Finite Element Analysis Final Report: A Lightweight Tandem Bicycle Frame

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Introduction

A lightweight tandem frame for two passengers was designed and tested in various studies using the finite element method. Four iterations were conducted with changing designs and materials to identify the ideal tandem frame that is both durable and comfortable to use. The benchmark used was a natural frequency of more than 30Hz and a lifespan of at least one million loading cycles.

Methods

Initial Setup

The model was designed in SolidWorks and built using the swept feature. In order to produce a quality model that can be used in the real world, attention was paid to perfect the pipes. The swept cut feature was used to ensure that the cross sections of the pipes did not protrude.

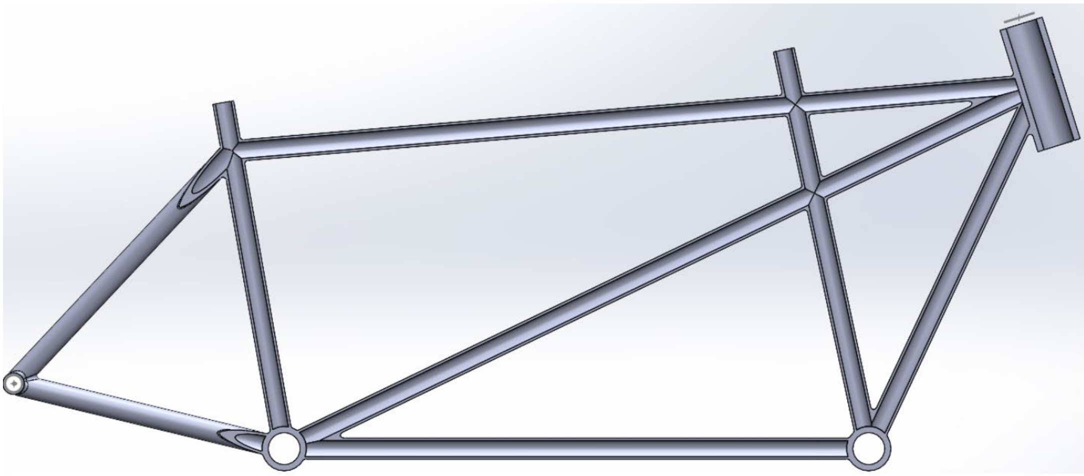


Figure: Cross-sectional view of tandem frame

Assumptions

All members were assumed to be manufactured perfectly with a uniform thickness and no surface imperfections such as cracks or scratches. The pipes were assumed to be welded perfectly and all joints were filleted by 5mm to imitate weldment. Due to constraints in budget and manufacturing techniques, the real frame might not be manufactured as perfectly, leading to areas of increased stress concentration that were unforeseen in this study. Since all materials used for the model were metals, a linear-elastic approach was taken and the assumption was made that strain levels that would cause yielding would not be reached during the study.

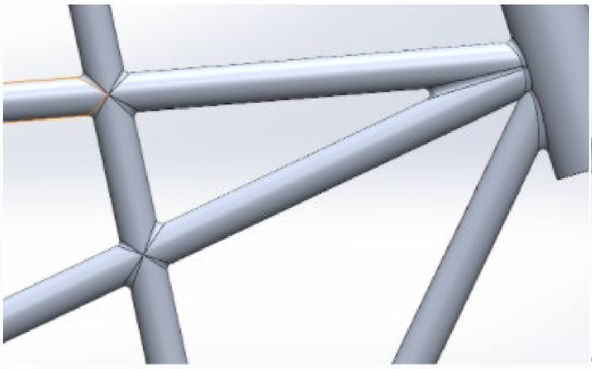


Figure: Filleted joints to mimic weldment

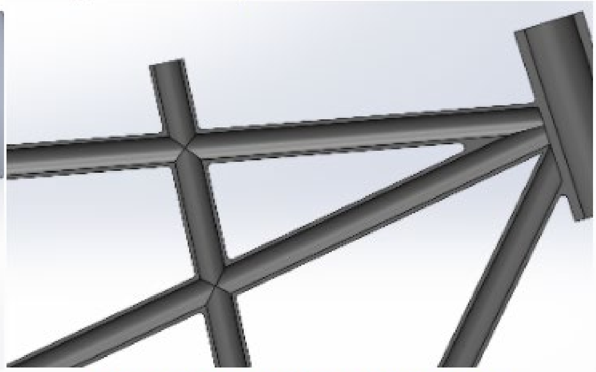
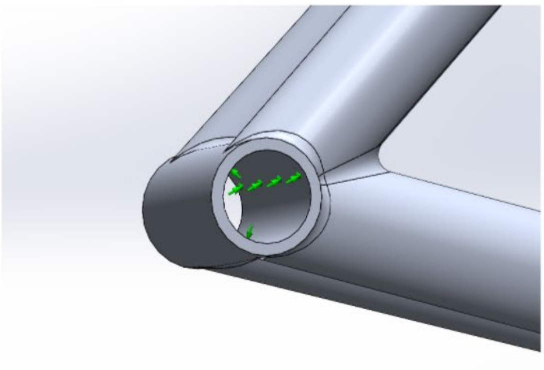
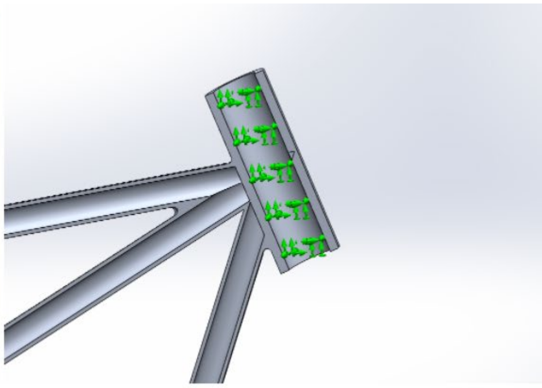


Figure: Constant pipe thickness

Boundary Conditions
The model was constrained in two areas. The inner surface of the fork shell was fixed since it will not undergo any translation or rotation when the tandem bike is moving. The rear wheel bearing is hinged as the rear wheel will rotate along its axis.

Figures: Fixed hinge at rear wheel bearing and fixed geometry at the inner surface of fork shell

Loads
The model accounts for two passengers riding the tandem, each having a mass of 150kg. Instead of applying the weight of the passengers normal to the ground, it was applied along the slanted seats. The effect of this simplification was negligible since the small angle of the slanted seats, which is 10°, will only result in a 3kg difference. Furthermore, 150kg is an upper bound of the average weight of an adult, with generous margin. Remote forces of 750N were also applied at a distance away from each crank shell surface. These forces represent the pedalling force of the passengers.

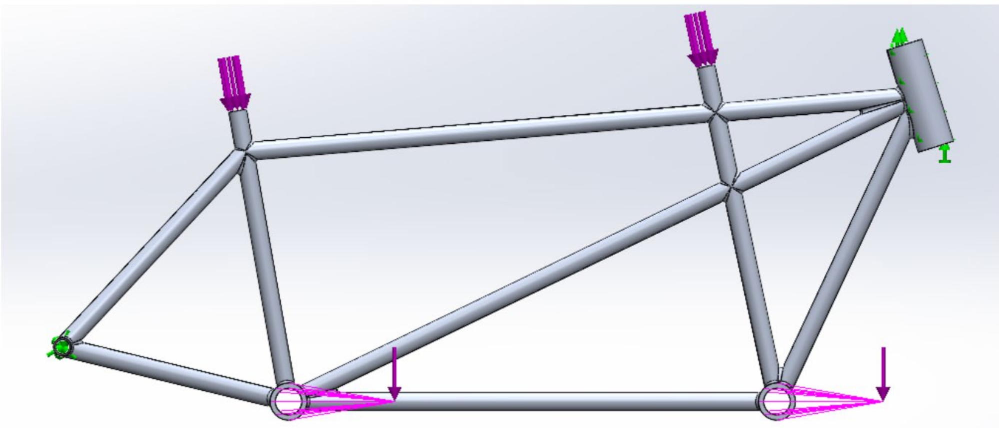


Figure: Arrows in purple indicate applied loads on the model

Two loading scenarios were studied in separate static simulations, with the first being when the passengers pedal on the right side, and the other on the left. The magnitude of the remote masses was held constant. Since this is a tandem bike, their pedalling will be in sync. A third scenario of when the passengers are just sitting on the bike without pedalling was not evaluated as the study aimed to determine the performance of the tandem in the worst-case scenario.

Fatigue Simulation Parameters

The two static simulations with different loading conditions as mentioned were used as events for the fatigue simulation to find the cycle peaks. To ensure a product lifespan of over ten years, one million loading cycles were applied. By obtaining the peaks of the two loads, the worst possible damage can be found. SolidWorks determined the alternating stress and compared it with the S-N curve of the material. If the alternating stress is above the S-N curve, failure has occurred. In the case where the S-N curve for the material was not available, SolidWorks obtained the alternating stress by applying the Gerber correction method on the mean stress found. The study produced the fatigue life of the bike and indicated areas that would face damage.

Frequency Simulation Parameters

The same boundary conditions and loads were applied for the frequency studies. To adhere to safety standards, the resulting fundamental frequency was compared against the minimum benchmark of 30Hz.

<u>Results</u>

Sanity Checks

Quick sanity checks were conducted to evaluate the rationality of the results. The rough dimensions of the model were determined by first modelling the diameter of a typical bicycle wheel, and then constraining the length and height of the overall model according to dimensions of tandem frames found online. The general shape of the frame was inspired by existing frames in the market. When analysing values from different studies, the units and orders of magnitude were compared against each other. Additionally, any displacement below 5mm was deemed as reasonable and non-catastrophic. Finally, the results of the studies were viewed in true scale to determine its reliability by eye.

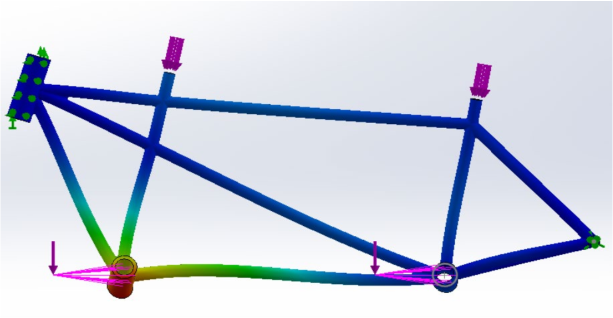


Figure: Exaggerated displacement of model

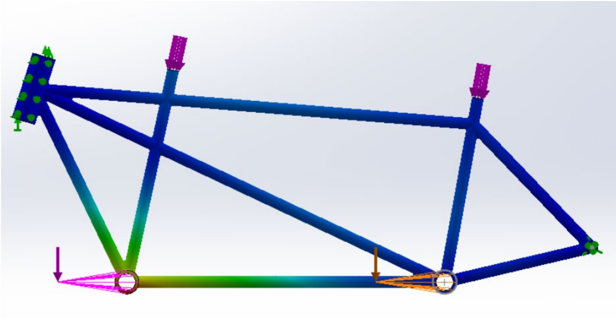


Figure: True displacement of model

Mesh Refinement

During the initial static study, a curvature-based mesh was applied with an element size of 5mm throughout the model. Upon analysing the von Mises stress plot and identifying problematic areas, mesh control was applied using smaller element sizes. The use of mesh control was vital in the study as too large mesh sizes could result in overlooked areas that could have high stress concentrations which might cause catastrophic failure of the product, whereas applying a small mesh size to the whole model is too time-consuming and costly. Mesh control was applied to regions with the highest stress, highlighted in red, which was found to be the filleted edge between the front seat pipe and the crank shell.

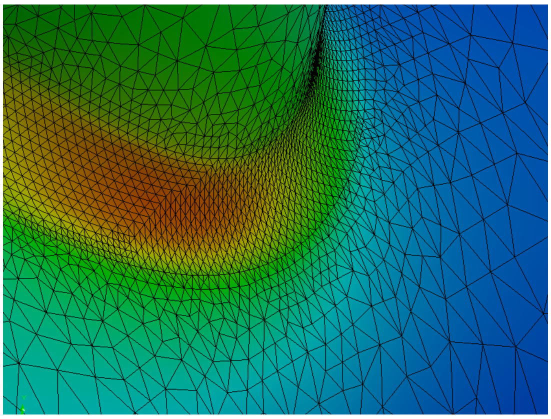


Figure: Varying mesh sizes from 5mm to 0.40mm applied on the model

By reducing the mesh size to 0.40mm in this region, the maximum von Mises stress of the model increased from 53.76MPa to 56.87MPa.

Table: Maximum von Mises stress with varying mesh element sizes

Element Size (mm)	Maximum von Mises Stress (MPa)
5	53.76
2	56.44
0.5	56.80
0.45	57.24
0.4	56.87

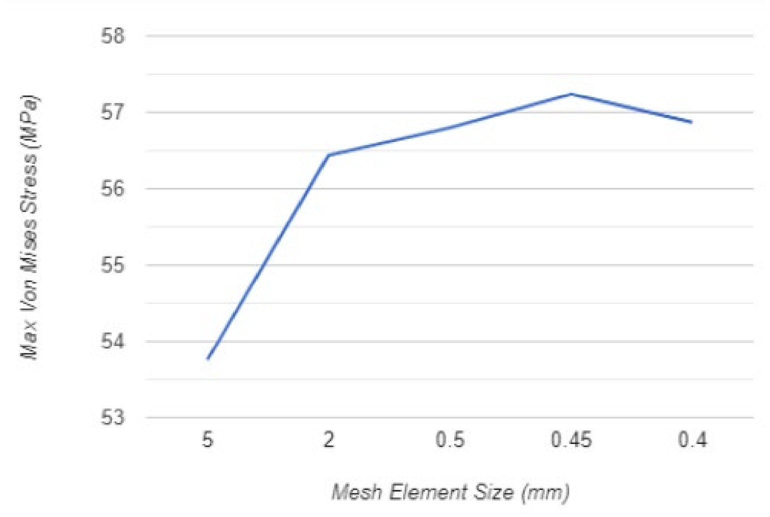
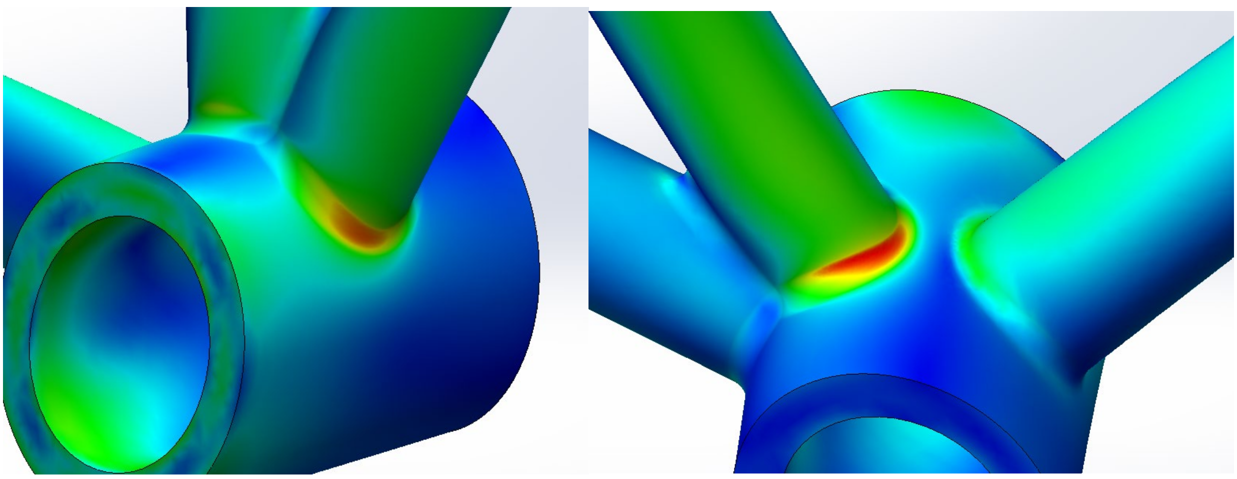


Figure 1: Mesh convergence plot

It was recognized that the accuracy of the study could be improved further with a more ideal mesh size of 0.10mm, however due to limitations in computational power, this was not feasible. The mesh convergence plot also showed that stopping at 0.4mm was reasonable as the graph was approaching the true value of the von Mises stress.

Model 1: Initial Design with Aluminium Alloy
Aluminium 7075-T6 was set as the material of the tandem. After applying mesh control as mentioned, the static study saw a maximum von Mises stress of 56.87MPa at the filleted joint between the front seat and crank shell. The rest of the model had fairly low stresses and were marked in blue. The maximum displacement of the model was found to be 2.29mm, which was a small amount, indicating that the model was stable and the cyclists would not detect it.


Figures: Highest stress concentration occurred at the filleted area of the crank shell joint

The two loading conditions of pedalling on both sides of the tandem produced maximum von Mises stresses of 56.87MPa and 57.10MPa respectively. The resulting fatigue analysis demonstrated that there were no specific areas of concern for damage as fatigue life was constant throughout the model, perhaps due to a well-chosen fillet size at the joints. The model was able to withstand 40 million loading cycles, which is well above the benchmark of one million cycles.

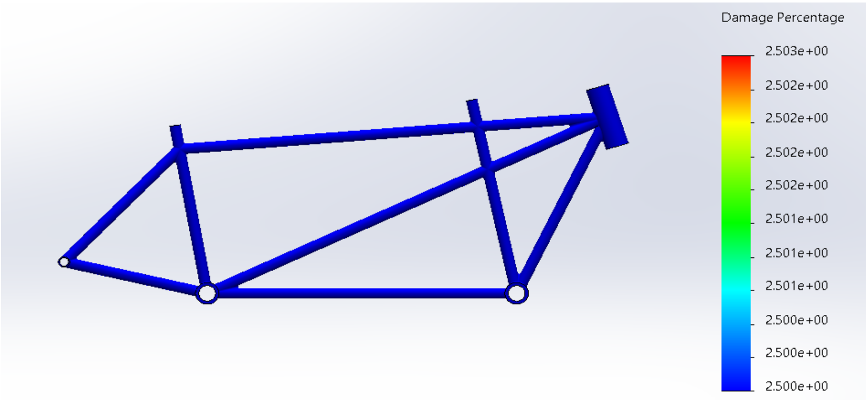
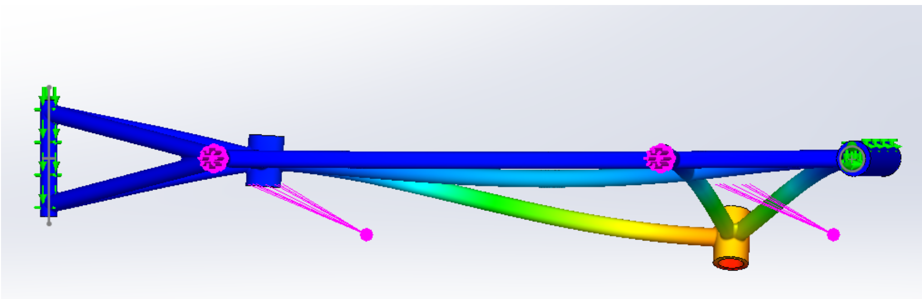


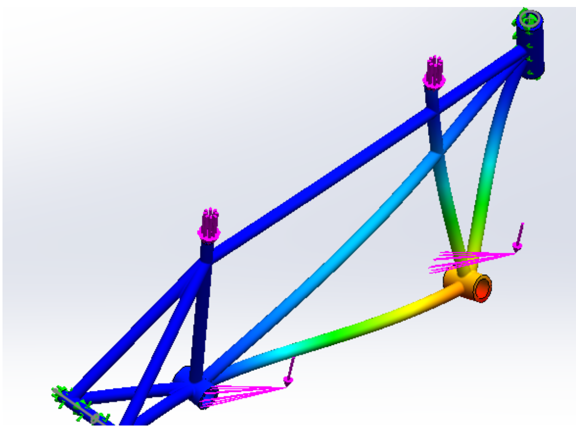
Figure: Model in blue, meaning no damage occurred after one million loading cycles

A frequency analysis indicated that the fundamental frequency of the model was 58.904Hz, which is above the minimum requirement of 30Hz. The fundamental vibration frequency of the model is derived from the eigenvalues, which resulted from the stiffness matrix and mass distribution of the model. It is vital that the natural frequency of the tandem frame is not in the range of that of the human body as resonance will cause large oscillations, resulting in passenger discomfort. In extreme long-term cases, the human body could even be damaged. Seeing that the human body in a sitting posture has a frequency of 4-6Hz [1], the tandem frame has successfully prevented this.

Table: Natural frequencies of Model 1

Mode	Frequency (Hz)
1	58.904
2	104.17
3	163.92
4	186.77
5	192.19

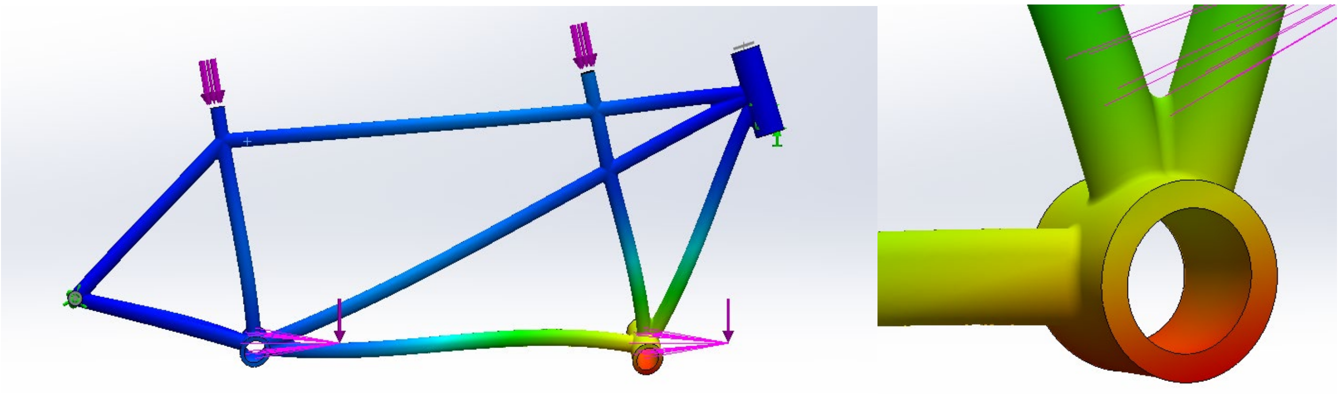




Figures: Top and angled view of the model after displacement caused by fundamental frequency

Model 2: Initial Design with Titanium Alloy

Titanium alloy (Ti-6Al-4V) was then used as the material, while the design of the frame remained constant. The static simulation showed that this model had a maximum von Mises stress of 56.75MPa, which is only 0.12MPa less than the same model made of Aluminium alloy. This improvement was lower than expected. It had a maximum displacement of 1.56mm at the front crank shell.



Figures: Exaggerated view of the displacement and the area with the largest displacement

The results from the fatigue simulation were identical to that from Model 1. No damage occurred in the model and the fatigue life was found to be 40 million cycles, which is an impressive amount. From the S-N curve, the alternating stress is 189.085MPa at one million loading cycles. This means that the model had an alternating stress much lower than that, seeing that it has 40 times the lifespan.

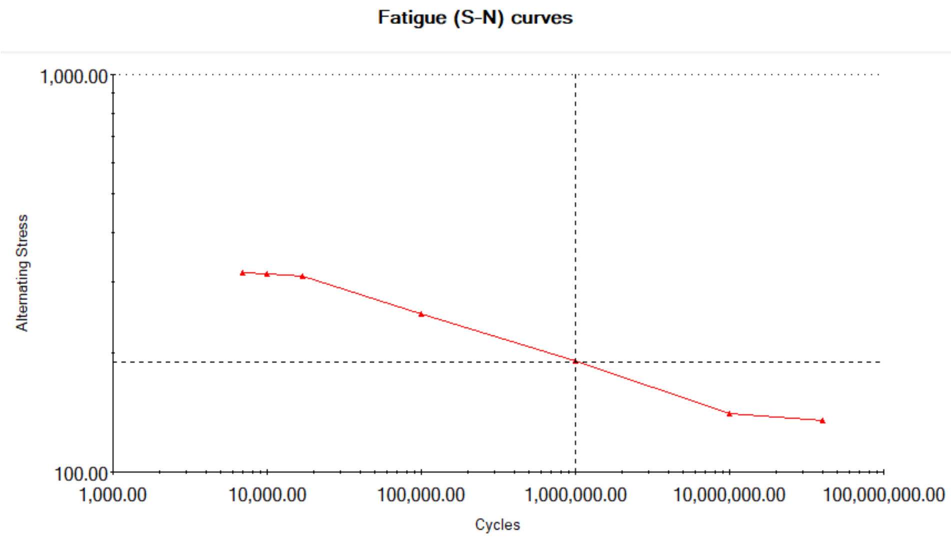


Figure: S-N Curve of Ti-6Al-4V [2]

The frequency simulation produced a fundamental frequency of 58.904Hz, equivalent to that found in Model 1. This was not surprising since the values from the other studies were also largely similar to before.

Model 3: Revised Design with Aluminium Alloy

A key to increasing the natural frequency of the model is to increase the structure stiffness or decrease its mass. Thus, a revision to the model was made. The stiffness of the model was increased by increasing the thickness of the pipes from 4mm to 6mm, which resulted in a 4mm decrease in the internal diameter. A static analysis showed that the maximum von Mises stress was 54.57MPa, which is 2.11MPa lower than Model 1, indicating a successful iteration. The now stiffer model also resulted in being more stable, with a lower maximum displacement of 1.85mm. Similar to the previous two iterations, failure from fatigue did not result and the model also had a lifespan of forty years, equivalent to forty million cycles.

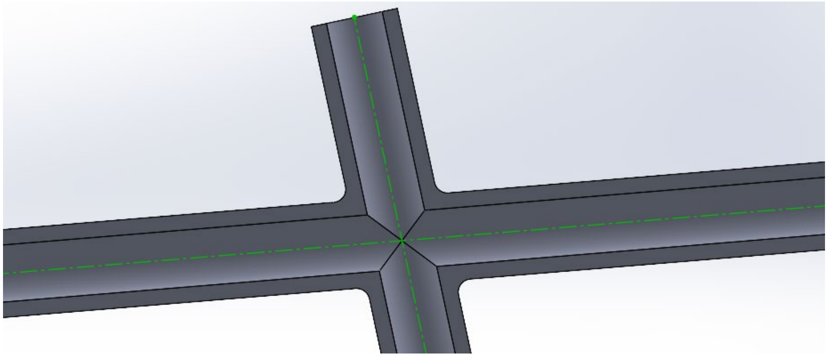


Figure: Revised model with increased pipe thickness

The frequency simulation verified the success in the model revision as this new model had a fundamental frequency of 59.794 Hz, 0.89Hz greater than previous models. This means that the stiffness of the model had been increased successfully without overly increasing its mass such that it offsets it. With a larger fundamental frequency, the tandem is able to provide a much smoother ride to users and meet higher safety standards.

Model 4: Revised Design with Titanium Alloy
The final study saw a maximum von Mises stress of 54.64MPa and maximum displacement of 1.24mm, which is the lowest among the iterations. Unsurprisingly, the alternating stresses throughout the model fell below the S-N curve again, meaning that no damage will occur after one million loading cycles. The frequency study resulted in a fundamental frequency of 57.4 Hz, which was lower than expected.

Discussion

Table: Comparison of Models using Key Parameters

Model	Maximum Von Mises Stress (MPa)	Maximum Displacement (mm)	Fundamental Frequency (Hz)	Fatigue Life (Years)
1	56.87	2.29	58.904	40
2	56.75	1.56	58.904	40
3	**54.57**	1.85	**59.794**	40
4	54.64	**1.24**	57.4	40

In terms of fatigue life, all iterations performed equally well, indicating that the model was built well with the intention of minimising stress concentrations throughout the body. Model 3, which is the model made of Aluminium alloy with increased pipe thickness, scored the lowest maximum von Mises stress and highest fundamental frequency. On the other hand, Model 4 had a lower maximum displacement and only a marginally larger maximum stress than iteration 3 by 0.07MPa. However, upon comparing the cost of the materials, Model 3 reigned superior due to its drastic savings in cost.

Table: Price of the two materials [3]

Material	Price per kg (£)
Aluminium Alloy, 7075-T6	3.945
Titanium Alloy, Ti-6Al-4V	19.25

In theory, all iterations proved to be suitable for real world production and use, within the limitations of the finite element method. Discretisation error could be prevented by conducting further mesh convergence studies and utilising smaller mesh element sizes. To diminish the modelling error and minimise the disparities between the theoretical predictions and the physical phenomenon, more features of the problem should be considered. For example, modelling the pedalling forces as a constant force greatly simplifies the problem, as the forces often vary in intensity in real life and can increase or decrease abruptly at the rider's whims. Further, some bike riders tend to stand up while pedalling, resulting in a significantly larger stress on the crank shells. Additionally, the product could be misused, for instance a third passenger could attempt to stand on the rear wheel bearings while the tandem is moving. Failure to account for such scenarios could lead to catastrophic failure of the tandem.

Nevertheless, FEA provides a powerful solution for field problems and is essential in ensuring quality and efficiency in the creation of a product.

References

[1] Wu Ren, Bo Peng, Jiefen Shen, Yang Li and Yi Yu. *Study on Vibration Characteristics and Human Riding Comfort of a Special Equipment Cab*. 2018. Available from: https://www.hindawi.com/journals/js/2018/7140610/

[2] Available from: SolidWorks 2020 Software

[2] Available from: Granta Edupack 2020 Software