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Performance All-Terrain Vehicle

Rear Axle Design

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Introduction

Problem Statement

During this project, we were tasked with designing an after-market, high-performance rear axle for an ATV. During its function, an ATV endures high loading conditions in all directions which arises many technical problems to consider when designing a high-performance component. As an initial prerequisite, there are specific dimensions that needed to be satisfied in the interest of compatibility with a wide variety of existing ATV models. When considering the components that interact with the ATV, we first have the choice of bearings which supports the axle through the connection of the swingarm. The bearings spacing is restricted to the central 200mm of the axle.

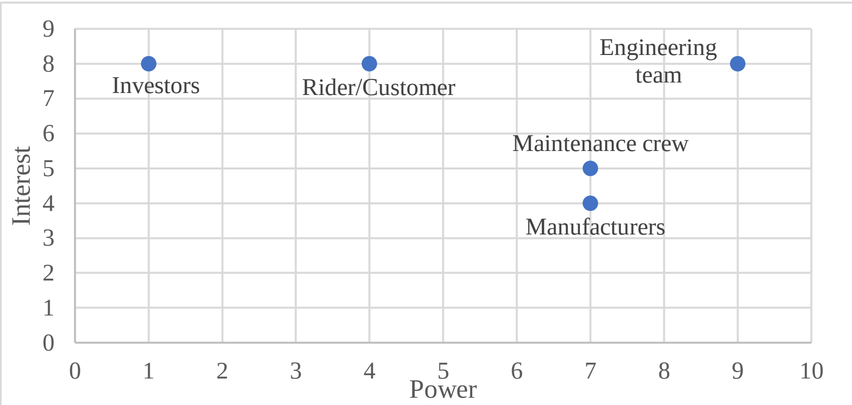

Figure 1A: Stakeholder power/interest chart

Table 1A: Technical data

	Value
Mass of ATV	~170kg
Power at crank	40HP / 30kW
@ engine speed of	6000 rpm
Min. drop height	0.5m
Suspension travel	228.6mm
Gearbox ratios (1st)	7.497
Gearbox ratios (5th)	2.352
Target sprocket ratio	3:1
Disc Brake Diameter	200mm
Service interval	300 hrs

Furthermore, to provide rotational power output from the gearbox to the axle, the size of the sprocket pair which acts parallel to the ground was to be determined, with a required chain pitch of 15.875mm and an ideal target ratio of 3 between the driver and driven sprocket. We, however, also had to look at where the disk brakes need to be situated in our design. The design of the brake or its hub was not required as our clients will fit their disk onto the axle. The brakes must lie within a region 50mm immediately to the right of the bearings, so that it provides enough space for the hub to be fitted. These components, along with the hub for the wheel must be appropriately fixed in place using determined fixing and tolerancing methods. Further technical data that we needed to adhere to are displayed in Table 1A (seen above), which assumes the mass of the ATV alongside the power provided by the engine at the crankshaft and the engine speed. These values will aid in finding the specific loading conditions of the axle through force analysis to determine the appropriate geometry of the shaft. The stakeholder groups are represented in Figure 1A, illustrating their interest and influence over specific areas of the project – including manufacturing, operating and overall quality control.

Priorities

When considering what factors make an axle 'high-performance', there are multiple criteria that we can optimise to achieve a superior product compared to a stock axle. For example, performance directly involves the strength and mass of the shaft which is dictated by the material we select. During selection, the ideal choice of material will have an optimum balance between multiaxial strength, minimal mass, and cost for our stakeholders. By analysing the maximum stress (yield stress) for a range of materials ensures that the axle will be able to withstand the appropriate loading conditions during its function. Furthermore, designing the geometry of the shaft will also directly influence the amount of stress being received at critical nodes on the axle. This leads to another factor we can maximise which is safety and reliability. If we think about during its operation, an ATV prolongs constant impact from high-speed jumps and aggressive terrain, so it is in our clients' best interests to ensure that our product does not sustain failure until the required service interval (300 hrs). From analysing the stakeholders' priorities and preferences, we have opted for a more strength and lightweight oriented package, where we are satisfied with providing the most performance available at a greater cost.

Product Design Specification

Table 2A: Local PDS for the axle

Importance	Requirement	Target/Limit
High	The shaft must be able to support the combined mass of the ATV and the rider	170kg ATV + 50-100kg rider = 230-270kg
High	The bearings should be able to withstand heavy radial loads	The axle should have spherical roller bearings
High	Driven sprocket should be made to accommodate the appropriate chain pitch required	The sprocket should be able to attach to a chain of pitch 15.875mm
High	Driven sprocket should be made to accommodate the appropriate target ratio required	The sprocket should be able to attach to a chain of target ratio 3
High	The bearings must fit well on the shaft with one bearing allowed to move horizontally and another rigidly fixed	The bore diameter of the bearings must match the diameter of the shaft (70mm)
High	The drive sprocket must fit well on the shaft with no vibration	The bore diameter of the sprocket must match the diameter of the shaft (60mm)
High	The disk brake must fit well on the shaft with no vibration	The bore diameter of the disc brake must match the diameter of the shaft (60mm)
Medium	The bearings must be an appropriate distance apart from each other	Position the left and right bearing 200mm apart from each other
Medium	The disk brake must not take up much horizontal distance	The width of the disk brake component should take up a horizontal distance of 50mm maximum
High	Must be heavier than a stock axle	Mass of ATV axle should be greater than a stock axle
High	The axle should be resistant to mud/dirt	The axle should be resistant to mud/dirt and should not experience any form of corrosion
High	The axle must be resistant to water such as rain	The axle should be resistant to water and should not experience any form of corrosion
High	Must have greater strength than a stock axle	Must be made of a material that has stronger material properties than a stock axle
High	The shaft should be able to withstand a little impact caused by dirt particles or rocks	The material should not experience any plastic deformation upon impact from mud or rocks
High	The wheel hub of the wheel should be fitted well with the axle	A taper lock with transition fit will be used to ensure the wheel is secured onto the axle
High	The sprocket must be fitted well on the shaft with good rigidity	Circlips will be installed at either side of the sprocket with a transition fit to ensure it doesn't experience any unwanted movement
High	The bearings must be fitted appropriately on the shaft	The bearings will be fitted using circlips at either side of both bearings. One of the bearings will have circlips tightly fitted so it doesn't move, and the other bearing will have circlips fitted in a way that allows horizontal movement for the bearing
High	The disc brake must be fitted well on the axle	The disc brake will be fitted using a locking ring
High	The axle should not be a single-use system	The axle should be able to produce the same function repeatedly without a drop in performance
High	The bearings, disk brake, sprocket must work reliably on the ATV shaft on each use	They should be made out of a reliable material and positioned in a way that doesn't interfere with each other
High	The axle system should work well and be able to withstand the vibration produced by the engine of the ATV	The components on the axle should be fixed securely on the shaft and work as one system.
High	The installation and use of the ATV axle should be safe for the rider	Must be made out of a reliable material and assembled in a way that does not cause components of the shaft to fail or fall out of place
Medium	Component replacement should not cause any collateral damage to other components	Axle should maintain 266.8 rpm before and after maintenance
High	During operation, components should remain fixed to the axle and maintain functionality	Tolerances for components must be adhered to

Analysis

Free Body Diagrams, Bending Moments & Shear Forces

To begin with, basic free body diagrams of the system were drawn both in perpendicular and parallel orientation relative to the ground. These contained all forces and reactions from components onto the axle, aligned with preliminary distances. The length of the axle and distance between bearings were fixed at 850mm and 200mm respectively, whereas an assumption was made to place these components in symmetry about the centre line of the axle. This was done in order to make calculations simpler, however in reality these components would be offset in optimal locations relative to other parts of the ATV. The system diagram, perpendicular and parallel free body diagrams can be seen in Figure 3A, 3B and 3C respectively.

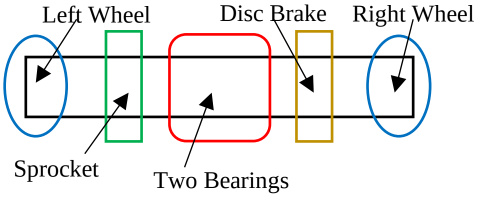

Figure 3A: System diagram

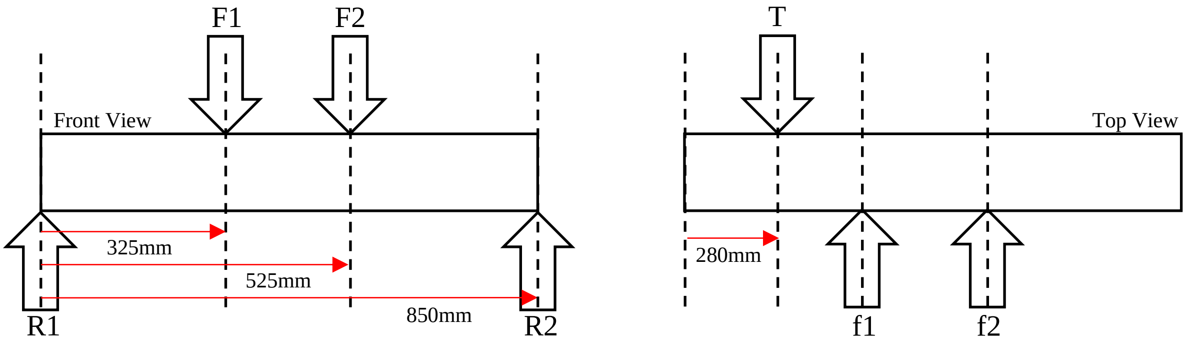

Figure 3B, 3C: Free body diagrams depicting perpendicular & parallel forces on axle

[Front View: R1, R2 = reaction forces at the right and left wheel. F1, F2 = forces on the bearings from weight; Top View: T is the force on the axle from the sprocket whilst f1, f2 are the reaction loads at the bearings in response to T.]

Secondly, the nodes at which forces were to be evaluated were chosen – these were selected to be at the position of each of the components, that way the stresses and loads could be evaluated at the key points. In total there were six components, hence six nodes. Based on these FBD's, general expressions for the bending moments and shear forces could be found at the respective nodes, hence two sets of graphs were drawn to show the variation of these forces and moments for the perpendicular and parallel loads – they can be found below in figure 3D & 3E respectively. The light blue line illustrates the load on the bearings as uniformly distributed. Initially both point loads and a uniformly distributed load were considered, however point loads were used in analysis due to the simpler nature of calculations, hence less margin for error. In reality, a UDL would be more realistic, hence it is reflected in the safety factor calculations in following parts of the report.

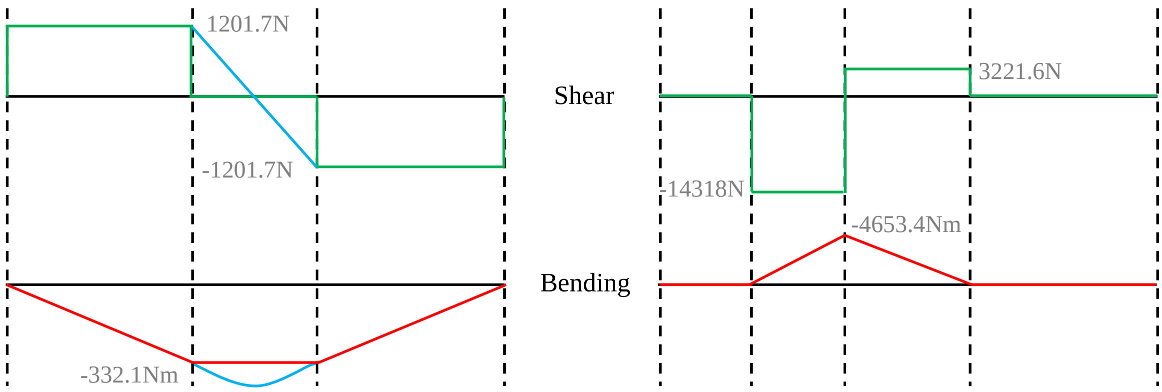

Figure 3D, 3E: Bending moment & shear force diagrams for both Figure 3B & 3C respectively

It was at this point where initial estimates and assumptions were made for the dimensions of the axle in order to commence calculations. The original shaft diameter was chosen as a constant 70mm, with step ups then being introduced at regions where components would be – from 70mm to 75mm to 80mm being the largest. This was due to knowing that bearings are the primary transition point for loads between the ATV and the axle, therefore would be under the largest stress. These were then changed once the components had actually been selected – the highest step was changed to 70mm as the maximum bearing bore was this much, the second step to 60mm due to the maximum bore value of the sprocket and finally 57.5mm for the final step. This was not in response to a component however a larger step was not necessary and would have contributed to a much higher stress between steps, hence was minimised to 2.5mm between the two.

Additionally, estimates were made for the driven sprocket diameter and the total static weight acting on the axle. The sprocket estimate was based on a given, fixed disc brake diameter – knowing that the disc brake and sprocket are usually of similar size – and again through online searches rough sizes appeared to be 250mm. Since the number of teeth was not decided upon, an estimate of 230mm was chosen. The mass of a person was stated to be between 50 – 100kg and the average of these, 75kg, was used as the dummy weight. With the total mass of the ATV being 170kg, a net mass of 245kg was acting on the axle.

Load Calculations

Since load calculations were required at each node, a spreadsheet was created in order to create a more autonomous process to allow changes to be reflected instantly amongst other values. The first load to be calculated was due to the net mass on the axle – 245kg. Multiplying this by $g = 9.81\ ms^{-2}$ a force of $2403.45N$ was calculated. Due to the symmetry of the design, it was assumed that the load was split equally through both bearings, in turn maintaining equal and opposite reaction forces at both wheels. This is represented in the first row of Table 3A below.

Table 3A: Table of vertical and horizontal loads on axle at each node

Vertical	Notation	R1		B1Y	B2Y	R2
Static Load	N	1021.725		-1021.725	-1021.725	1021.725
Shear	N	1021.725		0.000	-1021.725	0.000
Bending	Nm	0.000		-332.061	-332.061	0.000
Horizontal	Notation	R1	FS	B1X	B2X	R2
Static Load	N	0.000	-14318.100	17539.670	-3221.570	0.000
Shear	N	0.000	-14318.100	3221.570	0.000	0.000
Bending	Nm	0.000	0.000	4653.350	0.000	0.000
Torque	Nm	-536.930	-1073.860	0.000	0.000	-536.930

Also, Table A shows the other loads and moments at each node, including shear force, bending moment and torque. Shear and bending values were calculated through the use of taking cuts between each node and solving for equilibrium. These values are reflected on Figure 3D and 3E at their respective nodes. The final quantity was torque at the sprocket. This was calculated using the following equations:

$$g.b.r_1 = \frac{\omega_{engine}}{\omega_{driving\ sprocket}} \tag{1}$$

$$\frac{\omega_{driven}}{\omega_{driving}} = \frac{r_{driving}}{r_{driven}} \tag{2}$$

$$P = T\omega_{driven} \tag{3}$$

where; g.b.r_1 = 1st gearbox ratio, ω is the angular velocity [rad/s], T is the torque [Nm] and P is the power [W]

The values given in the requirements were: 1st gearbox ratio = 7.497, power at engine = 30kW, sprocket ratio ($r_{driving}/r_{driven}$) = 1/3, ω_{engine} = 6000 rpm = 628.3 rad/s

Finally, the torque was calculated by combining equations (1) through (3), giving equation (4):

$$T = \frac{P \times g.b.r_1 \times r_{driven}}{\omega_{engine} \times r_{driving}} = \frac{30000 \times 7.497 \times 3}{6000 \times \frac{2\pi}{60}} = 1073.86Nm \tag{4}$$

This in turn gives the static parallel load on the sprocket when divided by the radius of the sprocket as per the equation $Torque = Force \times radius$, coming to 14318N.

Stress Calculations

From these loads, stresses at each node were calculated including torsional, bending, shear and principal. Standard equations were used for torsional and bending and equations (5) & (6) for principal stress and max shear stress respectively:

$$\sigma_{11}, \sigma_{22} = \frac{1}{2}(\sigma_x + \sigma_y) \pm \sqrt{\frac{1}{4}(\sigma_x - \sigma_y)^2 + {\tau_{xy}}^2} \tag{5}$$

$$\tau_{max} = \sqrt{\frac{1}{4}(\sigma_x - \sigma_y)^2 + {\tau_{xy}}^2} \tag{6}$$

where; σ

The principal stresses at each node are represented in Table 3B below:

Table 3B: Principal stresses on axle at each node

σ_{11}	N/m^2	14607880.18	23903046.53	966456.21	0.00	0.00	14409803.70
	MPa	14.61	23.90	0.97	0.00	0.00	14.41
σ_{22}	N/m^2	-14214413.06	-29100437.31	0.00	-306512.81	0.00	-14409803.70
	MPa	-14.21	-29.10	0.00	-0.31	0.00	-14.41

Although values are lower than expected the pattern and distribution of loading matched expectations for the most part. Principal stresses at both wheels are similar, which signifies that stress is spread well across both wheels and a single wheel is not maintaining the majority of the reaction forces. The highest occurs at the sprocket which is the main component transferring power from the engine to the axle hence it is expected to be under a large load. Nonetheless, the next calculations were that of safety factors, which would then allow material selection to commence.

Safety Factors Calculations & Material Selection

The safety factor and hence the maximum allowable stress was determined using equation (7) which is based on yield stress:

$$Allowable\ Stress = \frac{Yield\ Strength}{N_y}, where\ N_y = bcdk \tag{7}$$

where; b = fatigue factor, c = shock factor, d = factor of safety & k = stress concentration factor

Fatigue factor was chosen as 1.5 since the load will vary between compression and tension, shock factor was chosen as 1.5 as there is mitigation such as suspension, however shock will still occur due to impacts. Factor of safety consisted of X and Y, where X measures manufacture and operation and Y measures impact and seriousness of failure. They were both chosen by the Pugsley Method, with realistic expectations maintained. X was chosen with A = very good, B = fair & C = fair as we expect materials and manufacturing to be of utmost quality whereas the engineers do not have control over the loads on the final product – this resulted with X = 1.9. Y was chosen with D = very serious & E = serious. Since this is a performance ATV that uses expensive components, both economic impact and danger to the operator are considered to be high – this resulted with Y = 1.5. Hence, factor of safety was 2.85. Stress concentration factors were also chosen loosely based on the ratio of step-up diameter to step down. These were D/d = 1.03 from node 1 to 2 and D/d = 1.1 from node 2 to 3. These values of D/d were rounded up/down from the actual ratios of our selected axle diameters. To have smoother transitions between step-ups, these values of D/d would be better represented and directly used to form new diameters i.e. D(node 2) = 1.03*D(node 1) & D(node 3) = 1.10*D(node 2). These also gave corresponding values which were substituted into a given formula to find K at each node. These values of K and the resulting safety factors are presented below in Table 3C:

Table 3C: Stress concentration and safety factors

Safety Factors	K	1.870	1.645	1.000	1.000	1.645	1.870
	S.F	11.992	10.549	6.413	6.413	10.549	11.992

The safety factors at nodes 1 & 2 are very high, indicating less margin for error and a more expensive region of the axle. They are also higher as there are step ups between these nodes, whereas the lower safety factors at nodes 3 & 4 are along an axle region of constant diameter, where there are only minor grooves for fixings. Overall, having high safety factors limits the permissible stresses at each node by a large proportion, hence reducing material selection to only the best materials.

A total of four materials were used and compared against each other based on stresses and allowable stresses at each node. The materials were; low, medium & high carbon steel and stainless steel. The safety factors ruled out low and medium carbon steel as these were either sub-par or broke even with the stresses in the axle. Table 3D shows the final two materials and their allowable stresses at each node:

Table 3D: Material selection

High Carbon Steel	Yield	MPa	490.00	490.00	490.00	490.00	490.00	490.00
	Shear Yield	MPa	163.33	163.33	163.33	163.33	163.33	163.33
	Allowable	MPa	40.86	46.45	76.41	76.41	46.45	40.86
Duplex Stainless Steel SAF 2205	Yield	MPa	440.00	440.00	440.00	440.00	440.00	440.00
	Shear Yield	MPa	146.67	146.67	146.67	146.67	146.67	146.67
	Allowable	MPa	36.69	41.71	68.62	68.62	41.71	36.69

Although both materials suffice the stresses calculated in the axle and support the maximum stresses, the allowable strength offered by High Carbon Steel is approximately 10% greater at all nodes, hence this is the material that was chosen for the axle. Given the priority of the ATV was performance, it is more sensical to select both a stronger and more valuable material, regardless of the corrosion resistance properties of stainless steel.

Transmission and Bearing Selection

Bearings

Choice of bearing**:** Duplex angular contact ball bearings

This choice of bearing withstands radial, combined, and axial loads in both directions well. It can handle high axle speeds and function at high accuracy all the time. It is also immune to vibration and impact caused by the ATV and therefore possesses good rigidity.

Values:

- Loadings: Radial load (F_r) = 14.3 kN | Estimated axial load (F_a) = 7.15 kN
  Radial load was calculated using the calculated horizontal load and the axial load is an assumption which uses an estimation of half the magnitude of radial load.
- Rotation rate: 266.8 rpm
- Operation environment: Pronounced shock loads | Safety factor (S_0) = 1.5
- Size: Shaft diameter is 70mm
- Life/Reliability: 25,000 hours at 266.8 rpm = 400,200,000 cycles
  Estimated total number of hours the axle will be at work: 25,000 hours.

Calculation:

Bearing life equation: $L_{10} = \left(\frac{C}{P}\right)^p$ (8)

Where: L_{10} = No. of cycles bearing should work (in millions) | C = Basic dynamic load rating (kN)

P = Equivalent dynamic bearing load (kN) | p = exponent specific to ball bearings (3)

Equivalent static bearing load: $P_0 = F_r + Y_0 \cdot F_a = 14.3 + (0.52 \text{ x } 7.15) = 18.018$ kN (9)

Where: F_r = Radial load (kN) | Y_0 = double-row angular ball bearing constant (0.52) | F_a = axial load (kN)

Basic static load rating: $C_0 = S_0 \cdot P_0 = 1.5 \text{ x } 18.018 = 27.027$ kN (10)

Where: S_0 = safety factor for pronounced shock loads (1.5) | P_0 = equivalent static bearing load (kN)

Equivalent dynamic bearing load: Since $f_0 = \frac{F_a}{C_0} \leq e$; $\therefore P = F_r + Y_1 \cdot F_a$ (11)

$\frac{7.15}{27.027} \leq 0.8$; $\therefore P = 14.3 + (0.78 \text{ x } 7.15) = 19.877$ kN

Where: f_0 = calculation factor | e = calculation factor for 32 A bearing series (0.8) | P = equivalent dynamic bearing load (kN) | Y_1 = calculation factor (0.78)

Substituting all values into equation (8): $L_{10} = \left(\frac{C}{P}\right)^p \rightarrow 400.2 = \left(\frac{C}{19.877}\right)^3$

Result

$\therefore$ Basic dynamic load rating (C) = 146.5 kN

Choice of bearing based on load rating C: *3314 A with 70mm bore diameter 150mm outer diameter.

(SKF Group, 2016)

Justifications

The duplex angular contact ball bearings were chosen based on its ability to handle loads in multiple directions more reliably than other bearings available. An expensive and high-performance bearing was chosen due to the unpredictable and high-impact environment of the axle since it's fixed onto a performance ATV. Equation 8 was then used to calculate the basic dynamic load rating (C) using calculated values such as rotation rate, radial loads, etc. Equation 9 to 11 was used from the SKF catalogue for bearings to calculate all the variables in the bearing life equation. An iteration that was conducted was the change in axle diameter where the bearings were fixed. Initially the chosen diameter was 35mm however the calculated basic dynamic load rating is 146.5 kN which requires the bearings to have a bore diameter of 70mm for the chosen *3314 A bearing. Therefore, the axle diameter at the bearings area was iterated from 35mm to 70mm to allow for a rigid fixing of both bearings.

Transmission

Type of chain: Simplex chain

Chain pitch: 15.875mm

Sprocket ratio: 3:1

Two sprockets were chosen that have a 3:1 ratio where one sprocket has 13 teeth and the other has 38. The following decisions are based off the type of chain, sprocket ratio and chain pitch which were all provided in the design brief.

Selection power: $P = P_{trans} \cdot f_1 \cdot f_2 = 30 \text{ x } 2.1 \text{ x } \frac{19}{13} = 92.1$ kW (12)

$f_2 = \frac{19}{No.\ of\ teeth\ on\ driving} = \frac{19}{13}$ (13)

Where: P_{trans} = transmission power (30 kW) | f_1 = safety tooth factor for heavy driven machine shocks with moderate driver shocks (2.1) | f_2 = safety factor

Details

Lubrication: Oilbath type 3
Maximum tensile strength: 22 kN
Chain selection: 10B-1
No. of teeth: 38

Pitch Circle Diameter (PCD): 192.24 mm
Top Diameter: 199.1 mm
Bore Min/Max: 20mm/60mm
Boss Diameter: 100 mm

(Renold, 2021)

Justifications

Multiple iterations of sprocket selection process took place as the initial assumptions had to be changed in order to create the optimal sprocket design. Since the target ratio was 3, the initial sprocket selection went for an exact ratio of 3 – with 19 teeth on the driving sprocket and 57 on the driven. Although this was the perfect ratio of 3, the caveat was differing materials of cast iron for the driven and steel for the driving. This would not be practical due to the differing material properties between the axle, chain, and sprockets. The hardness of each of these materials being different would cause varying wear in all components, being more unpredictable and hence not being able to accurately model it. This is why the final selection was to go with two steel sprockets of 13 and 38 teeth on the driving and driven respectively. These were the closest to the target ratio of 3 in the given section of the catalogue. The rim diameter was also estimated to allow a safe selection of the driven sprocket – average rims for ATV's were approximately 12-13 inches or 304-330mm, with the tires adding a lot more thickness. The difference between sprocket and wheel diameters must be very large in order to prevent damage on ground impacts hence a PCD of 192mm is beyond half the diameter of what the full wheel would be, therefore resulting in a safe selection. Finally, the first iteration included a triplex chain, which was then changed to a simplex chain due to being far too complex for a simple ATV. Triplex chains are used in the railway industry, quarrying, agriculture vehicles and other heavy machinery, which is far beyond the scale of a simple performance ATV.

Fixings and Geometry

Wheel Hub

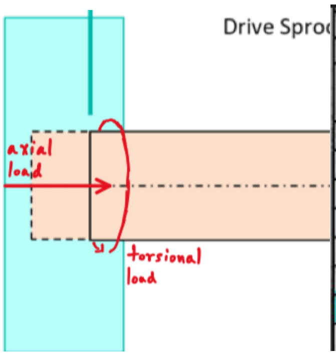

Figure 4A: FBD with acting loads

Table 4A: Tolerancing
[57.5mm shaft diameter]

Nominal sizes		Tolerance	
Over	To	H7	n6
mm	mm	0.001mm	0.001mm
–	3	+10 0	+10 +4
3	6	+12 0	+16 +8
6	10	+15 0	+19 +10
10	18	+18 0	+23 +12
18	30	+21 0	+28 +15
30	40	+25 0	+33 +17
40	50		
50	65	+30 0	+39 +20
65	80		

Dimensioning of the hub recess:

From table, H7 tolerances = $^{+30}_{0}$

∴ Maximum limit of size = 57.5 + 0.030 = 57.530

Dimensioning of the hub:

From table, n6 tolerances = $^{+39}_{+20}$

∴ Maximum limit of size = 57.5 + 0.039 = 57.539

& Minimum limit of size = 57.5 + 0.020 = 57.520

Loads: 2 degrees of freedom with axial load acting on the hub from horizontal impact (if any) on the wheels and torsional load from the rotation of each wheel on the hub.

Fixing method and stress: Taper locks fitted on the axle inside the hub of each wheel. Although it is expensive, this fixing method is very strong for high load applications, it can also easily be assembled and disassembled should the wheel need changing. The tolerancing chosen is a transition fit with drive. This fixing method is heavy duty so that the wheel does not experience any unwanted vibration, it also restricts any unwanted axial and rotational vibration. There are no notches in the shaft that need to be cut to fit the taper lock and therefore doesn't increase stress concentrations significantly.

Sprocket

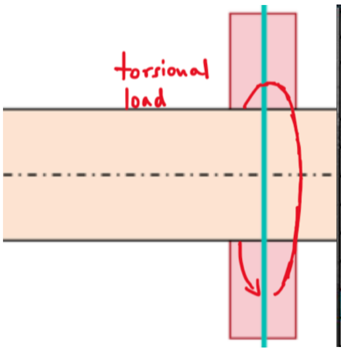

Figure 4B: FBD with acting loads

Table 4B: Tolerancing [60mm shaft diameter]

Nominal sizes		Tolerance	
Over	To	H7	n6
mm	mm	0.001mm	0.001mm
–	3	+10 0	+10 +4
3	6	+12 0	+16 +8
6	10	+15 0	+19 +10
10	18	+18 0	+23 +12
18	30	+21 0	+28 +15
30	40	+25 0	+33 +17
40	50		
50	65	+30 0	+39 +20
65	80		

Dimensioning of the sprocket recess:

From table, H7 tolerances = $^{+30}_{0}$

∴ Maximum limit of size = 60 + 0.030 = 60.030

Dimensioning of the sprocket:

From table, n6 tolerances = $^{+39}_{+20}$

∴ Maximum limit of size = 60 + 0.039 = 60.039

& Minimum limit of size = 60 + 0.020 = 60.020

Loads: 1 degree of freedom with torsional load acting on the sprocket from the rotation of the chain.

Fixing method and stress: Circlips are fitted on each side of the sprocket which fits in a groove. Circlips are cheap and very easy to assemble and disassemble should the sprocket need replacing. The circlips are pretty strong but not very, this is tolerable since it's a simplex chain, a duplex or triplex chain would need a stronger fixing method. The tolerancing chosen is a transition fit with drive. This is relatively tight to prevent the circlips from moving in their groove, however this tolerance still allows for the circlips to be easily replaced. The shaft will experience quite a significant increase in stress concentration due to the grooves cut into the shaft.

Bearings

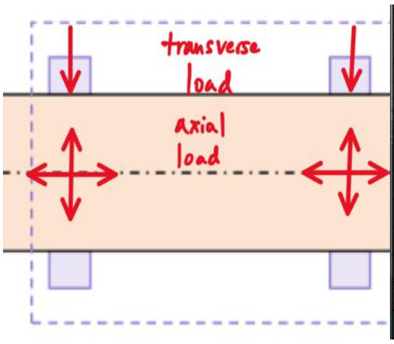

Figure 4C: FBD with acting loads

Table 4C: Tolerancing [70mm shaft diameter]

Nominal sizes		Tolerance		Tolerance	
Over	To	H7	k6	H7	g6
mm	mm	0.001mm	0.001mm	0.001mm	0.001mm
–	3	+10 0	+6 +0	+10 0	-2 -8
3	6	+12 0	+9 +1	+12 0	-4 -12
6	10	+15 0	+10 +1	+15 0	-5 -14
10	18	+18 0	+12 +1	+18 0	-6 -17
18	30	+21 0	+15 +2	+21 0	-7 -20
30	40	+25 0	+18 +2	+25 0	-9 -25
40	50				
50	65	+30 0	+21 +2	+30 0	-10 -29
65	80				

Dimensioning of the bearing recess:

From table, H7 tolerances = $^{+30}_{0}$

∴ Maximum limit of size = 70 + 0.030 = 70.030

& Minimum limit of size = 70 + 0.000 = 70.000

Dimensioning of the left bearing:

From table, k6 tolerances = $^{+21}_{+2}$

∴ Maximum limit of size = 70 + 0.021 = 70.021

& Minimum limit of size = 70 + 0.002 = 70.002

Dimensioning of the right bearing:

From table, g6 tolerances = $^{-10}_{-29}$

∴ Maximum limit of size = 70 – 0.010 = 69.990

& Minimum limit of size = 70 – 0.029 = 69.971

Loads: 2 degrees of freedom with axial and radial loads acting on both bearing in multiple directions. There is also a transverse load acting on the top of both bearings from the weight of the rider, suspension, etc.

Fixing method and stress: Circlips are fitted on each side of both bearings. The left bearing has circlips that hold the bearing firmly in place, the right bearing has circlips fitted in a way that allows the right bearing to experience some horizontal motion. Circlips are cheap and very easy to assemble and disassemble should the bearings need replacing due to wear. The circlips have a fair enough strength which is tolerable since the double row angular ball bearings can withstand high loads. The tolerancing for the left bearing is a transition fit with easy keying and the right bearing is a clearance fit with close running. The left bearing is fixed in place with no horizontal movement while the right bearing has a relatively looser fit so that it can move horizontally between its circlips. The grooves cut into the shaft for the four circlips with cause a significant increase in stress concentration.

Disc Brake

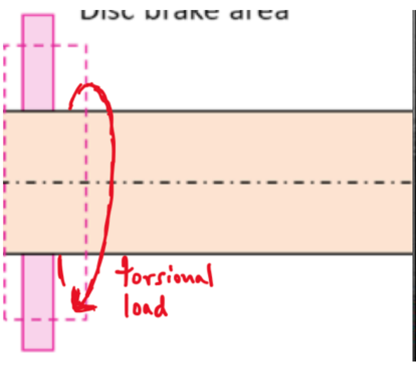

Figure 4D: FBD with acting loads

Table 4D: Tolerancing [60mm shaft diameter]

Nominal sizes		Tolerance	
Over	To	H7	n6
mm	mm	0.001mm	0.001mm
–	3	+10 0	+10 +4
3	6	+12 0	+16 +8
6	10	+15 0	+19 +10
10	18	+18 0	+23 +12
18	30	+21 0	+28 +15
30	40	+25 0	+33 +17
40	50		
50	65	+30 0	+39 +20
65	80		

Dimensioning of the disc brake recess:

From table, H7 tolerances = $\begin{matrix}+30\\0\end{matrix}$

∴ Maximum limit of size = 60 + 0.030 = 60.030

& Minimum limit of size = 60 + 0.000 = 60.000

Dimensioning of the disc brake:

From table, n6 tolerances = $\begin{matrix}+39\\+20\end{matrix}$

∴ Maximum limit of size = 60 + 0.039 = 60.039

& Minimum limit of size = 60 + 0.020 = 60.020

Loads: 1 degree of freedom with torsional load acting from the pad of the disc brake that stops the rotational of the wheel.

Fixing method and stress: A locking ring is used to secure the disc brake in place at either side. Although it is expensive, this fixing method can handle high load and can easily be assembled and disassembled should the brake pad need replacing. The tolerancing for the disc brake is a transition fit with drive. This allows the disc brake to be replaced, if need be, but also holds the brake well in place with no horizontal movement. There is no major increase in stress concentration in the shaft as no grooves need to be cut in however the locking ring itself has high a high stress concentration where both semi-circles connect.

Design Evaluation

Now that we have finalised our design with the geometry of all the components completely determined, we can evaluate the criteria in our PDS (Table 2A) against our design. When looking at each criterion, we will conclude whether or not we have successfully upheld our requirements.

Performance

In our PDS, each one of our performance requirements has high importance. Therefore, it is integral that each requirement is achieved, as we have also previously mentioned in our introduction that we have opted towards a performance-specific package. Overall, each requirement for performance has been satisfied. With the material selection of high carbon steel for our shaft along with the supporting stress calculations (maximum allowable stress of 76.41 MPa), the axle was able to completely support the combined mass of the ATV and rider. By selecting duplex angular contact bearings, the radial loads were calculated to have a basic static load rating of 27.027 kN, which compared to our estimated radial load through the shaft, was calculated to be 14.3 kN. Thus, allowing the bearings to hold significantly more load than it requires, satisfying our second requirement. The sprocket was chosen to have a ratio of 3 between the driver and the driven with 13 teeth and 38 teeth respectively. An agreed chain pitch of 15.875 mm was met, satisfying the performance requirements for the sprocket.

Size & Weight

In our PDS, the size and weight category had the most requirements we had to comply with. As we have finalised our geometry, the bearing on the side of the brake disk has a small gap between the bearings and the circlips fixing to allow for lateral movement and any deviation in size due to thermal expansion. The bearings to the side of the sprocket, however, is fixed rigidly by the circlips to prevent any sort of movement. Furthermore, designing the shaft so that the inner diameters of the driven sprocket and the disk brake match exactly with the diameters of the shaft at their respective situations, enable those components to fit firmly to prevent any form of vibrations. Requirement 10 states that our performance axle must weigh more than that of a stock axle. The mass of our axle can be calculated using $\rho = m/V$. The selection of high carbon steel gives us an average density of $4355.0\ kg/m^3$

(MatWeb, n.d.) and using Fusion 360, the volume was determined to be $2.612 \times 10^{-3} m^3$. Therefore, the mass of our shaft was calculated to be 9.407 kg, when compared to a stock ATV shaft (Ningbo Jun An Group Inc., 2021) with a length of $0.95\ m$ and an average diameter of $0.0347\ m$, the volume of the stock shaft is lower at $8.984\ m^3$. The price of the stock rear axle is about $50, so we can assume that the material used is cheaper and thus has a lower structural strength and density than high carbon steel. Therefore, the overall mass of our performance axle is higher, and that requirement is satisfied.

Environment

When choosing the material for our axle, we needed to ensure that the axle will be resistant to water in the case of riding during the rain and also resistant to mud and dirt. Due to the high amounts of carbon in the material, the corrosion resistance is appropriate for the terrain during the operation of the ATV. (Beck, 2020)

Materials

When comparing the strength between the selected high carbon steel material for our axle, between aluminium which is conventionally used in stock ATV rear axles, we can analyse the stresses each material withstands. Initially, we can compare the hardness between each material from data produced during the Vickers hardness test. HCS has a hardness of 182 – 748 (MatWeb, n.d.) whereas aluminium has a reading of exactly 15 (MatWeb, n.d.). We can easily see that HCS has a greater resistance to plastic deformation upon impacts, which is advantageous for an ATV. Overall, the materials requirements have been settled.

Mechanism

This section of requirements primarily covers fixings. For our design of the shaft, we have implemented a taper lock for our wheel hub, which provides a very strong fitting for high load applications. Both the bearings and the sprocket were secured using a circlip, which is relatively strong and enough for a simplex chain and a double row angular contact bearing. A locking ring was used the secure the disk brake in place. This fixing method provides a high load tolerance with no lateral movement. Each requirement for the mechanism was satisfied, however, due to cost considerations, we were only limited to circlips fittings for the bearings and sprocket.

Reliability/Quality

When designing the axle, we took into consideration that the shaft should last for a service interval of 300 hours. By carefully selecting the optimal bearing type, axle material, fixings, and geometry, we can confidently say with stringent testing and supporting calculations that the axle will comfortable last 300 hours of constant use without failure.

Safety

When designing the shaft, we incorporated a designated place for each of the fixings to slot into to ensure that during installation and operation, the axle is safe to use up to the mentioned 300 service hours. All factors including the material, fixings and tolerances ensure complete safety for the user during use.

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