Tandem Bicycle

01412280

Tomáš Kňaze

1 Introduction

The aim of the assignment is to design a tandem bicycle frame for a bicycle hiring company. The bicycle has to be suitable for use by two adults that will be sitting in a fore and aft arrangement.

1.1 Design requirements

The bicycle frame has to fulfil four requirements as stated in the brief:

Lightweight – decreasing the frame weight results in easier handling of the bicycle when in use. Less material is used during the production of the bicycle which is desirable for the manufacturer.

Comfortable – The frame vibrates when a bicycle is in use. This can result in whole body vibrations and ultimately in discomfort for the users. The natural frequency of the frame when in use has to be higher than 30 Hz as stated in the brief.

Sturdy – It is required for the frame to carry two people, weighting 100 kg each. The effective life of the frame is required to be at least 10 years. This is translated into loading cycles for the purpose of the analysis - force of 1kN on each pedal has to be withstood for 1 million cycles.

Unisex - The frame is required to be unisex in order to appeal to a large spectrum of users.

The rough geometry of the frame was given in the brief as shown in Figure 1. The overall length of the bicycle was constrained to be in the range of 2 to 3 metres. The wheel diameter is 800 mm. A fork shell and two crank shells have to be included, with the profile diameter of 70 mm and 10 mm thickness. The diameter of the tubes is not allowed to exceed 40 mm.

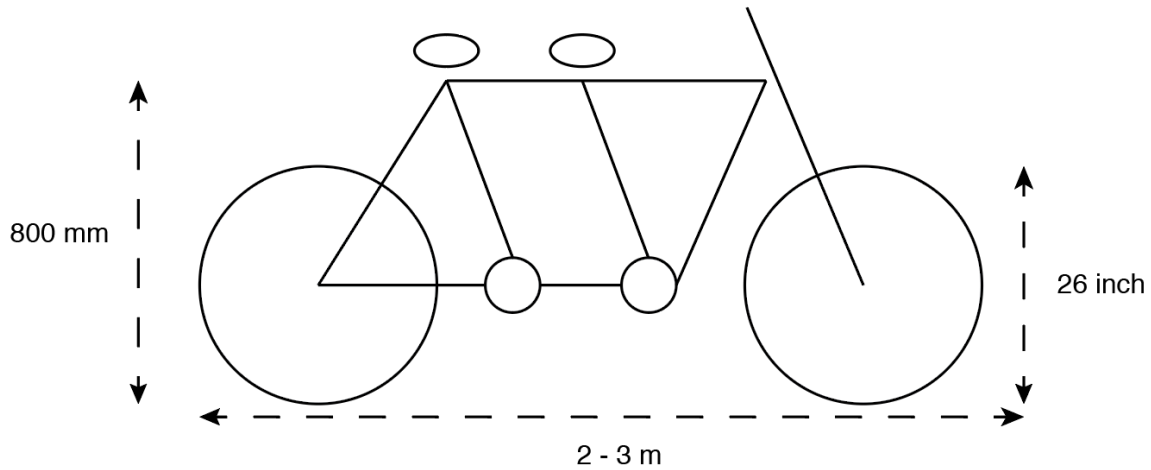

Figure 1

2. Method

Current tandem bicycle frame designs were examined mainly the number of tubes and tube orientation. These inspired the design of the first model. FEA analysis was then performed on the model in Solidworks. The loading conditions that the frame will have to withstand were stated in the brief and were used to perform the studies. Static study was carried out to determine whether the frame is able to withstand the required load. Two events were created, one for when the pedalling force is applied on the right pedal and one for when the pedalling force is applied on the left pedal. These events were used to carry out the fatigue analysis to determine the effective life of the bicycle frame. Frequency analysis was performed to find the natural frequency of the frame for the given loading conditions.

The frame design was adjusted based on the results of the FEA studies, with the aim to minimise weight, increase the natural frequency and fatigue life.

2.1 Modelling Assumptions

The two riders will be applying the force on the pedals perfectly in phase. In reality this will not be true and there will always be a phase difference between the two forces applied on the pedals.

The frame members are joint using welds, these are modelled using fillets of diameter in range of 5 mm to 10 mm. there will be boundary between the materials

Two crank shells are included in the design of the frame. These are modelled as hollow cylinders with a 70 mm external diameter, 10 mm wall thickness and a 100 mm length.

The fork shell is modelled as a hollow cylinder with a 70 mm external diameter, 10 mm wall thickness and a 200 mm length.

Perfect connection of the tubes at the welds is assumed. This would not be true as in reality the grains of the two pieces of metal are not aligned perfectly.

The simulation assumes perfect surface of all the tube members. In reality this will not be true, there will be imperfections in the tube surfaces that will act as stress concentrators and will ultimately decrease the effective life.

2.2 Analysis Parameters

In order for the simulation to run successfully, all the variables in the equilibrium equation (Figure 2) have to be defined.

$$[K]\{D\} = \{R\}$$

$$[K] - structure\ stiffness\ matrix$$

$$\{D\} - boundary\ conditions$$

$$\{R\} - forces\ applied$$

Figure 2

2.3 Boundary conditions

Two boundary conditions were implemented when carrying out the simulations. It was assumed that the inside surface of the fork shell is fixed – it is not able to translate or rotate. The inside surface of the rear wheel bearing was assumed to be hinged – it is able to rotate but not translate. These values were used by Solidworks to populate the {D} matrix.

2.4 Loading

To ensure that the frame design is conservative, analysis was carried out with high loading values that represent the worst case scenario. It was assumed that the bicycle is used by two people, each weighting 100 kg, exerting a force of 0.981 kN on both of the seat joints. 1 kN of remote force is exerted by both of the riders at the same time on the pedals which are located (200 x + 150 z) [mm] from the centres of the crank shells at the time the force is exerted.

2.5 Material

Aluminium alloy (7075 – T6) was used for the first set of analyses of the three frame design iterations. The second set of analyses was carried out to test the models made out of Magnesium alloy. It offers excellent fatigue resistance, denting and buckling resistance, and a high damping capacity. This results in a much smoother ride. (Precisiontandems.com, 2019) Material properties of magnesium alloys necessary to run a fatigue analysis are not defined in the Solidworks material library. Custom material with the properties of a magnesium alloy was created. Tensile strength of 335 MPa, Compressive strength of 240 MPa and Yield strength of 240 MPa were used. (Arthur, 2018) The values are for Magnesium Alloy ASM AE81 that is used by Allite to produce bicycle frames.

2.6 Fatigue Simulation

The loads applied onto the pedals oscillate from 0 to the maximum of 1 kN and back to 0. The S – N curve predefined in the Solidworks material library is defined such that the stress

is alternating around zero. The stress due to pedalling in the case of the bicycle will be oscillating above zero – the stress ratio is 0. To account for this the Gerber correction method was used because of it suitability for ductile materials. Load applied by the cyclist when the pedal is at the topmost position is zero. It then increases to maximum when the pedal is at a horizontal position and then decreased to zero when the pedal is at the bottom position. For the following half-cycle there is no load applied onto the pedal when it rises to the top as the other foot is applying force onto the other pedal. The cycle is then repeated. (Figure 3) To simulate this, two load events were created, each with the force of 1kN and with 500 000 cycles as the frame has to be able to withstand 1 million cycles. A downside to this is that Solidworks is not able to calculate the expected number of cycles that the part will be able to withstand when two loading events are added into the fatigue analysis. Predicted fatigue life was compared to fatigue life obtained using the “find peaks” method. Results from the alternating stress analysis were 22% larger. The alternating stress is computed using the Stress intensity to obtain more conservative results.

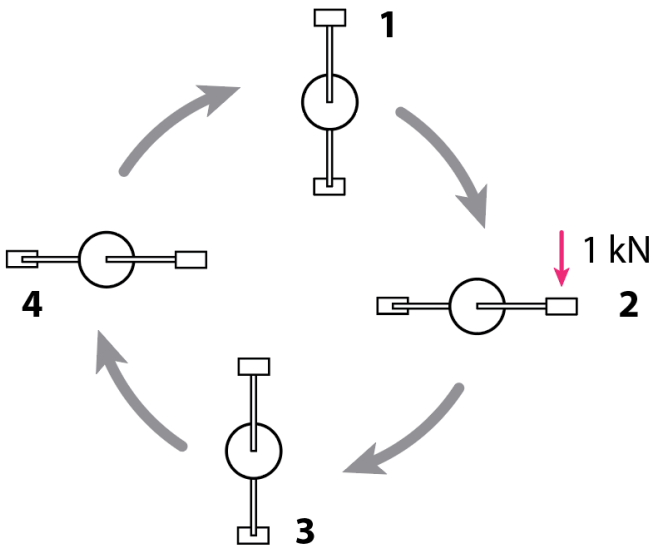

Figure 3

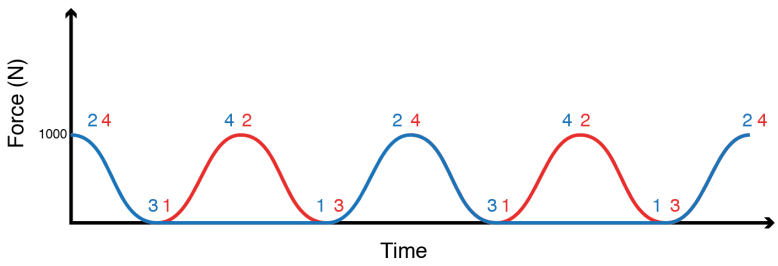

Figure 4

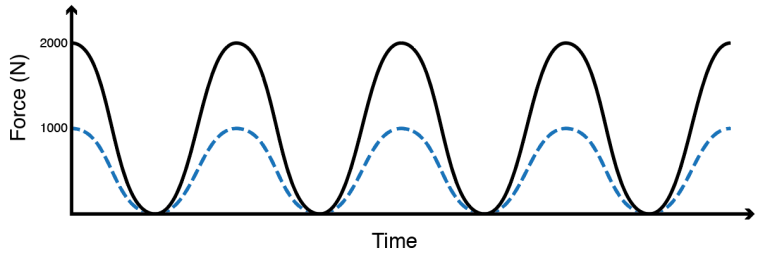

Figure 5

Frequency simulation

The natural frequency modes are calculated based on the forces on the seats and two remote loads on the pedals. The Direct sparse solver was used for the frequency analysis because of its accuracy.

Results

Aluminium Model 1

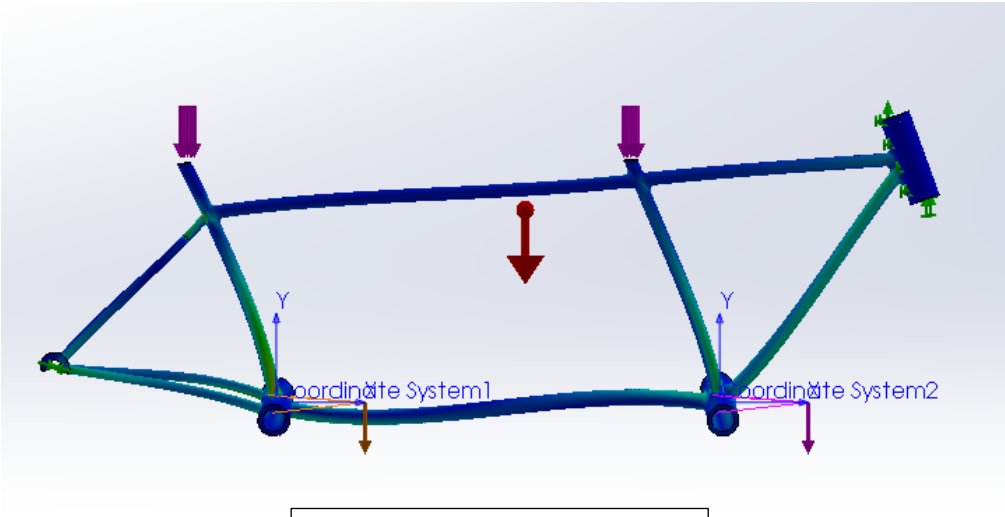

Figure 6

	STATIC				
MESH	Von Mises	Stress intensity	Yield Strength	Max displacement	
10 - 2mm	1.22E+08	1.39E+08	5.00E+08	6mm	
8 - 1.6mm	1.33E+08	1.51E+08	5.00E+08	6.13mm	
6 - 1.2mm	1.22E+08	1.39E+08	5.00E+08	6.13mm	
5 - 1mm	1.44E+08	1.64E+08	5.00E+08	6.138mm	
5-1mm + mesh c	1.31E+08	1.48E+08	5.00E+08	6.138mm	

FATIGUE	FREQUENCY (Hz)				
	1	2	3	4	5
No damage	39.989	51.62	74.7	114.93	116.72
No damage	39.985	51.582	74.69	114.91	116.69
No damage	39.972	51.514	74.672	114.88	116.68
No damage	39.965	51.481	74.654	114.83	116.59
No damage	39.97	51.481	74.657	114.8	116.54

Aluminium Model 2

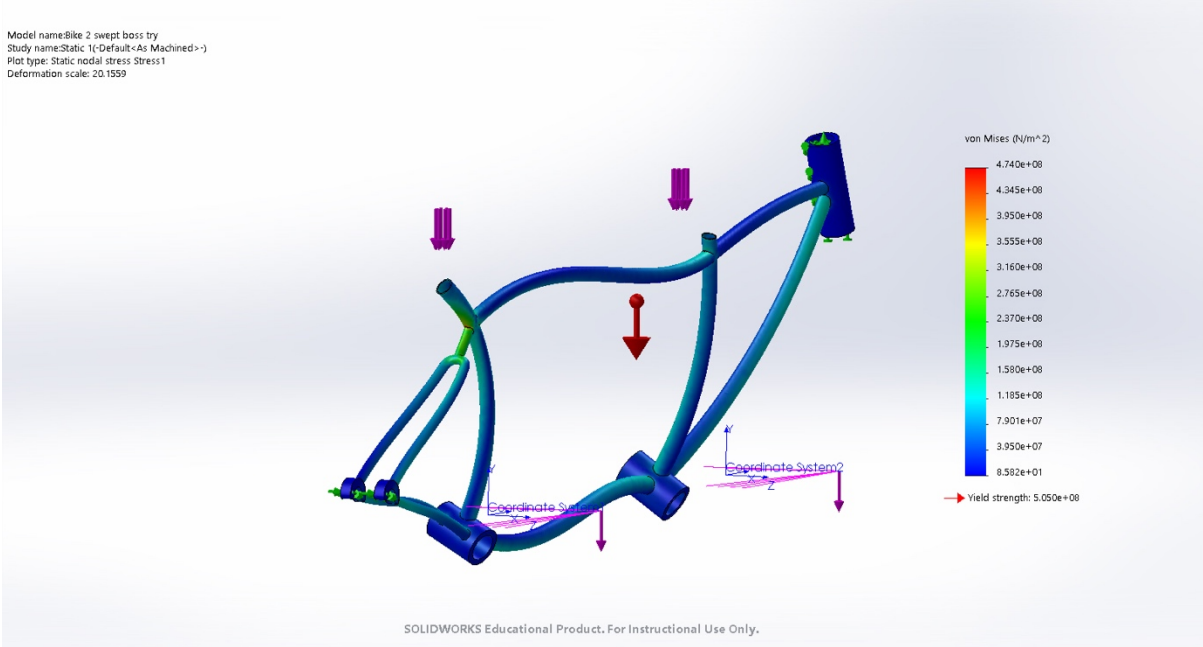

Figure 7

	STATIC				
MESH	Von MIses	Stress intensity	Yield Strength	Max displacement	
10 - 2mm	1.96E+08	2.24E+08	5.00E+08	17.39	
8 - 1.6mm	1.85E+08	2.08E+08	5.00E+08	17.43	
6 - 1.2mm	1.87E+08	2.11E+08	5.00E+08	17.58	
5 - 1mm	1.87E+08	2.11E+08	5.00E+08	17.57	

FATIGUE	FREQUENCY				
	1	2	3	4	5
3726 % - failure	30.932	245.96	65.317	92.183	108.38
failure	30.932	245.96	65.317	92.183	108.38

Aluminium Model 3

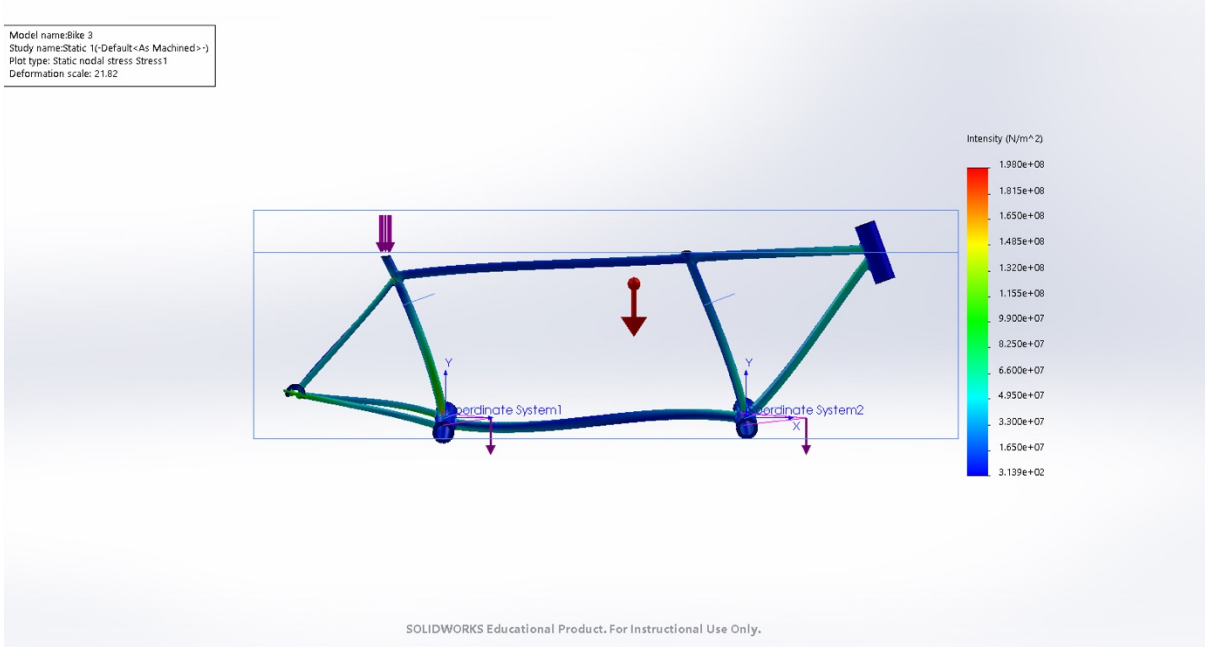

Figure 8

	STATIC				
MESH	Von MIses	Stress intensity	Yield Strength	Max displacement	
10 - 2mm	1.61E+08	1.86E+08	5.00E+08	9.434	
8 - 1.6mm	1.76E+08	1.99E+08	5.00E+08	9.692	
6 - 1.2mm	1.84E+08	2.09E+08	5.00E+08	9.723	
5 - 1mm+meshc	1.59E+08	1.80E+08	5.00E+08	9.412	

FATIGUE	FREQUENCY				
	1	2	3	4	5
No Damage	34.839	51.103	67.7	125.06	133.12
No Damage	34.52	51.049	67.728	125.48	133.62
No Damage	34.51	50.944	67.658	125.39	133.3

Magnesium Model 1

	STATIC				
MESH	Von MIses	Stress intensity	Yield Strength	Max displacement	
10 - 2mm	1.30E+08	1.49E+08	2.40E+08	9.836	
8 - 1.6mm	1.32E+08	1.50E+08	2.40E+08	9.85	
6 - 1.2mm	1.31E+08	1.48E+08	2.40E+08	9.84	

FATIGUE	FREQUENCY				
	1	2	3	4	5
no damage	41.14	51.2	76.49	115.8	118.22
no damage	41.159	51.168	76.482	115.77	117.93
no damage	41.161	51.17	76.491	115.79	

Magnesium Model 2

	STATIC					
MESH	Von MIses	Stress intensity	Yield Strength	Max displacement		FATIGUE
10 - 2mm	4.76E+08	5.44E+08		17.1mm		failure
8 - 1.6mm	5.20E+08	6.00E+08		17.61mm		failure
6 - 1.2mm	6.46E+08	6.60E+08		17.53mm		failure
5 - 1mm+meshc	6.44E+08	6.30E+08		17.53mm		failure

Magnesium Model 3

	STATIC				
MESH	Von MIses	Stress intensity	Yield Strength	Max displacement	
10 - 2mm	2.68E+08	2.81E+08		17.1mm	
8 - 1.6mm	2.73E+08	2.90E+08		17.61mm	
6 - 1.2mm	2.71E+08	2.89E+08		17.53mm	
5 - 1mm+meshc	2.71E+08	2.88E+08		17.53mm	

FATIGUE	FREQUENCY				
	1	2	3	4	5
no damage	38.239	49.712	64.728	121.719	139.22
no damage	38.212	49.69	64.723	121.723	139.23
no damage	38.201	49.678	64.72	121.72	139.19

Mesh size

Analysis of the models started with meshing at maximum element size of 10mm and minimum element size of 2mm. First couple of runs have shown that interpolation with larger sized elements are unnecessary as the results converged only at maximum element sizes smaller than 10mm. In case when the results were not converging at maximum element

sizes smaller than 0, mesh control was applied and areas with large damage percentages were interpolated to locally create a finer mesh. The maximum aspect ratio of the mesh was checked to ensure that it is below 40. The percentage of Jacobian distorted elements was monitored to ensure no elements are distorted.

Mesh type

Curvature based meshed was used as the majority of the bicycle frame is curved and would be insufficiently represented by a linear mesh. Quadratic interpolation was utilised to discretise the complex geometries into tetrahedrons. 4 Jacobian points in a tetrahedron were used. A trial run comparing 4 and 16 Jacobian point tetrahedrons. The former yielded a value of 2.084e8 N/m^2 and the latter 2.083 N/m^2. This is a difference of 0.05% and was decided to be non-beneficial as using 16 Jacobian points increases the runtime of the studies.

Static Simulation

Von Mises stress and Stress intensity were calculated. The Stress intensity was used to consider the worse case scenario and keep the design process conservative.

Sanity Checks

The static study of the models showed that the frame would displace under load by 5 to 15 mm. This seems reasonable and acted as a sanity check. Convergence of the obtained values with increasing mesh density showed whether the results are reasonable.

Discussion

The analysis performed has shown that MODEL 1 was unnecessarily overengineered, reaching a safety factor of 4.1. Based on the results of the fatigue analysis, the design would be able to function indefinitely under the condition stated in the brief. This was due to the large tube diameter and wall thickness, which also resulted in large weight of the model – 2.71kg. There were no large stress concentrations and so the same tube arrangement was carried onto MODEL 2.

The tube diameter and wall thickness was decreased for MODEL 2 to make the model less overengineered and lighter. The safety factor decreased to 2.23 and the mass to 1.76 kg. Fatigue analysis has shown that the effective life of the model will be significantly shorter than 1 million cycles. The Static analysis has shown that there is a significant stress concentration under the back seat.

The tube diameter of tubes in model 3 was increased to 30mm and wall thickness was changed to 1mm. This resulted in weight of the model increasing slightly to 1.89 kg. Two

tubes were used to connect the rear wheel bearing to the back seat tube to avoid the stress concentration from second model. This design iteration has a safety factor of 2.7 and will withstand the cyclic loading for 1 million cycles.

The analysis performed on the magnesium alloy frames has shown properties close to aluminium, with the internal stresses around 20% larger and 10% higher natural frequencies. The weight of the heaviest magnesium frame is 1.8kg, 40% lighter than the weight of the same aluminium frame which can be a great advantage to some users.

References

Figures 1 – 8: Made by the author

Arthur, D. (2018). *Allite launches 'Super Magnesium' alloy: lighter and stiffer than aluminium and less expensive than carbon fibre*. [online] road.cc. Available at: https://road.cc/content/tech-news/248587-allite-launches-super-magnesium-alloy-lighter-and-stiffer-aluminium-and [Accessed 21 Mar. 2019].

Precisiontandems.com. (2019). *Magnesium is the lightest structural metal currently available in theworld*. [online] Available at: https://www.precisiontandems.com/cat_files/paketa/paketamagnesium.htm [Accessed 21 Mar. 2019].