Kang Yang - Finite Element Analysis Final Project – 2023

Designing A Lightweight Tandem Bicycle Frame

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1 Introduction

The design of a tandem bike frame was aided by the use of Finite Element Analysis (FEA) with ANSYS, which will be discussed in this study.

The following specifications for the frame were listed in the brief:

- it should be lightweight.
- its natural frequencies should be larger than 30Hz to avoid discomfort due to whole body vibrations.
- its effective life should be at least 10 years (equivalent to one million loading cycles)

Examined four frames, considering two design iterations and two materials, aluminium and titanium. Each frame was tested for static analysis, natural frequency and fatigue life, enabling a thorough performance comparison.

2 Methods

2.1 Initial Model

Some geometric specifications for the frame were specified in the brief, and these requirements constituted the guiding dimensions for the overall design. These were attained throughout both design rounds and are as follows:

- Overall length of the frame should be between 1.5 and 2 metres.
- The diameter of the wheels is 26 inches.
- The height of the seat joints from the ground is 800mm.

Its adherence to existing frame standards was a key was made, and then tubes were modelled according to the following design factor in order to make it usable in the actual world [1]. Using SolidWorks, a completely dimensioned drawing standards:

Feature	Dimension
Head tube angle	63°
Seat tube angle	70°
Outer tube diameter	34mm
Tube thickness	4mm
The crank shell & fork shell outer Diameter	50mm
The crank shell & fork shell thickness	10mm
The crank shell length	100mm
The fork shell length	200mm
Outer Chainstay Diameter	20mm
Chainstay Thickness	3mm
Fillet (Weld) radius	10mm

Table 1: Dimensional requirements for design

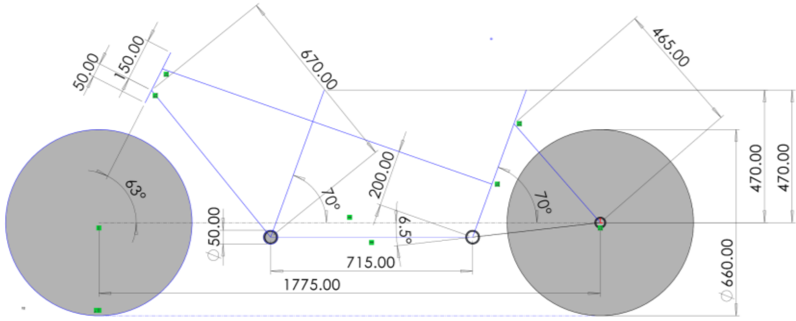

Figure 1: SolidWorks sketch of Frame Version 1

Figure 2: 3D SolidWorks model of Frame Version 1

2.2 Assumptions

The following assumptions were made for this study:

- The material was assumed to be homogeneous, isotropic, and flawless.
- No deterioration due to external factors or safety factor were considered.
- Riders' mass was set at 100 kg, and gravity at 9.81 m/s².
- Pedalling force was directly transferred to crankshafts, ignoring inertial force.
- Symmetry wasn't used, as force was applied to one side of the frame.
- Pedals were assumed to be synchronized.
- Riders' centres of mass were above the seat post profiles' middle.
- Only rider-applied weights were considered, ignoring external forces like wind resistance and bumps.
- Perfect interference assumed for pedalling force transfer to crank shells.
- Gravitational force was negligible in fatigue analysis.

Material	Density	Yield strength
Aluminium alloy	2.77e-06 kg/mm³	280.0 MPa
Ti-5Al-2.5Sn-0.5Fe	4.484e-06 kg/mm³	808.3 MPa

Table 2: Aluminium and titanium alloy material properties

2.3 Boundary Conditions

2.3.1 Fork Shell

The fork shell's inner surface boundary condition was 'Fixed'. Nodes on this surface have no degrees of freedom, preventing translation and rotation in all directions. This condition is used because the fork shell is attached to a ground-linked fork and does not bend because the fork goes through it.

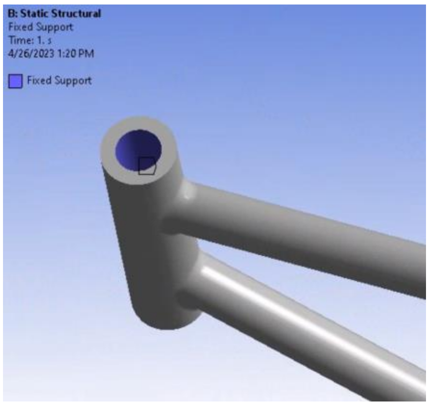

Figure 3: Fixed condition on fork shell

2.3.2 Rear fork mounts

In the analysis, the 'Remote Displacement' feature was applied to the rear wheel bearing's inner surface, allowing only one rotational degree of freedom. The cylindrical face can therefore only turn around its own axis. But nodes are still unable to spin. The reason for this is because the bearing utilised in the actual world would be allowed to spin, which would allow the frame to bend around that axis.

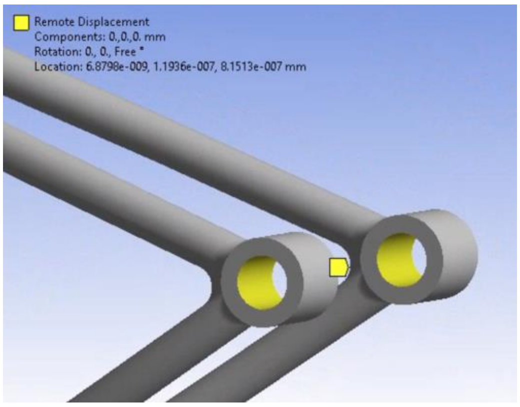

Figure 4: Remote displacement condition on rear dropouts.

2.4 Loading Conditions

2.4.1 Rider Mass

The frame could be subjected to one of two conceivable loads. As stated before, it was anticipated that each rider weighed 100 kg, hence the upper surfaces of both seat joints were subjected to a vertical force of 981N.

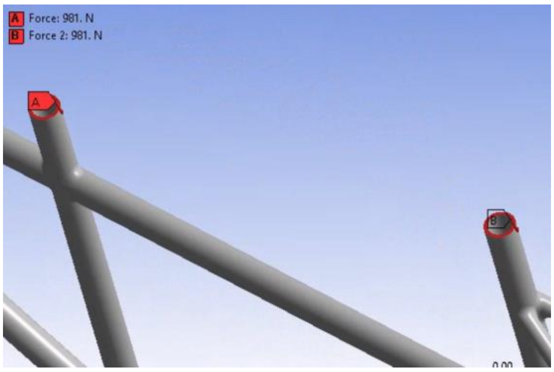

Figure 5: Weight of riders acting on seat posts

2.4.2 Pedalling Force

The force (remote load) generated by a rider pressing the pedals was depicted as a 700N load operating on the crank shell at a distance of 200mm in front of it and 100mm to its side (where the pedal would be on a bicycle).

The cyclists are exerting equal effort to move the bike ahead and are cycling at the same speed. This is a reasonable assumption to make because synchronising chains are typically connected on tandem bikes[2].

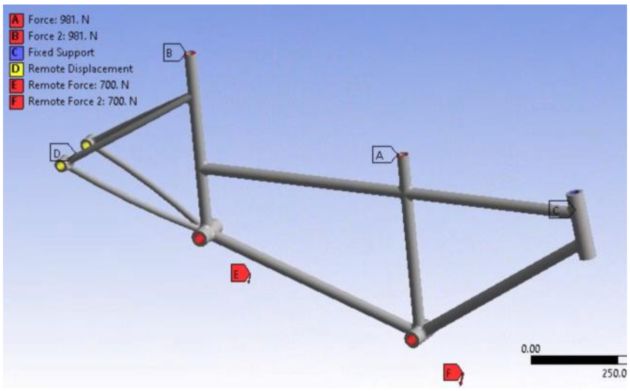

Figure 6: Pedalling force exerted by riders (tag E and F)

2.5 Static Simulation Properties

The static study takes into consideration both the rider's weight and maximal pedalling force, as the maximum stress value in any given cycle is the most useful value. As the maximal pedalling force only occurs on one side of the bicycle, symmetry is not utilised. The stress value we calculate for is the von Mises Stress, which considers three principal stresses and provides a single failure criterion:

$$\sigma_v > S_y$$

Where σ_v is von Mises Stress and S_y is the yield strength of the material at the point where it begins to deform plastically. Von Mises Stress is applied because the utilised materials are ductile metals.

2.6 Frequency Simulation Properties

In a frequency simulation for a bicycle frame, various loading scenarios can be evaluated for their effect on the structure's natural frequencies and mode shapes. Here are two potential outcomes:

1. Considering only the pedalling force. Because cyclists are not permanently attached to the seat, so their weight might not have the same impact as a fixed mass would.
2. Taking into account both pedalling force and rider weight.

The natural frequency must be greater than 30 Hz to prevent distress from whole-body vibrations. This criterion was evaluated using the default first 6 mode shapes.

2.7 Fatigue Simulation Properties

In a bicycle frame fatigue analysis, the loading conditions must account for both the static load of the cyclist's weight and the cyclic load of the pedalling force. The pedalling motion can be represented as cyclic loading, where the initial stress ratio ($R = \sigma_{min}/\sigma_{max}$) is assumed to be 0, with stress rising from 0 to its maximum value before falling back to 0 without ever becoming negative. However, the constant weight of the cyclists increases the mean stress value, resulting in an R value greater than 0.

Using a stress ratio of 0 in the fatigue analysis can be a conservative approach, especially when it is challenging to accurately represent the combined effects of the oscillating pedalling forces and constant cyclist weight in ANSYS. This conservative assumption can help ensure the safety and durability of the design, even if it is not the most precise representation of real-world loading conditions.

To simplify and strengthen the pedalling scenario, one million oscillation stresses were consolidated on the right side of the bicycle. Although this may not be the most accurate representation of reality, it is sufficient to identify design flaws and potential failure points.

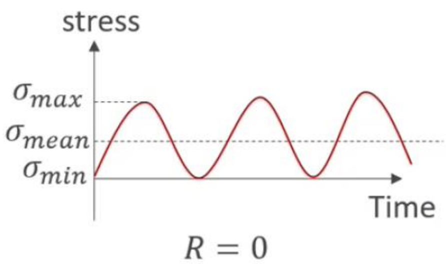

Figure 7: Characteristic of the zero based stress ratio

3 Results

3.1 Sanity Checks

Before investing a significant amount of computing resources in mesh refinements and simulations, it was essential to ensure that the model contained no glaring defects.

3.1.1 The Frame Mass

The tandem cycle frame's calculated mass of 6.1 kilograms falls within the acceptable range for an aluminium alloy frame[3]. This mass was determined using the volume method in ANSYS, which provides an accurate representation of the overall mass of the frame. Although tandem bike frame weight can vary based on design, material, and construction quality, a comparison with similar projects indicates that this weight is acceptable[4].

3.1.2 Meshing

ANSYS reported element aspect ratio, which were corrected during the mesh refinement phase. This ensured that tiny features had no distorted elements or misplaced edges. The process of mesh refinement and the resolution of these errors will be elaborated on in a later section.

3.1.3 Static Simulation

A static simulation was conducted with a mesh element size of 14mm and the aluminium material. The results were: the maximum stress was 69.4MPa, which is in the same order of magnitude as the yield stress (280.0MPa), which provides some reassurance that the design is not too weak and is worth pursuing; the location of the maximum stress was at joints, which makes sense given that discontinuities are stress raisers; and the maximum displacement was 2.4mm, which was ideal in real life situation.

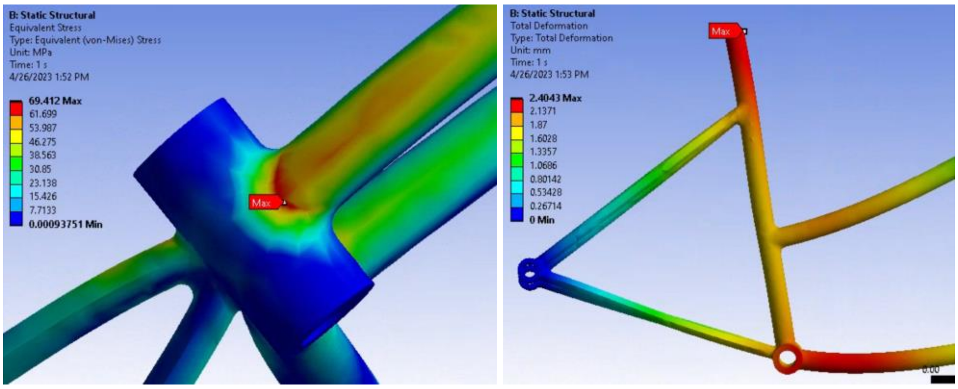

Figure 8: Location of the max stress (left) and true scale of max displacement (right)

3.2 Mesh Refinement

In order to determine the optimal size of the global mesh, a mesh size study was conducted. The mesh size was reduced from 16mm to 4mm in 2mm increments. The results showed that stress increased as the mesh size decreased. The maximum stress converged at an 8mm mesh size.

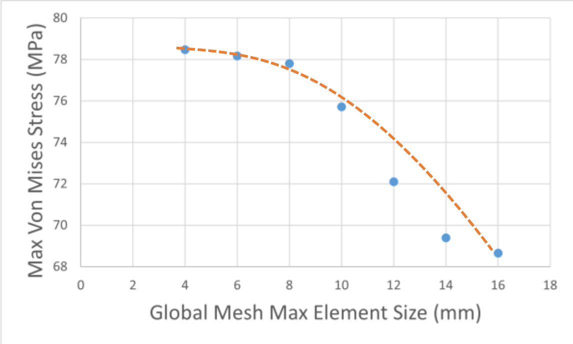

Figure 9: Maximum von Mises stress for different global mesh sizes

To improve the accuracy of the simulation results, mesh control was applied to areas of high stress in the filleted portions. It was observed that the location of maximum stress was consistently located at the filleted portions. By applying a 6 to 1mm mesh sizing to the part with the highest stress, the updated stress values were analysed. The maximum stress converged when using an 2mm controlled mesh size, indicating that this mesh size provides an appropriate balance between accuracy and computational efficiency.

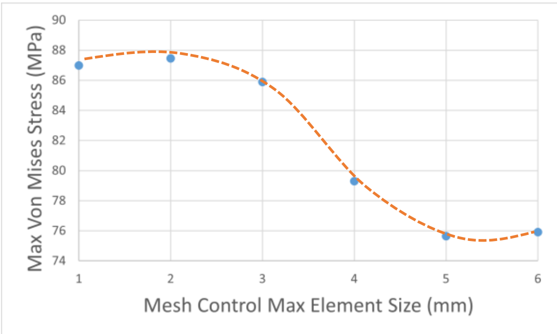

Figure 10: Graph of maximum von Mises stress while using mesh control

The maximum global size of the final mesh was 8mm, while the control mesh size was 2mm. This would yield extremely accurate results in the high-stress areas that are most crucial to the analysis, while reducing simulation durations.

3.3 Initial Design Results

3.3.1 Static Study

The maximum stress of the aluminium and titanium frame were 87.5 MPa.

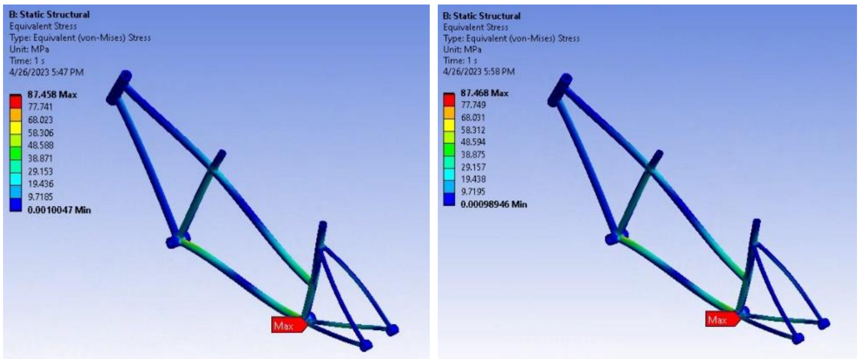

Figure 13: Static analysis of aluminium alloy frame (left) and titanium alloy (right)

3.3.2 Frequency Study

Considering only the pedalling force, the first natural frequency of the aluminium frame was 62.8Hz, while that of the titanium frame was 60.62Hz.

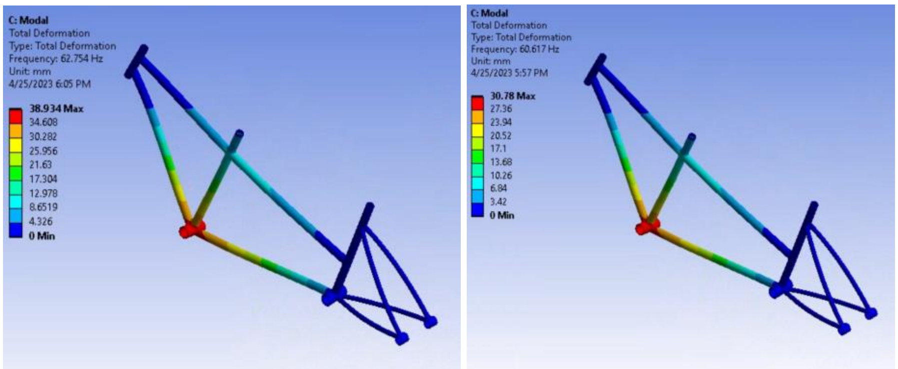

Figure 14: Frequency analysis (pedalling only) of aluminium (left) and titanium alloy frame (right)

Taking into account both pedalling force and rider weight, the first natural frequency of the aluminium frame was 63.4Hz, while that of the titanium frame was 61.1Hz.

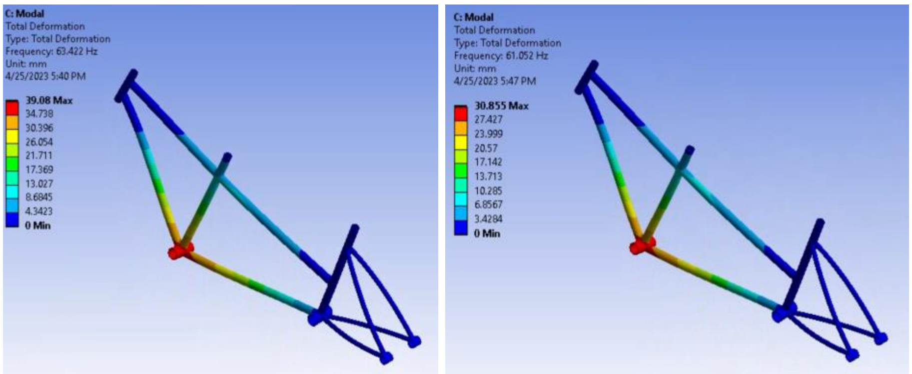

Figure 15: Frequency analysis of aluminium (left) and titanium alloy frame (right)

It can be concluded that the riders' weight has minimal impact on the natural frequency of the frame. This is due to the fact that the riders are not rigidly attached to the seat, allowing for some freedom of movement and reduced influence on the frame's frequency characteristics.

Titanium alloy has a lower natural frequency than aluminium alloy due to its higher density and lower stiffness. The combination of these factors results in a lower natural frequency for titanium compared to aluminium.

3.3.3 Fatigue Study

The applications aluminium and titanium alloy to the frame are all allowed it to readily withstand one million loading cycles, indicating that it can be used without significant risk of failure minimum for 10 years.

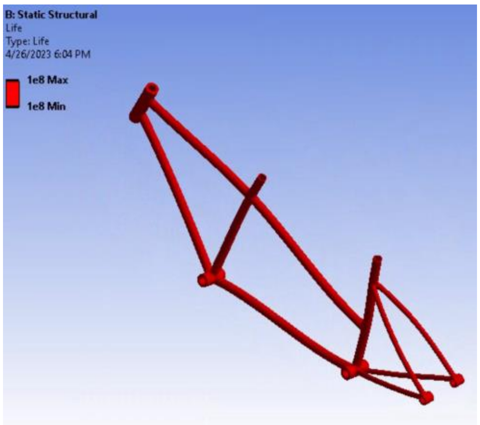

Figure 16: Fatigue analysis outcomes for both the aluminium the titanium alloy frame

In order to check the reliability of the fatigue life, the alternating stress has been calculated:

$$S = \frac{\sigma_{max} - \sigma_{min}}{2}$$

Where σ_{max} is the max stress and σ_{min} is the minimum stress.

The calculated alternating stress for the bike frame is 43.5 MPa. By comparing this value to the S-N curve (cyclic stress) for the Aluminium alloy provided by ANSYS, the simulation's reliability can be confirmed.

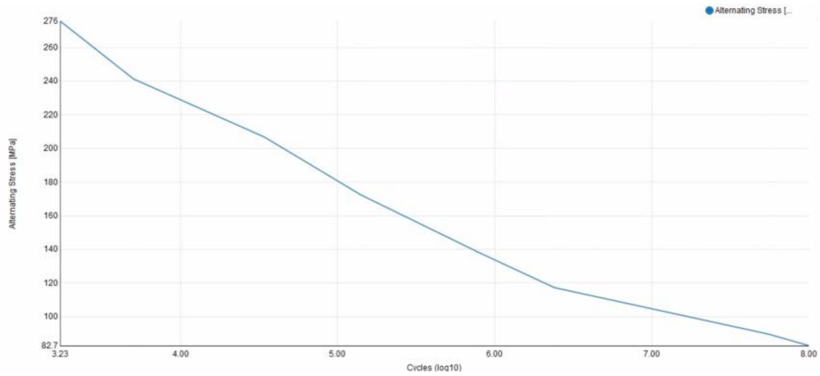

Figure 17: S-N curve of aluminium alloy

3.4 Design Changes

3.4.1 Support Tube

Adding a tube between the chainstays could potentially improve the performance of the bicycle frame by more uniformly distributing stress and minimising the maximum stress in the area under the second rider's crank shell (refers to figure 8).

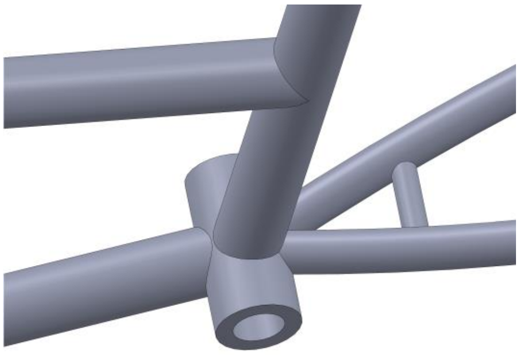

Figure 17: The tube between chainstays

3.4.2 Frequency Improvement

The second objective is to increase the natural frequency of the bike frame. This can be achieved either by increasing the stiffness or by reducing the mass, as indicated by the natural frequency equation:

$$w = \sqrt{\frac{k}{m}}$$

where k is the stiffness and m is the mass.

According to the equation for the stiffness of a cantilevered tube, k is proportional to I/A, where I represents the moment of inertia, and A is the cross-sectional area of the tube. The ratio (I/A) can be simplified to:

$$\frac{1}{4}(R_o^2 + R_i^2)$$

where R_o is the outer radius of the tube, and R_i is the inner radius.

Therefore, the modification consists of increasing the diameter of all pipelines while decreasing their wall thickness to increase the natural frequency. The outer diameter of the chainstay and seatstay tubes was increased from 20mm to 30mm, while the wall thickness was reduced from 3mm to 2mm. The diameter of all other tubes (except the fork and crank shell tubes) was increased from 34mm to 40mm, while their wall thickness was reduced from 4mm to 3mm.

3.4.3 Weight Reduction

To design a lightweight bike, it's crucial to balance performance improvement and weight reduction. Identifying low-stress frame areas helps optimize weight without compromising performance. Initial analysis shows that the circled tubes below experience less stress, making them suitable for weight-saving modifications. The tube connecting the fork shell will maintain its 34mm diameter with a 2mm reduction in wall thickness, while the seatstay tubes will maintain their 20mm diameter but 2mm wall thickness.

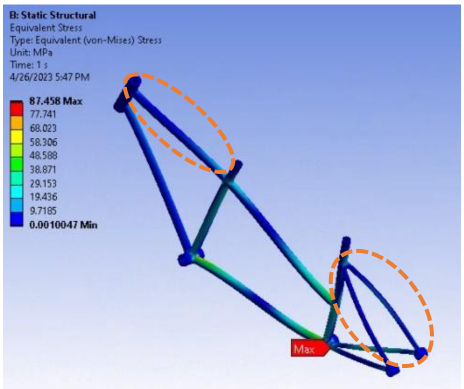

Figure 18: Tubes experience less stress

3.5 Revised Design Results

3.5.1 Static Study

The maximum stress of the aluminium and titanium frame decreased to 81.3MPa and 81.2 MPa, respectively.

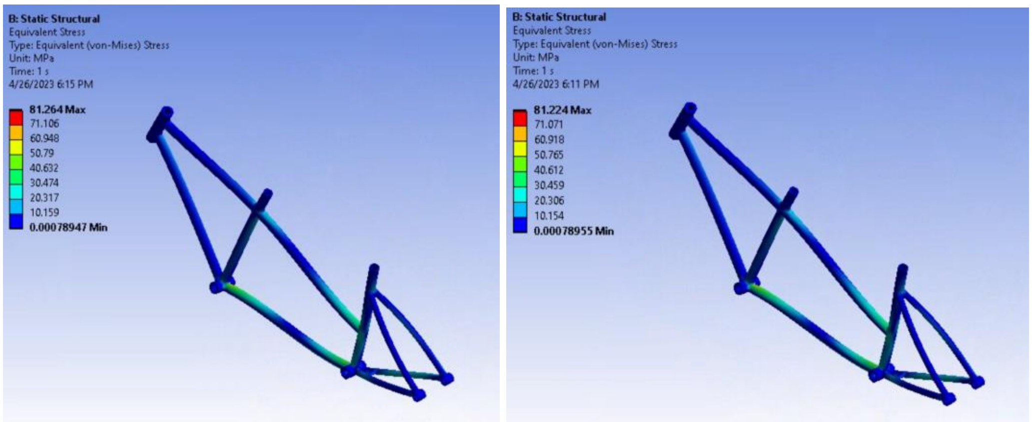

Figure 19: Static analysis of aluminium (left) and titanium alloy frame (right)

3.5.2 Frequency Study

The natural frequency of the aluminium frame was 71.2Hz, while the natural frequency of the titanium frame was 68.4Hz.

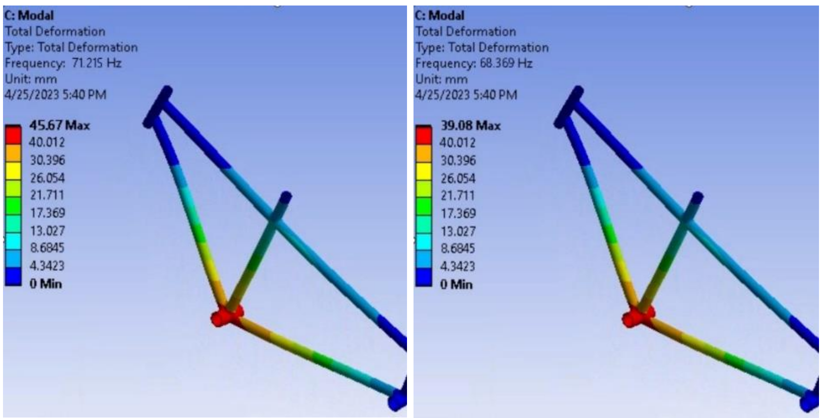

Figure 20: Frequency analysis of aluminium alloy (left) and titanium alloy frame (right)

3.5.3 Fatigue Study

The applications aluminium and titanium alloy to the frame are all allowed it to readily withstand one million loading cycles.

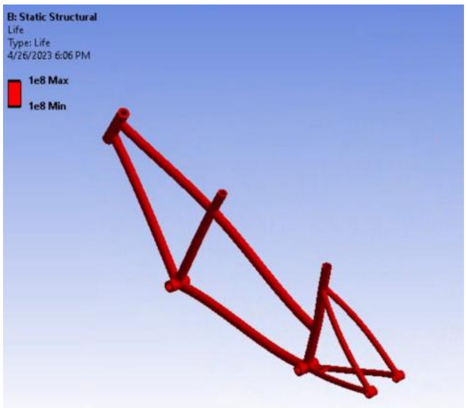

Figure 21: Fatigue analysis for both frames.

3.6 Results Summary

Design Version	Mass (g)	Max von Mises Stress (MPa)	Natural Frequency (Hz)	Life Span (cycles $\times 10^4$)
1st Aluminium	6167	87.5	62.8	>100
1st Titanium	9632	87.5	60.6	>100
2nd Aluminium	6123	81.3	71.2	>100
2nd Titanium	9563	81.2	68.4	>100

Table 3: Table of results for all variations

4 Discussion

In both iterations, the bike frames met all requirements, with the aluminium frame offering a significant weight advantage over the titanium frame, despite the latter's better fatigue life and stiffness.

The FEA Analysis provided valuable insights, but there were inherent limitations due to modelling error from the CAD representation, discretization error from meshing, and assumptions made regarding manufacturing perfection and loading. Even with these limitations, the results served as a robust basis for comparison between the frames, aiding in the decision-making process throughout the design and development stages.

Despite limitations, results provide a solid comparison between frames. ANSYS is useful for real-world simulations, but balancing computational efficiency and accuracy is key. Optimal mesh sizes, material properties, and boundary conditions help improve designs while considering trade-offs.

Further enhancements can be explored by studying how to minimize the weight while maintaining a reasonable balance between weight reduction and performance.

References

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