*International Journal of Advanced Technology and Engineering Exploration, Vol 12(125)*
*ISSN (Print): 2394-5443 ISSN (Online): 2394-7454*
*http://dx.doi.org/10.19101/IJATEE.2024.111100413*

**Research Article**

# Fatigue life enhancement of air disc brake caliper using finite element analysis

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Received: 31-March-2024; Revised: 23-March-2025; Accepted: 26-March-2025


## Abstract
***Fatigue failure, often caused by repeated loading and unloading, is a critical concern in the operation of vehicles and machinery, often leading to more severe consequences than other types of failures. This study aims to enhance the fatigue life of an air disc brake (ADB) caliper through simulation using the finite element method (FEM). Software tools such as ANSYS and an Excel-based fatigue analysis tool were employed to calculate fatigue life and maximum principal stress. Key parameters evaluated include the factor of safety and maximum principal stress, which were used to identify weak regions in the caliper geometry. The results indicate that regions high stress (HS2), HS3, and HS4 exhibit structural weaknesses. Specifically, the factor of safety at HS4 in the baseline design, as determined by finite element analysis (FEA), was 0.91—lower than the 0.95 observed for the present caliper. This reduced factor of safety indicates a higher risk of fatigue failure in the baseline design. Following design modifications to regions HS1, HS2, HS3, and HS4, the factor of safety improved to 0.98, demonstrating enhanced performance in these critical areas. Comparative analysis between the baseline and modified designs shows a significant improvement in the caliper's fatigue life. Geometric refinements, especially those that improved stress flow lines, played a key role in extending fatigue life. Changes in the caliper's topology directly influenced stress distribution and fatigue performance in critical regions.***

## Keywords
***Finite element analysis, Air disc brake caliper, Fatigue life, Stress distribution, Factor of safety, Design optimization.***

## 1.Introduction
Braking systems play a critical role in vehicle safety, particularly in preventing accidents and minimizing damage during emergencies. For effective accident prevention, brake components must be designed to ensure maximum braking power, shorter stopping distances, and enhanced safety, all while maintaining optimum brake efficiency. A reliable and robust brake system is essential for improving overall vehicle safety. Automotive components, including brakes, are regularly subjected to external forces, lateral deflection, and torsional stresses. These stresses can induce significant strain in the components, making properties like modulus of elasticity and rigidity crucial to performance. Since these components—particularly in heavy-duty vehicles—are frequently subjected to cyclic loading, they are prone to fatigue damage over time [1–3]. Fatigue failure occurs due to repeated loading and unloading, a common scenario in vehicles and machines under regular operation. In fact, fatigue-related failures are often more severe and unpredictable compared to other failure types, necessitating precise analysis and design improvements.

In recent years, significant efforts have been made to calculate fatigue life and perform stress analysis using both experimental methods and computer aided engineering (CAE) simulations. Among various braking systems, air disc brakes ($ADB_S$) are primarily used in heavy vehicles such as trucks and buses. These systems consist of several components, including the carrier, brake pads, bridge, brake lever, and bearings. One of the critical components is the caliper, typically made of ductile cast iron, chosen for its excellent fatigue resistance and mechanical durability for repeated stress. Despite advancements in brake systems, the caliper in $ADB_S$ remains a vulnerable component due to the cyclic nature of the loading applied during brake operation. The cyclic loading, caused by the continuous application and release of the brake pedal, accelerates fatigue damage in the caliper, potentially leading to failure. The

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challenge, as noted in several studies [4, 5], lies in accurately predicting fatigue life and improving caliper durability through design optimization while optimizing weight and material performance.

The motivation for this study stems from the need to enhance the fatigue life of the brake caliper to ensure greater reliability in heavy vehicles. Previous research has focused on either experimental fatigue testing or CAE simulations, but there is a need to integrate these approaches for more accurate predictions and design improvements. The main objective of this study is to extend the service life of the caliper used in $ADB_S$ through design optimization using finite element method (FEM) simulations. Tools such as ANSYS 19.2 and Excel-based fatigue analysis software were employed to calculate service life and maximum principal stress. The study investigates key parameters, including the factor of safety and maximum principal stress, to identify the weakest regions in the caliper's geometry. Additionally, a comparative analysis is conducted between baseline and modified caliper designs to validate the simulation results and highlight the impact of design modifications on fatigue performance.

The contributions of this paper include:
1. A detailed investigation of the fatigue failure in brake calipers using FEM based simulations.
2. Design modifications to critical regions of the caliper to improve fatigue resistance.
3. A comparative study of baseline and modified designs to validate the improvements.
4. Insights on the effect of stress flow lines and geometry on fatigue life.

The remainder of this paper is organized as follows: Section 2 reviews the relevant literature on fatigue analysis and brake system design. Section 3 describes the methods and materials used for FEM simulation and fatigue life analysis. Section 4 presents the key results. The section 5 presents the discussion part and section 6 concludes the paper, highlighting the improvements achieved and potential future work.

## 2.Literature review

The literature reveals the impact of design modifications and material properties on brake caliper performance. Previous studies focus on stress analysis, thermal performance, and modular design but do not provide specific insights into fatigue life improvements through FEM simulations.

Yadav et al. [6] conducted a study which deals with the design, analysis, and manufacturing of a brake disc of a two-wheeler whose strength-to-weight ratio is high. The thermal and structural analysis of the modified brake disc is performed using finite element analysis (FEA) and has been compared with the brake disc made of stainless steel to validate the results, and it has been found that the composite brake disc gives better results than the steel brake disc.

Mirajkar et al. [7] studied the performance improvement of an original equipment manufacturer (OEM) brake caliper through design modifications. The primary objective was to enhance braking efficiency by altering the caliper's geometric design. The simplified design of the caliper enables easy assembly and disassembly of the component. It also helps to improve the braking efficiency. Wagh [8] concentrated on the analysis and optimization of caliper. The structural as well as thermal analysis of conventional caliper was made and new modular design concept was introduced and again structural as well as thermal analysis was performed to improve the performance. This aims on how design concept of modular caliper affects the overall performance. Ingale et al. [9], studied a conceptual design of a brake caliper for an all-terrain vehicle (ATV), primarily focusing on reducing the size and weight without compromising its strength, stiffness and low piston drag. They attempted to design a floating brake caliper considering various parameters and analysing it for its performance at normal conditions neglecting the thermal effects.

Kulkarni et al. [10] studied the impact of design factors of disc brake rotor on the braking performance. This study aimed to analyze the structural stability and cooling characteristics of brake rotors with varying geometries. It was observed that disc brake rotors featuring different geometric patterns—such as holes, slots, or grooves—as well as various vented designs between the discs, contributed to weight reduction and enhanced cooling efficiency during braking. These variations also have impact on its lifespan. Maijer et al. [11] developed process model of an A356 brake caliper for fatigue life prediction. By using a thoroughly developed process models the fatigue performance of a preliminary brake caliper is predicted. The analysis technique was used to evaluate the design of aluminium alloy brake caliper. Mathematical models were described for the processing steps of casting, heat treatment, machining and in service performance. Patil and

Malagi [12], studied the casting simulation of ADB parameter mapping through the amalgamation of casting simulation and FEA results. FEA was performed but it was not enough because it does not take into account the variable material quality and casting defects so casting simulation is also performed here casting simulations may investigate the impacts of several casting factors like melt temperature, velocity, design of the gate and runner system, riser design, mild design. This simulation of casting can be utilized to reduce the casing defects. Fatchurrohman and Chia [13] studied the performance of hybrid nano-micro reinforced mg metal matrix composites brake caliper by using simulation approach. Most of the automobiles are having a grey cast iron caliper but here magnesium metal matrix composite is used for the weight reduction. In this paper hybrid Mg- metal matrix composite caliper performance was compared with the conventional grey cast iron caliper; all this was done by the FEA ANSYS software. Pishdad [14], studied the advanced design of brake calipers. This research project is concerned with assessment as well as the optimization of the caliper and the caliper is of racing car. Caliper design is optimized to decrease the mass and increase the stiffness of the caliper. Asymmetric design is gained which is result of topology optimization.

Saravanan [15] studied the improvement of fatigue strength of ductile iron components meant for automotive applications through foundry processes. Ductile iron is mostly used in automotive applications due to its good fatigue resistance. This study shows how the various foundry processes can improve this fatigue strength of components made up of ductile iron. Plantan and Lantz [16] developed air disk brake caliper pre-stressing method and pre-stressed caliper apparatus. This concentrates on the mainly the Pre-stressing ADB caliper which is made up of cast iron for this required pre-stressing equipment is used, hence to produce the compressive residual stress in the caliper. Which causes the cast iron material to deform plastically, but not enough to cause the failure like cracking? Pre-stressing causes to create a residual compressive stress these stresses in the cast iron caliper lower the mid-point of the stress range the plastically-deformed region of the caliper and thereby lowers the peak stress seen in this region, increasing fatigue life. Jorgensen [17] developed the design of nature-inspired components with tailored geometry for high mechanical performance., The purpose of this thesis is to study the effect of adding tailored geometries in specimens to strengthen them and increase their fatigue resistance. The results show that adding the specific geometries plays an important role increasing the fatigue life by redistributing the stress and better utilizing the lesser-stressed parts of the specimens.

Taylor et al. [18] studied the notch fatigue behaviour in cast iron using a fracture mechanics approach. In this study the notch sensitivity is checked with the help of a testing the notched specimens for this the fractures mechanics approach are used here. In the fracture mechanics approach here the notch is considered as a crack and the initiated crack length is also considered at the notch root analytical approach. Collini and Pirondi [19] analysed the fatigue strength of ferritic ductile iron castings. The fatigue crack growth is simulated by FEA using the virtual crack closure technique. It is concluded that crack takes more time to initiate rather than to propagate, and the local defect morphology plays a crucial role in it.

Park et al. [20] performance the experimental study to improve the thermodynamic performance of a ventilated brake disc by securing structural stability and verified through bench experiments. They have proposed that the heat dissipation performance and driving safety can be improved by using replaced brake discs.

Stender et al. [21] introduced new method to derive instability growth rates from vibration time-series data. Results show that the proposed approach is robust and applicable in a highly automated fashion to large data sets. The data analysis reveals that the final acoustical squeal intensity develops independently of the instability growth rates. Xiang et al. [22] conducted friction braking tests of blocks with different heights. The results show that blocks with greater height exhibit greater eccentric wear, resulting in stronger friction-induced vibration and noise (FIVN). The block inclined state causes contact pressure to concentrate at the leading edge, which is the main cause of eccentric wear.

Kim et al. [23] experimentally investigated the relationship between thermo–elastic–plastic deformation during braking and the residual stress in brake discs. The present experimental study provides insights into factors that affect thermo–elastic–plastic distortion in brake discs as well as methods that can reduce the distortion of brake discs in the design stage. Zeng et al. [24] studied the brakes used for railway vehicles. The weak points of the structure were identified by taking random response analysis, and the corresponding power spectral density spectrums of stress were obtained. The result shows that, the Dirlik's model and Steinberg's model can provide more exact results and be suitable for vibration fatigue failure prediction of brake caliper. Hui et al. [25] studied performances of C/C–Sic brake discs under extreme operating conditions, including cyclic high- temperature and emergency braking of heavy-duty tracked vehicles. It is observed that when the friction film was completely covered, the coefficient of friction (COF) decreased and tended to be stable and consistent, suggesting that C/C–Sic brake discs can provide a sufficient and stable braking force at different speeds.

Zhu et al. [26] conducted a field test and a FEA of disc brake squeal. The field test and FEA show the disc brake finite element prediction model can correctly predict the frequency of brake noise. Finite element results show that brake pressure is the most critical factor causing brake noise. The increase in braking pressure will directly lead to the disc brake system to become more unstable. Veeman et al. [27] reviewed the various brake composite materials' characteristics and their mechanical and tribological performance. The tribological study includes the various terminologies used in friction and wear using pin-on-disc, dynamometer and field studies. Wu et al. [28] investigated the effects of different damping components on the interfacial dynamic response and tribological properties of a braking system. The results indicate that the vibration response, noise characteristic, and interface contact state are affected by the deformation behavior and damping properties of the damping components. It is found that the rubber component plays a significant role in improving the interface contact state of the braking system. While the Mn-Cu damping component has an excellent performance in FIVN suppression.

Tyflopoulos et al. [29], explored the benefits and limitations of topology optimization using the housings of a front and a rear brake caliper as a case study. It is observed that the weight of the optimized calipers reduced by 41.6% compared to commercial calipers. Nejad and Liu [30] proposed a new method for evaluating the fatigue crack closure load under the ixed-mode loading conditions. The results of the new method for fatigue life are compared with experimental tests and numerical method that results in acceptable achievements. Radu et al. [31] determined the fatigue usage factors for Findley and Matake stress-based criteria for mine hoist drum brake system subjected to cyclic maneuver braking. Tonolini et al. [32] proposed two laser-cladding coatings to improve the wear performances of cast iron discs brake. The wear mechanism of the different tested conditions was investigated by means of Scanning Electron Microscope analyses on the sample worn surfaces, and the results were related to the friction coefficient and to the wear rate. Laser-cladding was found to be effective in reducing the consumption of disc brake materials.

The literature reveals that FEA study of brake components has been routinely investigated. However, study on brake caliper for fatigue life cycle improvements has not been undertaken. Hence, the present numerical study is performed on the caliper of ADB for fatigue life cycle improvement by using FEA technique. The purpose of this study is to compare the baseline and modified design of brake caliper of ADB in terms of the factor of safety and fatigue life cycle. This study will analyze how enhancements in caliper geometry improve the fatigue life cycle.

## 3.Methods

The main aim of the study is to enhance the life of caliper of ADB. Hence, to improve fatigue life, main changes are done in the design of the caliper. The improved fatigue life of the caliper is compared with the present caliper design and the results obtained using commercial ANSYS software. The fatigue life of the component is affected by many factors, geometry is one of them. Geometrical parameters have a significant impact on the fatigue strength of the components. This mainly happens because of the changes in stress flow lines. The more crowded the flow lines responsible for force distribution, the greater the stress produced. When these flow lines are smoother or less crowded, lesser the stress is produced. Hence, this principal is used for the geometrical modifications. In this study, the average brake force is calculated, as it is not feasible to determine the exact force acting on the caliper over its lifespan due to variations in applied braking force, which is not constant at all times.

### 3.1Design analysis procedure

Computer aided design (CAD) modelling is performed in the Creo software. Firstly, the CAD model of the baseline caliper model was prepared in the Creo software. Then this CAD model was analysed in the ANSYS workbench to find out the critical regions in the caliper where there is high stress (HS) concentration. Then the critical region in the caliper is defined according to the factor of safety. The geometry modifications were performed on that critical region according to the principal of the stress flow. Again, the analysis of this modified caliper was performed on the ANSYS workbench [33–35]. Structural analysis was performed on the ANSYS workbench and fatigue life [33–35] was calculated by using excel based fatigue tool.

The fatigue life of both the baseline and modified calipers was compared to evaluate improvements. The methodology followed in this study includes the following steps:

1. CAD modeling of the ADB caliper in Creo.
2. Structural analysis of the baseline caliper model using ANSYS Workbench.
3. Determination of maximum principal stress.
4. Estimation of fatigue life using an Excel-based fatigue analysis tool.
5. Implementation of geometric design modifications to the caliper in Creo.
6. Structural analysis of the modified caliper in ANSYS Workbench.
7. Determination of maximum principal stress for the modified model.
8. Estimation of fatigue life for the modified design using the Excel-based tool.
9. Comparison of fatigue life between the baseline and modified caliper designs.

### 3.2Finite element model

The ADB assembly primarily consists of several components, including the caliper, carrier, internal mechanism, brake pads, and disc. However, to reduce simulation cost and time, components other than the caliper were simplified by defeaturing the CAD model. Computational analysis is conducted under extreme condition of brake pads that is brake pads are considered at the extreme worn-out condition. For the numerical analysis, brake pads with back plate, disc, piston, lever, bridge, cylinder bearings are considered.

Meshing is performed to discretize the model into the number of elements to get the proper solution and proper distribution of the brake force over the whole assembly. The mesh size of 12 mm is given for all the components of ADB. A smaller mesh size increases the simulation run time; therefore, a mesh size of 12 mm was selected to reduce computational time while maintaining acceptable accuracy. Initially, the caliper was meshed with a 1 mm mesh size to identify high-stress locations. However, after evaluating runtime requirements and accuracy, a 12 mm mesh size was selected for the overall model. A finer mesh size of 1 mm was applied at high-stress locations by assigning a sphere of influence to these critical areas.

The above simplified assembly of ADB as shown in *Figure 2* is taken for the FEA. The present analysis is carried out on the total wear of the pad material. The thickness of the pad material is 2.1 mm. The caliper is de-featured by removing machined symbols to simplify the simulation. The components considered in the assembly include the disc, piston, bridge, brake caliper, and worn-out pads. The extreme condition of the ADB is considered for the analysis.

Brake force applied is 248 KN. The material selected for the caliper is spheroidal cast iron. Spheroidal cast iron was selected due to its excellent fatigue resistance, ductility, and ability to absorb stresses under cyclic loading conditions. These properties make it particularly suitable for components like caliper, which experience significant stress variations during braking cycles. The high tensile strength and ductility allow the material to endure repeated loading without fracturing, while the material's yield strength and modulus of elasticity ensure that it can withstand the applied stresses with minimal deformation.

The specific material properties are as follows:

- Yield strength: 352 MPa
- Tensile strength: 570 MPa
- Young's modulus: 167 GPa
- Poisson's ratio: 0.266

In the context of our fatigue analysis, these material properties were critical in determining the stress-life behavior of the caliper. The simulations accounted for these properties to accurately predict stress distribution and fatigue life across various regions of the caliper. The lower factor of safety observed in the baseline design was primarily due to localized stress concentrations in regions where the material's yield strength was exceeded. By modifying the design to reduce these stress concentrations, the factor of safety was increased to above 1, preventing fatigue failure.
The main aim of the simulation is to find out the maximum principal stresses at the various locations on the caliper and to modify the regions of the caliper which are causing the failure of the caliper. This helps increase the fatigue life and fatigue strength of the caliper. HS locations are locations with weaker geometry which need to be modified. The location of the brake force is centre of piston bolt pretension is applied to get to touch the real deformation of the

caliper according to the real action or the movement of the caliper in real world. The total brake force is 248 kN and Young's Modulus is 167 GPa. Applied the bolt pretension force, which is negative in magnitude to get the proper action of the caliper. The applied total force is divided into half to get the applied equally by each piston respectively on the pad and achieve the proper braking action. The model of caliper of ADB is shown in *Figure1*. The simplified ADB assembly is shown in *Figure 2*.

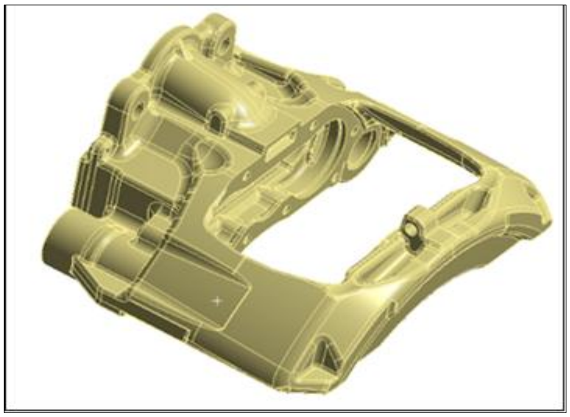

**Figure 1** ADB caliper

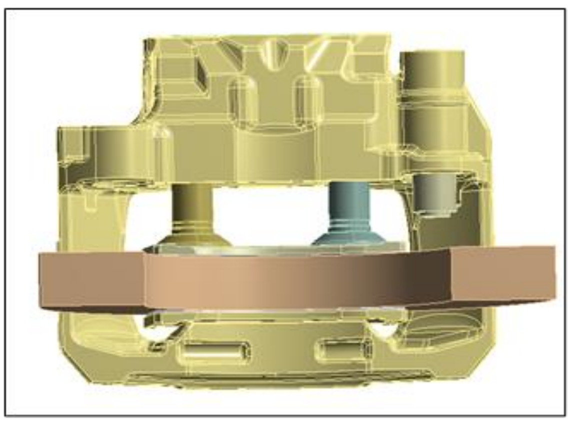

**Figure 2** Simplified ADB assembly

### 3.3Hardware and software requirements:

**Hardware Requirements:**

- Processor: A multi-core processor with high clock speed (e.g., Intel Core i7 or AMD Ryzen 7) is recommended for efficient FEM simulations.
- Random access memory (RAM): At least 16GB of RAM is necessary for handling large models and complex simulations. More RAM can significantly improve performance.
- Storage: A solid-state drive (SSD) is preferred for faster data access and loading times.
- Graphics Card: A dedicated graphics card with at least 4GB of VRAM can accelerate certain operations within the FEM software. However, it's not strictly necessary for most fatigue analysis tasks.

**Software requirements:**

Following software's are used for analysis in the present study

**FEA Software:** ANSYS is used as a FEA tool for structural analysis.

**Fatigue analysis tool:** Excel-based tool is used for data analysis and calculations related to fatigue life.

**CAD Software:** Creo software is used to create and modify the 3D model of the caliper.

**Pre- and post-processing tools:** For generating high-quality finite element meshes and visualizing the results and analysing the stress distributions ANSYS software is used.

## 4.Results

In this section, the simulation results under different conditions are presented and discussed.

### 4.1Simulation results for baseline model

The baseline model was simulated to identify regions requiring modification to improve fatigue life, and the results are presented in *Figure 3*.

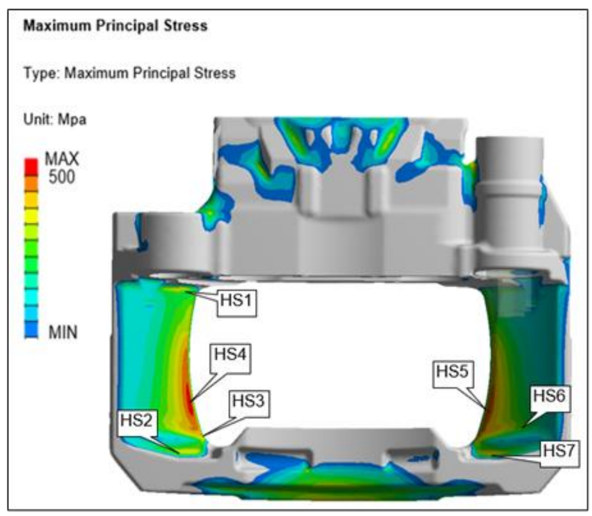


**Figure 3** HS at the failure locations

**Fatigue life measurement:**

Fatigue analysis methods are generally categorized into two main approaches: the stress-life (S–N) approach and the strain-life (ε–N) approach. The S–N approach is suitable for high-cycle fatigue (HCF) conditions, where materials experience a large number of cycles under relatively low stress levels. It is based on the relationship between the stress amplitude (S) and the number of cycles to failure (N). The ε–N approach, or strain-life method, is more appropriate for low-cycle fatigue, where components are subjected to higher stress or strain levels, often causing plastic deformation. This method considers the relationship between strain amplitude (ε) and the number of cycles to failure (N).

Fatigue life calculation is performed by using stress life approach; this is because caliper is tested for high number of cycles. In HCF approach plastic deformation is less. Number of cycles to be tested mainly depends upon the customer requirements.

**Stress-life approach:**
If the stress is below $\sigma_e$ (the endurance limit or fatigue limit), the component has effectively infinite life. Computational results using the stress–life approach is only informative when the degree of plastic straining is negligible, as HCF, where plastic strains are relatively small. The stress and fatigue life curve are shown in *Figure 4*. This figure shows the variation of stress amplitude ($\sigma_a$) with cycles to failure ($N_f$). It shows that the number of cycles to failure increases with lower stress amplitude.

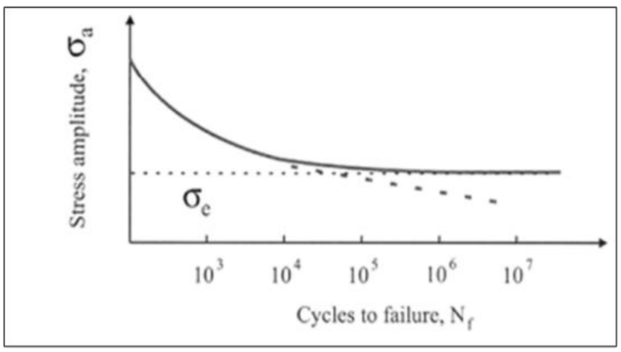


**Figure 4** Stress and fatigue life curve

## 4.2Fatigue life estimation using an excel-based fatigue analysis tool

Input for excel based fatigue tool

1. Relative stress gradient:

The relative stress gradient can be calculated in two ways. The first method involves using ANSYS APDL to directly compute the stress gradient. The second method involves calculating the stress gradient using Equation 1, with the required values obtained from simulation results in ANSYS Workbench.

$$x' = \frac{1}{\sigma max} \times \frac{d\sigma}{dx} \qquad (1)$$

Where,

$x'$=Relative stress gradient
$\sigma max$=Maximum principal stress
$d\sigma$= Difference in stress at that location between two nodes
dx= Distance between two nodes

2. Plasticity Factor ($K_p$)

The plasticity factor given in Equation 2 is defined as the ratio of the load at the elastoplastic stage to the load at the elastic limit, particularly at the critical cross-section where the material first reaches its yield strength. This calculation is carried out in two steps. In the first step, the structure is analyzed in the elastic region, and the load at which the material reaches its yield strength is determined—this is referred to as the elastic limit load. In the second step, a nonlinear analysis is performed to determine the plastic limit load, which is the load value at the final converged substep in the elastoplastic region. The plasticity factor is then obtained as the quotient of the plastic limit load to the elastic limit load. This factor helps in assessing the reserve strength capacity of a structure beyond the elastic range.

$$K_p = \frac{Lp}{Lf} = \frac{\text{Load for reaching Yield stress over the whole cross section}}{\text{Load for first time reaching Yield stress at geometry}} \qquad (2)$$

3. Maximum Principal stress- The maximum principal stress is used as the basis for the fatigue criterion in this study, as the principal stress theory is appropriate for brittle materials. In contrast, von Mises stress theory is typically used for ductile materials. The maximum principal stress values were obtained from ANSYS Workbench at the identified high-stress locations.
4. Minimum Principal stress- When the brake is applied, stress is generated in the caliper due to the braking force. However, when the brake is released, some residual stresses may still remain in the structure. For the purpose of this analysis, these residual stresses are assumed to be negligible and are considered as zero. Therefore, the minimum principal stress is taken as zero in the fatigue analysis.
5. Number of cycles- Total number of cycles required for caliper fatigue testing.
6. Material properties- Young’s modulus, tensile strength, and yield strength of the selected material

**Output from the Excel-based fatigue analysis tool**
The primary parameters obtained from the Excel-based fatigue analysis tool are the factor of safety and the fatigue life (in terms of load cycles). These

parameters are critical in identifying the regions of the caliper that require design modifications. The fatigue life results for the baseline model are presented in *Table 1*.

The S–N curves for the high-stress regions HS2, HS3, and HS4 of the baseline model are shown in *Figures 5*, *6*, and *7*, respectively. The simulation results indicate that these regions require design modifications, as they fail under the applied braking force and do not meet the defined load cycle criteria. These regions exhibit a lower factor of safety, suggesting weaker geometrical characteristics.

**Table 1** Fatigue life results for baseline model

| Locations | HS1 | HS2 | HS3 | HS4 | HS5 | HS6 | HS7 |
|---|---|---|---|---|---|---|---|
| Factor of safety | 1.08 | 0.98 | 1.00 | 0.91 | 1.03 | 1.07 | 1.04 |
| Total life cycles | 1,10,253 | 45,275 | 53,635 | 25,397 | 66,301 | 1,17,245 | 79,128 |

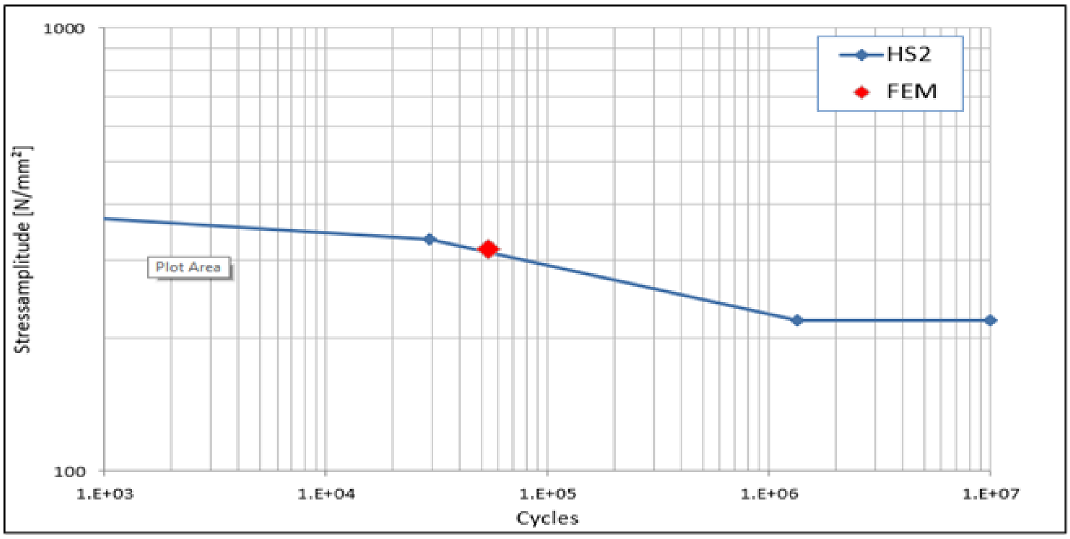


**Figure 5** S-N curve for region HS2 at baseline model

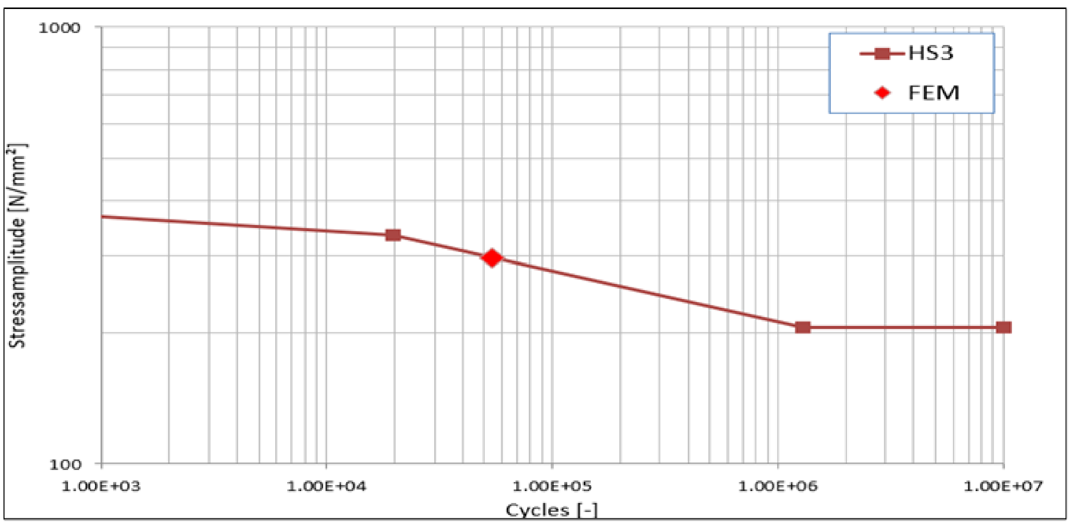


**Figure 6** S-N curve for region HS3 at baseline model

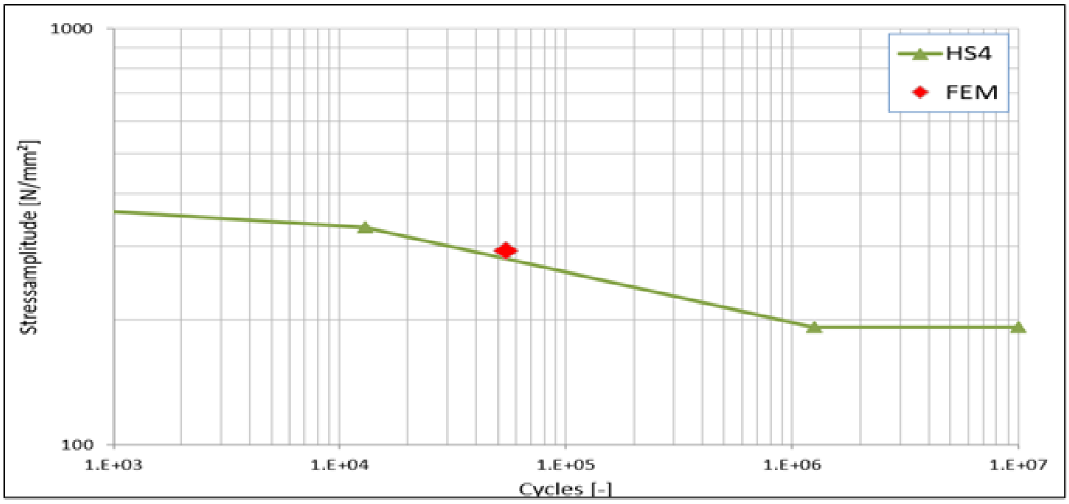


**Figure 7** S-N curve for region HS4 at baseline model

The S–N curves reveal the following:

- In HS2, the point lies above the endurance limit line, indicating a higher stress amplitude and a factor of safety less than 1, marking it as a critical area.
- In HS3, the point again falls below the acceptable threshold, with stress amplitude exceeding safe limits, suggesting that the factor of safety is also less than 1.
- HS4 shows the highest stress amplitude among the three regions, confirming it as the weakest section of the design.

From the S–N plots, it is evident that stress amplitude decreases as the number of cycles increases, which is consistent with typical fatigue behavior. Therefore, modifications to the geometry of HS2, HS3, and HS4 are necessary to improve fatigue life and enhance the overall fatigue strength of the caliper design.

### 4.3Topology improvement for design modifications

*Figure 8* shows the cross-sectional view of the critical regions HS2 and HS3 in the modified caliper model, highlighting the improved topology in these areas. Several iterations were carried out, considering various parameters that influence stress distribution in these regions. The geometry of these regions involves variable radii, and there is an interdependence between them—modifying one radius affects the others, which in turn influences the stress concentration at those locations. Therefore, a thorough and systematic evaluation was essential to achieve an optimal design. The detailed design of experiments (DOE) for the critical regions is given in *Table 2.*

Topology improvement involves the analysis of stress flow lines within the regions of interest. Various geometric features and cross-sectional areas significantly influence the behavior and direction of these flow lines. To achieve effective improvements, a detailed evaluation of the geometry is essential to ensure uniform stress distribution and minimize critical stress concentrations.

A (DOE) study was conducted to reduce stress and improve the factor of safety. Multiple design experiments were performed based on criteria affecting geometric changes, as outlined in *Table 2*. The modified model stress (MMS) was compared with the baseline model stress (BMS) to evaluate the effectiveness of each modification.

Through the DOE process, stress levels were successfully reduced, and favourable results were obtained. This comparative approach helped minimize the number of required iterations. Each subject design was continued only if it yielded positive results; otherwise, the next variation was considered. Critical regions HS2, HS3, and HS4 were geometrically modified, resulting in the desired factor of safety and improved fatigue life. The iterative process revealed a strong interdependence between the high-stress regions—modifying one region affected the stress distribution in others. Therefore, it was essential to evaluate all high-stress zones simultaneously during the design modification process. As a result, region HS1 was also modified to a significant extent to balance stress distribution and achieve the required performance improvements in HS2, HS3, and HS4.

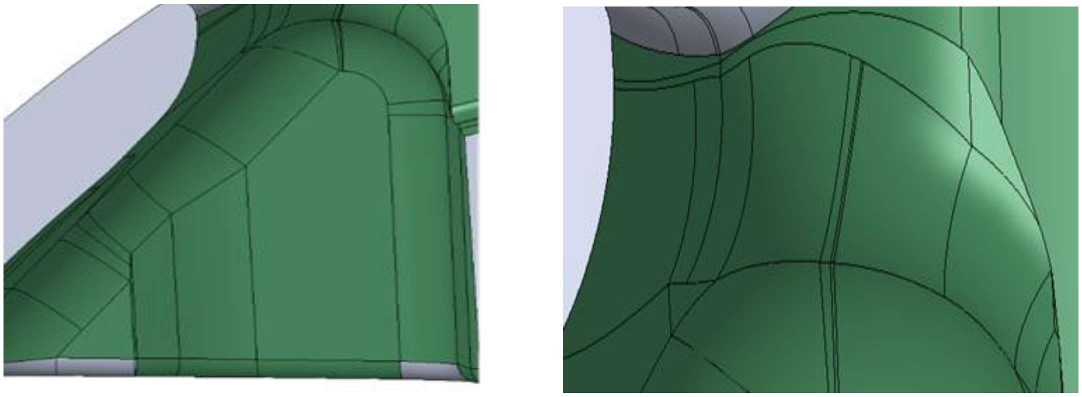

**Figure 8** Design Modifications at the region HS2, HS3

**Table 2** Design of experiments

| Parameter | Comparison of stresses with baseline model and Modified model | Remarks |
|---|---|---|
| Increase in the thickness | Check if MMS <= BMS | Increase in the thickness will result in increase in material Design modifications are performed on the core and changes will be modified on Caliper. (Designing and assembly of Caliper is performed on basis of actual sand-casting process of Caliper used for the manufacturing) Continue the subject for positive result from comparison otherwise checking to next subject |
| Decrease in thickness | Check if MMS <= BMS | Decrease in the thickness will result in decrease in material, this affects to increase a stress at that cross-section |
| Increase in radius | Check if MMS <= BMS | Increase in radius will result in decrease in material, this affects to decrease a stress at that region |
| Decrease in radius | Check if MMS <= BMS | Decrease in radius will result in increase in material, this affects to increase a stress at that region |
| Alteration in dimensions | Check if MMS <= BMS | Greater angles result in the smooth stress flow lines-decrease in stress value, whereas smaller angles result in tight flow lines-increase in stress value. |
| Increase the Notch area | Check if MMS <= BMS | Increase in notch area results in decrease in stress at the notch occupying area if compared with smaller notch area |
| Decrease the Notch area | Check if MMS <= BMS | Decrease in notch area results in increased stress at the tip of notch. |
| Stress concentrators Like fillet, sharp corners | Check if MMS <= BMS | Sharp corners result in crowded stress flow lines. Proper fillets result in smooth stress flow line. Evaluates the change in stress at its neighbour region. |
| Notch sensitivity | Check if MMS <= BMS | Notch results in increased stress but the dimensions of notch like angle and radius affects the stress area and its neighbouring area (Selection of notch dimensions depend on various iteration results and experience) |
| Gradual transitions in geometry | Check if MMS <= BMS | Alterations in edges result in reduced inflation at geometry which causes smooth transition |
| Accumulation of Extra Material | Check if MMS <= BMS | Removal of extra material due to manufacturing processes like welding, forming, Machining is required to reduce residual stresses but sometimes also depend on the customer requirement. |

### 4.4Results obtained from the Excel-based fatigue analysis tool

The results of fatigue life for modified model are shown in *Table 3.*

**Table 3** Fatigue life results for modified model

| Locations | HS1 | HS2 | HS3 | HS4 | HS5 | HS6 | HS7 |
|---|---|---|---|---|---|---|---|
| Factor of safety | 1.16 | 1.00 | 1.05 | 0.98 | 1.03 | 1.07 | 1.04 |
| Total life cycles | 1,97,902 | 53,853 | 81,756 | 47,549 | 66,301 | 1,17,245 | 79,128 |

### 4.5S-N curves for HS regions

The S–N curves for the high-stress regions HS2, HS3, and HS4 of the baseline model are shown in *Figures 9*, *10* and *11* respectively. In the *Figure 10*, the point representing HS3 lies below the S–N curve, indicating that the stress amplitude has decreased, and the number of cycles to failure has increased. This results in an improved fatigue life and a higher factor of safety. Similarly, for HS4 region as shown in *Figure 11*, the point lies on the S–N curve, and although it is not below the line, the stress amplitude is still lower compared to the baseline model. As a result, the fatigue life is enhanced, and a greater factor of safety is achieved.

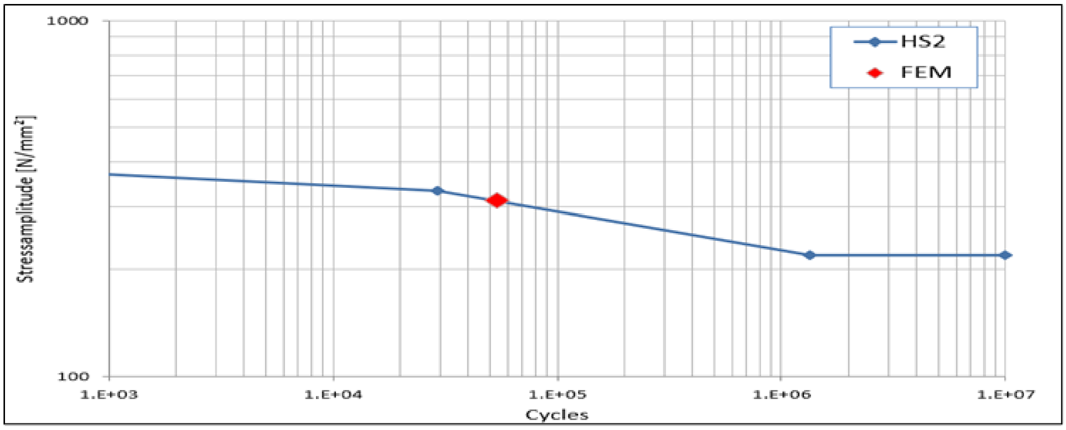


**Figure 9** S-N curve for region HS2 in the modified model

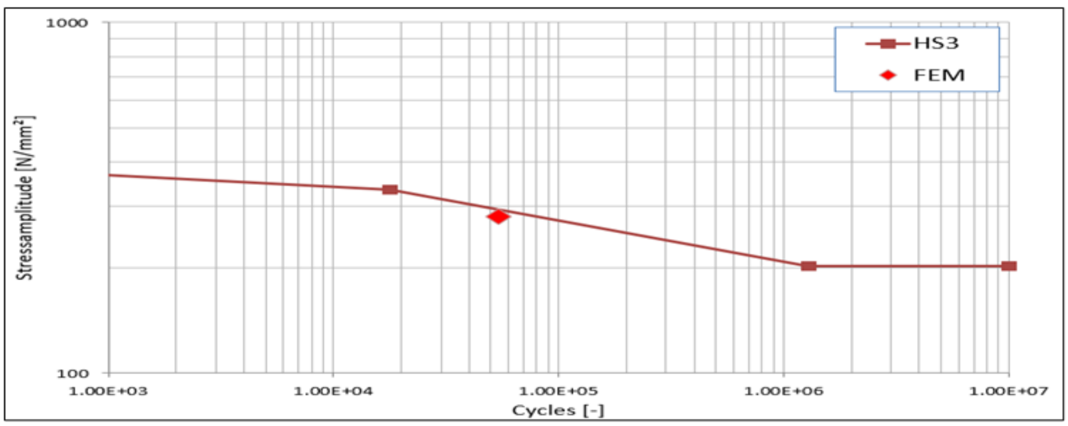


**Figure 10** S-N Curve for region HS3 in the modified model

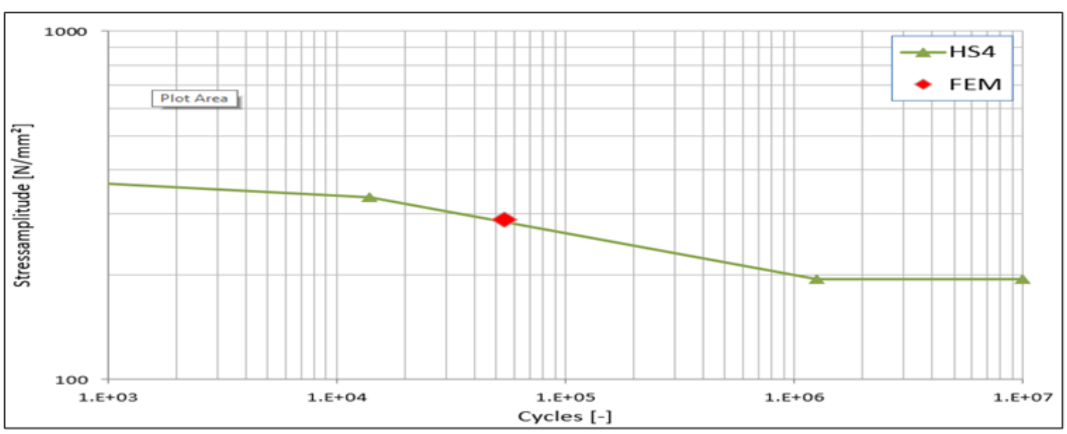


**Figure 11** S-N curve for region HS4 in the modified model

Design modifications were made to regions HS1, HS2, HS3, and HS4 to improve the fatigue life of the caliper. A comparison of the factor of safety and number of load cycles between the baseline and modified models is presented in *Table 4.* The primary improvements in the caliper's geometry were guided by an analysis of stress flow lines, which played a crucial role in identifying critical regions. The topology of the caliper significantly influences these stress flow lines, and any changes in geometry directly affect the stress distribution—ultimately impacting the fatigue performance of the structure.

**Table 4** Comparison of factor of safety for baseline model and modified model

| Regions | Factor of safety for baseline model | Factor of safety for modified model |
|---|---|---|
| HS1 | 1.08 | 1.16 |
| HS2 | 0.98 | 1.00 |
| HS3 | 1.00 | 1.05 |
| HS4 | 0.91 | 0.98 |

The analysis identified region HS4 as the primary location for crack initiation and fatigue failure in the caliper. The results were compared with a present caliper design that has demonstrated reliable performance in the field. The factor of safety at position HS4 for the present caliper, based on test results, is 0.95, whereas the baseline design, as obtained from FEA, shows a lower factor of safety of 0.91. This reduced value in the baseline model indicates a higher risk of failure due to weaker geometry, potentially resulting in a shorter fatigue life that may not meet application requirements.

To address this, design modifications were implemented, leading to an improved factor of safety of 0.98 at HS4 in the modified caliper design. This value is higher than both the baseline and the present caliper designs, indicating improved durability. Furthermore, the modified design also demonstrated improved fatigue performance in other critical regions, including HS1, HS2, and HS3, when compared to the present caliper. The fatigue life obtained from the analysis confirms a significant improvement in the modified caliper design relative to the baseline model.

## 5.Discussion

The study demonstrates that geometric modifications in the caliper significantly enhance fatigue life by reducing stress concentrations in critical regions. Design iterations focused on optimizing stress flow lines, which led to increased factors of safety in previously weak areas, particularly HS2, HS3, and HS4. The effectiveness of these improvements was validated through a comparative analysis of MMS and BMS, highlighting substantial reductions in stress amplitude and improved fatigue performance. A comparative analysis with present design, which has proven reliable in the field, was conducted. The factor of safety at HS4 in the present design was 0.95, whereas the baseline model had a lower value of 0.91. This indicated potential failure due to weaker geometry. After implementing the design modifications, the factor of safety improved to 0.98, surpassing both the baseline and the present caliper values. Additionally, the improved design demonstrated better fatigue performance in other regions (HS1, HS2, HS3), confirming the positive impact of stress flow optimization. The study further shows that changes made to one high-stress region influence others, indicating interdependence among geometrical features. Therefore, a comprehensive design approach was essential.

**Limitations**

Despite its valuable insights, the study has certain limitations which are as under:

- Material Considerations: The simulation assumes ideal material properties, without accounting for manufacturing inconsistencies, casting defects, or heat treatment variations.
- Loading Conditions: The FEA does not incorporate real-world conditions such as fluctuating loads, temperature changes, or environmental effects, which could influence fatigue behavior.
- Manufacturing Constraints: Practical challenges in implementing the design changes on a large scale were not considered. Machining tolerances, surface finish, and material variability may affect actual performance and stress distribution.
- Lack of Experimental Validation: The conclusions are based entirely on FEM simulations and Excel-based fatigue analysis. Direct experimental validation is necessary to confirm these findings.
- Limited Benchmarking: There is insufficient comparison with experimental fatigue tests or alternative simulation models, which would help strengthen the validity of the conclusions.

Given the above, while the design modifications show promise in simulations, real-world factors may influence the fatigue life of the caliper. In particular, critical regions with variable radii and interconnected geometry (e.g., HS2 and HS3) are sensitive to manufacturing deviations, which could alter local stress distributions and compromise fatigue strength. A complete list of abbreviations is listed in *Appendix I*.

## 6.Conclusion and future work

The study successfully achieves its goal of improving the fatigue life of a brake caliper used in ADBs through finite element simulation and design optimization. Using ANSYS and an Excel-based fatigue tool, several high-stress regions (HS1, HS2, HS3, HS4) were analyzed and modified to improve factor of safety and life cycles. Simulation results show that these regions had factor of safety values below 1 in the baseline design, indicating a high risk of fatigue failure. Modifications improved this, with HS4 identified as the primary site for crack initiation, and its factor of safety improved from 0.91 to 0.98. The results confirm that changes in topology and stress flow lines significantly enhance fatigue resistance. The modified caliper outperformed both the baseline and the present caliper across all key stress regions. To enhance the reliability and applicability of the findings, future studies should consider several key areas. First, there is a need to conduct real-world fatigue tests to validate the simulation outcomes and confirm the improvements observed in the modified caliper design. Additionally, sensitivity analysis should be performed to quantify the effects of material inconsistencies, surface defects, and manufacturing tolerances on fatigue performance. Material optimization is another important area, where alternative materials or surface coatings can be explored to further improve fatigue resistance. Future research should also include operational load studies that simulate dynamic braking conditions, accounting for thermal effects, impact loads, and varying environmental factors. Finally, the manufacturing feasibility of the proposed design modifications should be analyzed to ensure they can be practically implemented without significantly increasing production costs or complexity.

### Acknowledgment

The authors gratefully acknowledge the support provided by Kolhapur Institute of Technology's College of Engineering, Kolhapur (Maharashtra), India, and Knorr-Bremse Technology Centre, Pune, in carrying out this work.



### Conflicts of interest

The authors have no conflicts of interest to declare.

### Data availability

The data will be made available by the corresponding author upon reasonable request.

### Author's contribution statement

**Darshan Desai:** Data collection, conceptualization, writing - original draft, software and simulation, analysis and interpretation of results. **Udaysinh S. Bhapkar**: Conceptualization, investigation, data duration, writing-original draft, writing- review and editing draft manuscript preparation. **Kiran V. Chaudhary:** Study conception, design, data collection, supervision, manuscript reviewing and editing. **Swati Jadhav:** Study conception, design, data collection, supervision, manuscript reviewing and editing

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## Appendix I

| S. No. | Abbreviation | Description |
|---|---|---|
| 1 | ADB | Air Disc Brake |
| 2 | ATV | All-Terrain Vehicle |
| 3 | BMS | Baseline Model Stress |
| 4 | CAD | Computer Aided Design |
| 5 | CAE | Computer Aided Engineering |
| 6 | COF | Coefficient of Friction |
| 7 | DOE | Design of Experiments |
| 8 | FEA | Finite Element Analysis |
| 9 | FIVN | Friction-Induced Vibration and Noise |
| 10 | FEM | Finite Element Method |
| 11 | HCF | High-Cycle Fatigue |
| 12 | HS | High Stress |
| 13 | MMS | Modified Model Stress |
| 14 | OEM | Original Equipment Manufacturer. |
| 15 | RAM | Random Access Memory |
| 16 | SSD | Solid-State Drive |