

Numerical Method-Based Study on Deformation and Fatigue Life of SKD 11 Dies and Punches

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Article Information	ABSTRACT
Article History Received : June 03, 2026 Revised : June 12, 2026 Published : July 04, 2026	Progressive die stamping is critical in automotive component manufacturing, yet fatigue failure of dies and punches remains a key challenge that directly impacts tool life and production continuity. This study aims to evaluate the structural integrity and comparative fatigue life of four steel for cold die (SKD) 11 compound progressive die components: die button, punch pierce, die blank, and punch blank, to translate the findings into practical maintenance scheduling criteria. This study presents a sequential finite element analysis (FEA) framework, comprising static structural analysis followed by stress-life fatigue assessment to evaluate the structural integrity and cyclic service life of SKD 11 tool steel components in a compound progressive die for automotive hanging bracket stamping. Three-dimensional models were analyzed in ANSYS Workbench using tetrahedral meshing with convergence verified at <5% variation in peak von Mises stress. Cutting forces were derived from the formulation $P = L \times t \times \sigma_B$, with mean stress correction applied via the Goodman approach using empirical SKD 11 S-N data. Static analysis confirmed all components satisfy structural safety criteria, with von Mises stresses ranging from 286 MPa to 979 MPa, all below the SKD 11 yield strength of 1,540 MPa and maximum deformation within 0.023 mm. Fatigue analysis revealed a pronounced life disparity: the die blank exhibited a minimum cycle life of 69,215 cycles, while the punch pierce showed the most critical behavior at 827 cycles. The punch pierce requires inspection every 500–800 cycles, while future research should investigate fillet geometry optimization and surface coating effects to further extend punch service life.
Keywords: <i>Dies Stamping;</i> <i>Fatigue Life;</i> <i>FEA;</i> <i>SKD 11.</i>	

INTRODUCTION

The automotive component manufacturing industry requires highly reliable production tools, especially in the metal forming process using dies and punches (Arohman et al., 2022; Lianny et al., 2022; Zhang et al., 2023). Components such as hanging brackets are structural parts that must be produced with precision and consistency, so the quality of the dies and punches used greatly determines production efficiency and tool life (Safril et al., 2024; Solih et al., 2023; Su et al., 2022; Xu et al., 2022). In the context of vocational education, die design practicums are an important means for students to understand the relationship between design, analysis, and manufacturing realization. Therefore, this research is urgent to ensure that the designed dies and

punches are not only producible but also have adequate resistance to repetitive workloads (Solih et al., 2023).

A number of previous studies have evaluated the structural performance of dies and punches through both simulation and experimental approaches. On the simulation side, static and fatigue analysis using finite element methods has been widely applied to verify design safety and predict service life (Motte & De Waele, 2024; Zhang et al., 2023). ANSYS based FEA has demonstrated the ability to capture stress distribution and deformation in die components with sufficient accuracy for engineering decision making (Kaya et al., 2022; Shaheen et al., 2020). The foundational design principles for blanking and piercing die punches established by (Avadhani et al., 2017) have been further validated and extended by recent numerical studies (Kim et al., 2023). On the experimental side, (Akiniwa et al., 2006) characterized the very high cycle fatigue behavior of SKD 11 Steel for Cold Die grade 11, a high-chromium cold work tool steel standardized under JIS G 4404 and equivalent to AISI D2 revealing that its fatigue resistance is orientation-dependent, with longitudinal specimens outperforming transverse specimens across the 500–900 MPa stress amplitude range. More recently, (Masuda et al., 2024) experimentally investigated fatigue crack growth behavior in AISI D2 tool steel under surface coating conditions, demonstrating that uncoated specimens exhibit fatigue strengths consistent with the Akiniwa dataset and providing further empirical grounding for S-N based life prediction in this material class.

However, a critical research gap remains in the existing literature. While prior studies have applied FEA to isolated die components (Avadhani et al., 2017; Zhang et al., 2023) or focused solely on static strength verification (Kaya et al., 2022; Solih et al., 2025), no study has performed a systematic comparative fatigue life analysis across geometrically distinct die and punch components within a compound progressive die system. In particular, no study has incorporated SKD 11's orientation-dependent S-N behavior as empirically characterized by (Akiniwa et al., 2006) with Goodman mean stress correction to produce component-specific fatigue life predictions. Furthermore, the translation of such numerical findings into practical maintenance scheduling criteria has not been addressed in prior work.

The present study addresses this gap by making three specific contributions: (1) a systematic comparative fatigue life analysis of all four geometrically distinct components of a compound progressive die using SKD 11; (2) integration of SKD 11's empirically characterized orientation-dependent S-N behavior with Goodman mean stress correction for physically representative fatigue prediction; and (3) translation of quantitative FEA results into component specific maintenance scheduling criteria, providing practical design guidance beyond what prior static only analyses offer. Accordingly, this study evaluates the structural integrity and cyclic service life of SKD 11 die button, punch pierce, die blank, and punch blank components through sequential static and fatigue FEA, with the goal of identifying component-specific fatigue critical zones and establishing quantitative criteria for maintenance scheduling in compound progressive die stamping operations.

RESEARCH METHODS

This study was conducted using a sequential FEA-based numerical simulation approach implemented in ANSYS Workbench 2025 R2 (ANSYS Inc., Canonsburg, PA, USA). The three-dimensional solid models of the die button, punch pierce, die blank, and punch blank were developed using CAD software based on the compound progressive die design documentation for hanging bracket production, with geometric dimensions derived from the associated die design practicum course at Polytechnic STMI Jakarta.

The following simplifying assumptions were adopted throughout the simulation: (1) all materials were treated as homogeneous, isotropic, and linearly elastic; (2) thermal effects and residual stresses from the manufacturing process were neglected; (3) dynamic inertia effects were excluded, as the stamping operation frequency is sufficiently low to justify a quasi-static loading assumption; and (4) each component was analyzed independently under conservative isolation conditions, without modeling inter-component contact or assembly pre-load effects.

The sequential simulation workflow proceeded in two stages: first, static structural analysis was performed to obtain the von Mises stress distribution, total deformation, and safety factor for each component under the applied cutting force; second, the resulting stress field was used as the basis for stress-life fatigue analysis to predict the cyclic service life of each component.

Static Finite Element Analysis (FEA)

Static simulations were performed using ANSYS Workbench software with a finite element analysis (FEA) approach (Shaheen et al., 2020; Solih et al., 2025). The designed three-dimensional models of the dies and punches were entered into the simulation, with a refined mesh selection in critical areas such as sharp corners and fillets. Table 1 shows the SKD 11 parameters used in the simulation (Mac et al., 2022; Shaheen et al., 2020).

Table 1. SKD 11 Mechanical Properties

No.	Mechanical Properties	Description
1	Elastic Modulus	210000 Mpa
2	Yield Strength	1540 Mpa
3	Ultimate Tensile Strength	2340 MPa

The mechanical properties listed in Table 1 are consistent with values reported across the literature for SKD 11 and its international equivalents. (Shaheen et al., 2020) adopted an elastic modulus of 207,000 MPa and yield strength of 1,520 MPa for AISI D2 tool steel in compound die FEA within 1.4% and 1.3% of the present values, respectively. (Masuda et al., 2024) reported a UTS range of 2,280–2,350 MPa for heat-treated AISI D2 specimens, consistent with the 2,340 MPa adopted here. Additionally, (Mac et al., 2022) characterized SKD 11 in a machining study and confirmed an elastic modulus of approximately 210,000 MPa under standard heat treatment conditions. The close agreement between the present material model and these independent sources, with maximum deviation below 2% across all properties which confirms that the SKD 11 parameters adopted in this study are representative of the material's typical behavior in hardened condition and suitable for use as FEA inputs.

Boundary conditions were defined as follows: a fixed support constraint was applied to the entire bottom face of each component, restricting all translational and rotational degrees of freedom at that surface. The cutting force derived from Equation (1) was applied as a uniformly distributed force over the active cutting surface of each component. No lateral constraints were imposed on the side faces, representing free body conditions during the cutting stroke. Individual component analyses were performed under conservative isolation assumptions, without modelling intercomponent contact friction.

The simulation results, in the form of von Mises stress distribution, total deformation, and safety factor against the material yield limit, were used to assess whether the design met the structural strength requirements (Xu et al., 2022; Yuan et al., 2024). If the maximum stress exceeded the material's UTS value, the design was considered unsafe and needed to be revised.

Fatigue Analysis

The next step is fatigue analysis to estimate the service life of components (Zhang et al., 2023). This analysis is performed using two approaches: first, using the fatigue module in ANSYS Workbench with the stress-life (S-N curve) method, and second, using fatigue analysis in static structure to capture more complex dynamic load characteristics (Kim et al., 2023; Motte & De Waele, 2024). The S-N curve data for SKD 11 originally characterized by (Akiniwa et al., 2006) and subsequently confirmed by (Masuda et al., 2024) using AISI D2 tool steel of equivalent grade which serves as the primary empirical reference for cycle life estimation in this study (Akiniwa et al., 2006; Masuda et al., 2024; Motte & De Waele, 2024), with average stress correction using the Goodman approach (Böhm et al., 2020). The fatigue simulation parameters were defined as follows: a fully pulsating compressive load cycle was assumed with a stress ratio of $R = 0$ (minimum stress = 0, maximum stress = peak cutting force magnitude), representing a zero-to-

compression pulsating load cycle consistent with the operating condition of stamping dies under repeated press stroke. Mean stress correction was applied using the Modified Goodman criterion within the ANSYS fatigue module, referencing the SKD 11 S-N dataset from (Akiniwa et al., 2006) as the primary empirical input. The target number of cycles was set to 10^6 , consistent with the conventional high-cycle fatigue boundary. The simulation results, in the form of fatigue life estimates (number of cycles to failure), damage accumulation, and critical crack locations, were used to assess the design's resistance to material fatigue (Avadhani et al., 2017; Bang et al., 2022).

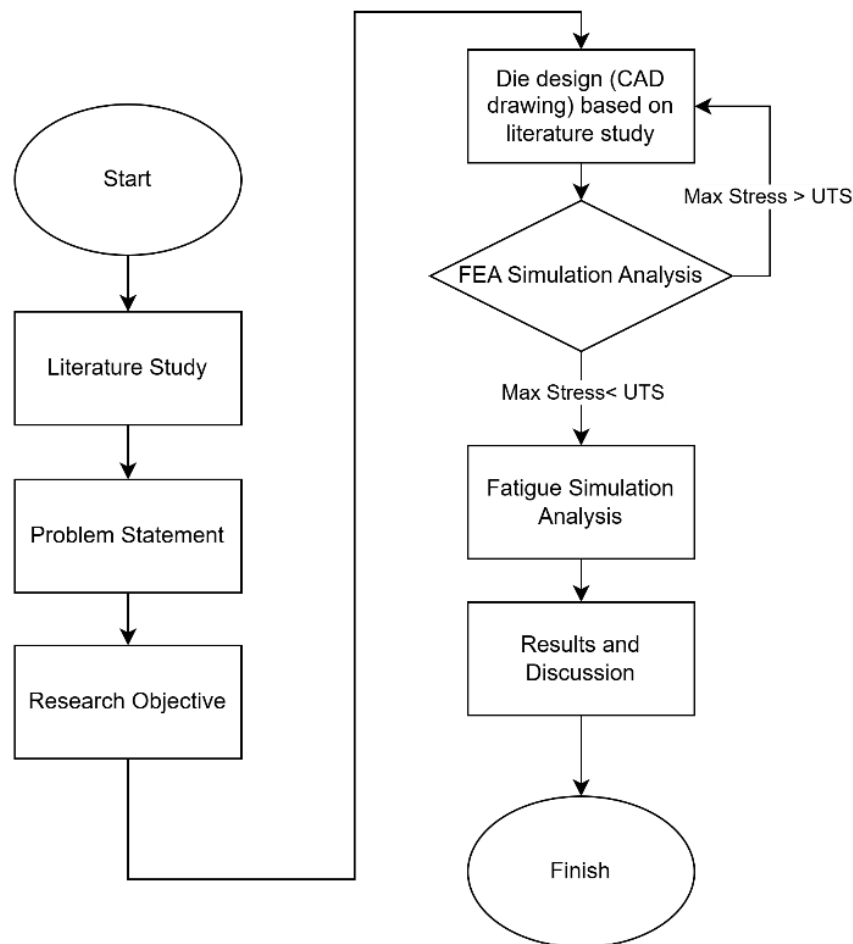


Figure 1. Research Flowchart

Validation

The final stage is validation. Due to the unavailability of full-scale press testing facilities on campus, direct experimental punching tests could not be performed. As an alternative, indirect validation was conducted using a comparative approach against published benchmark data (Kaya et al., 2022).

The fatigue life predictions obtained from the FEA simulation were benchmarked against the empirical S-N dataset for SKD 11 reported by (Akiniwa et al., 2006; Motte & De Waele, 2024). The S-N graph for SKD 11 material studied the relationship between cyclic stress amplitude and the number of cycles until failure (Masuda et al., 2024; Su et al., 2022). They found the Longitudinal orientation (SKD11-L) has higher fatigue resistance than transverse orientation (SKD11-T), with maximum number of cycles up to 1.0×10^{11} when the stress amplitude at 500 MPa and the lowest number cycle down to 105 at 900 MPa. Some specimens did not fail at low stresses, indicating a long potential service life. This data is an important reference in estimating the cycle life of SKD 11 dies and punches (Akiniwa et al., 2006; Mac et al., 2022).

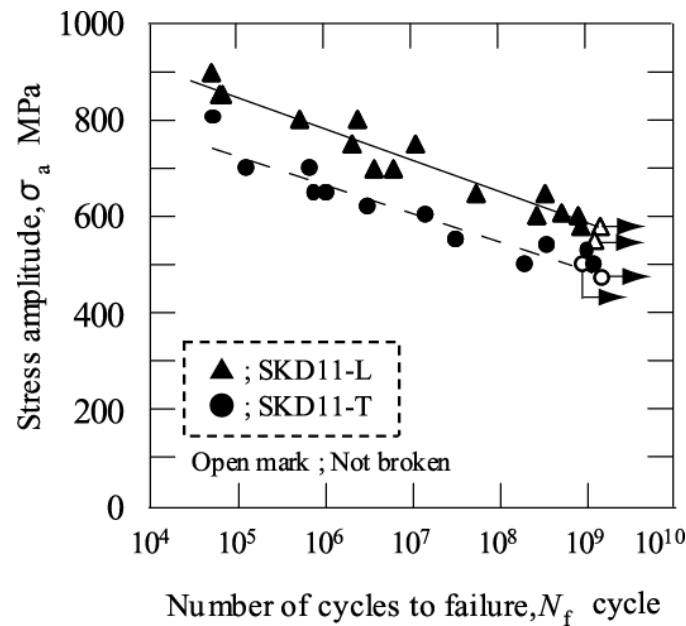


Figure 2. Fatigue Characteristics of SKD 11 Steel

(Source: Akiniwa et al., 2006)

The 500-900 MPa stress amplitude domain documented in the S-N curve, within which Akiniwa et al. (Akiniwa et al., 2006; Motte & De Waele, 2024). report a monotonic reduction in cycle life following the Basquin power-law relationship (Böhm et al., 2020; Motte & De Waele, 2024). The fatigue life hierarchy observed in the simulation (Die Blank > Punch Blank > Die Button > Punch Pierce) is qualitatively consistent with this empirical trend: components operating at lower stress levels exhibit substantially longer predicted service lives, as expected from the S-N characteristic of SKD 11 material (Masuda et al., 2024). This indirect validation approach follows the methodology adopted by Kaya et al. (Kaya et al., 2022) for tool life estimation in progressive die stamping when direct experimental testing is infeasible.

Methods

The design of the dies and punch in Figure 3 for the production of the hanging bracket are realized in the form of a three-dimensional model using CAD software (Jundi & Alaiwi, 2024; Yang & Hinduja, 2023). This design is based on a literature study, considering geometric aspects that support structural strength and ease of the manufacturing process (Su et al., 2022; Xu et al., 2022).

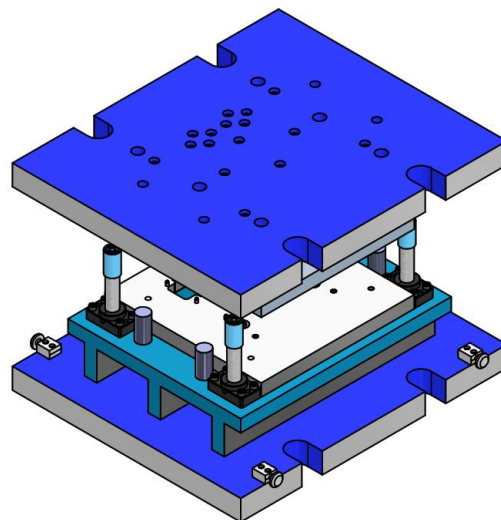


Figure 3. 3D Die Assembly

This study uses cutting force as a reference for the load acting on the dies and punches. Cutting force is calculated based on the following equation:

$$P_{cutting} = L \times t \times \sigma_B \quad (1)$$

Where L is the circumference of the cutting feature, t is the plate thickness of 0.9 mm, and σ_B is the shear stress of the plate material assumed to be 80% of the ultimate tensile strength (UTS) value (Avadhani et al., 2017; Shaheen et al., 2020). With a material UTS of 27 kgf/mm², the value obtained is 21.6 kgf/mm². The cutting force calculation results for the die and punch designs are then converted to N by multiplying gravity 9.8 m/s² shown in Table 2, and these values are used as force inputs in static simulations and fatigue analysis.

Table 2. Punch and Die Cutting Forces

No.	Part	L (mm)	t (mm)	σ_B (kgf/mm ²)	P (kgf)	P (N)
1	Die Button	78.5	0.9	21.6	1526	14970
2	Punch Pierce	78.5	0.9	21.6	1526	14970
3	Die Blank	325.6	0.9	21.6	6329	62087
4	Punch Blank	451.1	0.9	21.6	8770	86034

In the numerical simulation, the meshing process was performed using the tetrahedron method, which is suitable for the complex geometry of dies and punches (Solih et al., 2025; Yuan et al., 2024). The body sizing was set in the range of 0.003 to 0.007 m, depending on the size and complexity of each component. To improve accuracy, refinement is performed on critical edges, particularly in the punch and die areas that are in direct contact with the cutting process, so that the stress distribution in the contact zone can be identified in greater detail. Details of the meshing settings and simulation parameters used can be seen in Table 3. Mesh convergence was verified by progressively refining the element size at critical stress locations until the variation in peak von Mises stress between successive refinements was less than 5%, consistent with the convergence criterion recommended for structural FEA of tool steel components (Kim et al., 2023). The final mesh configurations presented in Table 3 correspond to the converged state.

Table 3. Meshing Parameter

No.	Part	Body Sizing	Nodes	Elements
1	Die Button	0.003	93370	63802
2	Punch Pierce	0.003	52100	35472
3	Die Blank	0.007	78231	52115
4	Punch Blank	0.007	50235	34788

RESULTS AND DISCUSSION

Die Button

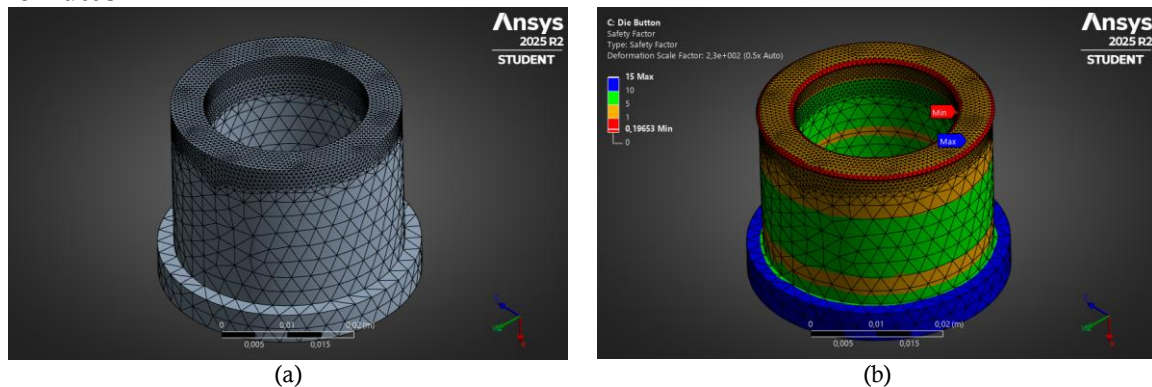


Figure 4. Die Button Analysis. (a) Meshing of Die Button. (b) Safety Factor of Die Button

The die button exhibited a maximum deformation of 0.0075 mm and an average of 0.0038 mm, both well within structurally acceptable limits. The maximum von Mises stress of 846 MPa corresponds to a stress utilization ratio of 54.9% relative to the SKD 11 yield strength of 1,540 MPa, indicating that the component operates with substantial structural reserve. This relatively moderate stress level is attributable to the die button's annular geometry, which distributes the reaction force over a larger contact perimeter compared to the punch components.

Fatigue analysis revealed a minimum cycle life of 1,326 cycles at the outer and inner diameter contact zones regions of maximum stress concentration due to the direct cutting interface. However, the average fatigue life of 971,000 cycles approaches the 10^6 upper boundary of the simulation, indicating that the bulk of the component volume operates well within the high-cycle fatigue regime of SKD 11. This result is consistent with (Shaheen et al., 2020), who reported that die components in compound piercing configurations exhibit substantially higher average fatigue lives than their punch counterparts due to more favorable load distribution geometry. The average safety factor of 5.3 confirms overall structural integrity. The minimum safety factor of 0.2 at the sharp edges is consistent with known FEA stress singularity behavior at geometric discontinuities and does not represent actual material failure.

Punch Pierce

The punch pierce exhibited the highest maximum deformation among all components at 0.023 mm, and the highest peak von Mises stress at 979 MPa, corresponding to a stress utilization ratio of 63.6% relative to yield strength. This elevated stress level is a direct consequence of the punch pierce geometry: as a slender cylindrical component bearing the full compressive cutting force over a relatively small cross-sectional area, it experiences substantially higher stress density than the die components. This geometric disadvantage is compounded by the direct material contact at the cutting outer diameter, which generates localized stress concentration at that interface.

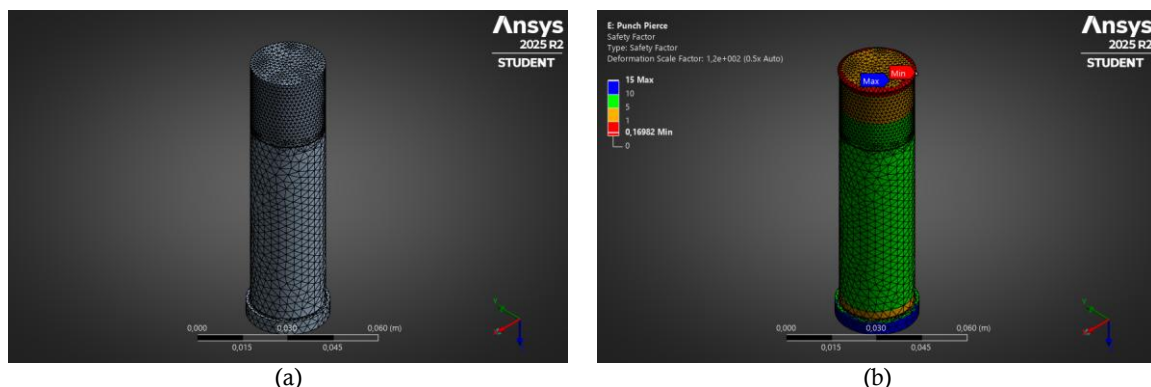


Figure 5. Punch Pierce Analysis. (a) Meshing of Punch Pierce. (b) Safety Factor of Punch Pierce

Consequently, the punch pierce exhibited the most critical fatigue behavior, with a minimum cycle life of only 827 cycles at the cutting-contact outer diameter. Referencing the SKD 11 S-N characteristic documented by (Akiniwa et al., 2006), a peak stress of 979 MPa falls within the steep region of the curve, where cycle life degrades rapidly with increasing stress amplitude, consistent with the Basquin power-law relationship. This is further supported by (Kim et al., 2023), who reported that tool components with stress utilization ratios exceeding 60% of yield strength exhibit fatigue lives two to three orders of magnitude shorter than lower-utilization counterparts, a pattern directly reflected in the present results. Based on these findings, periodic inspection of the punch pierce is recommended after every 500–800 operational cycles. The average safety factor of 5.44 confirms overall structural adequacy, while the minimum value of 0.17 at sharp edges is consistent with known FEA stress singularity behavior at geometric discontinuities and does not represent actual material failure.

Die Blank

Figure 6 shows the die blank simulation result a maximum deformation of 0.012 mm with an average of 0.001 mm, both within structurally acceptable limits. The von Mises stress analysis

shows a maximum value of 286 MPa and an average of 14.34 MPa. This is significantly below the yield strength of the SKD 11 material, confirming that the die blank design possesses adequate structural resistance to the working loads.

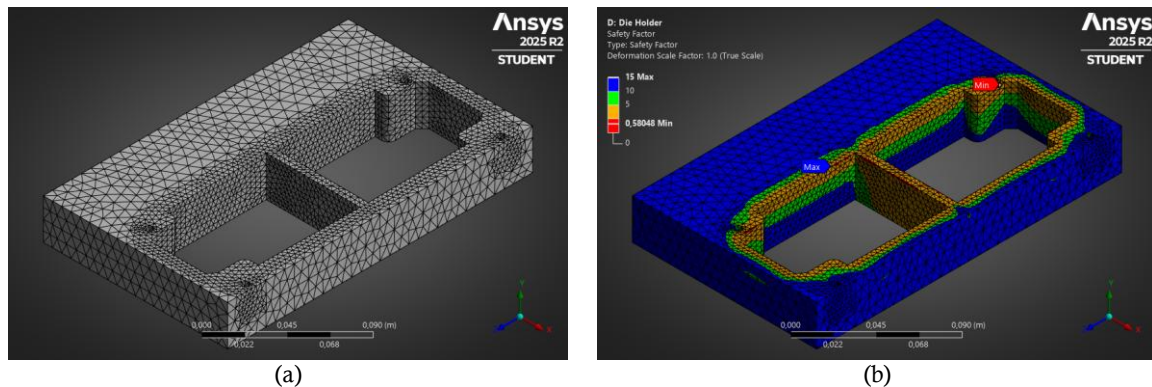


Figure 6. Die Blank Analysis. (a) Meshing of Die Blank. (b) Safety Factor of Die Blank

Consistent with this low stress level, the die blank achieved the highest minimum fatigue life of 69,215 cycles, occurring at the lateral blank edge where geometric asymmetry promotes slight stress concentration and an average fatigue life approaching 10^6 cycles, the upper boundary of the simulation output. Referencing the SKD 11 S-N dataset by (Akiniwa et al., 2006), a peak stress of 286 MPa resides well below the 500 MPa lower boundary of the characterized stress amplitude range, placing the die blank firmly within the high-cycle fatigue regime where cycle life is sustained and degradation is minimal. The average safety factor of 14.34 is the highest among all components, confirming superior structural reserve. The minimum safety factor of 0.58 at the sharp corners is consistent with known FEA stress singularity behavior at geometric discontinuities and does not represent actual material failure.

Punch Blank

The simulation results for the punch blank show a maximum deformation of 0.021 mm, with an average of 0.003 mm. This negligible deformation confirms that the component maintains dimensional integrity under operational cyclic loading. The von Mises stress analysis shows a maximum value of 661 MPa and an average of 16.6 MPa. This stress level is safe because it is still below the yield strength of the SKD 11 material under the applied pressing load.

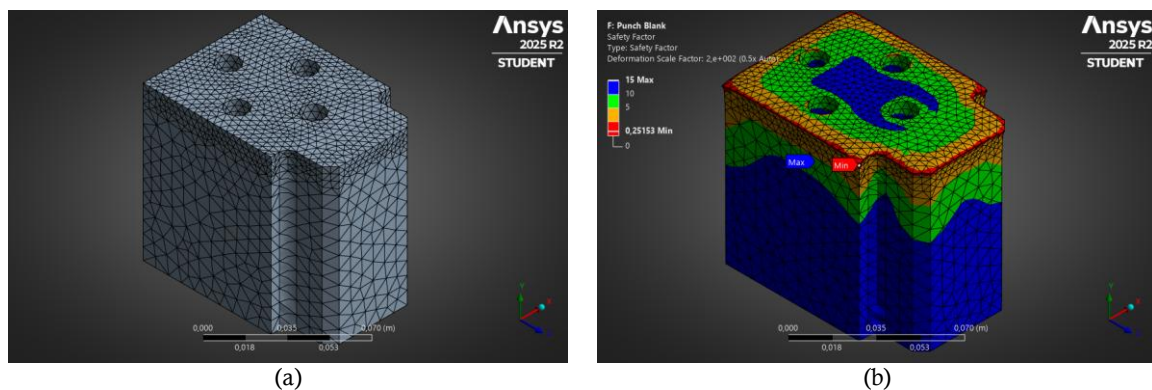


Figure 7. Punch Blank Analysis. (a) Meshing of Punch Blank. (b) Safety Factor of Punch Blank

Fatigue analysis revealed a minimum cycle life of 3,045 cycles at the blanking contact edge, the first contact point during the blanking stroke. This localized reduction in fatigue life is primarily attributable to the absence of a fillet radius at the punch blank cutting edge, which promotes stress concentration at the sharp corner geometry. This finding is consistent with established tool design principles reported by (Avadhani et al., 2017), who demonstrated that the introduction of even a small fillet radius (0.1–0.3 mm) at punch cutting edges can reduce local stress concentration by 15–25%, with a proportional improvement in predicted fatigue life. Based on the present results, the addition of a fillet at the cutting edge is recommended in future design iterations to extend the

punch blank service life. Periodic inspection is recommended after every 2,500–3,000 operational cycles. The average safety factor of 12.44 confirms overall structural integrity, while the minimum value of 0.25 at the sharp corner is consistent with known FEA stress singularity behavior at geometric discontinuities.

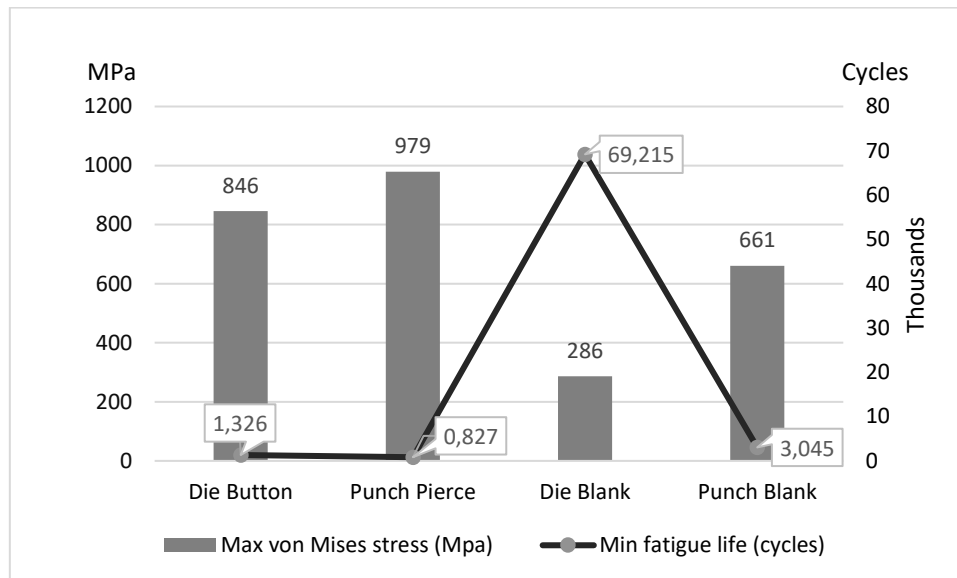


Figure 8. Combined bar and line plot of maximum Von Mises stress and minimum fatigue life for each component.

Figure 8 presents a comparative overview of the minimum fatigue life and maximum von Mises stress across all four components. A clear inverse relationship is observed: components with higher peak von Mises stress consistently exhibit shorter minimum fatigue life. The punch pierce, with the highest stress of 979 MPa, recorded the shortest fatigue life at 827 cycles, while the die blank, with the lowest stress of 286 MPa, achieved the longest minimum life at 69,215 cycles. This trend is physically consistent with the S-N characteristic of SKD 11 documented by (Akiniwa et al., 2006), wherein higher stress amplitudes correspond to accelerated fatigue degradation following the Basquin power-law relationship. The pronounced disparity between die and punch components underscores the importance of component-specific maintenance scheduling rather than a uniform tool replacement interval.

Table 4. Static Analysis of Die Button, Punch Pierce, Die Blank, and Punch Blank

No.	Part	Max Deformation (mm)	Max von Mises (MPa)	Min Fatigue Life (cycles)	Avg Safety Factor
1	Die Button	0.0075	846	1,326	5.30
2	Punch Pierce	0.0230	979	827	5.44
3	Die Blank	0.0120	286	69,215	14.34
4	Punch Blank	0.0210	661	3,045	12.44

The selection of SKD 11 as the tool material is further justified by its documented superior wear resistance and fatigue performance relative to conventional tool steels. (Masuda et al., 2024) demonstrated that heat-treated AISI D2, the international equivalent of SKD 11 exhibits a fatigue strength of approximately 900 MPa at 10^5 cycles, declining to approximately 650 MPa at 10^7 cycles under uncoated conditions. The peak stresses observed in the present simulation (286–979 MPa) span both the short-life and long-life regions of this range, consistent with the pronounced fatigue life disparity observed across the four components. These findings reinforce that material selection alone is insufficient to guarantee adequate fatigue life without concurrent attention to component geometry and stress distribution.

Overall, the simulation results confirm that die components exhibit substantially higher fatigue resistance than punch components, consistent with their more favorable load distribution geometry and lower stress utilization ratios. This fatigue life hierarchy: Die Blank (69,215 cycles minimum) > Punch Blank (3,045 cycles) > Die Button (1,326 cycles) > Punch Pierce (827 cycles) is qualitatively consistent with findings by (Shaheen et al., 2020), who reported analogous service life disparities between die and punch components in compound progressive die configurations.

Furthermore, the result aligns with the orientation-dependent S-N behavior of SKD 11 documented by (Akiniwa et al., 2006): components operating at lower stress amplitudes (die blank at 286 MPa, die button average at 49 MPa) reside in the high-cycle region of the curve, while those at higher stress amplitudes (punch pierce at 979 MPa) correspond to the steep short-life region. These comparative findings provide a physically consistent and empirically grounded basis for the maintenance scheduling recommendations presented in the conclusion.

CONCLUSION

This study applied a sequential FEA framework, a static structural analysis followed by stress-life fatigue assessment to evaluate the structural integrity and cyclic service life of four SKD 11 compound progressive die components for automotive hanging bracket stamping.

Static analysis confirmed that all components satisfy structural safety criteria, with peak von Mises stresses ranging from 286 MPa to 979 MPa. All below the SKD 11 yield strength of 1,540 MPa and maximum deformations within 0.023 mm, indicating negligible geometric distortion under operational loading.

Fatigue analysis revealed a pronounced service life disparity governed by stress utilization ratio. The die blank achieved the highest minimum cycle life (69,215 cycles; stress utilization 18.6%), while the punch pierce exhibited the most critical behavior (827 cycles minimum; stress utilization 63.6%). The punch blank recorded an intermediate minimum life of 3,045 cycles, limited by stress concentration at the fillet-absent cutting edge. These results are physically consistent with the orientation-dependent S-N characteristic of SKD 11 documented by (Akiniwa et al., 2006).

From a practical standpoint, punch components particularly the punch pierce and punch blank require periodic inspection after every 500–800 and 2,500–3,000 operational cycles, respectively, while die components may operate under extended service intervals. The addition of a fillet radius at the punch blank cutting edge is recommended in future design iterations to improve fatigue life. This study further demonstrates that integrated static-fatigue FEA constitutes an effective and resource-efficient methodology for die design evaluation in manufacturing education environments where full-scale press trials are unavailable.

This study has several limitations that should be acknowledged. First, the absence of full-scale press testing means that fatigue life predictions are based solely on indirect validation against published S-N data, without direct experimental confirmation. Second, inter-component contact friction and dynamic press inertia were excluded from the simulation model, which may result in conservative stress underestimation under actual operating conditions. Third, surface coating effects, known to significantly influence fatigue life in tool steel applications (Masuda et al., 2024) were not considered.

Future research should address these limitations by: (1) conducting experimental fatigue testing on SKD 11 specimens under loading conditions representative of the stamping process; (2) incorporating contact mechanics and dynamic effects in the FEA model; (3) investigating the effect of surface coating (e.g., TiN, TiCN) on the fatigue life of punch components, particularly the punch pierce; and (4) extending the analysis to include wear prediction alongside fatigue life estimation for a more comprehensive tool life assessment.

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