışı üzerinde önemli bir etkisi olduğunu göstermektedir: Küçük boşluklar nispeten düzgün gerilme dağılımlarını korurken, boşluğun bir miktar artması çekme baskın bölgelerden basma baskın bölgelere doğru dramatik bir kaymayı tetiklemektedir. Bu geçiş, uygulanan yüke oldukça duyarlıdır ve maksimum gerilme bölgesinde doğrusal olmayan değişimlere neden olmaktadır. Daha büyük boşluklar ise temas gerilmelerinin oldukça yerleşmesine yol açarak basma gerilmelerini keskin bir şekilde artırmakta; yüzey hasarı, plastik deformasyon ve yorulma başlangıcı için potansiyel kritik bölgeleri ortaya çıkarmaktadır. Bu bulgular, kulak-pim bağlantılarındaki kritik gerilme davranışının hem geometriye hem de yükleme koşullarına dinamik olarak bağlı olduğunu göstererek, geleneksel bağlantı tasarımı varsayımlarını sorgulamaktadır. Çalışma, boşluk ile uygulanan yük arasındaki etkileşimi nicel olarak değerlendirerek, dengeli gerilme dağılımı sağlanması ve kulak-pim bağlantılarının dayanıklılık ve güvenilirliğinin artırılması için pratik bir rehber sunmaktadır.

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1. INTRODUCTION

Pin-loaded lug connections are widely used in mechanical and structural engineering to transfer loads between structural members [1-6]. Because the load transfer occurs through localized contact between the pin and the lug bore, the stress distribution in this region plays a critical role in determining the strength and durability of the joint [7]. Consequently, understanding the factors that influence the maximum stress in pin-lug connections is essential for reliable structural design [8]. In an ideal pin-loaded lug connection, the pin fits tightly into the lug hole, and the load is mainly transferred through shear in the pin and bearing pressure on the lug surface [7]. In practical engineering applications, however, manufacturing tolerances and assembly requirements inevitably introduce an initial clearance between the pin and the lug bore [9]. The presence of this clearance modifies the contact conditions between the pin and the lug, leading to changes in the pressure distribution along the contact surface and potentially increasing the stress concentration near the contact edge [8-10]. Such stress concentrations may accelerate wear of the pin surface and the lug hole wall, which is often a dominant failure mechanism in pin-lug joints [9-10].

The mechanical behavior of pin-lug connections has attracted considerable research interest over the past decades. Early analytical investigations were mainly based on the two-dimensional theory of elasticity and the Michell polar solution for annular geometries, providing valuable insight into the stress field in pin-lug systems [11]. Simplified analytical models based on curved beam theory were later introduced to describe the load transfer mechanism in pinned connections, although these approaches often lack sufficient accuracy for detailed stress analysis [2, 6, 12]. Several studies have also examined the effect of initial clearance on the stress field in pinned connections. Persson [13] provided one of the earliest analytical treatments of this problem and demonstrated that clearance significantly influences the contact conditions between the pin and the lug. Subsequent studies extended this analysis and highlighted the progressive development of the contact arc as the applied load increases [9]. Nevertheless, many analytical approaches rely on simplified assumptions that limit their applicability to realistic geometries and practical engineering conditions.

Design specifications and previous research have proposed several simplified models to describe the contact pressure distribution between the pin and the lug, including uniform pressure models, sinusoidal pressure models, and Hertzian contact models [14-16]. Although these models provide useful approximations, they generally neglect important aspects such as the combined effects of clearance, geometry, and deformation of the pin under load. As a result, accurately predicting the maximum stress in pin-lug joints with clearance remains challenging [1, 9].

With the advancement of computational techniques, the finite element method has become an effective tool for investigating the stress distribution and contact behavior in pin-lug connections. Numerical studies have shown that the location and magnitude of the maximum stress in the lug depend on several parameters, including the applied load, lug geometry, and the magnitude of the initial clearance [2, 7-9, 17].

Despite the considerable amount of research conducted on pinned connections, the influence of initial clearance on the maximum stress in lug-pin joints still require further investigation, particularly for practical configurations. Therefore, the present study focuses on evaluating the effect of initial clearance on the maximum stress occurring in a lug-pin joint. The analysis is performed using finite element analysis, which enables a detailed examination of the contact stress distribution and its dependence on the magnitude of the clearance.

2. LUG – PIN JOINT

Figure 1 illustrates a typical pin-lug connection used to transfer load between structural members through a pinned joint mechanism. Such connections are widely used in mechanical systems, aerospace structures, and steel constructions where efficient load transfer and rotational freedom are required [1, 13, 14, 16, 17].

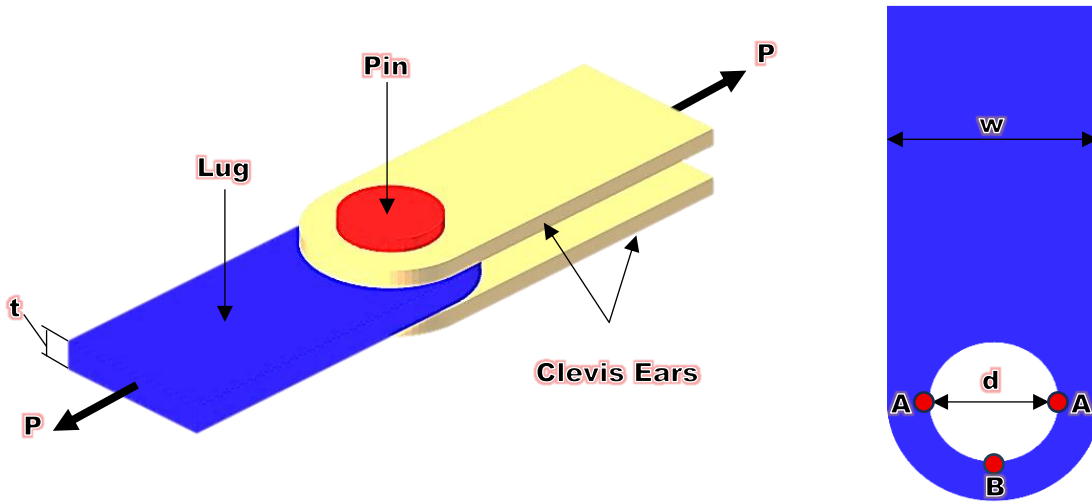


Figure 1. Geometry of a typical pin-lug connection: (a) 3-dimensional schematic of the assembly with the pin, lug, and clevis ears constituting a double-shear configuration, and (b) Geometric definition of the lug with lug width w and hole diameter d

As shown in Figure 1, the assembly consists of three primary components: the lug, the pin, and the clevis ears. The lug is a plate-like structural element containing a circular hole at its end. The pin passes through this hole and acts as the main load-transferring element between the connected members. The clevis ears are two parallel plates placed on both sides of the lug. The pin passes through the holes of the clevis ears and the central lug, creating a double-shear connection that links the components and enables load transmission. In pin-lug connections, the applied load transfers from the lug to the pin mainly through bearing (contact) stresses that develop along the interface between the pin surface and the lug hole. The distribution of these stresses depends on several factors, including the applied load, the geometric characteristics of the lug, and the presence of initial clearance between the pin and the lug bore [9, 13]. These factors influence the extent of the contact region and the resulting stress distribution around the hole.

Stress concentrations generally develop along the boundary of the lug hole, where the load is transferred from the lug to the pin [8]. The maximum stress may occur either within the pin-lug contact region or near the hole edge just outside the contact arc, depending on the loading conditions and the magnitude of the clearance. These localized stresses are critical because they may initiate local yielding, wear, or fatigue damage in the lug. The geometric parameters of the lug considered in this study also appear in Figure 1. The parameter w denotes the lug width, measured across the plate perpendicular to the loading direction, while d represents the diameter of the lug hole, which corresponds to the pin diameter in a neat-fit configuration. These parameters significantly influence the stress distribution in the lug and therefore constitute important design variables in the analysis of pin-lug connections. Note that Point A and Point B in Figure 1 represent the maximum tensile and maximum compressive stress locations, respectively [2, 8, 9, 16, 18, 19].

3. FINITE ELEMENT ANALYSIS

The stress distribution in the pin-lug connection was investigated using the finite element method. The numerical analyses were performed using Abaqus/Standard [20], which employs an implicit solution procedure suitable for static contact problems. A three-dimensional (3D) model was adopted to accurately represent the contact interaction between the pin and the lug and to capture the stress distribution around the lug hole. The geometry of the pin-lug assembly was discretized using HyperMesh, which was used to generate the computational mesh of the model. Particular attention was given to the mesh density in the vicinity of the lug hole, where high stress gradients are expected due to the contact interaction between the pin and the lug. The analyses were conducted assuming linear elastic material behavior. In addition, geometric nonlinearity was activated in the simulations to account for the influence of large displacements and the progressive evolution of the contact region between the pin and the lug during loading. This consideration is important in contact problems such as pin-lug connections, where the extent of the contact arc may change as the load increases.

The material properties, geometric parameters, and loading conditions used in the numerical simulations are summarized in Table 1.

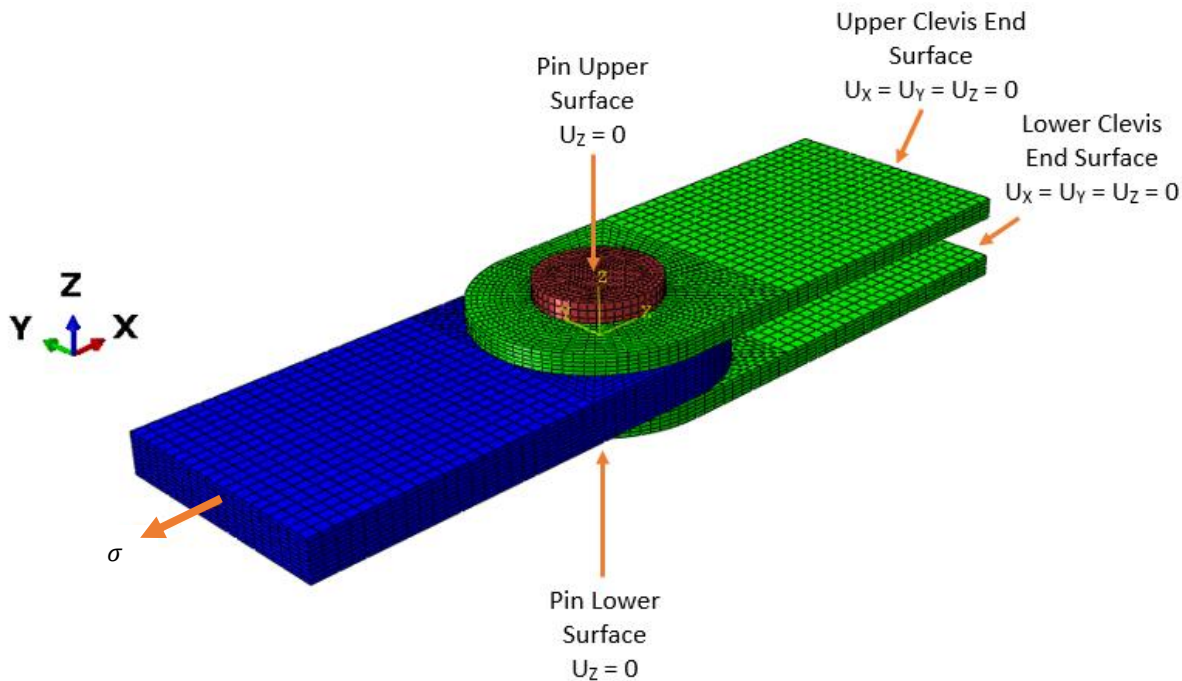
Table 1. The material properties, geometric parameters, and loading conditions used in the numerical simulations

Parameter	Symbol	Value	Unit	Description
Young's Modulus	(E)	200000	MPa	Elastic modulus of the material used for both the pin and the lug, representing the stiffness of the material.
Poisson's ratio	(ν)	0.3	–	Ratio of transverse strain to axial strain under uniaxial loading, typical for structural steel.
Lug width	(w)	60	mm	Distance across the lug measured perpendicular to the loading direction.
Hole diameter	(d)	30	mm	Diameter of the lug hole, corresponding to the pin diameter in a neat-fit configuration.
Lug Thickness	(t)	10	mm	Thickness of the lug plate through which the pin passes.
Applied Load	(P)	5000	N	External force applied to the lug to represent the loading condition of the pin-lug connection.

3.1. Finite Element Model, Loading and Boundary Conditions

The three-dimensional finite element model developed for the lug-pin joint assembly is illustrated in Figure 2. The model consists of the lug, the clevis, and the connecting pin, all represented using solid finite elements to accurately capture the stress distribution and load transfer within the joint. A structured mesh was employed for the discretization of the geometry, with a relatively finer mesh around the pin-lug contact region and the hole boundaries, where high stress gradients are expected. This mesh refinement ensures improved numerical accuracy in regions that are critical for evaluating stress concentration and structural performance.

As shown in Figure 2, appropriate boundary conditions were applied to represent the physical constraints of the assembly. The upper clevis end surface and the lower clevis end surface were fully constrained in all translational directions, such that $U_X = U_Y = U_Z = 0$. These constraints simulate the condition in which the clevis is rigidly connected to a supporting structure, preventing any rigid body motion of the system. To properly represent the mechanical behavior of the pin within the joint, displacement constraints were applied on the upper and lower surfaces of the pin. Specifically, the displacement in the thickness direction was restricted by imposing $U_Z = 0$ on both the pin upper surface and the pin lower surface. This constraint prevents axial movement of the pin while still allowing the transfer of load between the contacting components.


Figure 2. Finite element model of pin-lug joint with boundary and loading conditions

Instead of applying a concentrated external force, a uniform pressure was applied on the free end surface of the lug, as indicated in Figure 2. The applied pressure corresponds to a nominal stress defined by $\sigma = P/wt = 8.33 \text{ MPa}$. This pressure-based loading approach ensures a uniform stress distribution across the loaded surface and represents the tensile loading condition commonly used in lug joint analyses [2, 7, 9].

In essence, the finite element model together with the specified loading and boundary conditions shown in Figure 2 provides a realistic numerical representation of the lug–pin joint system, allowing a detailed evaluation of stress distribution and stress concentration behavior within the joint. Additionally, the material used for the lug in the finite element model is assumed to be G550 structural steel, which is commonly used in high-strength thin-walled and connection applications. In the present study, the material is modeled as linear elastic with an elastic modulus of 200000 MPa and a Poisson's ratio of 0.3, which are typical values reported for structural steels [21]. Since the analysis is limited to elastic stress evaluation, no plasticity or yield behavior is considered in the material definition.

It is important to emphasize that, in the present study, both the lug and pin materials are intentionally modeled as linear elastic. This modeling choice is consistent with the classical stress concentration framework of Peterson's stress concentration factors [8], where stress magnification due to geometric discontinuities is evaluated strictly under elastic conditions. Since the main objective of this work is to isolate and quantify the effect of initial clearance on the elastic stress distribution in lug–pin connections, the inclusion of material nonlinearity and plastic deformation is beyond the scope of the present analysis and is therefore not considered in the finite element model.

3.2. Meshing and Element Type

The finite element model presented in Figure 2 was discretized using three-dimensional solid elements to represent the geometry and to capture the stress distribution within the lug–pin joint assembly. In this study, the C3D8R element type was employed. This element is an eight-node linear brick element with reduced integration and hourglass control, and it is widely used in three-dimensional structural analyses due to its computational efficiency and its ability to provide reliable results in problems involving contact and stress concentrations. [22]. A structured meshing approach was applied to the model to maintain good element quality. Special attention was given to the contact region between the pin and the lug hole, where higher stress gradients are expected, as shown in Figure 2. In this region, the mesh was refined to improve the accuracy of the stress prediction. In contrast, relatively larger elements were used in areas farther from the contact interface to reduce computational cost while maintaining sufficient accuracy.

The final mesh consisted of 34,144 elements and 39,659 nodes. The quality of the mesh in the critical contact region was carefully controlled, resulting in an element aspect ratio of approximately 1.365, which is within acceptable limits for reliable numerical analysis and helps maintain stable and accurate stress calculations. To ensure that the mesh density was adequate, a mesh convergence study was conducted based on the maximum stress values obtained from the analysis. The mesh was progressively refined and the corresponding maximum stress results were compared. Convergence was considered achieved when further mesh refinement produced only negligible changes in the maximum stress values, indicating that the selected mesh provides sufficiently accurate results for the analysis of the lug–pin joint model shown in Figure 2.

3.3. Contact and Clearance

The load transfer within the lug–pin assembly occurs primarily through contact between the pin and the surrounding structural components. Therefore, an appropriate contact formulation was implemented in the finite element model to accurately represent the interaction between these parts [9]. In the present study, surface-to-surface contact pairs were defined to model the mechanical interaction between the contacting bodies [9]. Specifically, contact was established between the pin and the lug hole surface, as well as between the pin and the inner surfaces of the clevis ears. This approach enables the realistic simulation of load transfer through bearing contact within the joint. A small-sliding contact formulation [20] was employed for all contact interfaces. This formulation assumes that the relative sliding between the contacting surfaces remains limited during loading and is widely used for structural contact problems where the expected relative motion is small. The use of small-sliding contact improves numerical stability and computational efficiency while maintaining adequate accuracy for the present analysis.

To investigate the influence of assembly tolerances on the mechanical behavior of the joint, several clearance values between the pin and the lug hole were considered. Clearance represents the initial radial gap between the pin surface and the inner surface of the lug hole before loading is applied. Variations in this gap can significantly affect the contact pressure distribution, stress concentration, and load transfer characteristics within the joint.

A total of six clearance configurations were analyzed in the present study. The selected clearance values were 0, 1×10^{-4} , 1×10^{-3} , 1×10^{-2} , 1×10^{-1} , and 5×10^{-1} . These values were chosen to represent a wide range of possible assembly conditions, from a perfect interference-free fit to relatively large clearances. For clarity and consistency throughout the analysis, each clearance configuration was assigned a specific sample designation, as summarized in Table 2.

Table 2. Clearance values considered in the simulations and corresponding sample designations

Sample Designation	Clearance Value (mm)
C0	0
C1	1×10^{-4}
C2	1×10^{-3}
C3	1×10^{-2}
C4	1×10^{-1}
C5	5×10^{-1}

In order to isolate the influence of clearance on the structural response of the joint, frictional effects were neglected, and a frictionless contact condition was assumed for all contact pairs. This assumption eliminates tangential resistance at the interface and allows the analysis to focus on the normal contact pressure and the resulting stress distribution within the lug–pin assembly. This contact modeling strategy enables a systematic evaluation of how different pin–hole clearance values influence the stress distribution and contact behavior within the lug–pin–clevis assembly under the applied loading conditions.

4. RESULTS AND DISCUSSION

The numerical values of the maximum stresses obtained for different clearance values in the lug–pin joint are presented in Table 3. These stresses were extracted from the normal stress contours shown in Figure 4, illustrating the normal stress distribution in the lug for various pin–hole clearances. As depicted in Figure 1, the maximum tensile stress occurs at Point A, while the maximum compressive stress is observed at Point B. These results are consistent with the stress patterns reported in the relevant literature [2, 8, 9]. The variation of these stresses with clearance is graphically represented in Figure 3, showing the maximum tensile stress at Point A and the maximum compressive stress at Point B as the clearance increases.

Table 3. Effect of clearance on maximum tensile stress and maximum compressive stress at the critical locations

Clearance (mm)	Max Tensile Stress (MPa)	Max Compressive Stress (MPa)
0	37.97	22.32
1×10^{-4}	38.43	22.29
1×10^{-3}	41.48	22.37
1×10^{-2}	52.24	30.06
1×10^{-1}	58.25	104
5×10^{-1}	58.61	187.9

Note: Compressive stresses are taken positive

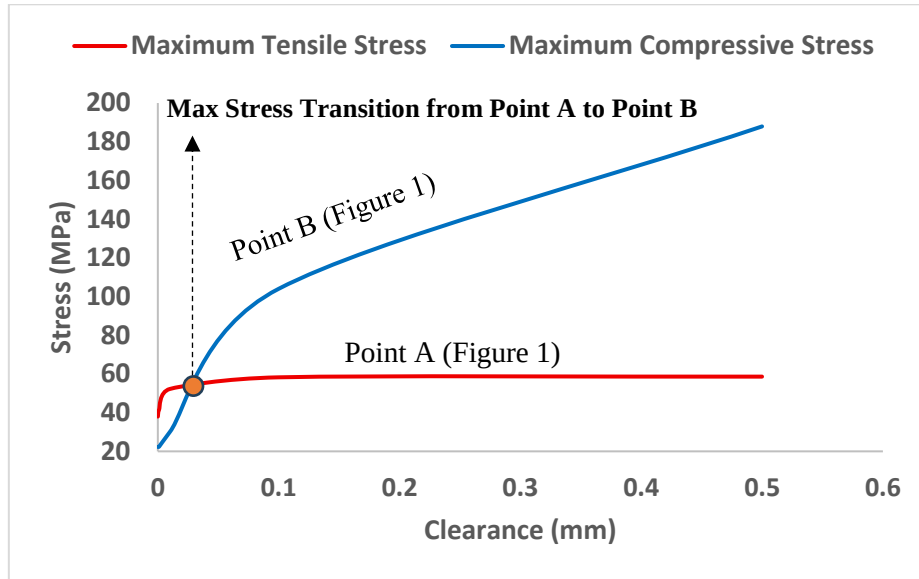
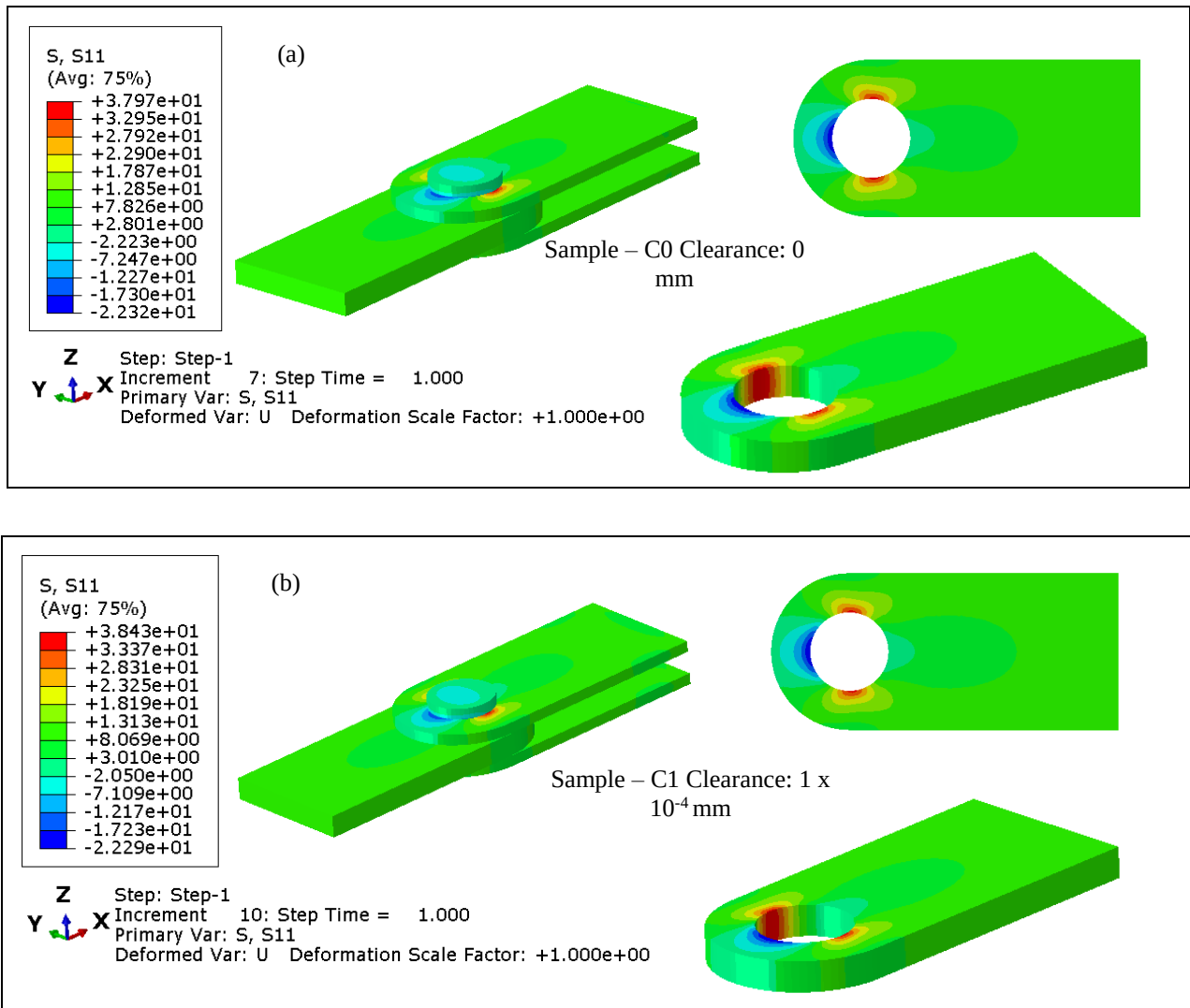
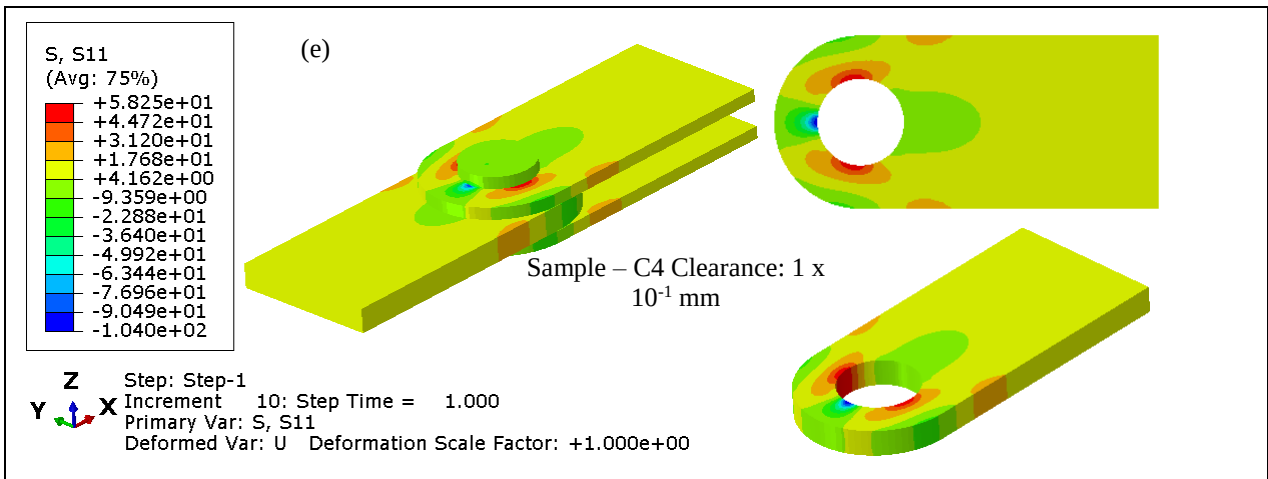
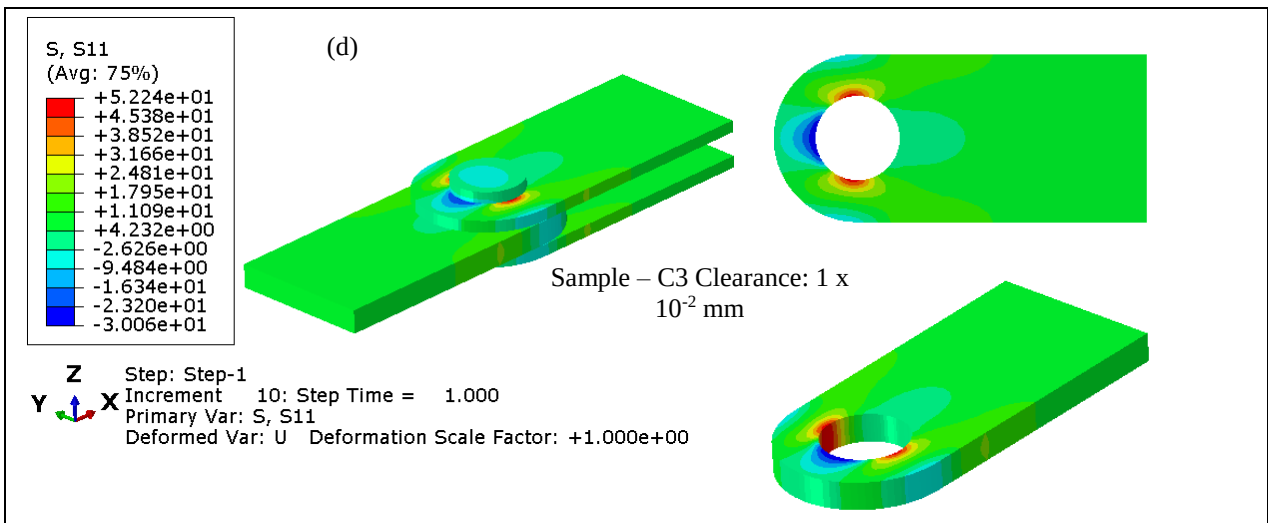
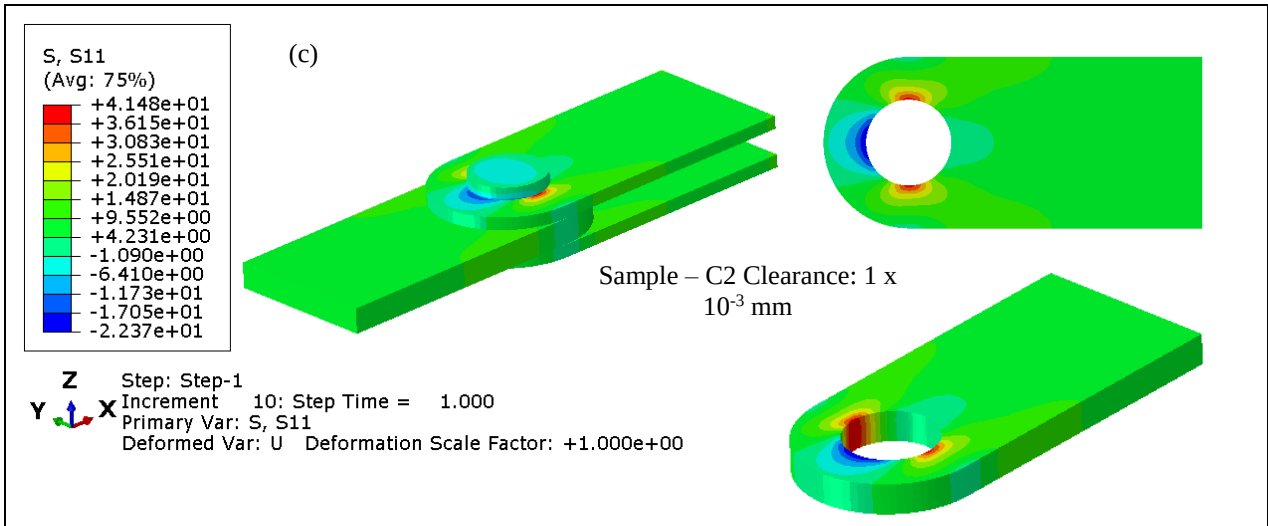


Figure 3. Variation of maximum tensile stress (Point A) and maximum compressive stress (Point B) with increasing clearance





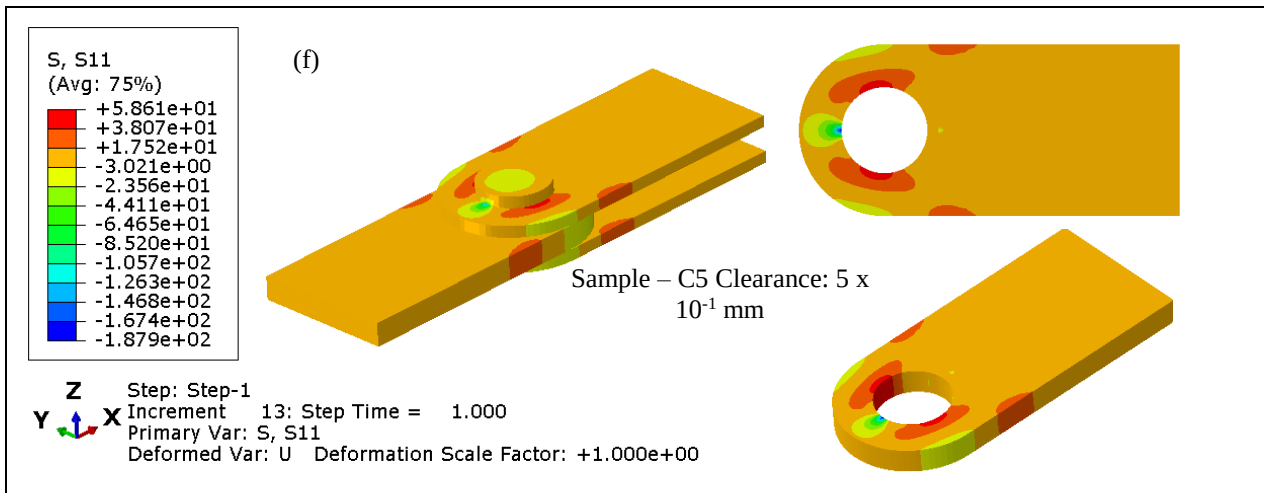


Figure 4. Normal stress distribution in lug for various clearances, (a) C0, (b) C1, (c) C2, (d) C3, (e) C4, and (f) C5

As evident from the results presented in Table 3 and Figure 4(a), for the zero-clearance condition, the maximum tensile stress at Point A is 37.97 MPa, while the compressive stress at Point B is 22.32 MPa. Under this condition, the pin fits tightly within the lug hole, resulting in a relatively uniform contact distribution along the hole surface. Consequently, the load is transferred more evenly, leading to lower localized contact stresses. Additionally, when a very small clearance ($1 \times 10^{-4} - 1 \times 10^{-3}$ mm) is introduced, only minor changes in the stress values are observed, as reported in Table 3. The corresponding normal stress contours presented in Figure 4(b) and Figure 4(c) indicate that the overall stress distribution remains very similar to the zero-clearance case. The tensile stress at Point A increases from 37.97 MPa to 41.48 MPa, corresponding to an increase of approximately 9.2%. On the other side of the coin, the compressive stress at Point B increases only marginally from 22.32 MPa to 22.37 MPa, representing an increase of about 0.2%. These results indicate that extremely small clearances have a negligible influence on the stress distribution, and the load transfer mechanism within the lug–pin joint remains largely unchanged.

As the clearance increases to 1×10^{-2} mm, the tensile stress at Point A rises to 52.24 MPa, representing an increase of approximately 37.6% relative to the zero-clearance case. On the other hand, the compressive stress at Point B increases to 30.06 MPa, corresponding to an increase of about 34.7%. The corresponding normal stress contours in Figure 4(d) reveal that the load transfer between the pin and the lug becomes more localized, highlighting the emergence of stress concentrations around the contact region [2, 9, 23]. This behavior can be attributed to the gradual reduction in the effective contact area: as the pin no longer fully engages the lug hole, the stress distribution becomes increasingly uneven, leading to higher local tensile and compressive stresses [2]. This transition marks the onset of a more critical stress state, signaling the sensitivity of lug–pin joints to even modest increases in clearance.

A more pronounced change occurs when the clearance reaches 1×10^{-1} mm. At this stage, the tensile stress at Point A rises to 58.25 MPa, corresponding to an increase of 53.4% compared with the zero-clearance case. On the other hand, the compressive stress at Point B escalates sharply to 104 MPa, representing an increase of approximately 366% (about 4.7 times) relative to the initial value. The normal stress contours in Figure 4(e) reveal that the pin now engages the lug over a smaller area, resulting in highly localized contact stresses. Interestingly, the transition of the maximum stress from Point A to Point B may occur at clearances smaller than 1×10^{-1} mm, highlighting the sensitivity of the joint to clearance.

At the largest clearance considered (5×10^{-1} mm), the tensile stress at Point A increases slightly to 58.61 MPa, an overall rise of 54.3%, whereas the compressive stress at Point B rises dramatically to 187.9 MPa, an increase of approximately 742% (about 8.4 times) over the zero-clearance case. The corresponding stress contours in Figure 4(f) illustrate extreme localization of the contact region, highlighting the critical sensitivity of lug–pin joints to large clearances and the potential for accelerated contact-induced damage and fatigue.

An important observation from Table 3 and Figure 3 is the transition of the location of the maximum stress from Point A to Point B with increasing clearance. At small or zero clearance, the maximum stress in the lug is primarily tensile at Point A, which is associated with the stress concentration around the hole boundary. However, as the clearance increases, the contact between the pin and the lug becomes more localized, causing the maximum stress to shift to Point B, where the compressive contact stresses dominate.

This behavior is clearly illustrated by the stress contours in Figure 4. With increasing clearance, the contact between the pin and the lug occurs over a smaller region of the hole surface. As a result, the load is transferred through a highly localized contact zone, leading to a significant increase in compressive stress at Point B. Meanwhile, the tensile stress at Point A increases more gradually because it is governed by the overall load transfer through the lug section rather than purely by the local contact condition.

Consequently, the results indicate that clearance has a strong influence on the stress distribution in lug–pin joints, particularly on the compressive contact stresses. Excessive clearance results in highly localized contact loading and a shift of the critical stress location from Point A to Point B, which may promote surface damage, plastic deformation, and fatigue crack initiation at the pin–lug interface. In other words, the results clearly suggest that clearance has a significantly greater influence on compressive contact stresses than on tensile stresses in the lug body.

Based on these findings, multiple simulations were performed to determine the clearance at which the maximum tensile stress at Point A equals the maximum compressive stress at Point B. The results of this parametric investigation are presented in Figure 5, which clearly identifies the specific clearance corresponding to this stress transition. These results provide a quantitative basis for understanding the shift of the maximum stress location from Point A to Point B and highlight the critical influence of clearance on the stress distribution and contact behavior of lug–pin joints.

As shown in Figure 5, the results of the parametric simulations reveal that the maximum tensile stress and the maximum compressive stress reach equal magnitudes at a clearance of 0.0331 mm. This condition is particularly significant from a design perspective, as both critical stress regions become equally dominant at this clearance value. As shown in Figure 3 and confirmed by the results presented in Figure 5, one of the key findings is that for the case of $d/w = 0.5$, point A should be considered in the design for clearance values smaller than 0.0331 mm, whereas point B becomes critical for clearance values greater than 0.0331 mm.

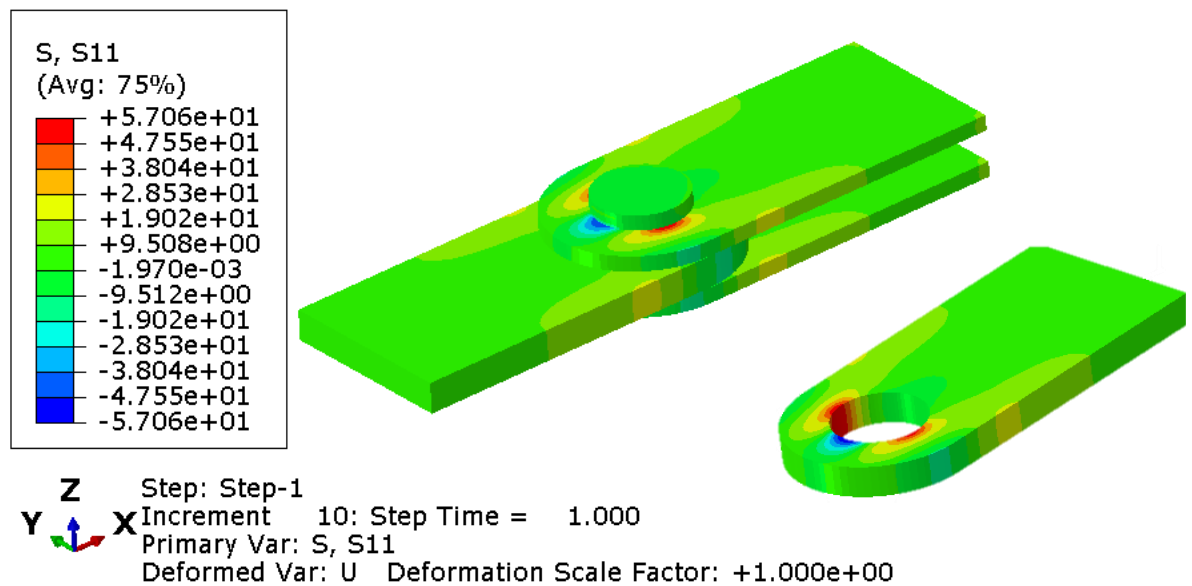


Figure 5. Normal stress distribution in the lug at a clearance of 0.0331 mm with equal maximum tensile and compressive stresses at of $P = 5000$ N

Although the critical clearance value is identified as shown in Figure 5, this value is valid for the specific case of $d/w = 0.5$ and a loading condition of $P = 5000$ N. As reported in the relevant literature, the magnitude of the applied load also influences whether the maximum stress occurs at Point A or at Point B. In other words, the location of the maximum stress varies depending on the magnitude of the applied load and the d/w ratio [2, 19]. This behavior can be attributed to the dependence of the stress concentration factor on both the magnitude of the applied load and the clearance value [2, 19].

To further investigate this behavior, the simulations were repeated by varying the applied load from 1200 N to 36,000 N while maintaining the clearance at 0.0331 mm. The corresponding normal stress distributions obtained from these simulations are presented in Figure 6. The results in Figure 6 are summarized in Table 4, which details the changes in maximum tensile and compressive stresses at a fixed clearance of 0.0331 mm under varying loads. As presented in Table 4, the maximum tensile and

compressive stresses at a constant clearance of 0.0331 mm exhibit a striking dependence on the applied load. Notably, at 5000 N ($\sigma = 8.33 \text{ MPa}$), the tensile and compressive stresses converge to an identical value of 57.06 MPa, confirming the critical balance previously observed in Figure 5. Beyond this load, tensile stress grows more rapidly than compressive stress, reaching 343.7 MPa at 36000 N compared to 176.4 MPa for compressive stress, highlighting the nonlinear amplification of stresses with increasing load. These results emphasize a pivotal design insight: for loads below 5000 N, the compressive stress at point B governs, whereas for higher loads, tensile stress at point A becomes critical. The results presented in Figure 7 provide a clearer illustration of this behavior. Thus, this interplay between load magnitude and stress location underscores the essential role of stress concentration and clearance in defining the governing design criteria.

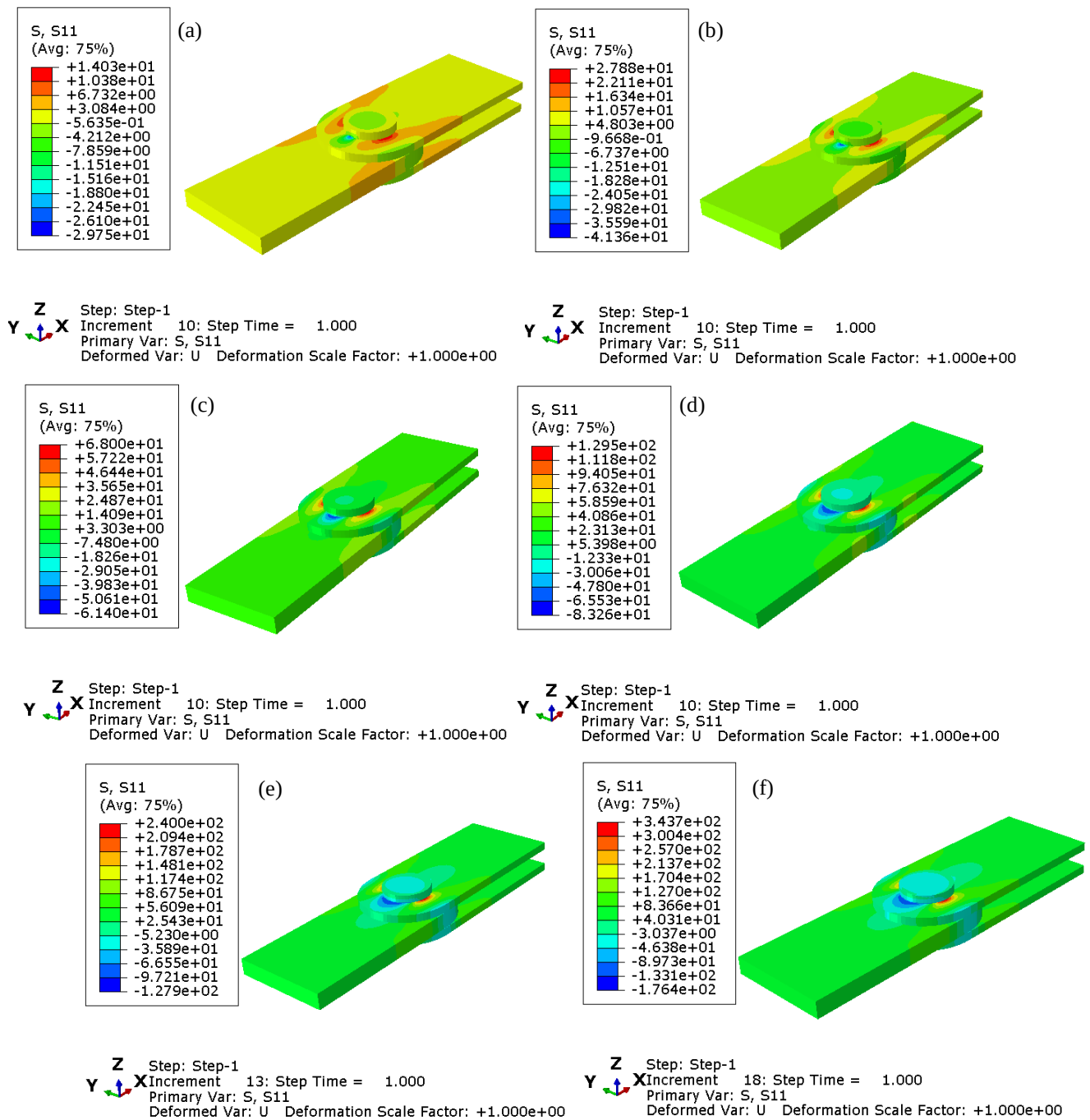
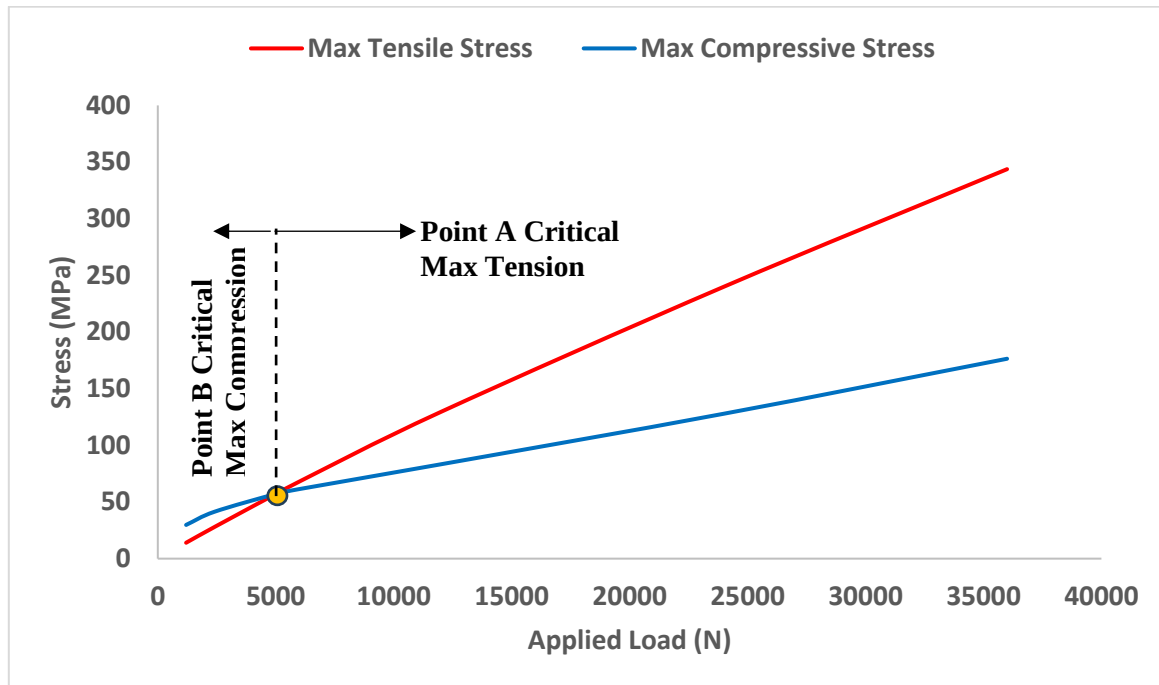


Figure 6. Normal stress distributions in the lug for various loads at a constant clearance of 0.0331 mm, (a) 1200 N, (b) 2400 N, (c) 6000 N, (d) 12000 N, (e) 24000 N, and (f) 36000 N

Table 4. Maximum tensile and compressive stresses at a constant clearance of 0.0331 mm under varying loads

Applied Load P (N)	Corresponding Applied σ (MPa)	Max Tensile Stress (MPa)	Max Compressive Stress (MPa)
1200	2	14.03	29.75
2400	4	27.88	41.36
5000	8.33	57.06	57.06
6000	10	68	61.4
12000	20	129.5	83.26
24000	40	240	127.9
36000	60	343.7	176.4

Note: Compressive stresses are taken positive.


Figure 7. Variation of maximum tensile and compressive stresses with applied load at a constant clearance of 0.0331 mm

Based on the results collectively presented in Figures 3 and 7, for a round-ended lug ($d/w = 0.5$) [18,19,24], which represents the most commonly used configuration in practice, the critical stress location depends strongly on the clearance. When the clearance exceeds 0.0331 mm, point B, corresponding to the maximum compressive stress, becomes critical, whereas for smaller clearances, point A, corresponding to the maximum tensile stress, governs. At the specific clearance of 0.0331 mm, loads below 5000 N render point B critical, while higher loads shift the critical location to point A. The most important finding is that the critical location varies dynamically with both the magnitude of the applied load and the clearance. For each load, there exists a specific clearance at which the maximum tensile and compressive stresses are equal, and conversely, there are clearances where the dominant stress shifts in the opposite manner. However, it should be noted that, since the present analysis is limited to the elastic regime, the identified critical regions represent potential stress concentration zones rather than actual failure or yielding locations. The maximum normal stress is used as a comparative indicator to evaluate the influence of initial clearance on the stress distribution within the lug-pin joint. This approach is consistent with classical design methodologies, where elastic peak stresses are commonly used to assess the severity of geometric discontinuities in structural connections. Furthermore, the coefficient of friction at the contact interfaces has not been explicitly defined in the present study, as the analysis is carried out under a simplified contact assumption to focus on the effect of initial clearance on elastic stress distribution. The inclusion of frictional effects may significantly influence local contact pressures and stress magnitudes, and in some cases may lead to stress levels closer to the material yield strength. Therefore, a more

detailed investigation incorporating frictional contact behavior and its potential contribution to nonlinear response is suggested as a topic for future work.

The values presented here are applicable to round-ended lug-pin connections with the specified geometric dimensions, whereas square-ended lug-pin configurations require further investigation. Additionally, for the round-ended lug, the lug radius is aligned with the center of the drilled hole. Cases where this alignment does not occur are also possible [8] and represent a separate topic for future research.

5. CONCLUDING REMARKS

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

- The stress distribution in lug-pin joints is highly sensitive to pin-hole clearance, with maximum tensile stress occurring at Point A and maximum compressive stress at Point B. While very small clearances (<0.001 mm) produce minimal changes, larger clearances lead to highly localized contact stresses, particularly at Point B, dramatically increasing the potential for contact-induced damage.
- For a round-ended lug with $d/w = 0.5$ under a load of 5000 N, the maximum tensile and compressive stresses become equal at a clearance of 0.0331 mm. This threshold marks a transition in the governing stress location, providing a quantitative reference for design optimization.
- Load magnitude strongly influences stress dominance. At the critical clearance, low loads (<5000 N) make the compressive stress at Point B critical, whereas higher loads shift the governing stress to tensile stress at Point A. This interplay highlights the combined effect of clearance and applied load in defining the joint's critical regions.
- Nonlinear stress amplification occurs with increasing clearance. Compressive stresses increase far more sharply than tensile stresses as clearance grows, with the highest compressive stress reaching over eight times the zero-clearance value. This nonlinear behavior underscores the risk of localized plastic deformation, surface damage, and fatigue initiation at the pin-lug interface.
- In practical applications, the governing stress location in round-ended lugs shifts with clearance: point A is critical at small clearances, while point B dominates at larger clearances. This dynamic behavior, influenced by both clearance and applied load, underscores the importance of precise clearance specification in lug-pin joints to ensure balanced stress distribution and reduce the risk of premature failure. Configurations such as square-ended lugs or cases with non-aligned pin-lug radii remain less explored and present opportunities for future research.

AUTHOR CONTRIBUTIONS

Osman Özenç: Supervision, Interpretation of the Simulation Results, Formal Analysis, Writing-Review and Editing. **Mehmet Akif Dündar:** Data Collection, Finite Element Simulations, Writing-Review and Editing. **Kazım Ercan:** Data Collection, Simulation, Drafting. **Mirali Nuraliyev:** Conceptualization, Analysis of Results, Review and Editing.

CONFLICT OF INTEREST

The authors declare no conflict of interest.

ETHICS

There is no ethical problem in the publication of this article.

REFERENCES

- [1] K. Iyer, "Solutions for contact in pinned connections," *Int. J. Solids Struct.*, vol. 38, no. 50–51, pp. 9133–9148, 2001.
- [2] A. Strozzi, M. Giacomini, E. Bertocchi, S. Mantovani, and A. Baldini, "Towards an analytical model of a pin-lug connection," *Int. J. Solids Struct.*, vol. 253, p. 111446, 2022, doi: 10.1016/j.ijsolstr.2022.111446.
- [3] D. Duerr, "Pinned connection strength and behavior," *Journal of structural engineering*, vol. 132, no. 2, pp. 182–194, 2006.
- [4] A. Strozzi, A. Baldini, M. Giacomini, S. Rivasi, and R. Rosi, "Maximum Stresses in a Taper-shanked Round-ended Lug Loaded by an Oblique Concentrated Force," *Strain*, vol. 43, no. 2, pp. 109–118, May 2007, doi: <https://doi.org/10.1111/j.1475-1305.2007.00321.x>.
- [5] R. N. Tolbert and R. M. Hackett, "Experimental investigation of lug stresses and failures," *Engineering Journal*, vol. 11, no. 2, pp. 34–37, 1974.

- [6] R. J. Grant, J. Smart, and P. Stanley, "A parametric study of the elastic stress distribution in pin-loaded lugs," *J. Strain Anal. Eng. Des.*, vol. 29, no. 4, pp. 299–307, Oct. 1994, doi: 10.1243/03093247V294299.
- [7] A. Strozzi and E. Radi, "Analytical tool assisting the designer of pin-lug connections," *Proc. Inst. Mech. Eng. C J. Mech. Eng. Sci.*, vol. 237, no. 23, pp. 5734–5742, Dec. 2023, doi: 10.1177/09544062231167264.
- [8] W. D. Pilkey, D. F. Pilkey, and Z. Bi, *Peterson's stress concentration factors*. John Wiley & Sons, 2020.
- [9] Y. Li, R. Huang, S. Zhao, and J. Wang, "Contact pressure analysis of pin-loaded lug with clearance," *Advances in Mechanical Engineering*, vol. 14, no. 6, p. 16878132221107476, Jun. 2022, doi: 10.1177/16878132221107475.
- [10] T. S. Ramamurthy, "New studies on the effect of bearing loads in lugs with clearance fit pins," *Compos. Struct.*, vol. 11, no. 2, pp. 135–150, 1989, doi: [https://doi.org/10.1016/0263-8223\(89\)90065-2](https://doi.org/10.1016/0263-8223(89)90065-2).
- [11] H. Reißner and F. Strauch, "Ringplatte und Augenstab," *Ingenieur-Archiv*, vol. 4, no. 5, pp. 481–505, 1933.
- [12] M. M. Frocht and H. N. Hill, "Stress-Concentration Factors Around a Central Circular Hole in a Plate Loaded Through Pin in the Hole," *J. Appl. Mech.*, vol. 7, no. 1, pp. A5–A9, Mar. 2021, doi: 10.1115/1.4008987.
- [13] A. Persson, *On the stress distribution of cylindrical elastic bodies in contact*. Chalmers University of Technology, 1964.
- [14] S. B. Liu and C. L. Tan, "Two-dimensional boundary element contact mechanics analysis of pin-loaded lugs with single cracks," *Eng. Fract. Mech.*, vol. 48, no. 5, pp. 711–725, 1994.
- [15] S. B. Liu and C. L. Tan, "Boundary element contact mechanics analysis of pin-loaded lugs," *Comput. Struct.*, vol. 53, no. 3, pp. 653–665, 1994.
- [16] N. L. Pedersen, "Stress concentration and optimal design of pinned connections," *J. Strain Anal. Eng. Des.*, vol. 54, no. 2, pp. 95–104, 2019.
- [17] R. J. Callinan, "Stress analysis of a lug loaded by a pin," 1977.
- [18] D. J. Cartwright and A. P. Parker, "Opening mode stress intensity factors for cracks in pin-loads joints," *Int. J. Fract.*, vol. 18, no. 1, pp. 65–78, 1982, doi: 10.1007/BF00034829.
- [19] A. Strozzi, A. Baldini, M. Giacomini, E. Bertocchi, and L. Bertocchi, "Maximum equivalent stress in a pin-loaded lug in the presence of initial clearance," *J. Strain Anal. Eng. Des.*, vol. 46, no. 8, pp. 760–771, Nov. 2011, doi: 10.1177/0309324711423587.
- [20] D. Systèmes, "Abaqus Documentation (Dassault Systèmes, Providence, RI)," 2014, Version.
- [21] N. D. Kankanamge and M. Mahendran, "Mechanical properties of cold-formed steels at elevated temperatures," *Thin-Walled Structures*, vol. 49, no. 1, pp. 26–44, Jan. 2011, doi: 10.1016/j.tws.2010.08.004.
- [22] D. Systèmes, "Abaqus Analysis User's Manual: Element Library.," Dassault Systèmes, 2016.
- [23] "References for the Design of Steel Structures," in *Structural Steel Design to Eurocode 3 and AISC Specifications*, 2016, pp. 34–48. doi: <https://doi.org/10.1002/9781118631201.ch2>.
- [24] R. G. Budynas and J. K. Nisbett, *Shigley's mechanical engineering design*, vol. 9. McGraw-Hill New York, 2011.