
**REDESIGN AN AMERICAN PTV GO-KART WITH
AVAILABLE METRIC COMPONENTS**



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ABSTRACT

An American “Personal Tracked Vehicle (PTV) Go Kart Build Plans” was purchased. The build plans used imperial components and the drawings were drawn in inches. Not all the required information was available in the build plans. Therefore, some components needed to be designed and some of the required components are not available in the local market. In order to build the PTV go kart, components needed to be imported from the United States of America. Importing components proved to be more expensive. The most economical option was to redesign with available metric components. Thus, this redesign project was carried out.

As part of this project, a suitable chain drive, belt drive, shafts, bearings and disc brake were redesigned. Suppliers' catalogues were searched for available components and then the suitable components were selected. Structural analysis was carried out to select suitable tubes for the frame. A new 3D model was drawn using metric SI units. Using that 3D model, potential problems were investigated and solutions were found. Moreover, possible improvements to safety and practical uses of the kart were investigated.

A new version of the PTV go kart was designed successfully using metric components. The selected components were available in the local market and easy to locate. The selected components were capable of supporting the maximum loads and this was proven by theoretical calculations.

Keywords Mechanical design, Go kart, Chain drive, Belt drive, Shaft, Disc brake

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1 INTRODUCTION

Engineers are continuously working on ways to improve existing designs and therefore designing is a key area in engineering. For a business to grow and to expand, it is important to develop new and innovative solutions. Redesigning of existing designs and their continuous development is vital because of the competitive market. Hence, a lot of time and effort is allocated for redesign and further development of the existing designs. Thus, innovative new design solutions are frequently discovered by design engineers.

This thesis is based on the prototype PTV (personal tracked vehicle) go kart build plans. Using the information from the purchased PTV build plans, a similar PTV go kart will be redesigned. The newly designed PTV go kart will be made using suitable metric components available in the local market. As part of this project, a suitable chain drive, belt drive, shafts, bearings and disc brake will be redesigned. Using the design calculations, suitable components will be selected from suppliers' catalogues. Structural analysis will be carried out to select suitable tubes for the frame. A 3D model of the PTV go kart will be drawn using SI units in millimetres. The 3D model can be used to analyse any potential problems and to see if the components will fit together properly.

This project tests the knowledge and abilities of a fourth-year mechanical engineering student in a real life situation. Moreover, the designer will have an opportunity to improve mechanical design knowledge, further establish existing skills and also have an opportunity to learn new design methods and techniques. The designer will consolidate problem solving skills in a real life design situation. This thesis only focuses on design and theoretical analysis of the PTV go kart. The manufacturing process might be carried out later on. As this is a practical design project, suitable components will be found by researching the suppliers' catalogues. By doing so, the researcher will familiarize himself with the suppliers and the products available in the market. Design calculations will be carried out to check if components are capable of supporting the maximum loads without failing. The skills developed during this project will be useful in working life as an engineer.

Safety is one of the major issues related to the PTV go kart, thus, ways to improve the safety will be investigated. Moreover, instead of using the PTV go kart just for recreational purposes, potential everyday life uses will be investigated.

2 BACKGROUND INFORMATION

The main difference between a normal go kart and a PTV go kart is that the PTV go kart has tracks instead of wheels and the wheels are located inside the track to guide it. The prototype PTV go kart is shown in Figure 2.1. The wheels in the front are used as the driving wheels and power is transfer to the tracks by friction between the driving wheel and the track surfaces. (PTV Build Plans 2009, 4.)

2.1 PTV Go Kart

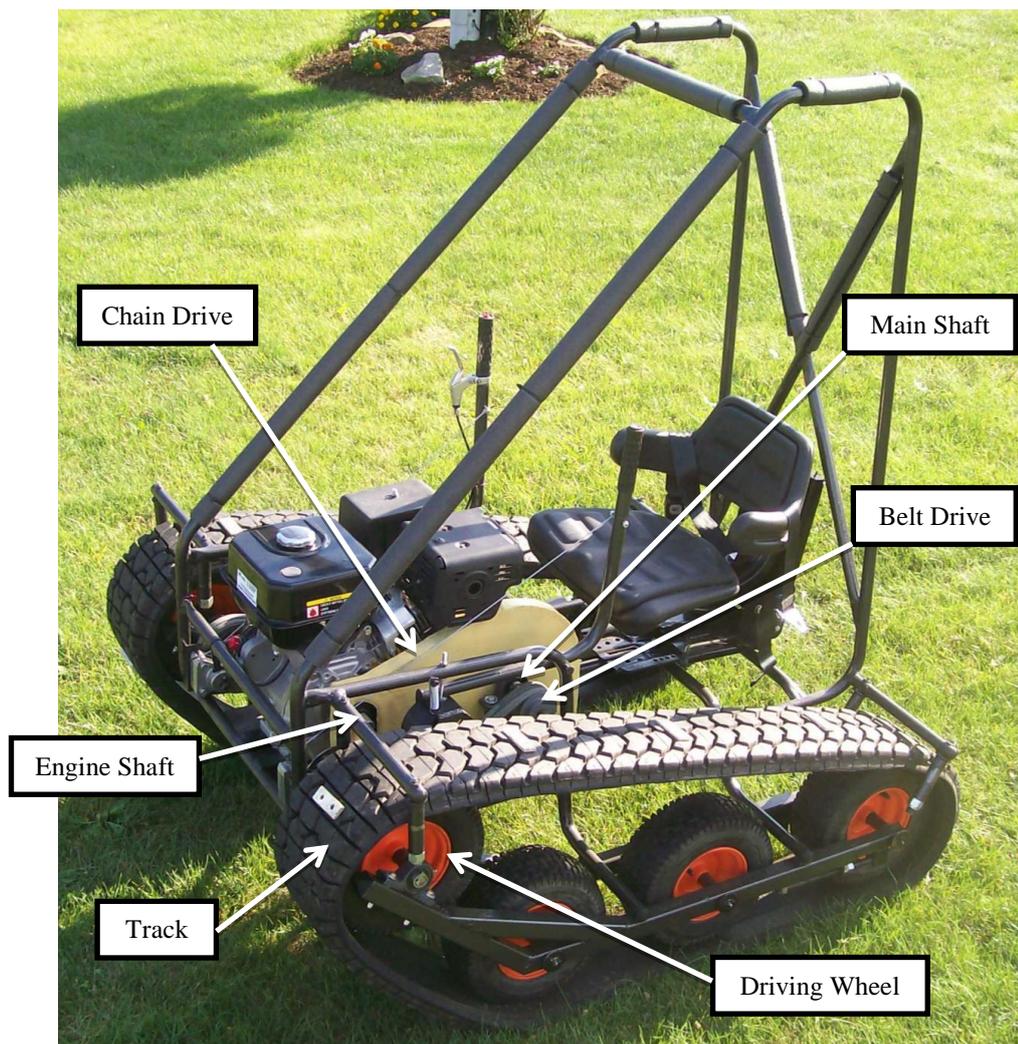


Figure 2.1 PTV go kart (PTV Build Plans 2009, 1)

The power from the engine is transferred to the main shaft by a chain drive system. The main shaft is clearly visible in the Figure 2.2. The main shaft transfers the power to the pulleys, which are mounted at both ends of the shaft. With the use of belt drives, the power is transferred from the main shaft to the driving wheels. The driving wheels transfer the power to the tracks using the friction between the two surfaces (friction drive). There will be power losses by the chain, belt and friction drives. The major power loss will occur by the friction drive. (PTV Build Plans 2009.)

2.1.1 PTV Controls

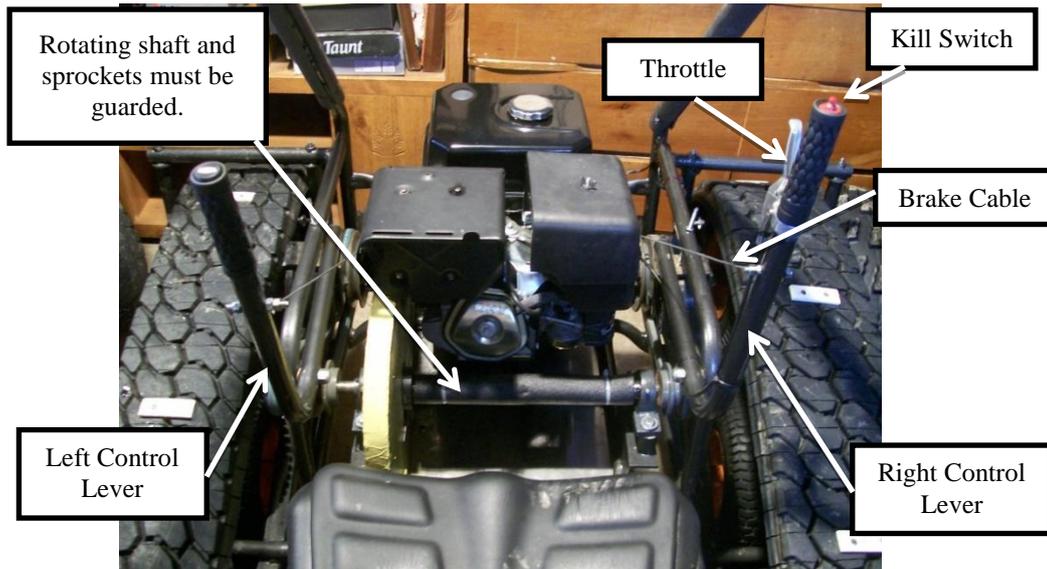


Figure 2.2 PTV controls (PTV Build Plans 2009, 15)

The controls of the PTV go kart are quite different compared to the normal go karts. The PTV go kart does not have a steering wheel. Two control levers are used instead of a steering wheel. Engine speed can be controlled using the throttle and pressing on the kill switch to stop the engine. When the engine is started, the main shaft will also start to rotate. There are two small pulleys mounted on either end of the main shaft and they will start to rotate. At the normal position of the control levers, the V-belts are loose on the belt drive systems. At this point, no power will be transferred to the driving wheels, thus, the kart is stationary. To drive the kart forward, both control levers need to be pushed forward. Kart speed can be controlled by the amount of pushing force applied to the levers. When driving forward, slight turns can be made by adjusting the applied force to the each control lever. For example, a slight turn to the left can be made by applying more force to the right lever and reducing the force on the left lever. The kart can be stopped by pulling back both control levers, which will apply the brake to the both driving wheels. (PTV Build Plans 2009, 15.)

In order to make a sharp turn at a slow speed, one control lever will be pulled back to apply brakes on that side track and the other control lever will be pushed forward to transfer the power to that side track. This causes the kart to turn using the locked side track as the centre of rotation. For example, a sharp turn to the left at a slow speed can be made by pulling back the left lever to lock the left side track and pushing forward the right lever to transfer power to the right side track. So the right side track will drive around the left side track using the left side track as the centre of rotation. It is extremely dangerous to make sharp turns when driving at a high speed. The build plans strongly suggests not making sharp turns at high speed as it could cause the kart to flip over and cause serious injuries or even death. All the rotating surfaces must be guarded properly to prevent accidents from occurring. Especially the main shaft and sprockets near the driver's legs must be guarded. (PTV Build Plans 2009, 15.)

2.1.2 PTV Track

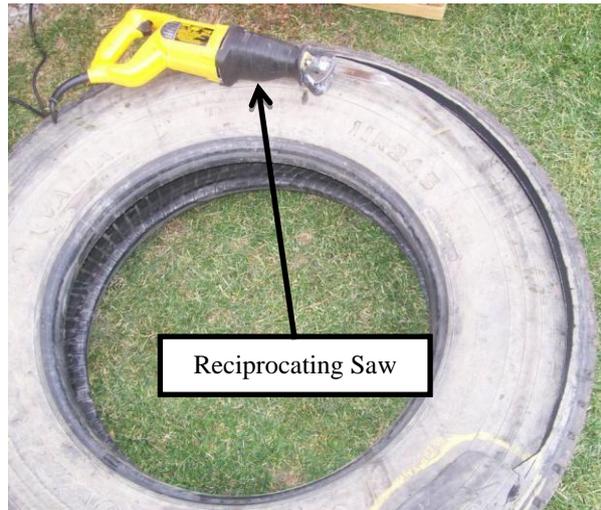


Figure 2.3 Tire sidewall cutting (PTV Build Plans 2009, 9)

The tracks of the PTV go kart are made by cutting off the sidewalls of approximately 42 inch diameter tires. Tires with damaged sidewalls cannot be re-treaded so these tires can be obtained free of charge. The sidewalls are cut off using a reciprocating saw. When sidewalls are sawed off, there might be sharp metal wires on the sides. So those sharp edge wires should be filed off to prevent any personal injuries from occurring. (PTV Build Plans 2009, 9.)



Figure 2.4 Track (PTV Build Plans 2009, 9)

The tires are very stiff so, cross slits are cut to reduce the stiffness and to make the track more flexible to bend. There are metal wires inside the tire. Those metal wires should not be cut as they will provide the required strength to the track. Whilst driving the PTV prototype, the driver may notice too much grip between tracks and the road; and as a result it will be difficult to turn the kart at a slow speed. Hence, anti-traction strips were introduced to reduce the grip. However, these strips increase the traction when driving in grass or wet conditions. (PTV Build Plans 2009, 9.)

2.1.3 PTV Driving Wheel and Track Tension



Figure 2.5 Driving wheel (PTV Build Plans 2009, 9)

Tubeless tires were used on the prototype. Metal screws are fitted in to the driving wheel to increase the friction between the driving wheel and the track. A tire sealer was used to stop the air leakage by the screws. These screws improve the power transmission from the driving wheel to the track and increase the kart performance. Also, this reduces the slipping affect whilst driving in wet conditions. (PTV Build Plans 2009, 9.)



Figure 2.6 Track tension screws (PTV Build Plans 2009, 11)

It is important to get the right track tension to get the best possible performance out of the kart. The track tension can be adjusted by the screws on the both sides of the wheel. The screws need to be tightened evenly to achieve the right track tension. (PTV Build Plans 2009, 11.)

2.1.4 PTV Seat

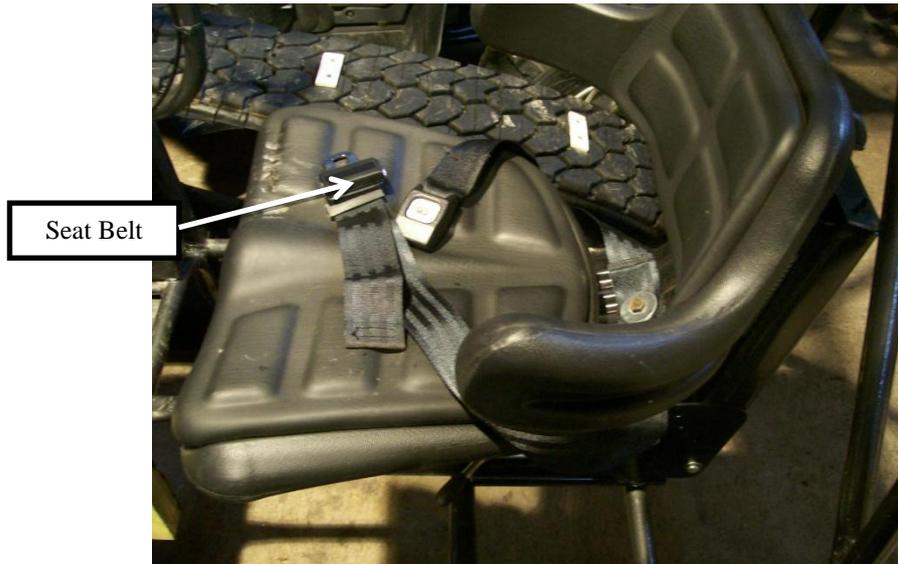


Figure 2.7 Seat (PTV Build Plans 2009, 13.)

As in Figure 2.7, a similar type of machinery seat can be used. The seat is to be mounted securely on the frame and a seat belt must be mounted with the seat. As safety is vital, the seat belt must be capable of holding the driver's weight in case of an accident. (PTV Build Plans 2009, 13.)

2.1.5 PTV Disc Brake

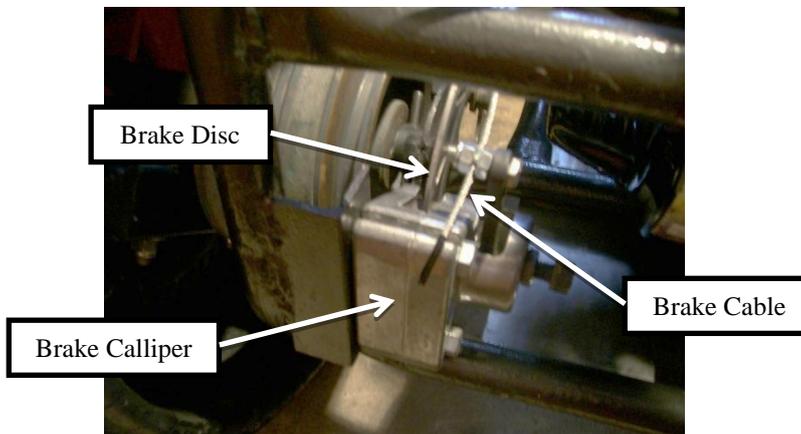


Figure 2.8 Disc brake (PTV Build Plans 2009, 12.)

The PTV go kart uses disc brakes. The disc is mounted on the drive wheel shaft, near the larger size pulley. The brake cable is connected to the control lever. At the normal position of the control lever, the cable is adjusted to be slightly tight so the brake calliper is breaking the disc. This is done to stop the kart from rolling in its' stationary position. When the control lever pushes forward, the calliper will loosen and let the disc to freely rotate. When the control lever pulls back, the brake calliper will clamp the disc to stop it from rotating. The brakes are used to stop the kart and to turn the kart. (PTV Build Plans 2009, 12.)

2.2 Reasons for Redesign

The main reason for redesign is that the components used in the PTV build plans cannot be found in the European market. Also, plans are drawn in inches and not all the required information is available. As some components are needed to be designed, designing the whole PTV go kart with available components is a practical and sensible decision. Also it would be economical as importing components from the United States proved to be expensive. The redesign will be made using available metric components so that finding any spare parts in the future will not be a problem.

2.3 Components Selection

Most of the components recommended in this thesis are selected from well recognised manufactures and are available in most of the outlets. Similar, more economical, components may be available from other manufacturers so it is up to the builder to decide which manufacturers' components should be used. Components such as an engine can be found for a cheaper price from used parts sellers. It is strongly recommended to use similar size components as use of different size components would affect the design calculations. If the builder decided to use a different size of components, it is their responsibility to redo the design calculations and to check the capabilities of the selected components.

2.4 CAD Modelling

CAD modelling is a major part of design engineering at present. There is number of different modelling software available in the market. For this thesis, AutoCAD and Pro Engineer (Creo Elements Pro) software is used. AutoCAD is typically used for 2D drawings and Pro Engineer is typically used for 3D modelling. 3D modelling and 3D assembling provides an opportunity to investigate potential problems which could occur during the actual assembly process. This way the problems can be found, then fixed and checked so that everything fits together adequately before the manufacturing begins. This is one of the main reasons that 3D modelling is popular.

3 REDESIGN PROCESS

The word redesign means that an original design of an object will be altered to make a new design to meet the new requirements. For this thesis, the original concept is taken from “Personal Tracked Vehicle (PTV) Go Kart Build Plans”. A redesign of the PTV go kart is carried out because most of the recommended components by PTV Build Plans could not be found in the local market. The purpose of the redesign is to build a similar PTV go kart with available metric components.

A mechanical engineering design can be done in two ways. One method is by using the required output and select components to meet the requirements. For example, if output of 400rpm required, then the input components are selected to get this required output. The other method is to select an input like a motor or engine and then select components capable of transmitting input power. In this method, the output is dependent on the selected components so changing the size of components will vary the output value. This thesis used the second method of design, thus an engine will be selected first.

3.1 Engine Selection

Commonly used and available go kart engines are searched for in the local market. Honda is a popular engine brand and available in most of the local shops. Thus, a Honda engine is selected. Using an engine from a less popular brand might be economical but the availability is not certain. Also buying a used engine is more economical than buying a new one.

As the PTV build plans used a 13HP horizontal shaft engine, a 13HP Honda GX390 is selected. This engine is commonly used in go karts. However, when selecting a chain, it is discovered that the standard chains are not capable of transfer the power safely (refer to Figure A1.1, Appendix 1/2). Therefore, a lower power engine is chosen for this project and a Honda GX200 6.5HP engine is selected. This engine is also commonly used in go karts and the required power can be transmitted with a relatively small chain sprocket.



Figure 3.1 Honda GX200 (Honda 2011)

3.2 Chain Drive Design

A suitable chain needs to be selected, which should safely transfer the engine's power to the main shaft.

- **08B-1** with **12.7mm** pitch chain is selected (refer to Appendix 1/2).

A suitable smaller sprocket capable of transmitting the required power needs to be selected.

- **23** teeth smaller sprocket is selected (refer to Appendix 1/3).

The PTV build plans used only 12 teeth smaller sprocket, which is clearly not capable of safely transmitting a 13HP engine's output power. This will increase the wear and reduce the life of the chain drive. Online research established that most of the racing karts use lower teeth smaller sprocket to get a higher speed ratio but they change the smaller sprocket and the chain before every race to prevent failures from occurring (Forchetto 2008).

The PTV build plans used a 70 teeth larger sprocket. In order to get a similar speed ratio as the PTV build plans, a 134 teeth larger sprocket is required. However, the PTV go kart has limited space available. If a 134 teeth sprocket is used, then the space available for the driver will be reduced. Therefore, a similar size larger sprocket as in the PTV build plans is selected.

- **70** teeth larger sprocket is selected.

The pitch diameters of the sprockets are calculated (refer to Appendix 1/4):

- **93.27mm** pitch diameter smaller sprocket.
- **283.07mm** pitch diameter larger sprocket.

The chain length and centre distance of the two sprockets are calculated:

- **91** pitches chain length is chosen (refer to Appendix 1/4).
- **265mm** centre distance is chosen (refer to Appendix 1/5).

The arc of contacts between sprockets and the chain are calculated (refer to Appendix 1/5):

- **138⁰** smaller sprocket arc of contact
- **222⁰** larger sprocket arc of contact

Use of a greater angle than 120⁰ for the smaller arc of contact is recommended by Mott (Mott 2006, 291). The calculated arc of contact is greater than 120⁰ thus this condition is met.

Source: GX200 is downloaded from www.3dcontentcentral.com

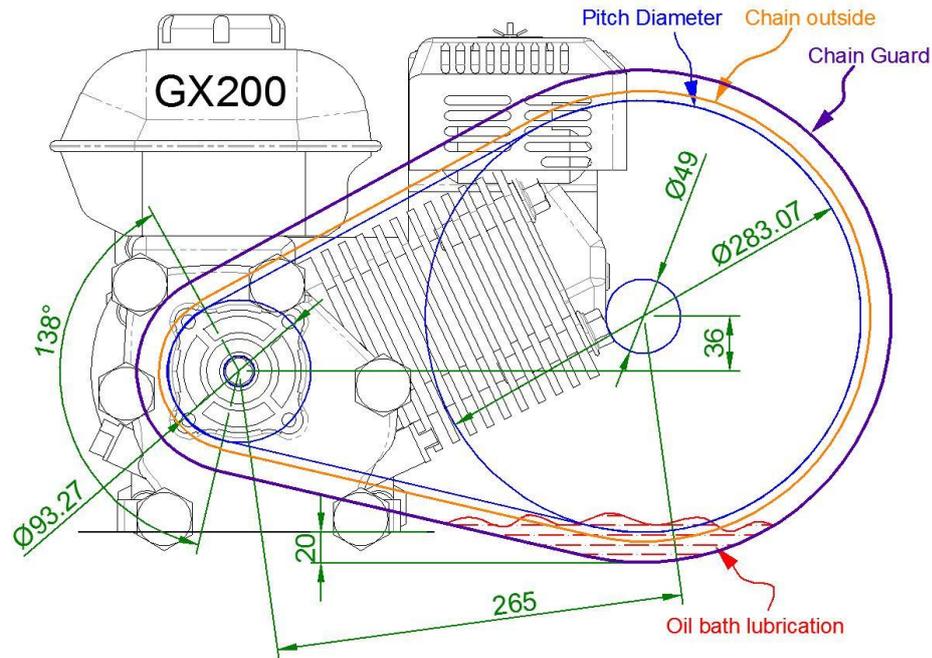


Figure 3.2 Chain drive

The centre of the larger sprocket had to be moved up 36 mm to keep the same clearance between the ground and the bottom of the PTV go kart. The bottom of the chain guard will be 20 mm lower than engine base. This will not reduce the PTV go kart's bottom clearance as the engine is mounted onto 27mm diameter pipes. With the selected centre distance, the maximum main shaft diameter is limited to 49mm. This should be considered when designing the main shaft. Rexnord recommended use of forced feed lubrication (refer to Appendix 1/3). Forced feed lubrication requires a motor and a pump. Oil bath lubrication is decided upon as it is the easiest way to lubricate the chain drive. However, the oil bath lubrication will not be as effective as the forced feed lubrication so the life of the chain drive will be reduced.

3.3 Belt Drive Design

The PTV go kart has two belt drives in either end of the main shaft. Both belt drives are the same. Belt drives are used for transmitting power from the main shaft to the driving wheel. Classic V-belts or Wedge belts can be used for the design. Wedge belts are popular at the moment. For belt drive design, available Australian standards SAA HB6 book is used. Australian standards are made using ISO, BS and AS standards. Also, the Australian standards calculation are in SI units so metric components can be selected using it. SAA HB6 book only has classic V-belt power rating tables. Wedge belts are more height than classic V-belts (SAA HB6 1999, 78). So V-belts have a bigger gap between the bottom of the belt and the pulley groove. Therefore, V-belts are better suited for PTV go karts because the bottom of the V-belt has less chance of touching the base of the pulley groove when force is applied on the V-belts. Thus classic V-belts are the first choice for the belt drive.

A classic V-belt cross-section is needed to be selected to transmit the required power.

- A-section V-belt is selected (refer to Appendix 2/1).

In order to select a suitable pulleys combination, an output speed is required. The output speed of the PTV build plans is approximately 380rpm. Therefore, a similar speed is preferred for the larger pulley.

- 380rpm larger pulley speed
- 1183rpm smaller pulley speed (refer to Appendix 2/1)

Using the speeds of the pulleys, a preferred speed ratio can be found.

- $r = 3.11$ (refer to Appendix 2/2)

Then, the required speed ratio can be found using available combination of pulleys.

Table 3.1 Pulley selection process (refer to Appendix 2/2 for more information)

D_{1A}	D_{2P}	D_{2A}	r_N	P_b	N	L_P	L_A	C
80	249	250	3.13	0.9	4.56	Maximum 3 groove pulley available		
85	265	250	2.94	Different speed ratio than required.				
		280	3.29	Different speed ratio than required.				
90	280	280	3.11	1.13	3.63	1482.52	1510	454.47
95	296	280	2.95	Different speed ratio than required.				
100	311	315	3.15	1.37	2.99	1558.92	1580	451.26
106	330	315	2.97	Different speed ratio than required.				

The heading descriptions of Table 3.1:

- D_{1A} Diameter of the available smaller pulley
- $D_{2P} = rD_1$ Diameter of the preferred larger pulley
- D_{2A} Diameter of the closest available larger pulley
- $r_N = \frac{D_{2A}}{D_{1A}}$ New speed ratio
- P_b Basic V-belt power rating (kW)
- N Numbers of belts required
- L_P Preferred length of the belt
- L_A Length of the closest available belt
- C Actual centre distance

From Table 3.1, 90mm and 280mm pulley combination is selected. This combination has the closest speed ratio required and the 90mm pulley is available with 4 grooves. Another option is the 100mm and 315mm pulley combination but the use of a smaller pulleys combination is preferred as only limited space is available.

- **90mm** smaller pulley pitch diameter
- **280mm** larger pulley pitch diameter

The centre distance of the belt drive system is fixed to 440.57mm by the chain drive system and the driving wheel. To transmit power, approximately four V-belts are required. The four V-belts on the pulleys have approximate width of 62mm. An idler with larger width than 62mm needs to be chosen. Fenner Drives flat belt idler F3B6280 has a width of 74mm and 152.4mm diameter. It is the smallest diameter idler with required width.

- **74mm** width and **152.4mm** diameter idler (Fenner Drives 2008, 8)
- **440.57mm** fixed centre distance (refer to Figure 3.3).

As per Table 3.1, the closest available belt length is 1510mm. The belt should be loose at the normal position of the PTV control levers. It was evident from the analysis that a 1530mm long belt will be better suited with the selected idler (Figure 3.3). If a smaller idler with required width could be found, then the 1510mm long belt can be used. Also, the actual numbers of belts required to transmit power is calculated.

- **1530mm** long **A**-section V-belts
- **4 A**-section v-belts are selected (refer to Appendix 2/3)

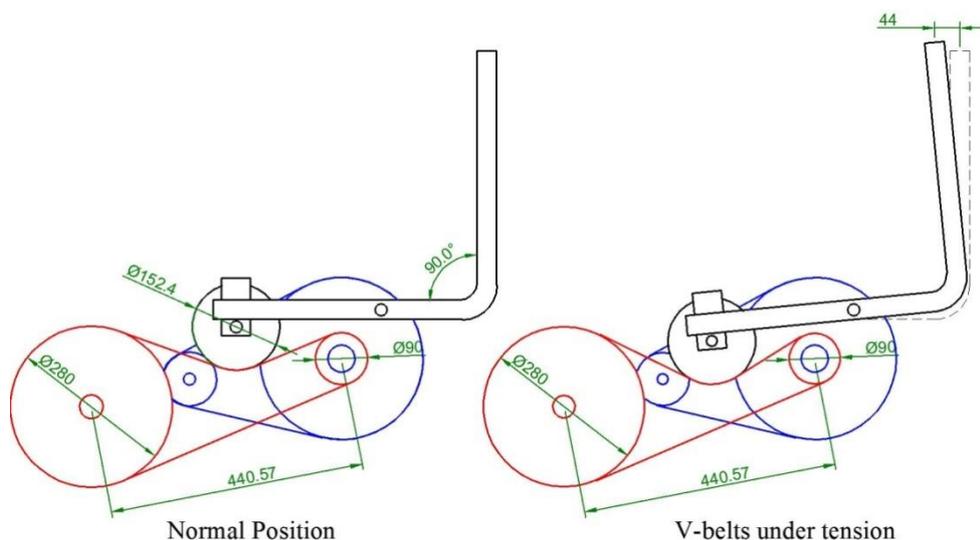


Figure 3.3 V-belt tension analysis

The V-belt pitch length is measured and found that the belt drive system will transmit the maximum possible power when the control lever is pushed forward by 44mm.

3.4 Main Shaft Design

Static and dynamic loadings are required to design a shaft. Analysing the selected components and the PTV build plans, required lengths to mount the components on the main shaft is found. The maximum possible shaft diameter that can be fitted through, without touching the engine is 49mm (refer to Figure 3.2).

3.4.1 Static Loading

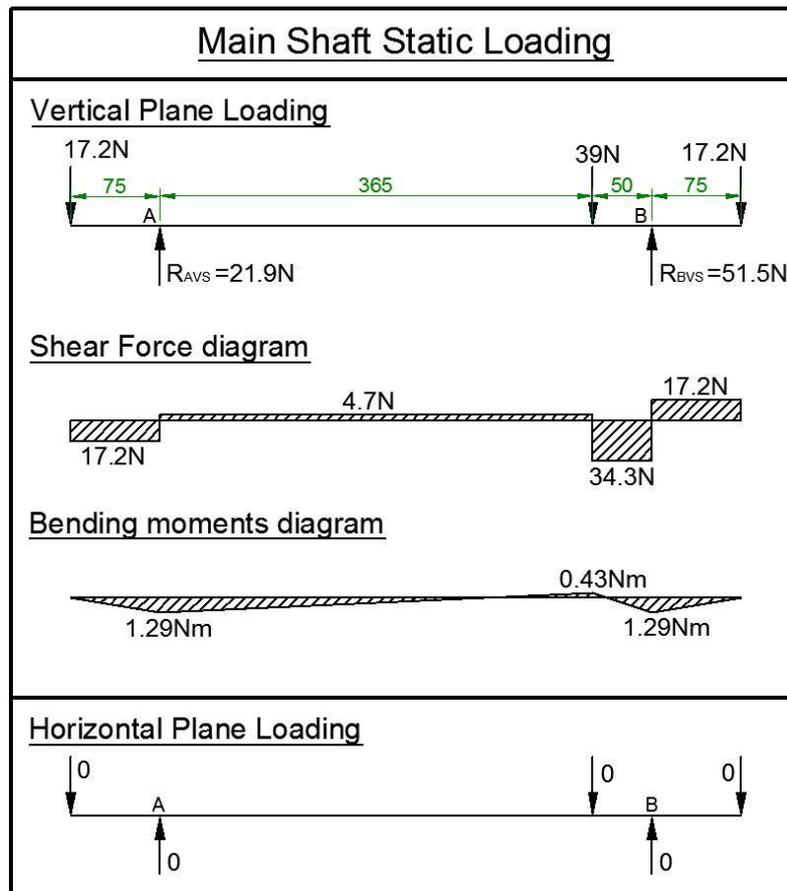


Figure 3.4 Static loading of the main shaft (refer to Table A3.1, Appendix 3/1)

The main shaft horizontal static loading is zero. This is because the chain drive is not transmitting any power and the V-belts are loose at static normal position.

3.4.2 Dynamic Loading

There are three critical dynamic loading scenarios. Critical dynamic loading occurs whilst:

1. Driving forward with a top speed
2. Turning sharp right
3. Turning sharp left

All three scenarios need to be investigated to design the shaft and to select the bearings.

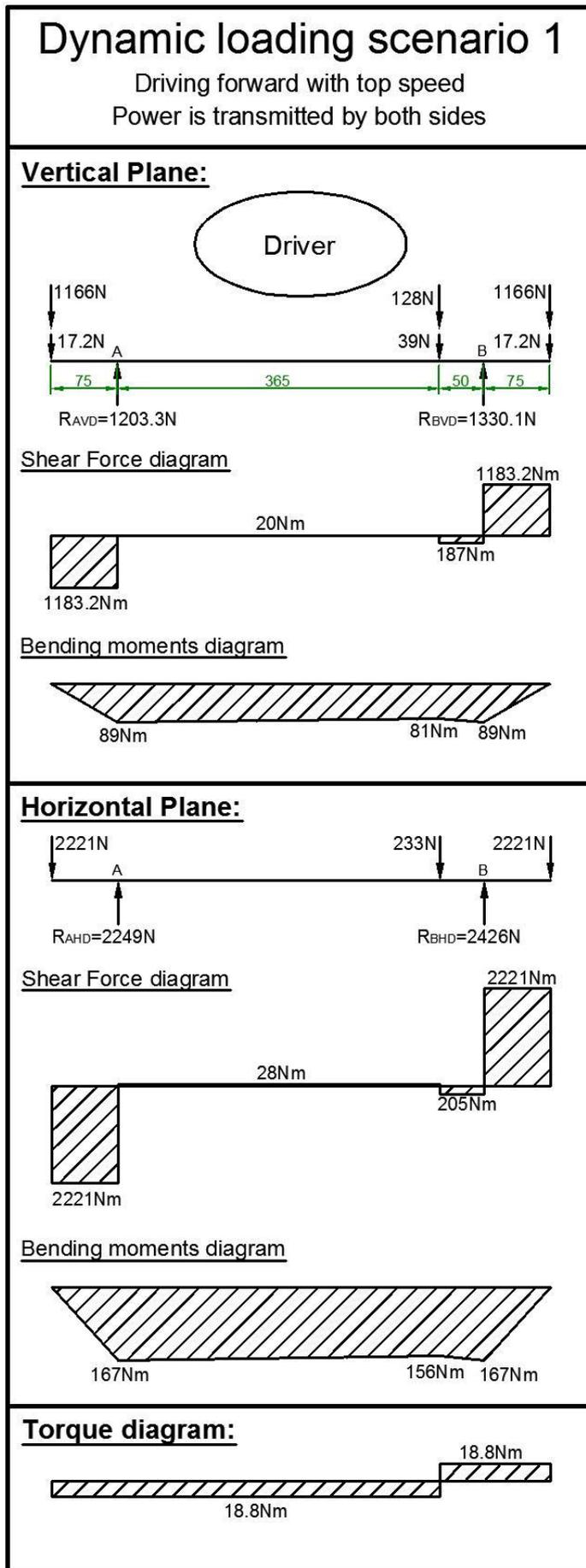


Figure 3.5 Dynamic loading scenario 1 (refer to Table A3.2, Appendix 3/3)

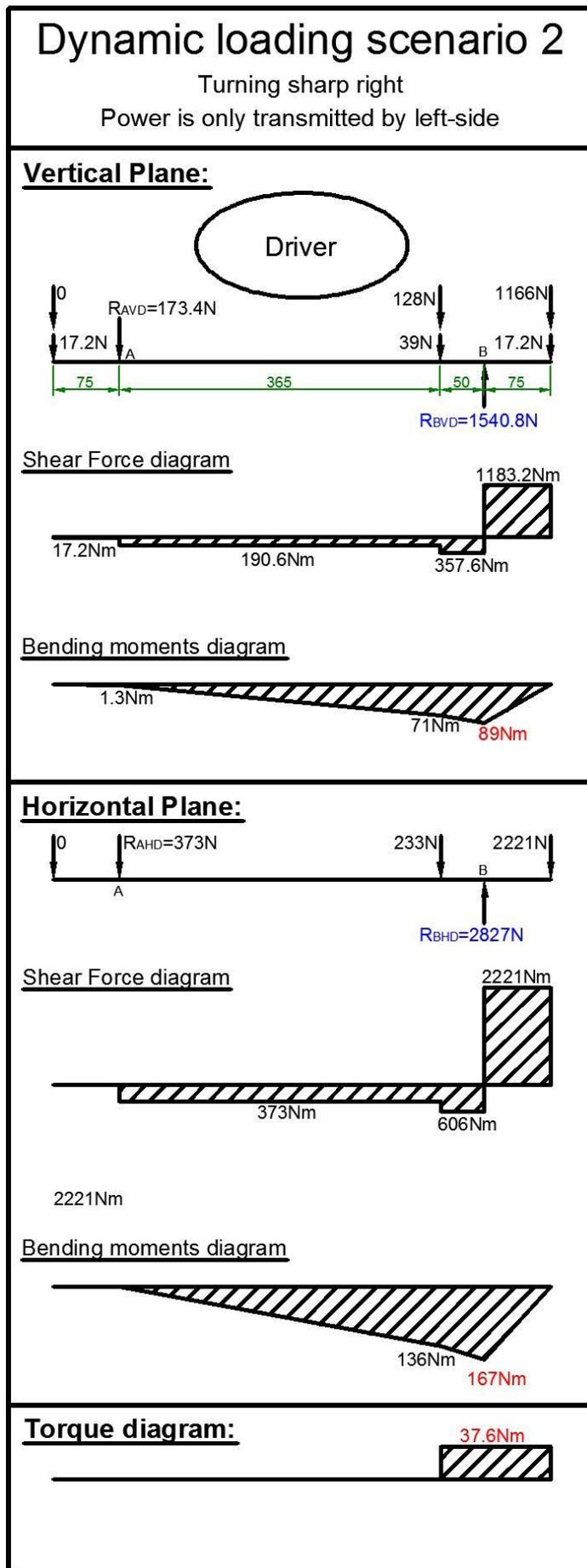


Figure 3.6 Dynamic loading scenario 2 (refer to Table A3.3, Appendix 3/3)

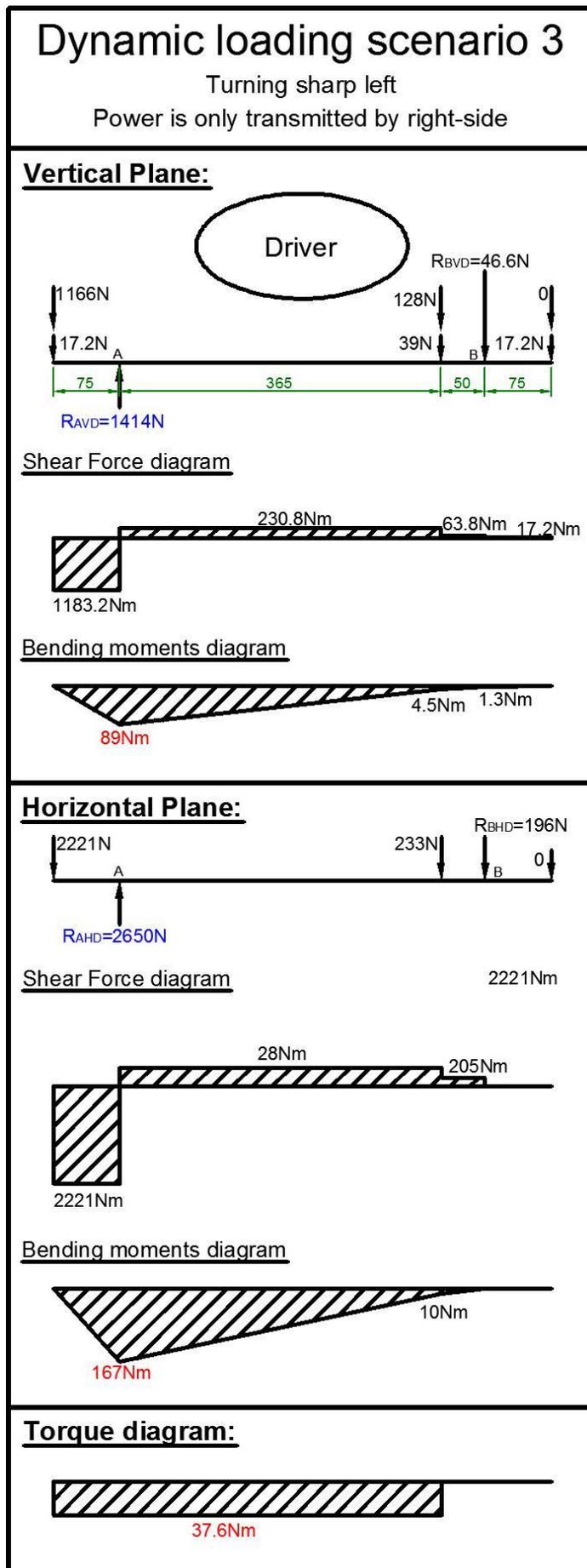


Figure 3.7 Dynamic loading scenario 3 (refer to Table A3.4, Appendix 3/4)

3.4.3 Main Shaft Design 1

The Australian standards are used to design the shaft. Refer to Appendix 3/4 for the shaft design method. Shaft diameters for the critical cross-sections are shown in Table 3.2. Different shaft cross-sections have different critical shaft diameters. The three different dynamic loading scenarios also have different critical shaft diameters. The minimum shaft diameter required to safely transfer power at all dynamic loading scenarios are highlighted (in yellow) in the Table 3.2.

Table 3.2 Critical shaft diameters (refer to Appendix 3/4)

Shaft cross-section under consideration (see Figure 3.8)	Point 1	Point 2	Point 3	Point 4	Point 5
Stress-raising factor (K)	1.85	1.65	1.85	1.65	1.85
D_{min} when driving forward (mm)	10.76	26.03	26.43	26.03	10.76
D_{min} when sharp right turn (mm)	0.00	4.94	25.33	26.09	13.56
D_{min} when sharp left turn (mm)	13.56	26.09	13.98	4.94	0.00

The AISI 1040 cold-drawn steel is chosen as the shaft material (refer to Appendix 3/4). The stress-raising factors used for different shaft cross-sections are shown in the Table 3.2. For the first design, pulleys and sprocket are mounted either by interference fits or using taper lock bushes. A decision is made to use a constant diameter shaft. The minimum constant shaft diameter required is 26.43mm (from Table 3.2). However, the closest bearing size available is 30mm so a 30mm constant diameter shaft is selected.

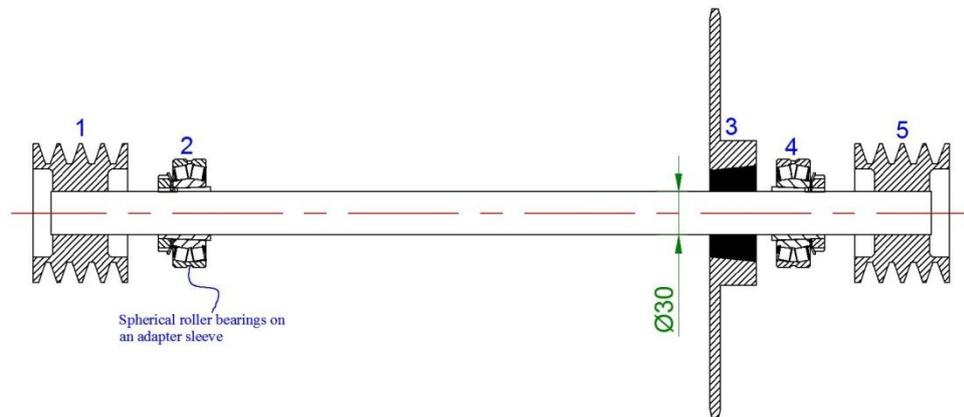


Figure 3.8 Main shaft design 1

This design uses two spherical roller bearings. The bearings can be mounted using their adapter sleeves. A taper lock bush can be used to mount the sprocket. The pulleys are interference fit mounted.

3.4.4 Main Shaft Design 2

In this design, the shaft is stepped and grooved to mount the bearings.

Table 3.3 Critical shaft diameters (refer to Appendix 3/4)

Shaft cross-section under consideration (see Figure 3.9)	Point 1	Point 2	Point 3	Point 4	Point 5	Point 6	Point 7
Stress-raising factor (K)	1.85	2.80	1.50	1.85	1.50	2.80	1.85
D_{min} when driving forward (mm)	10.76	31.04	25.22	26.43	25.22	31.04	10.76
D_{min} when sharp right turn (mm)	0.00	5.89	4.78	25.33	25.28	31.09	13.56
D_{min} when sharp left turn (mm)	13.56	31.09	25.28	13.98	4.78	5.89	0.00

Yellow highlighting in the Table 3.3 indicates the required minimum shaft diameter at different shaft cross-sections.

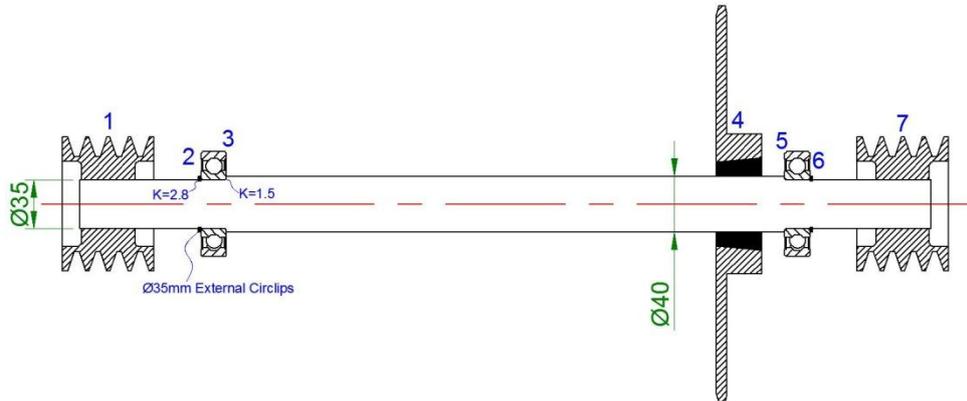


Figure 3.9 Main shaft design 2

3.4.5 Main Shaft Design 3

This design uses nuts and washers to mount the bearings.

Table 3.4 Critical shaft diameters (refer to Appendix 3/4)

Shaft cross-section under consideration (see Figure 3.10)	Point 1	Point 2	Point 3	Point 4	Point 5	Point 6	Point 7
Stress-raising factor (K)	1.85	1.50	1.50	1.85	1.50	1.50	1.85
D_{min} when driving forward (mm)	10.76	25.22	25.22	26.43	25.22	25.22	10.76
D_{min} when sharp right turn (mm)	0.00	4.78	4.78	25.33	25.28	25.28	13.56
D_{min} when sharp left turn (mm)	13.56	25.28	25.28	13.98	4.78	4.78	0.00

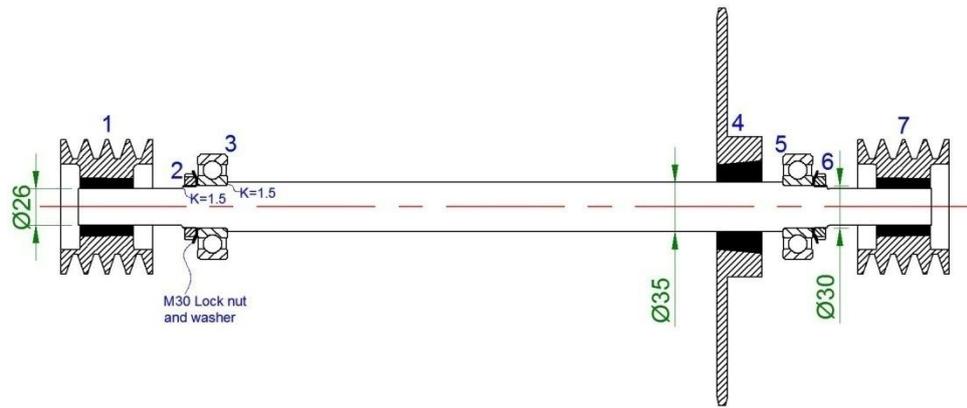


Figure 3.10 Main shaft design 3

The shaft is stepped at points 2, 3, 5 and 6. Calculated values in the Table 3.4 are the smaller side diameters of the steps at the points 2, 3, 5 and 6. The minimum shaft diameter at 2 and 6 could be 25.28mm (from Table 3.4). So a 26mm shaft diameter is selected for points 1 to 2 and 6 to 7 (see Figure 3.10). Bearings with inside diameter bigger than 26mm need to be chosen. The available bearing diameter is 30mm and the shaft diameter is also 30mm at those locations. M30 lock nuts and washers are selected to mount the bearings. A 35mm shaft diameter between points 3 and 5 is selected as the bearings will have shoulders to support.

3.4.6 Main Shaft Design 4

This design uses keys to mount the pulleys and the sprocket.

Table 3.5 Critical shaft diameters (refer to Appendix 3/4)

Shaft cross-section under consideration (see Figure 3.10)	Point 1	Point 2	Point 3	Point 4	Point 5	Point 6	Point 7
Stress-raising factor (K)	2.35	1.50	1.50	2.35	1.50	1.50	2.35
D_{min} when driving forward (mm)	11.39	25.22	25.22	28.63	25.22	25.22	11.39
D_{min} when sharp right turn (mm)	0.00	4.78	4.78	27.42	25.28	25.28	14.34
D_{min} when sharp left turn (mm)	14.34	25.28	25.28	14.86	4.78	4.78	0.00

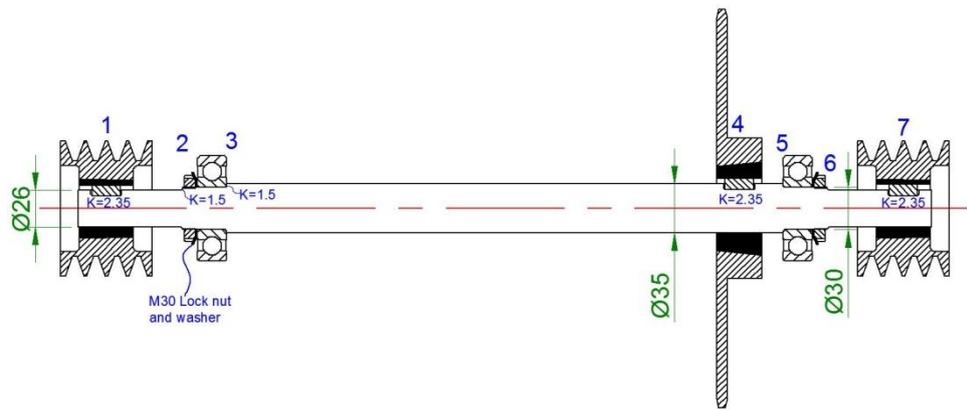


Figure 3.11 Main shaft design 4

The maximum torque on the main is 37.6Nm. The minimum key length required to transfer the torque is 7.05mm (refer to Appendix 3/6). Since the torque is not very large, the keys are not required to transmit the power.

All four shaft designs are capable of transmitting required power. Also, the diameter of these shafts are smaller than the maximum possible shaft diameter of 49mm (refer to Figure 3.2). Therefore, the shaft can be easily fitted. However, the shaft design 3 is considered as the best shaft design for this application.

- Shaft design 3 is preferred.

3.5 Bearings Selection for the Main Shaft

Bearings are needed to be selected for the required shaft diameter. The method of calculating dynamic loading on the bearing is shown in Appendix 4/1.

Table 3.6 Minimum required bearing load rating (refer to Appendix 4/2)

Loading	Bearing A		Bearing B	
	Ball bearing	Roller bearing	Ball bearing	Roller bearing
Static Load (C_0)	21.9N	21.9N	51.5N	51.5N
Dynamic Load (C)	21.0kN	17.3kn	22.6kN	18.6kN

Any bearing with higher than required dynamic and static loads can be selected. Table 3.6 shows the minimum required dynamic and static loads for both the ball and the roller bearings.

For the shaft design 3 and 4, a 30mm inside diameter bearing is needed. SKF ball bearing 6306 has a higher static and dynamic loading capability than required (refer to Deep groove ball bearings, single row, SKF n.d.) and two SKF ball bearing **6306** is chosen.

3.6 Disc Brake Design

Brakes are important part of any vehicle. When applied, brake will provide the capability of stopping a vehicle whilst travelling at a speed in a reasonable distance. It is vital that the driver is able to stop the vehicle, in an emergency situation to prevent an accident from occurring. Humans take few seconds to react and should be considered when designing the breaks.

The PTV go kart is not normally driven in the streets. The brakes are required to stop the vehicle, to turn the vehicle and to avoid any accidents with solid objects, e.g. trees. When driving at a top speed, it is decided that the PTV go kart should stop 1m from the point where the brakes are applied. Top speed of the kart is approximately 24 km/hr, assuming there are no power losses. However, the actual top speed will be less than this. The mass of the PTV go kart is estimated to be approximately 300kg.

- **44.5mm** minimum outside radius brake disc is required (refer to Appendix 5/2)

The PTV build plans recommends a use of a chain sprocket for the brake disc as it is more economical. The thickness of a normal disc brake is 6mm and the thickness of the closest bigger sprocket is 7.4mm ISO 08B-1 sprockets. Therefore, 50 teeth, 208.3mm pitch diameter sprockets are selected. Teeth will be machined off to make the 200mm diameter brake discs. The diameters of these brake discs are more than twice large than the minimum required diameter.

- **200mm** diameter brake disc.

3.7 Wheel Shaft Design

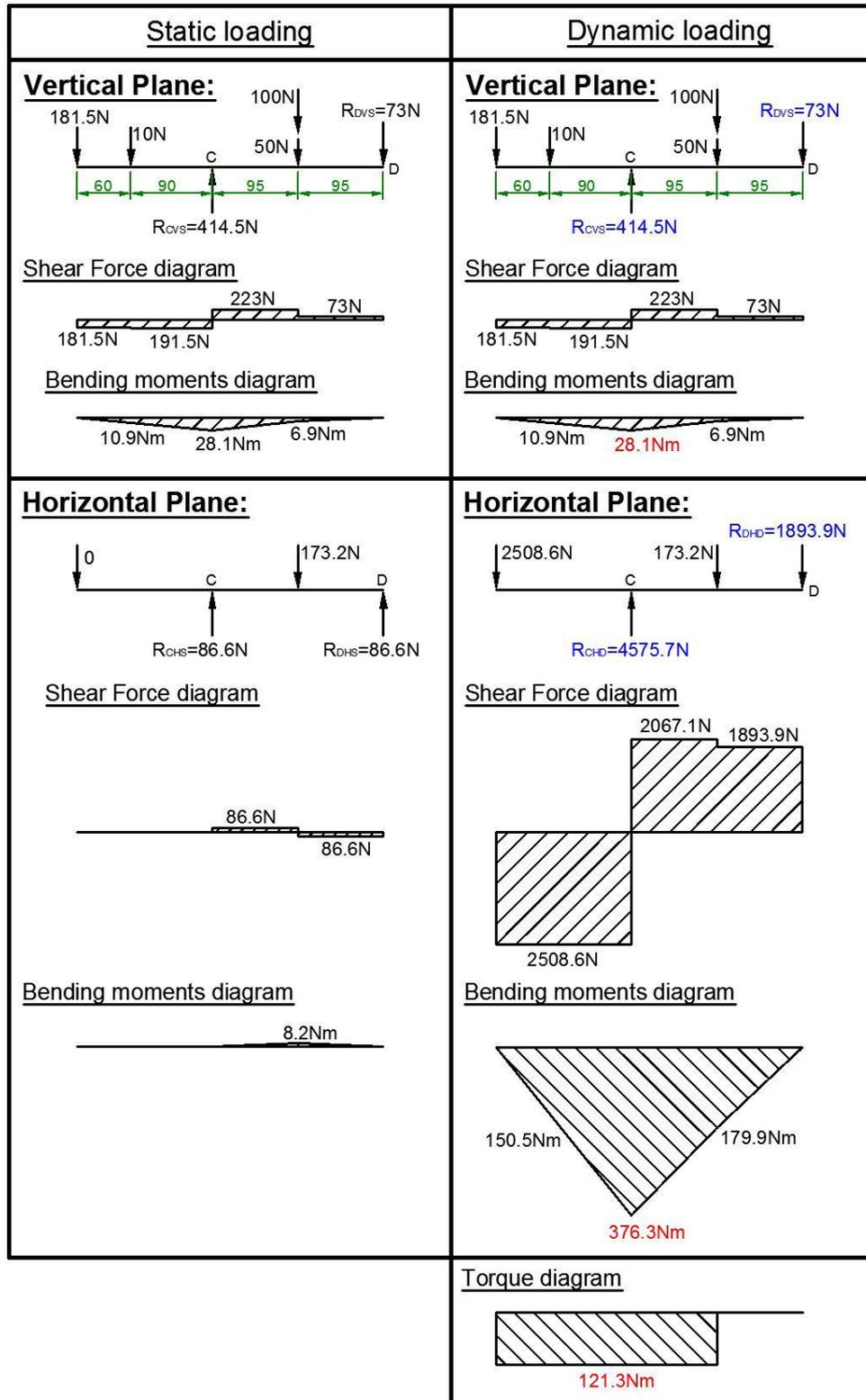


Figure 3.12 Static and Dynamic loadings (refer to Table A6.1 and Table A6.2)

During the dynamic loading, the critical loading occurs at point C. In the Figure 3.12, highlighted in red are the maximum bending moments and the torque on the shaft. The values highlighted in blue are the maximum loads on the bearing C and D.

Table 3.7 Critical shaft diameters (refer to Appendix 6/2)

Shaft cross-section under consideration (see Figure 3.13)	Point 1	Point 2	Point 3	Point 4	Point 5	Point 6	Point 7
Stress-raising factor (K)	1.85	2.35	1.50	1.95	1.85	1.95	1.65
Required minimum diameter (mm)	20.28	29.00	33.11	34.07	28.24	23.33	0.00

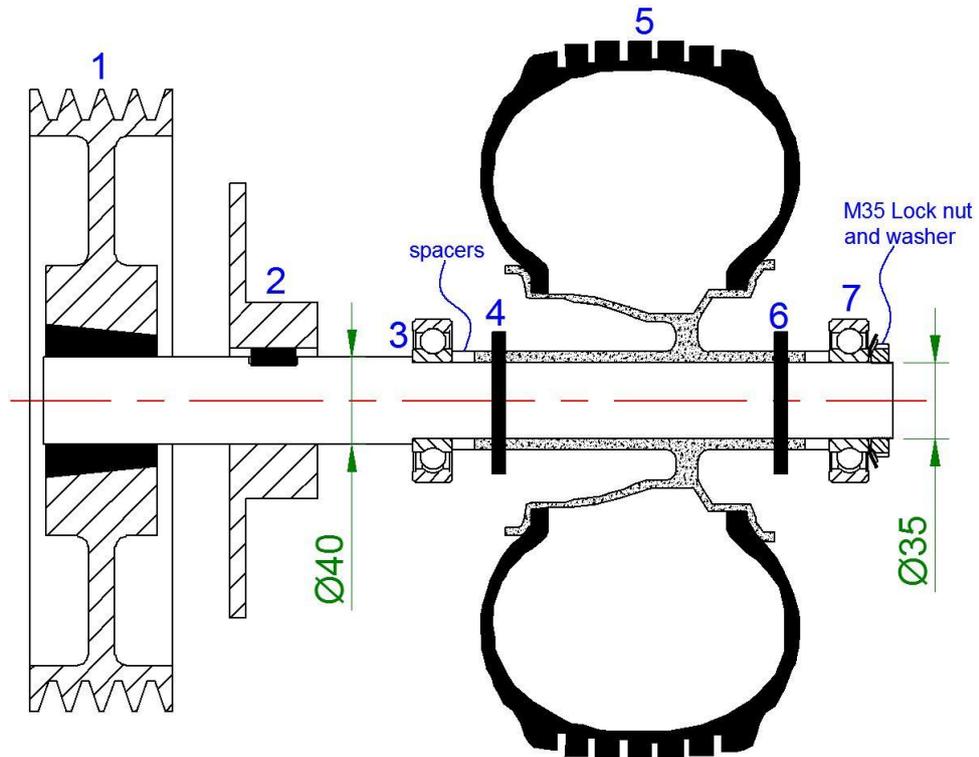


Figure 3.13 Wheel shaft design

The brake disc is mounted using a 10×8 key. The minimum key length is 12.5mm (refer to Appendix 6/3) so a longer 20mm length key is selected. A taper lock bush or interference fit could be used to mount the pulley. A shoulder and a spacer are used to support the bearing at point 3. Other bearing at point 7 is supported by a spacer and a lock nut. Pins are used to mount the wheel on the shaft. The minimum required pin diameter is 2.77mm and a larger 6mm diameter pins are selected.

3.8 Bearings Selection for Wheel Shaft

Table 3.8 Minimum required bearing load rating (refer to Appendix 4/2)

Loading	Bearing C		Bearing D	
	Ball bearing	Roller bearing	Ball bearing	Roller bearing
Static Load (C_0)	450.9N	450.9N	119.3N	119.3N
Dynamic Load (C)	22.0kN	18.8kn	9.09kN	7.77kN

The shaft diameter is 35mm at the locations of the bearings. Bearings with an inside diameter of 35mm needs to be selected. SKF ball bearing 6207 have higher static and dynamic loading capability than required (refer to Deep groove ball bearings, single row, SKF n.d.). Therefore, SKF ball bearing **6207** is selected.

- SKF **6207** ball bearings are selected for wheel shafts.

3.9 Structural Analysis

Three most critical tubes are selected for the analysis. If these tubes are capable of supporting the maximum loads then the other tubes are easily capable of supporting their loads. The selected tubes are simplified to make an analysable frame structure.

3.9.1 Middle Floor Tube

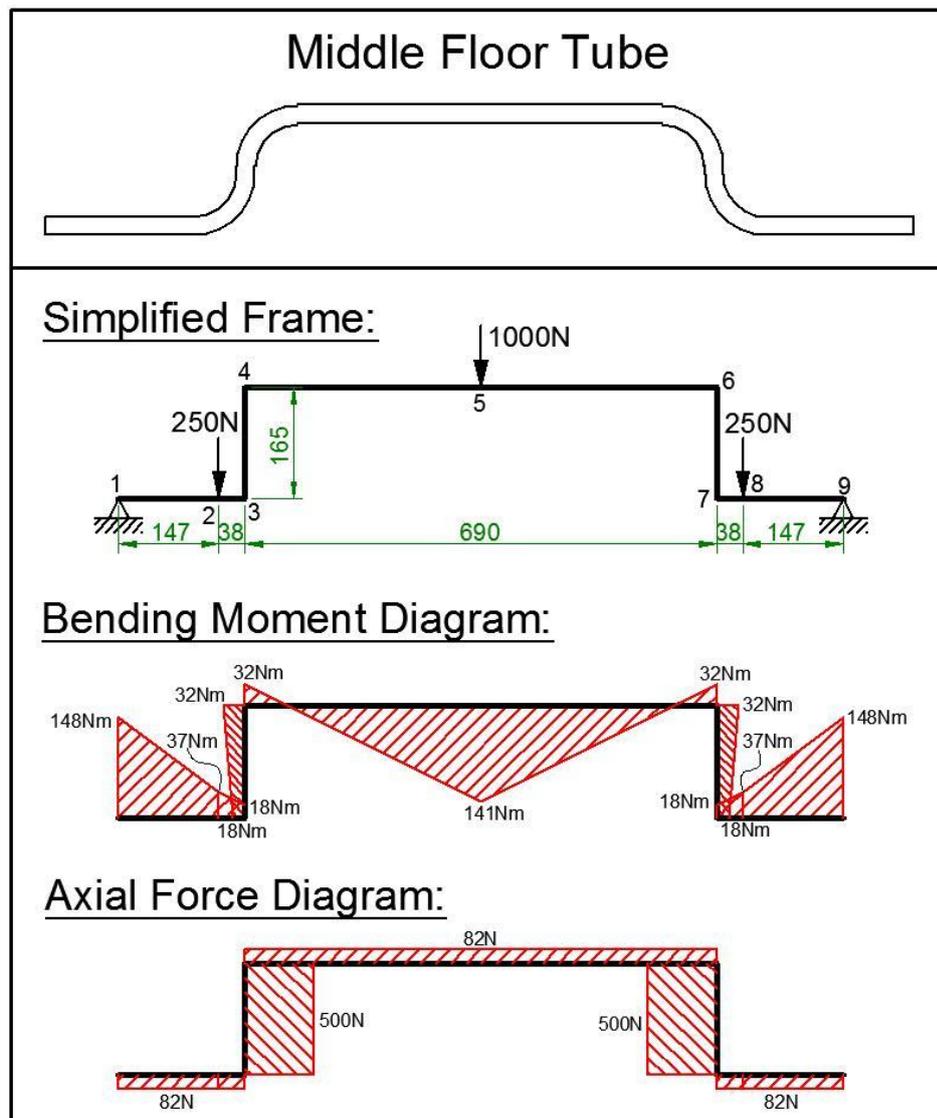


Figure 3.14 Middle floor tube

This middle floor tube is directly located underneath the driver. The driver's weight is supported by two floor tubes. Analysis is carried out assuming that approximately 1000N of the driver's weight is supported by the tube. Also another 250N loads are applied by the side frames.

A tube capable of supporting the require load needs to selected. Material yield strength of the tube is needed in order to select a tube size. Therefore, a **500MPa** yield strength Ruukki's hollow section tubes are chosen. A suitable tube size can be selected using the calculated minimum section modulus. The calculated minimum section modulus is **888mm³** (refer to Appendix 7/1). A tube with bigger section modulus than the calculated minimum section modulus should be selected. The 27mm outside diameter with 2.5mm thick circular hollow section tube has a 1070mm³ section modulus and a 192mm² cross-sectional area (Ruukki n.d.).

- **27mm** outside diameter and **2.5mm** thick tube is selected for the floor tubes.
- $S = 1070 \text{ mm}^3$
- $A = 192 \text{ mm}^2$

The bending and axial forces act in the tube at the same time (see Figure 3.14). Thus, the tube needed to be checked for the combine stresses. The combine normal stresses were calculated for the nodes using the selected tube's section modulus and the cross-sectional area (see Table 3.9).

Table 3.9 Combined normal stresses (refer to Appendix 7/2)

Tube locations (see Figure 3.14)	Node 1	Node 2	Node 3	Node 4	Node 5	Node 6	Node 7	Node 8	Node 9
Bending moments M (Nm)	148	37	18	32	141	32	18	37	148
Axial force F (N)	82	82	500	500	82	500	500	82	82
Combine normal stresses σ_c (MPa)	139	35	19	33	132	33	19	35	139

The middle floor tube has a maximum combine normal stress of 139MPa. The design stress (allowable stress 166.7MPa) is much greater than the combine normal stress. Thus, the floor tubes are capable of supporting the load safely.

3.9.2 Side Frame Tubes

Some of the driver's weight is supported by the side frames and it is shared between the two side frames. An assumption is made that 500N of driver's weight is supported by one side frame. When the control lever is pushed forward, forces will acts on the side frame. These forces are estimated to be 50N vertically upward and 10N horizontally forward (see Figure 3.15).

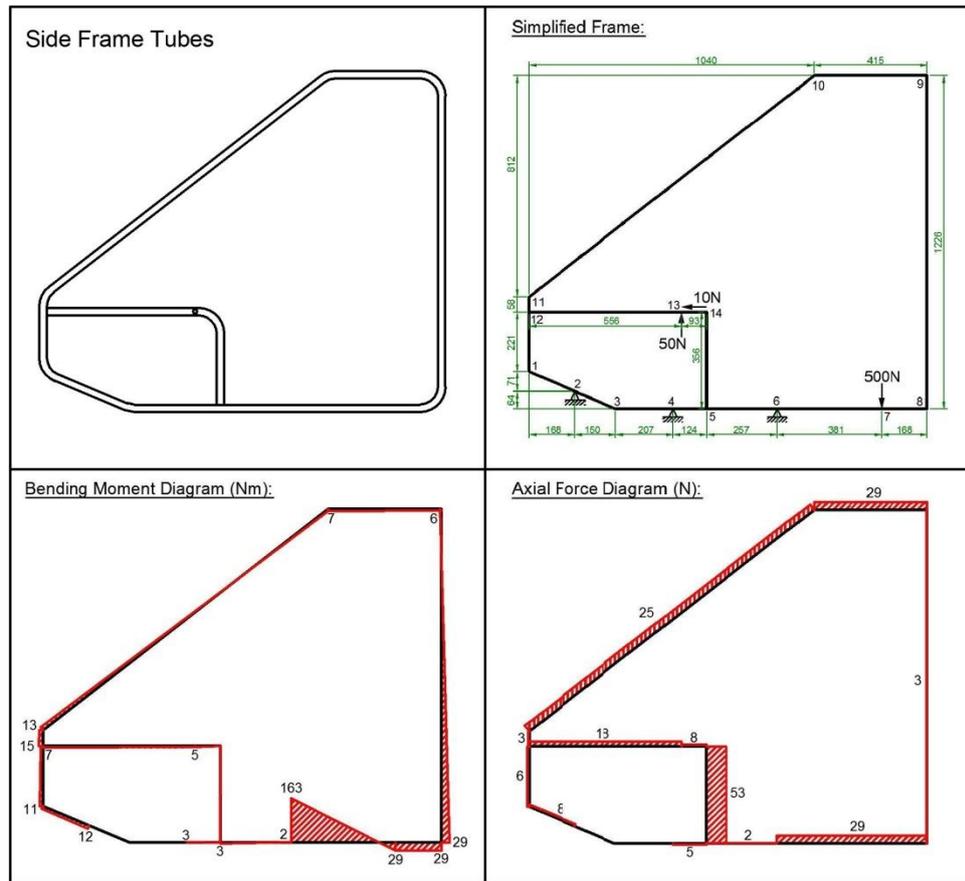


Figure 3.15 Side frame tubes

The maximum bending moment on the side frame is 163Nm at node 6 (Figure 3.15). The minimum section modulus is calculated to be 978mm^3 (refer to Appendix 7/1). The tube selected for the middle floor have a bigger section modulus than required (refer to page 25). Accordingly, a 27mm outside diameter with 2.5mm thick circular hollow section tube is selected. Then the combine stresses are calculated for all of the nodes using the selected tube's section modulus and the cross-section area.

Table 3.10 Combined normal stresses (refer to Appendix 7/2)

Tube locations (see Figure 3.15)	Node 1	Node 2	Node 3	Node 4	Node 5	Node 6	Node 7
Bending moments M (Nm)	11	12	0	3	3	163	29
Axial force F (N)	8	8	0	5	53	29	29
Combine normal stresses σ_c (MPa)	10.3	11.3	0.0	2.8	3.1	152.5	27.3
Tube locations (see Figure 3.15)	Node 8	Node 9	Node 10	Node 11	Node 12	Node 13	Node 14
Bending moments M (Nm)	29	6	7	13	15	5	0
Axial force F (N)	29	29	29	25	18	18	53
Combine normal stresses σ_c (MPa)	27.3	5.8	6.7	12.3	14.1	4.8	0.3

The side frame has a maximum combine normal stress of 152.5MPa and this is smaller than the design stress of 166.7MPa. Therefore, the side frame tubes are capable of supporting the loads safely.

3.9.3 Inside Wheel Tubes

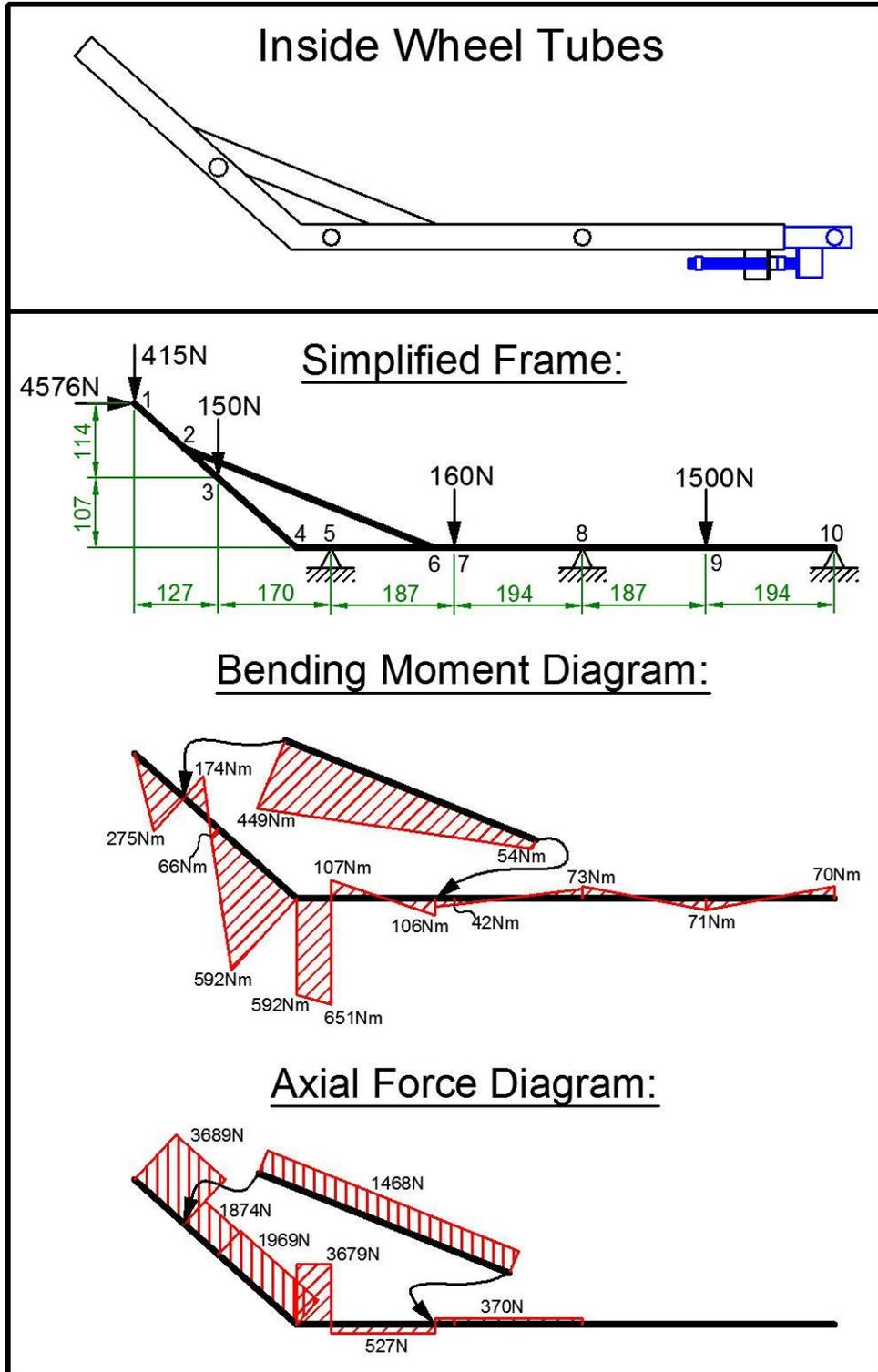


Figure 3.16 Inside wheel tubes

The bearing C is mounted at the node 1 and the loads on the bearing is supported by these tubes. For the node 1, the vertical and horizontal forces are taken from the Figure 3.12. The other loads on the tubes are approximated using the driver's weight, the components' weight and the tubes' weights.

The maximum bending moment on the inside wheel tubes is 651Nm at node 5 (Figure 3.16). Then the minimum section modulus is calculated to be 3905mm³ (refer to Appendix 7/1). The PTV build plans used square hollow sections tubes. Accordingly, square hollow sections tubes are selected for the inside wheel frame. The 40×40mm with 3mm thick square tube has a bigger section modulus than required.

- **40×40mm** with **3mm** thick square tubes are selected.
- $S = 4660 \text{ mm}^3$ (Ruukki n.d.)
- $A = 421 \text{ mm}^2$ (Ruukki n.d.)

Using the selected square tube's cross-section area and the section modulus, combine normal stresses are calculated.

Table 3.11 Combined normal stresses (refer to Appendix 7/2)

Tube locations (Figure 3.16)	Node 1	Node 2	Node 3	Node 4	Node 5	Node 6	Node 7	Node 8	Node 9	Node 10
M (Nm)	0	449	66	592	651	106	42	73	71	70
F (N)	3689	3689	1969	3679	3679	527	370	370	0	0
σ_c (MPa)	9	105	19	136	148	24	10	17	15	15

In the Table 3.11, σ_c is the combine normal stresses. The maximum combine stress on the inside wheel tubes is 136MPa and this is smaller than the design stress of 166.7MPa. Therefore, the inside wheel tubes are capable of supporting the loads safely.

After analysing the three most critical tubes, suitable tubes capable of supporting the maximum loads are selected. Therefore, all the other tubes with lower loads could be able to support their loads safely.

4 CAD MODELLING

There are two main reasons for carrying out CAD modelling. First, by supplying the required drawings, the building process will need minimum supervision from the designers. Second, to use as a design aid subsequently the designer can visualise the design and could see the potential problems occur during manufacturing and assembly process.

4.1 Components Drawing

The individual components are drawn in individual CAD files. The CAD models are drawn to the actual physical size of the component. Some of the components are available to download from online sources. The bearings and bearing housings are downloaded from SKF website. The idler is downloaded from Fenner Drive website. The engine and tires are downloaded from 3D Content Centre website.

4.2 Assembly

The individual components are assembled together to make the PTV go kart. The CAD assembling process is similar to the real practical assembly process.

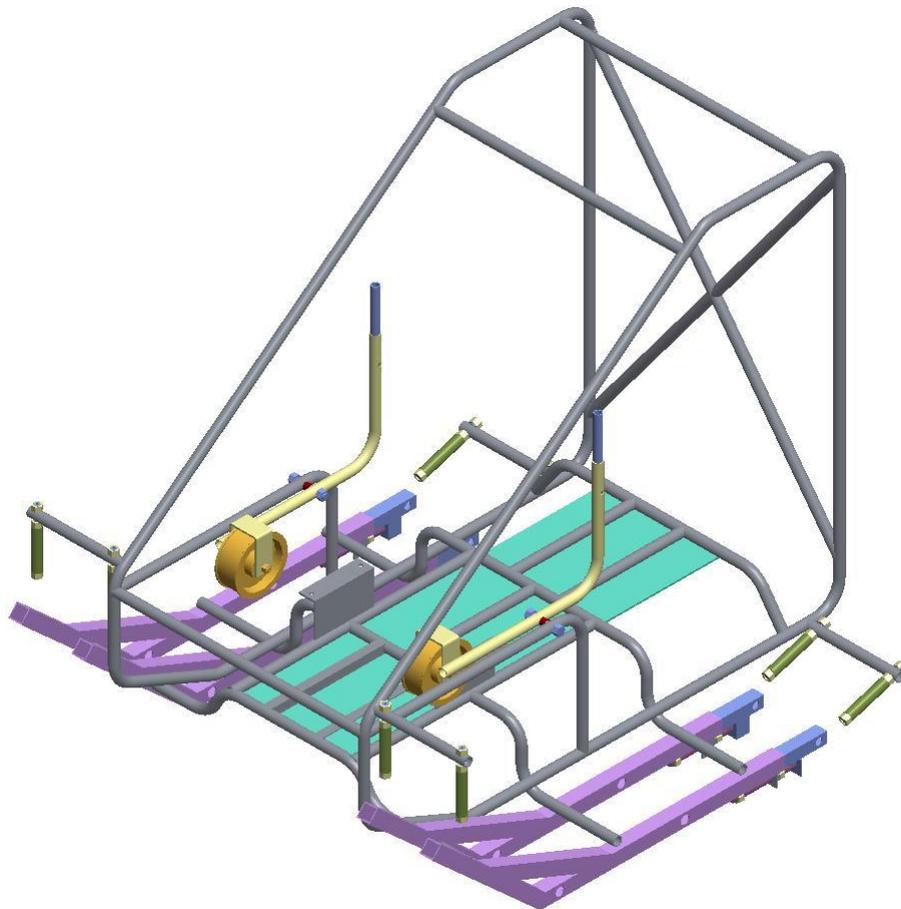


Figure 4.1 Frame assembly

First some sub-assemblies are assembled to simplify the assembly process. The frames are assembled using the tubes. Then the PTV go kart assembly is made using the sub-assemblies and components part drawings. Refer to Appendix 8 for detailed drawings.

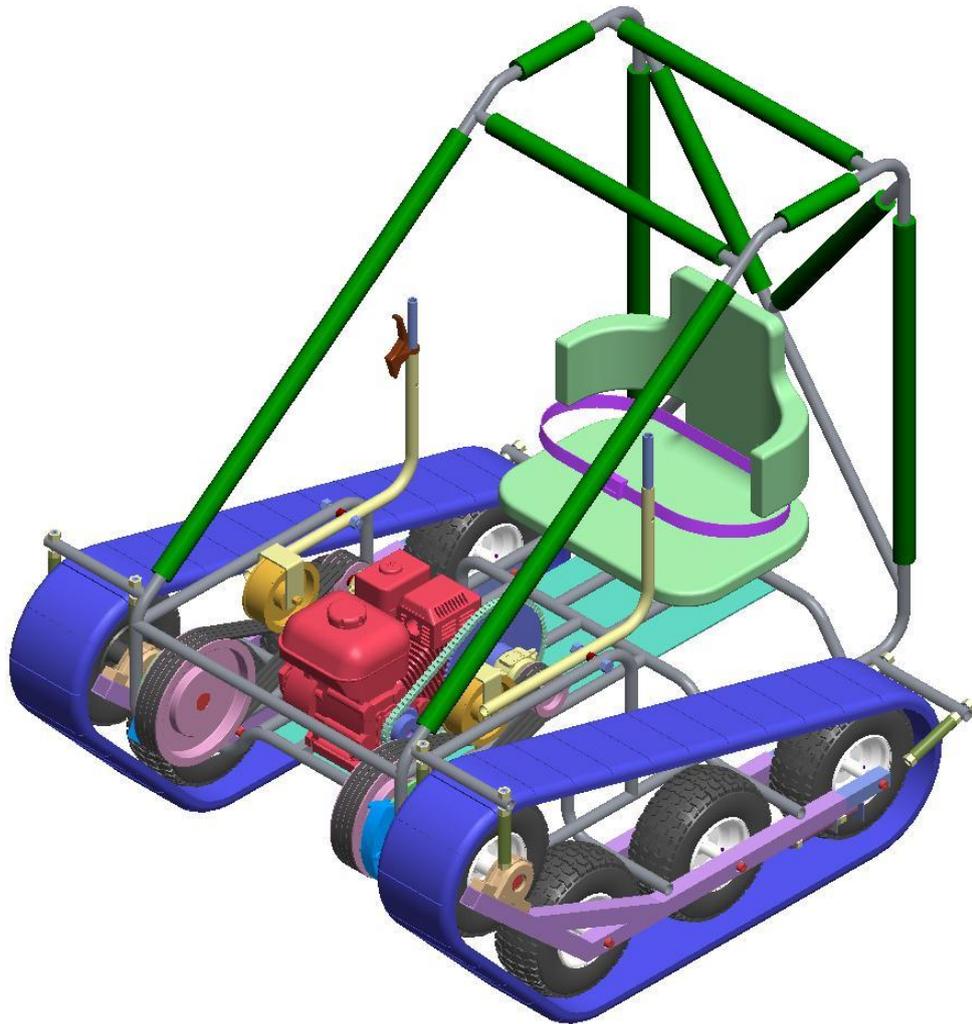


Figure 4.2 PTV go kart

During the CAD assembly process, few problems occurred as some components are overlapping each other. This means that it will be impossible to assemble the components in real life. Therefore, modifications had to be made to make it possible so that the components will fit together. Most of the problems that may occur during the prototype assembly process are solved. However, few other practical problems may occur as the real life assembling might be slightly different to the CAD assembling and not all the practical problems can be seen by CAD modelling. All these problems will be solved whilst building a prototype.

5 POSSIBLE IMPROVEMENTS

There is vast potential to improve the go kart. Plastic roof could be introduced so the kart can be driven during the rain without driver getting wet. Transparent acrylic sheet could be used as the windscreen and raincoat type waterproof fabric could be used for covering the sides. The engine and other components could be moved forward so the driver will have more leg room. A clutch could be placed between the engine shaft and the chain wheel to prevent the main shaft from turning during idle (PTV Build Plans 2009, 15). An accelerator foot pedal could be used instead of bicycle type hand operated brake. There are many possibilities but the safety is the most important factor that needs to be improved.

5.1 Safety

This kart needs a significant safety upgrade. The frame should be tested for the safety. The seat belt needs to be tested to see if it is capable of holding a person's weight. All the rotating objects should be guarded properly (PTV Build Plans 2009, 15). Not only should the rotating components be guarded from the driver's side, they need to be guarded from the other sides to prevent anyone from injuring themselves.

The major safety issues will occur if/when the driver tries to make a sharp turn whilst driving at a high speed. This could lead to the kart flipping over and causing serious injuries (PTV Build Plans 2009, 15). Two designs are proposed to prevent such an event.

5.1.1 Braking Structure

The first design recommends use of an additional braking structure (see Figure 5.1) which will sit behind the control levers. Both sides brake cables are connected to the new structure. This way if only one control lever is pulled back, brakes on the both driving wheels will be activated and the kart will be stopped. Springs can be used as parking brakes to pull back the new braking structure.

However, this design has one major problem. Even though brakes are activated by the new braking structure, the brakes and the V-belts could overheat if the full power is transmitted by a non-braking control lever. If slipping occurs on the power transmitted side brake, then the kart will turn. Therefore, this design needs to be thoroughly tested.

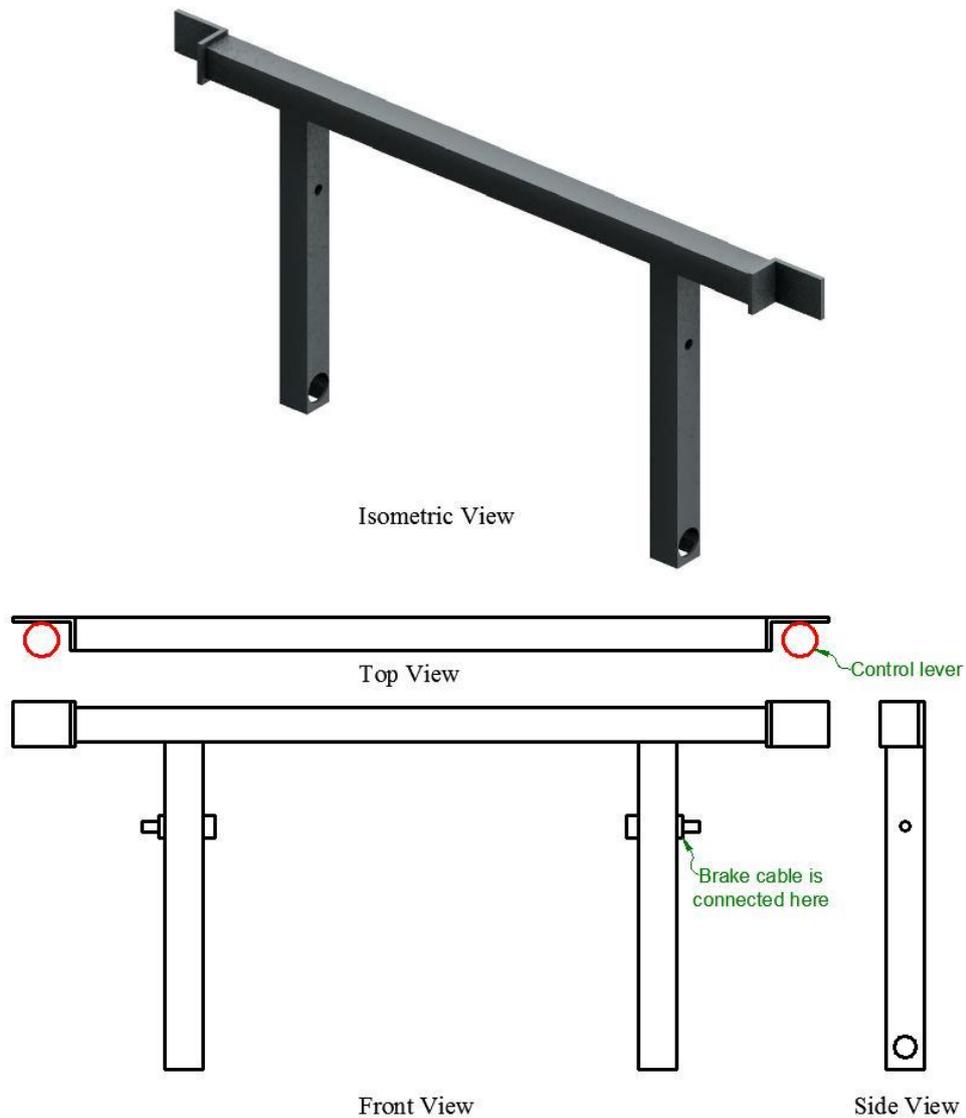


Figure 5.1 Safety structure design 1: – braking structure.

5.1.2 Limiting Structure

The second design uses a similar concept as before but the brake cables are connected to the control levers as normally do. This design has a maximum limit on how much distance the control levers can be apart (see Figure 5.2). If only one brake is applied, then the power transmission to the other side also reduces. This is because the limiting structure will pull back the non-braking control lever. This design is needed to be tested and the correct maximum limit of the control levers needs to be investigated. Theoretically 80mm is a good maximum limit. This way, the PTV go kart still has the ability to make sharp turns at a low speed. However, the flexibility of the PTV go kart will be reduced but the safety of the kart is superior. Returning of the limiting structure back to its normal position can be done using springs.

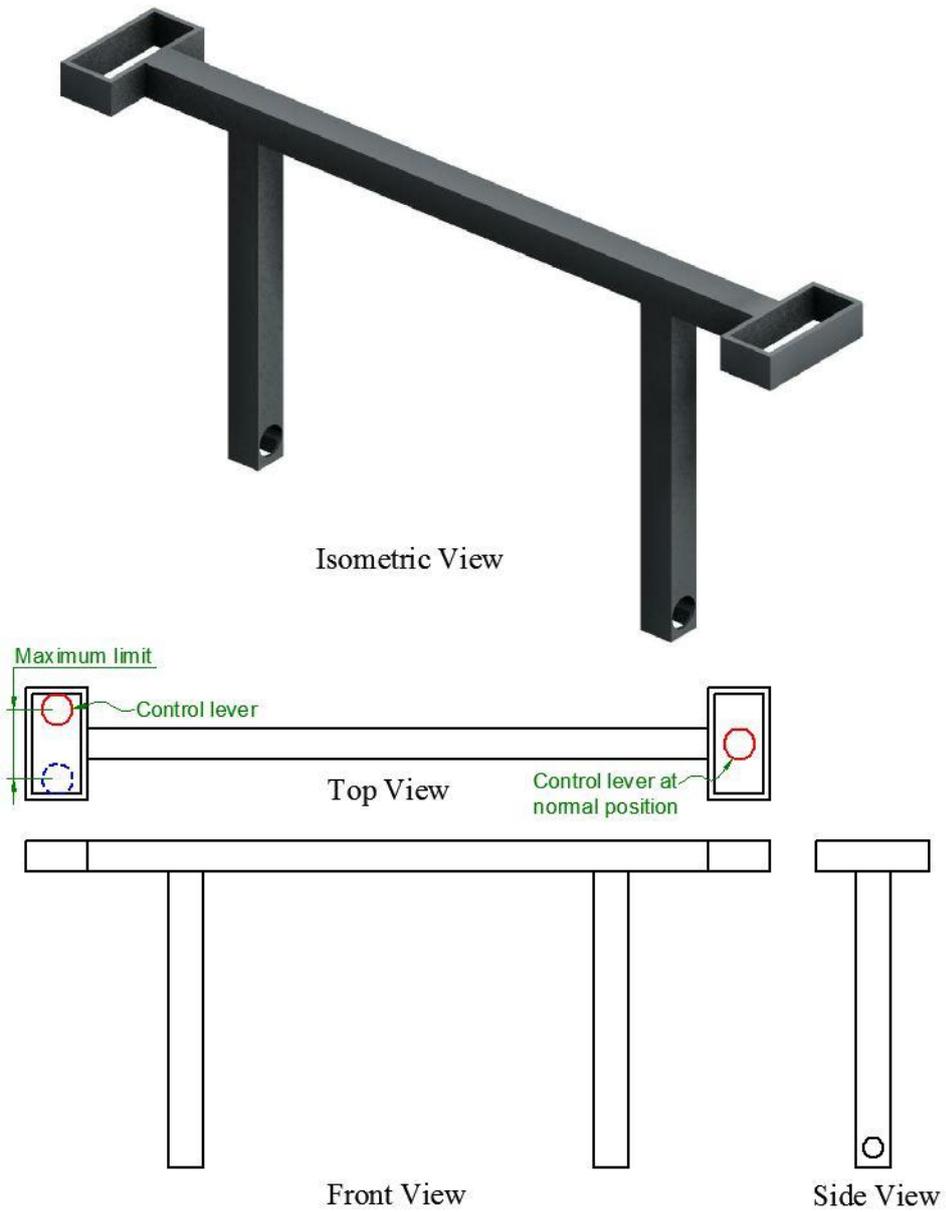


Figure 5.2 Safety structure design 2: – limiting structure.

The second design is a superior option as this design will not introduce any overheating problems and keep all the original functions (sharp turning) of the PTV go kart.

6 POTENTIAL PRACTICAL USES

The PTV go kart is used for recreation purposes as a racing kart at present. However, there is potential to get practical use out of it. It might be possible to mount a grass cutting blade in front of the kart and use it as a lawnmower. Another use would be to drag a heavy rock or tree branch from one place to another. Also it is possible to use the kart as a snow cleaner. The snow cleaner concept is investigated in detail as it will be ideal for household use.

6.1 Household Snow Cleaner

Slight modifications to the front of the PTV go kart needs to accommodate a snow cleaning attachment. Two metal pipes are welded on to the front of the kart. Two small rectangular metal plates are welded onto those pipes to mount the snow cleaning plate with bolts and nuts. Another pipe is welded between those two pipes to hang the snow cleaning plate. This modification to the front of the kart is shown in blue colour in Figure 6.1.

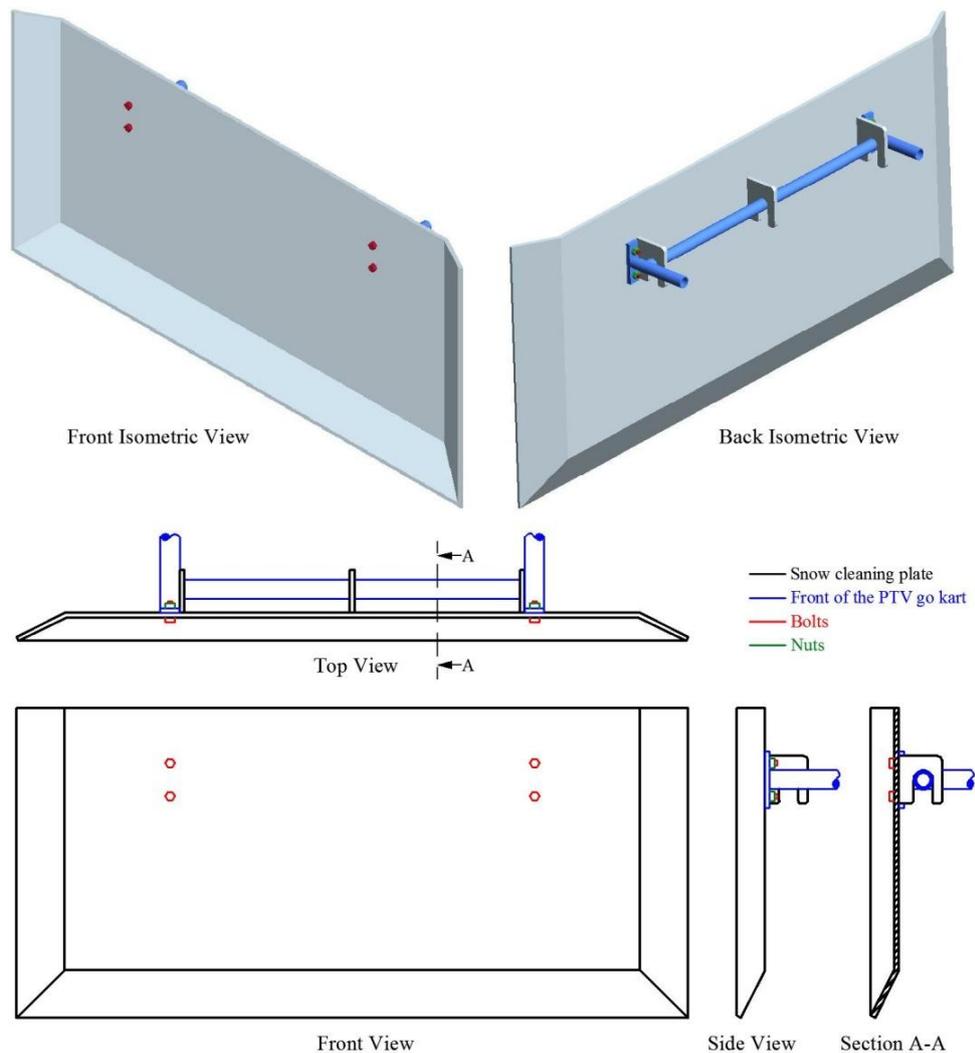


Figure 6.1 Snow cleaner

As illustrated in the Figure 6.1 a metal sheet is bent from ends to make a cleaning plate. Three hook shaped metal plates are welded to the back of the snow cleaning plate. This is to make the mounting of the snow cleaning plate easier as the snow cleaning plate can be located onto position with the hooks. After located the snow cleaning plate onto position, the plate will be mounted securely to the front of the PTV go kart using four bolts and nuts and the PTV go Kart is ready to be used as a snow cleaner. Snow will be pushed away by the snow cleaning plate. This is only a concept and if to be manufactured, a structural analysis should be carried out to check the capability of the snow cleaning plate and the front modifications of the PTV go kart.

This design can be improved by using door hinges to mount the snow cleaning plate onto the PTV go kart. The hinges allow the snow cleaning plate to rotate from the normal vertical position to forward direction. The snow cleaning plate cannot be rotate backwards as the kart frame is blocking the hinges from rotating backwards. It might be even possible to rotate the snow cleaning plate up to a horizontal position. A levering system can be designed to control the snow cleaning plate from the driver's seat. That way it is more flexible and there is no need for the driver to get out of the PTV go kart to control the snow cleaning plate.

As the friction of the snow is very low, slipping may occur whilst trying to push the snow. Small metal screws could be screwed into the track to increase the friction. The power might not be enough to push a large amount of show. Testing needed to establish the capabilities of the PTV go Kart as a household snow cleaner.

7 CONCLUSION

A PTV go kart was successfully redesigned using available metric components. Design calculations were carried out and suitable components were selected from the popular manufacturers. This PTV go kart has half the power as the original PTV build plans. Using the selected engine power, a suitable chain drive, belt drive, shafts, disc brake and bearings were redesigned. Four main shaft designs were made and design 3 is considered to be the best suited design for this application. Some assumptions were made when the actual values were unknown. These designs were theoretically capable of transmitting the maximum power.

Structural analysis was done and suitable frame tubes were selected. Tubes were simplified to theoretically analysable frames and then the calculations were carried out. The loads on the tubes were estimated conservatively so the tube can support more than the actual maximum loads. This PTV go kart should be reliable for some years so the manufacturing of the design is worthwhile. The Table 7.1 summarizes all of the selected components during the redesign process. By drawing a 3D model, most of the potential problems that could occur during the assembly process were found and resolved. Two safety structures were introduced and the second design was considered as the most practical solution. Potential snow cleaning concept was introduced to make the PTV go kart more useful.

Table 7.1 Selected components list

Component name	Manufacturer	Part number
Engine	Honda	GX200
Chain with 12.7 mm pitch	SKF	08B-1
23 teeth sprocket	SKF	PHS 08B-1TB23
70 teeth sprocket	SKF	PHS 08B-1TB70
A section V-belt	Fenner	A1530
90 mm diameter pulley	Fenner	031A0124
280 mm diameter pulley	Fenner	031A0324
Idler	Fenner Drives	F3B6280
Brake disc: 50 teeth sprocket	SKF	PHS 08B-1B50
Ø30 ball bearing	SKF	6306
M35 lock nut and washer	SKF	KM 6 and MB 6
Ø35 ball bearing	SKF	6207
M35 lock nut and washer	SKF	KM 7 and MB 7
Circular hollow section tube	Ruukki	27 × 2.5 mm tube
Square hollow sections tube	Ruukki	40 × 40 × 3 mm tube

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CHAIN DRIVE DESIGN

Engine:

Honda GX200

Net Power Output = **4.1 kW (5.5HP) @ 3600 rpm** (Honda 2011)

Maximum torque = **12.4 Nm @ 2500 rpm**

Design Power:

$$P_D = SF \times P_R \quad (\text{Mott 2006, 293})$$

Where:

P_D = Design power

SF = Service factor

P_R = Required output power

Output power is not known and it will be less than the power of the engine due to friction and other power losses. If a service factor is selected from Mott page 290, Table 7-8, the service factor is 1.4 for Moderate shock engine. However, a decision is made to use the engine power as the design power. Service factor will not be used as the engine power will be bigger than the output power and to select smaller the size sprocket.

$$P_D = P_{engine} = 4.1kW$$

Chain selection process:

Using the design power and the rotational speed (refer to Appendix 1/1), a suitable chain can be selected from Figure A1.1.

$$P_D = 4.1kW \quad \text{and} \quad n = 3600rpm$$

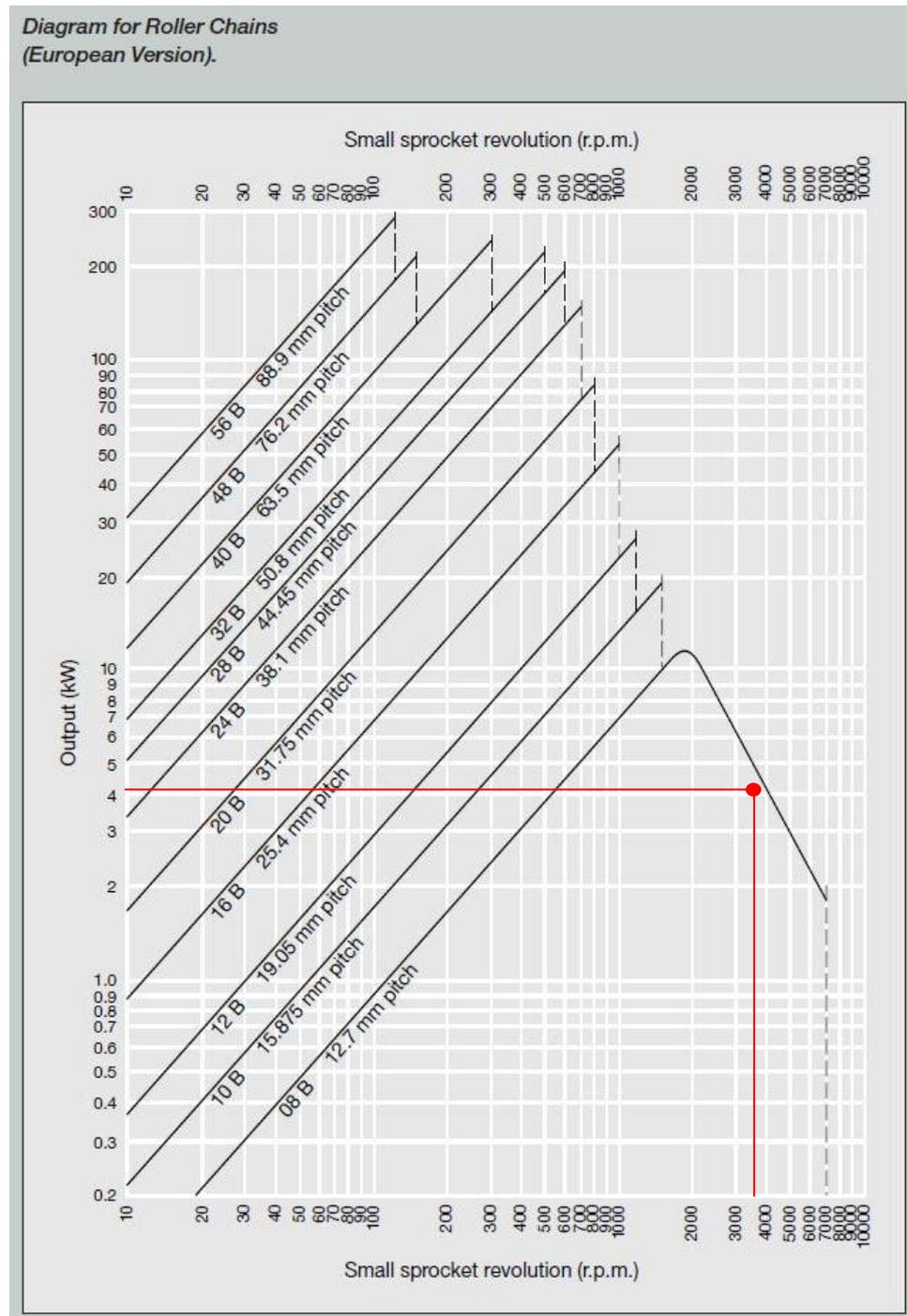


Figure A1.1 Chain Performance Diagram. (Rexnord n. d., 22)

Chain number: **08B** with **12.7mm** pitch chain is selected.

Small chain sprocket selection process:

Using the design power and the engine speed (refer to Appendix 1/1), required small sprocket is selected.

**5.1 Transmittable output's (kW)
for Rex-High-Capacity-Roller Chains 08 B - 1
12.7 mm pitch, European version** **DIN 8187**

Number of teeth	Reference Ø mm	Small sprocket revolution																			
		50	200	400	600	900	1200	1800	2400	3000	3500	4000	4500	5000	5500	6000	6500	7000	7500	8000	9000
		Hand lubrication		Drip lubrication			Oil bath lubrication			Forced feed lubrication											
13	53.10	0.23	0.80	1.50	2.16	3.11	4.02	4.99	3.24	2.32	1.84	1.51	1.27	1.07	0.93	0.82	0.72	0.65	0.58	0.53	—
14	57.10	0.25	0.87	1.62	2.33	3.37	4.36	5.58	3.62	2.59	2.06	1.68	1.41	1.21	1.04	0.92	0.82	0.72	0.66	0.59	—
15	61.10	0.27	0.93	1.75	2.52	3.62	4.70	6.19	4.02	2.88	2.28	1.87	1.57	1.33	1.16	1.02	0.90	0.81	0.72	—	—
16	65.10	0.29	1.00	1.87	2.70	3.88	5.03	6.82	4.42	3.17	2.52	2.06	1.72	1.47	1.27	1.12	0.99	0.89	0.80	—	—
17	69.10	0.31	1.07	2.00	2.88	4.15	5.37	7.47	4.85	3.47	2.76	2.26	1.89	1.62	1.40	1.23	1.09	0.97	0.87	—	—
18	73.10	0.32	1.14	2.12	3.07	4.42	5.72	8.13	5.28	3.78	3.00	2.46	2.06	1.76	1.52	1.33	1.18	1.06	0.96	—	—
19	77.20	0.35	1.21	2.26	3.25	4.68	6.06	8.75	5.73	4.10	3.26	2.67	2.23	1.91	1.65	1.45	1.28	1.15	1.04	—	—
20	81.20	0.37	1.27	2.38	3.43	4.95	6.41	9.25	6.19	4.42	3.52	2.88	2.41	2.06	1.78	1.57	1.39	1.24	1.12	—	—
21	85.20	0.38	1.35	2.52	3.62	5.22	6.76	9.75	6.66	4.77	3.78	3.09	2.59	2.22	1.92	1.68	1.49	1.33	1.21	—	—
22	89.20	0.41	1.42	2.64	3.81	5.48	7.10	10.25	7.14	5.11	4.06	3.32	2.78	2.37	2.06	1.81	1.60	1.43	—	—	—
23	93.30	0.42	1.48	2.77	3.99	5.75	7.45	10.75	7.63	5.46	4.33	3.55	2.97	2.54	2.20	1.93	1.72	1.53	—	—	—
24	97.30	0.45	1.56	2.90	4.18	6.02	7.80	11.25	8.13	5.82	4.62	3.78	3.17	2.71	2.34	2.06	1.82	1.63	—	—	—
25	101.30	0.47	1.62	3.03	4.37	6.29	8.15	11.75	8.67	6.19	4.91	4.02	3.37	2.88	2.49	2.19	1.94	—	—	—	—
28	113.40	0.52	1.83	3.42	4.94	7.12	9.25	13.25	10.25	7.33	5.82	4.77	3.99	3.41	2.96	2.59	2.30	—	—	—	—
30	121.50	0.57	1.98	3.69	5.32	7.67	9.92	14.33	11.33	8.13	6.46	5.28	4.42	3.78	3.27	2.88	—	—	—	—	—
32	129.60	0.61	2.12	3.96	5.71	8.22	10.67	15.33	12.50	9.00	7.12	5.82	4.88	4.17	3.61	3.17	—	—	—	—	—
35	141.70	0.67	2.33	4.37	6.28	9.08	11.75	16.92	14.33	10.25	8.13	6.66	5.58	4.77	4.13	—	—	—	—	—	—
40	161.90	0.77	2.70	5.04	7.26	10.42	13.58	19.50	17.50	12.50	9.92	8.13	6.82	5.82	—	—	—	—	—	—	—

Figure A1.2 Transmittable Output. (Rexnord n. d., 24)

The 23 teeth sprocket looks capable of transferring 4.1kW of power. The actual power transfer capability at 3600rpm can be found by interpolating values.

Interpolation:

$$\frac{P - 4.33}{3600 - 3500} = \frac{3.55 - 4.33}{4000 - 3500} \rightarrow P = 4.174 \text{ kW @ } 3600\text{rpm}$$

The 23 teeth sprocket power transmission capability is bit higher than the required design power of 4.1kW at 3600rpm.

Therefore, a 23 teeth sprocket is selected.

Lubrication:

Figure A1.2 recommend the use of forced feed lubrication for this chain drive.

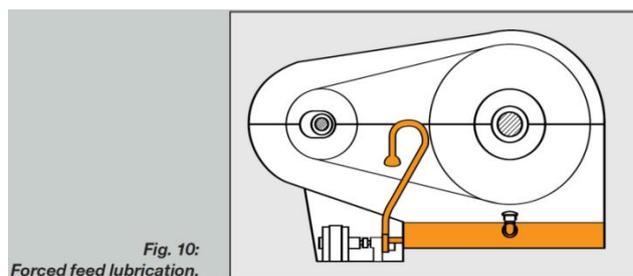


Figure 2.9 Forced feed lubrication. (Rexnord n. d., 20)

Speed ratio of the chain drive:

$$r = \frac{n_1}{n_2} = \frac{N_2}{N_1} \quad (\text{Mott 2006, 293})$$

Where: $n_1 = 3600$ rpm (smaller sprocket speed = engine speed)
 $N_1 = 23$ (smaller sprocket number of teeth)
 $N_2 = 70$ (larger sprocket number of teeth, from page 9)

Therefore: $r = \mathbf{3.04}$ (speed ratio)
 $n_2 = \mathbf{1182.86}$ rpm (larger sprocket speed)

This larger sprocket speed is approximately two times bigger than the PTV build plans' larger sprocket speed. However, the speed can be reduced by the belt drive to get the similar final output speed.

Pitch diameter of sprockets:

$$D = \frac{P}{\sin(180^\circ/N)} \quad (\text{Mott 2006, 291})$$

Where: $P = 12.7$ mm (chain pitch, from Appendix 1/2)

Therefore: $D_1 = \mathbf{93.27}$ mm (smaller sprocket pitch diameter)
 $D_2 = \mathbf{283.07}$ mm (larger sprocket pitch diameter)

Chain length:

$$L = 2C + \frac{N_2 + N_1}{2} + \frac{(N_2 - N_1)^2}{4\pi^2 C} \quad (\text{Mott 2006, 290})$$

In order to find the chain length, first the centre distance of the two sprockets should be estimated. To estimate the centre distance, a drawing is drawn. A Honda GX200 drawing was downloaded from the 3D Content Central website. It is important to keep the main shaft closest to the engine without any contact. The maximum main shaft diameter needs to be approximated and assumed to be 45mm. Then the centre distance is found to be 263 mm (refer to Figure 3.2). This value needs to be converted into chain pitches as the chain length equation uses number of teeth to calculate the length. The estimated centre distance is divided by chain pitch to convert the value into pitches.

$$C = 263 \text{ mm} = 20.71 \text{ pitches}$$

Thus, the calculated chain length: $L = 90.62$ pitches

As the chain length must be a whole number, calculated value is rounded to the nearest upper whole number. Therefore, more space is available to fit a bigger size shaft diameter.

$$L = \mathbf{91} \text{ pitches} = 1155.70 \text{ mm}$$

Actual maximum centre distance:

$$C = \frac{1}{4} \left[L - \frac{N_2 + N_1}{2} + \sqrt{\left(L - \frac{N_2 + N_1}{2} \right)^2 - \frac{8(N_2 - N_1)^2}{4\pi^2}} \right] \quad (\text{Mott 2006, 290})$$

$$C = 20.91 \text{ pitches} = 265.58 \text{ mm}$$

This C is the maximum centre distance of the two sprockets that could be apart with a 91 pitches long chain. At the maximum centre distance there is no sagging in either side of the chain (Mott 2006, 290). A centre distance of 265mm is selected for this design.

$$C = \mathbf{265 \text{ mm}}$$

Smaller sprocket arc of contact:

$$\theta_1 = 180^\circ - 2\sin^{-1} \left[\frac{D_2 - D_1}{2C} \right] \quad (\text{Mott 2006, 291})$$

$$\theta_1 = \mathbf{138.13^\circ}$$

Larger sprocket arc of contact:

$$\theta_2 = 180^\circ + 2\sin^{-1} \left[\frac{D_2 - D_1}{2C} \right] \quad (\text{Mott 2006, 291})$$

$$\theta_2 = \mathbf{221.87^\circ}$$

BELT DRIVE DESIGN

Design Power:

Maximum power on a belt drive system occurs when the PTV go kart makes a sharp turn. The chain drive will transmit the power to the main shaft and the main shaft transfer that power to the pulleys. The maximum power on one belt drive system will be equal to the chain drive's delivered power. However, this power is not known so an assumption is made that the chain drive will transmit all of the power from the engine without any losses. Thus, a decision is made to use the same design power as in the chain drive.

$$P_D = 4.1kW$$

Smaller pulley rotational speed:

Smaller pulley speed is equal to the larger sprocket speed as both of them are mounted on the main shaft.

$$n_1 = 1182.86 \text{ rpm (from Appendix 1/4)}$$

V-belt selection process:

V-belt cross-section is selected from the Figure A2.1, using the design power and the smaller pulley speed.

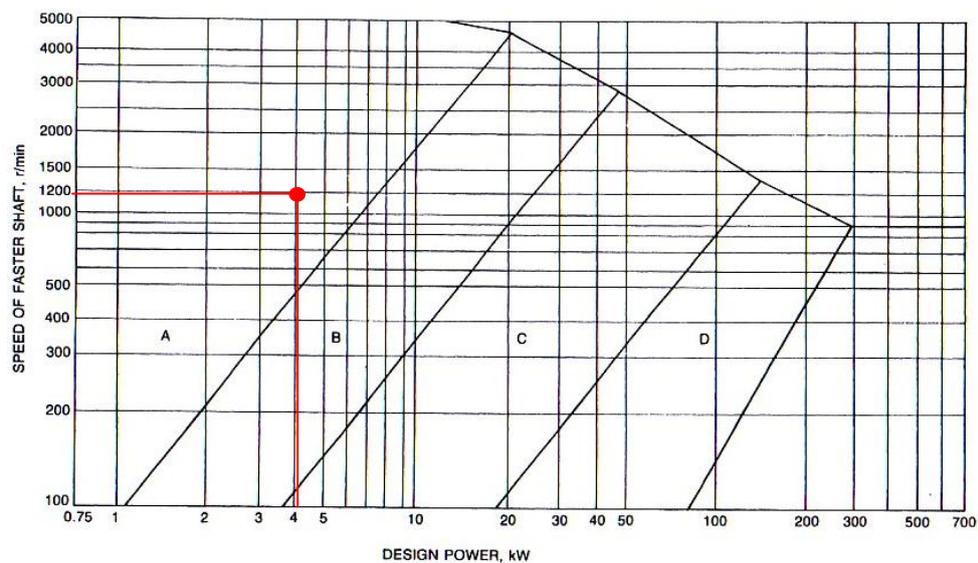


Figure A2.1 V-belt cross-section selection (SAA HB6 1999, 89)

The cross-section A is selected.

Speed ratio of the belt drive:

$$r = \frac{n_1}{n_2} = \frac{D_2}{D_1} \quad (\text{Mott 2006, 270})$$

Where: $n_1 = 1182.86$ rpm (smaller pulley speed, from Appendix 2/1)
 $n_2 = 380$ rpm (larger pulley speed, from page 11)

Therefore: $r = 3.11$

Pulley selection process:

A pulley combination that matches the required speed ratio is selected. However, only limited space is available to mount the pulleys. Therefore, smallest possible pulley combination should be selected.

Larger pulley pitch diameter:

$$D_2 = rD_1$$

New speed ratio:

$$r = \frac{D_2}{D_1}$$

Basic V-belt power rating:

The basic A-section V-belt power rating read out from SAA HB6, Table A10 on page 94. Smaller pulley speed and the diameter is use for reading the value.

Numbers of belts require:

Approximately numbers of belts required is found using the design power and the basic V-belt power.

$$N_{belts} = \frac{P_d}{P_{VB}}$$

This is not the actually required number of belt. Numbers of belt actually required will be found on the Appendix 2/3.

Preferred belt length:

$$L = 2C + 1.57(D_2 + D_1) + \frac{(D_2 - D_1)^2}{4C} \quad (\text{SAA HB6 1999, 86})$$

Where C is approximated centre distance. In this case C has a fix value of 440.57mm as the driving wheel and the larger chain sprocket already have fixed locations (refer to Figure 3.3).

Actual centre distance:

$$C = A + \sqrt{[A^2 - B]} \quad (\text{SAA HB6 1999, 86})$$

Where:

$$A = \frac{L}{4} - \pi \frac{(D_2 + D_1)}{8} \quad (\text{SAA HB6 1999, 86})$$

$$B = \frac{(D_2 - D_1)^2}{8} \quad (\text{SAA HB6 1999, 86})$$

Actual centre distance is found using the closest available belt length. The actual centre distance is the maximum centre distance of the pulleys. In this case the centre distance is already fixed. A V-belt with actual centre distance should be larger than the fixed centre distance to keep the V-belts loose at a normal position.

Actual V-belt power rating:

$$P_{VA} = (P_{VB} + P_r) \times C_\theta \times C_L$$

Using smaller pulley speed 1183rpm (from Appendix 2/2) and smaller pulley pitch diameter 90mm (from page 12), the basic belt power can be found from SAA HB6, Table A10 on page 94.

$$P_{VB} = 1.13kW \quad (\text{basic belt power, SAA HB6 1999, 94})$$

$$P_r = 0.15kW \quad (\text{speed ratio power increase, SAA HB6 1999, 94})$$

$$C_\theta = 1 \quad (\text{arc of contact factor, SAA HB6 1999, 91})$$

$$C_L = 0.98 \quad (\text{belt length factor, SAA HB6 1999, 91})$$

$$\text{Therefore: } P_{VA} = 1.25$$

Actual numbers of belts require:

$$N_{belts} = \frac{P_d}{P_{VA}}$$

$$\text{Where: } P_D = 4.1kW \quad (\text{design power, from Appendix 2/1})$$

$$\text{Therefore: } N_{belts} = 3.27$$

Thus **4 A-section V-belts** are selected.

MAIN SHAFT DESIGN**Static loading:**

Weights of components:

$$W_S = 39\text{N} \quad (70 \text{ teeth sprocket weight, SKFPTP 2006, 114})$$

$$W_{P1} = 17.2\text{N} \quad (90\text{mm pulley weight, Fenner 2008, 62})$$

$$W_{P2} = 181.5\text{N} \quad (280\text{mm pulley weight, Fenner 2008, 63})$$

$$W_{\text{disc}} = 10\text{N} \quad (\text{brake disc weight, SKFPTP 2006, 114})$$

$$W_{\text{tire}} \approx 50\text{N} \quad (13'' \text{ turf saver tire weight, Carlisle Tire Store n.d.})$$

Bearing reaction force:

Vertical and horizontal reaction forces of the bearings A and B can be calculated by using the force and bending moments:

$$\sum F = 0 \quad \text{and} \quad \sum M_A = 0$$

Resultant reaction force and bending moment can be found using Pythagorean Theorem:

$$F = \sqrt{F_V^2 + F_H^2} \quad \text{And} \quad M = \sqrt{M_V^2 + M_H^2}$$

Table A3.1 Static Loading

Shaft cross-section under consideration (refer to Figure 3.8)	Point 1	Point 2	Point 3	Point 4	Point 5
Vertical force: F_V (N)	17.2	21.9	39	51.5	17.2
Horizontal force: F_H (N)	0	0	0	0	0
Resultant force: F (N)	17.2	21.9	39	51.5	17.2
Vertical bending moment: M_V (Nm)	0	1.29	-0.43	1.29	0
Horizontal bending moment: M_H (Nm)	0	0	0	0	0
Resultant bending moment: M (Nm)	0	1.29	0.43	1.29	0

Dynamic loading:

Hub load due to chain drive:

$$F_C = \frac{2T_1}{D_1} = \frac{2T_2}{D_2} \quad (\text{Mott 2006, 538})$$

Where: $T_1 = 12.4Nm$ (maximum torque, from Appendix 1/1)
 $D_1 = 93.27mm$ (smaller sprocket, from Appendix 1/4)
 $D_2 = 283.07mm$ (larger sprocket, from Appendix 1/4)

Therefore: $F_C = 266N$ (chain drive hub load)
 $T_2 = 37.63Nm$ (main shaft torque)

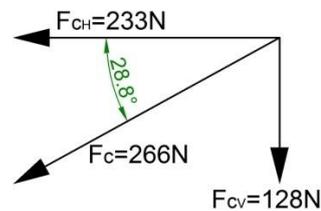


Figure A3-1 Chain Drive Hub Load on main shaft

The chain drive hub load is act at 28.8° downward from horizontal. The vertical and horizontal components of the hub load can be found using trigonometric functions (refer to Figure A3.1).

Therefore: $F_{CV} = 128N$ and $F_{CH} = 233N$

Hub load due to belt drive:

$$F_B = \frac{3T_1}{D_1} = \frac{3T_2}{D_2} \quad (\text{Mott 2006, 539})$$

Where: $T_1 = 37.63Nm$ (main shaft torque, from Appendix 3/1)
 $D_1 = 90mm$ (smaller pulley, from page 12)
 $D_2 = 280mm$ (larger pulley, from page 12)

Therefore: $F_B = 1254.3N$ (belt drive hub load)
 $T_2 = 117.1Nm$ (driving wheel torque)

The actual belt drive hub load is not known. The hub load will increase with the applied force by the control lever. Therefore, a safety factor of 2 is introduced.

Approximated hub load: $F_B = 2508.6N$ (safety factor = 2)

By analysing the Figure 3.3, found the smaller pulley hub load will act approximately 27.7° downward from horizontal.

Therefore: $F_{BV} = 1166N$ and $F_{BH} = 2221N$

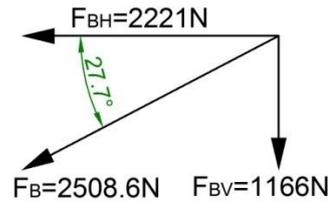


Figure A3-2 Belt Drive Hub Load on main shaft

Bearing reaction force:

Dynamic reaction forces can be found using the same equations in the static loading section (refer to Appendix 3/1).

Table A3.2 Dynamic loading scenario 1: Driving forward with top speed (see Figure 3.5)

Shaft cross-section under consideration (see Figure 3.8)	Point 1	Point 2	Point 3	Point 4	Point 5
Vertical force: F_V (N)	1183.2	1203.3	167	1330.1	1183.2
Horizontal force: F_H (N)	2221	2249.1	233	2425.9	2221
Resultant force: F (N)	2516.5	2550.7	286.7	2766.6	2516.5
Vertical bending moment: M_V (Nm)	0	88.7	81.4	88.7	0
Horizontal bending moment: M_H (Nm)	0	166.6	156.3	166.6	0
Resultant bending moment: M (Nm)	0	188.7	176.2	188.7	0
Torque: T (Nm)	18.82	18.82	18.82	18.82	18.82

Table A3.3 Dynamic loading scenario 2: Sharp right turn (see Figure 3.6)

Shaft cross-section under consideration (see Figure 3.8)	Point 1	Point 2	Point 3	Point 4	Point 5
Vertical force: F_V (N)	17.2	-173.4	167	1540.8	1183.2
Horizontal force: F_H (N)	0	-373.3	233	2827.3	2221
Resultant force: F (N)	17.2	411.6	286.7	3219.9	2516.5
Vertical bending moment: M_V (Nm)	0	1.29	70.9	88.7	0
Horizontal bending moment: M_H (Nm)	0	0	136.3	166.6	0
Resultant bending moment: M (Nm)	0	1.29	153.6	188.7	0
Torque: T (Nm)	0	0	37.6	37.6	37.6

Table A3.4 Dynamic loading scenario 3: Sharp left turn (see Figure 3.7)

Shaft cross-section under consideration (see Figure 3.8)	Point 1	Point 2	Point 3	Point 4	Point 5
Vertical force: F_V (N)	1183.2	1414.0	167	-46.6	17.2
Horizontal force: F_H (N)	2221	2650.5	233	-196.5	0
Resultant force: F (N)	2516.5	3004.1	286.7	201.9	17.2
Vertical bending moment: M_V (Nm)	0	88.7	4.5	1.29	0
Horizontal bending moment: M_H (Nm)	0	166.6	9.8	0	0
Resultant bending moment: M (Nm)	0	188.7	10.8	1.29	0
Torque: T (Nm)	37.6	37.6	37.6	0	0

Shaft diameter:

The Australian standards are used for the shaft design. Firstly some assumptions need to be made to select a shaft design formula from SAA HB6.

- Assume engine starts and stops 10 times per day.
- Assume uses 5hr per day and 2 days per week.

Therefore:

- Number of mechanism starts per year is 1040.
- Number of revolutions of shaft per year is 36.9×10^6 .

Using the mechanism starts and shaft revolutions, formula 4 is selected from SAA HB6, Table 2 in page 10.

$$D^3 = \frac{10^4 F_S}{F_R} \sqrt{\left[K_S K \left(M_q + \frac{P_q D}{8000} \right) \right]^2 + \frac{3}{16} [(1 + K_S K) T_q]^2}$$

Where:

Safety factor:

$$F_S = 1.2 \quad (\text{minimum recommended by SAA HB6 1999, 10})$$

Endurance limit:

$$F_R = 0.45 F_U \quad (\text{SAA HB6 1999, 6})$$

Mott page 49, Table 2-4 is recommended use of AISI 1040 cold-drawn steel as shaft material. Thus, select an AISI 1040 steel.

$$F_U = 552 \text{ MPa} \quad (\text{Tensile strength of AISI 1040 cold-drawn steel, from Appendix 3, Mott 2006, A-6})$$

$$\text{Therefore: } F_R = 248.4 \text{ MPa}$$

Size factor:

Size factor can be read from Fig. 1, page 11, SAA HB6. However, the shaft diameter is not known. Therefore, the diameter of a trial shaft is needed to find out.

Trial shaft diameter:

Maximum resultant bending moment and torque at a critical shaft cross-section. Critical shaft cross-sections are occurred at point 2 and 4 (see Figure 3.6 and Figure 3.7).

$$M_{max} = 188.7Nm \quad (\text{from Table A3.3 or Table A3.4})$$

$$T_{max} = 37.6Nm \quad (\text{from Table A3.3 or Table A3.4})$$

Equivalent Torque:

$$T_E = 1.15 \sqrt{M_{max}^2 + 0.75T_{max}^2} \quad (\text{SAA HB6 1999, 19})$$

$$\text{Therefore: } T_E = 221.3Nm$$

Using the equivalent torque, trial shaft diameter can be read from Fig A1, page 19, SAA HB6. The trial shaft diameter is read using a low strength steel line as the selected material's endurance limit is similar to low strength steel.

$$D_t = 19mm \quad (\text{Fig A1, SAA HB6 1999, 19})$$

Size factor can be read using the trial shaft diameter.

$$K_S = 1.17 \quad (\text{Fig. 1, SAA HB6 1999, 11})$$

Axial force at shaft cross-section under consideration:

There are no axial forces. $\therefore P_q = 0$

Bending moment at shaft cross-section under consideration:

M_q (see Table A3.2, Table A3.3 and Table A3.4)

Bending moment varies with different shaft cross-sections. Most critical bending moments are shown in Table A3.2, Table A3.3 and Table A3.4.

Torque at shaft cross-section under consideration:

T_q (see Table A3.2, Table A3.3 and Table A3.4)

Torque also varies with different shaft cross-sections.

Stress-raising factor:

There are many different kinds of stress-raising factors. The stress-raising factor is depending on the physical shape of the shaft and the method of components mounting onto the shaft.

Shaft fitted with rolling element bearings:

$$K = 1.65 \quad (\text{Fig 5, SAA HB6 1999, 13})$$

Shaft fitted component without keys or spline:

$$K = 1.85 \quad (\text{Fig 6, SAA HB6 1999, 14})$$

Stress raising factor for keys:

$$K = 2.35 \quad (\text{Fig 7, SAA HB6 1999, 15})$$

Stress raising factor for annular groove:

$$K = 2.8 \quad (\text{Fig 9, SAA HB6 1999, 17})$$

Stress raising factor for stepped shaft:

$$K = 1.5 \quad (\text{Fig 3 and 4, SAA HB6 1999, 12 and 13})$$

Stress-raising factor varies with different shaft cross-sections. See shaft design Table 3.2, Table 3.3, Table 3.4 and Table 3.5.

Keys:

Required size of the key can be select using the shaft diameter. **8mm** width and **7mm** high key is selected from SAA HB6, page 21. AISI 1020 cold-drawn steel is typically used for keys (Mott 2006, 500).

$$\begin{aligned} \text{Key} &= \mathbf{8 \times 7} && (\text{SAA HB6, 21}) \\ T &= 37.63Nm && (\text{from Appendix 3/2}) \\ D &= 26mm && (\text{shaft diameter, from Figure 3-11}) \\ N &= 3 && (\text{typical industrial applications, Mott 2006, 500}) \\ s_y &= 352MPa && (\text{yield strength of ALSI 1020 CD, Mott 2006, A-6}) \end{aligned}$$

Minimum required length of the key in shear:

$$L = \frac{2T}{\tau_d DW} \quad \text{and} \quad \tau_d = \frac{s_y}{2N} \quad (\text{Mott 2006, 499})$$

$$\text{Therefore: } \tau_d = \mathbf{58.7MPa} \quad \text{and} \quad L = \mathbf{6.17mm}$$

Minimum required length of the key in compression:

$$L = \frac{4T}{\sigma_d DH} \quad \text{and} \quad \sigma_d = \frac{s_y}{N} \quad (\text{Mott 2006, 500})$$

$$\text{Therefore: } \sigma_d = \mathbf{117.3MPa} \quad \text{and} \quad L = \mathbf{7.05mm}$$

BEARINGS SELECTION**Design Life:**

Mott is recommending use of 1500-5000 hours of design life for automotive. Therefore, select a 3000 hours design life.

$$L_d = 3000hr \quad (\text{Table 14-4, Mott 2006, 612})$$

$$L_{dm} = 213 \times 10^6 \text{ revolutions} \quad (\text{for main shaft})$$

$$L_{dw} = 68.4 \times 10^6 \text{ revolutions} \quad (\text{for wheel shaft})$$

Rotation Factor:

The inner races of the bearings are rotating.

$$V = 1 \quad (\text{Mott 2006, 613})$$

Design Load:

$$P_d = VR \quad (\text{Mott 2006, 613})$$

Where:

R is the maximum resultant dynamic load on the bearing.

$$R_A = 3004N \quad (\text{maximum load on bearing A, from Table A3.4})$$

$$R_B = 3220N \quad (\text{maximum load on bearing B, from Table A3.3})$$

$$R_C = 4597N \quad (\text{maximum load on bearing C, from Table A6.2})$$

$$R_D = 1896N \quad (\text{maximum load on bearing D, from Table A6.2})$$

Therefore:

$$P_{dA} = 3004N \quad \text{and} \quad P_{dB} = 3220N$$

$$P_{dA} = 4597N \quad \text{and} \quad P_{dB} = 1896N$$

Life Adjusted for Reliability:

$$L_{aR} = C_R L_{10} \quad (\text{Mott 2006, 616})$$

Where:

$$L_{10} = 10^6 \text{ revolutions} \quad (\text{with 90\% reliability, Mott 2006, 616})$$

Let's use 95% reliability.

$$C_R = 0.62 \quad (\text{Table 14-6, Mott 2006, 616})$$

$$\text{Therefore: } L_{aR} = 0.62 \times 10^6 \text{ revolutions}$$

Dynamic Loading:

$$C = P_d \left(\frac{L_d}{L_{AR}} \right)^{\frac{1}{k}} \quad (\text{Mott 2006, 611})$$

Where:

$$k = 3.00 \quad (\text{for ball bearings, Mott 2006, 606})$$

$$k = 3.33 \quad (\text{for roller bearings, Mott 2006, 606})$$

Therefore:

Ball Bearings:

$$C_A = 21.0kN \quad \text{and} \quad C_B = 22.6kN$$

$$C_C = 22.0kN \quad \text{and} \quad C_D = 9.1kN$$

Roller Bearings:

$$C_A = 17.3kN \quad \text{and} \quad C_B = 18.6kN$$

$$C_C = 18.8kN \quad \text{and} \quad C_D = 7.8kN$$

Static Loading:

$$C_{0A} = 21.9N \quad (\text{maximum static load on bearing A, from Table A3.1})$$

$$C_{0B} = 51.9N \quad (\text{maximum static load on bearing B, from Table A3.1})$$

$$C_{0C} = 450.9N \quad (\text{maximum static load on bearing C, from Table A6.1})$$

$$C_{0D} = 119.3N \quad (\text{maximum static load on bearing D, from Table A6.1})$$

DISC BRAKE DESIGN**Deceleration:**

Using a constant acceleration motion equation, required deceleration can be calculated:

$$V_2^2 = V_1^2 + 2aS$$

Where: $S = 1$ m (braking distance, from page 21)
 $V_1 = 24$ km/hr = 6.67 m/s (top speed, from page 21)
 $V_2 = 0$ (the PTV go kart is stopped)

Therefore: $a = -22.24$ m/s²

Force:

The required force to stop the PTV go kart from its top speed can be calculated using Newton's second law.

$$F = ma$$

Where: $M \approx 300$ kg

The PTV build plans approximated the mass to be 250kg. However, this design uses heavier components and pipes. Thus, the mass is approximated to be 300kg.

Therefore: $F = 6672$ N

Torque:

Assuming there are no power losses.

$T = 117.1$ Nm (from Appendix 3/2)

Disc radius:

Assume uniform wear will occur on the friction surfaces.

$$T = N\mu F \frac{(r_o + r_i)}{2} \quad (\text{Norton 2011, 998; Juvinall \& Marshek 2006, 716})$$

Where: r_o is the outside radius of the brake disc.
 r_i is the inside radius of the brake disc.

Also:

$$r_i = 0.577r_o \quad (\text{Norton 2011, 998; Juvinall \& Marshek 2006, 717})$$

Therefore:

$$r_o = \frac{2T}{1.577N\mu F}$$

Where:

$\mu = 0.25$ (dynamic friction coefficient of cast iron, Norton 2011, 997)

N is the number of friction surfaces. If both sides' brakes are applied, N will be 4 as each disc brake has 2 friction surfaces. If only one side brake is applied, N will be 2. However, applying only one side brake is extremely dangerous as the kart could flip over.

Therefore:

When $N = 4 \rightarrow r_o = 22.3mm$

When $N = 2 \rightarrow r_o = 44.5mm$

WHEEL SHAFT DESIGN

Static loading:

Hub load due to track tension:

Assume the hub load on the driving wheel due to track tension is 200N. By analysing the drawing on the PTV build plans, it was estimated that the hub load will act approximately 30° downward from the horizontal plane.

Therefore: $F_{TV} = 100N$ and $F_{TH} = 173N$

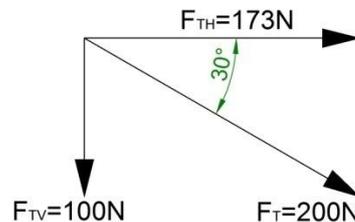


Figure A6.1 Hub Load due to track tension.

Bearing reaction force:

Bearing reaction forces can be found using the same equations in the Appendix 3/1.

Figure A6.1 Static Loading

Shaft cross-section under consideration (see Figure 3.8)	Point 1	Point 2	Point 3	Point 4	Point 5
Vertical force: F_V (N)	181.5	10	414.5	150	-73.0
Horizontal force: F_H (N)	0	0	86.6	173.2	86.6
Resultant force: F (N)	181.5	10	423.5	229.1	113.3
Vertical bending moment: M_V (Nm)	0	10.9	28.1	6.9	0
Horizontal bending moment: M_H (Nm)	0	0	0	-8.2	0
Resultant bending moment: M (Nm)	0	10.9	28.1	10.8	0

Dynamic loading:

Hub load due to belt drive:

$F_B = 2508.6N$ (from Appendix 3/2)

By analysing the Figure 3.3, found the larger pulley hub load will act approximately parallel to the horizontal.

Therefore: $F_{BV} = 0N$ and $F_{BH} = 2508.6N$

Figure A6.2 Dynamic Loading

Shaft cross-section under consideration (see Figure 3.13)	Point 1	Point 2	Point 3	Point 5	Point 7
Vertical force: F_V (N)	181.5	29	442.5	150	-82.0
Horizontal force: F_H (N)	2508.6	0	4575.7	173.2	-1893.9
Resultant force: F (N)	2515.2	29	4597.0	229.1	1895.6
Vertical bending moment: M_V (Nm)	0	10.9	29.8	7.8	0
Horizontal bending moment: M_H (Nm)	0	150.5	376.3	179.9	0
Resultant bending moment: M (Nm)	0	150.9	377.5	180.1	0
Torque: T (Nm)	117.1	117.1	117.1	117.1	0

Shaft diameter:

Minimum shaft diameter can be calculated using the same equations from the Appendix 3/4.

Size factor:

Size factor is found using the same method in the Appendix 3/5.

$$K_S = 1.3 \quad (\text{Fig. 1, SAA HB6 1999, 11})$$

Stress-raising factors:

Refer to Appendix 3/6 for stress-raising factors.

Stress raising factor for transverse hole:

$$K = 1.95 \quad (\text{Fig 10, SAA HB6 1999, 18})$$

Minimum required Pin diameter

$$d = \sqrt{\frac{4T}{\pi D \tau_d}} \quad (\text{Mott 2006, 508})$$

Where:

$$T = 117.1 \text{ Nm} \quad (\text{from Appendix 3/2})$$

$$D = 330 \text{ mm} \quad (\text{13" tire outside diameter, Carlisle Tire Store n.d.})$$

$$\tau_d = 58.7 \text{ MPa} \quad (\text{from Appendix 3/6})$$

Therefore: $d = 2.77 \text{ mm}$

Keys:

Key = **10 × 8** (selected from SAA HB6, 21)

$T = 117.1Nm$ (from Appendix 3/2)

$D = 40mm$ (shaft diameter, from Figure 3.13)

Minimum required length of the key in shear:

Minimum key length can be calculated using the equations in Appendix 3/6.

$$L = 10mm$$

Minimum required length of the key in compression:

$$L = 12.5mm \quad (\text{refer to Appendix 3/6})$$

STRUCTURAL ANALYSIS**Design stress (Allowable stress):**

$$\sigma_d = \frac{\sigma_y}{F_S} \quad (\text{Mott 2006, 188})$$

Where: σ_y is the yield strength of the tube's material. Ruukki's hollow section tubes are chosen. Tubes are available in different strengths. Optim HS 500 MH circular and square tubes are most commonly used in mechanical engineering applications (Ruukki n.d.). This type of tubes has minimum yield strength of 500MPa.

$$\sigma_y = 500\text{MPa} \quad (\text{yield strength, Ruukki n.d.})$$

$$F_S = 3 \quad (\text{safety factor, Mott 2006, 185})$$

Therefore: $\sigma_d = 166.7\text{MPa}$

Minimum section modulus:

$$\sigma_d = \frac{M_{max} y}{I} = \frac{M_{max}}{S} \quad (\text{Mott 2006, 106})$$

Where: M_{max} is the maximum bending moment on the tube.
S is the section modulus.

Middle floor tube:

$$M_{max} = 148\text{Nm} \quad (\text{maximum bending moment, from Figure 3.14})$$

The section modulus is calculated using the design power and the maximum bending moment.

$$\text{Therefore:} \quad S = 888 \text{ mm}^3$$

Side frame tubes:

$$M_{max} = 163\text{Nm} \quad (\text{maximum bending moment, from Figure 3.15})$$

$$\text{Therefore:} \quad S = 978 \text{ mm}^3$$

Inside wheel tubes:

$$M_{max} = 651\text{Nm} \quad (\text{maximum bending moment, from Figure 3.16})$$

$$\text{Therefore:} \quad S = 3905 \text{ mm}^3$$

Combined normal stress:

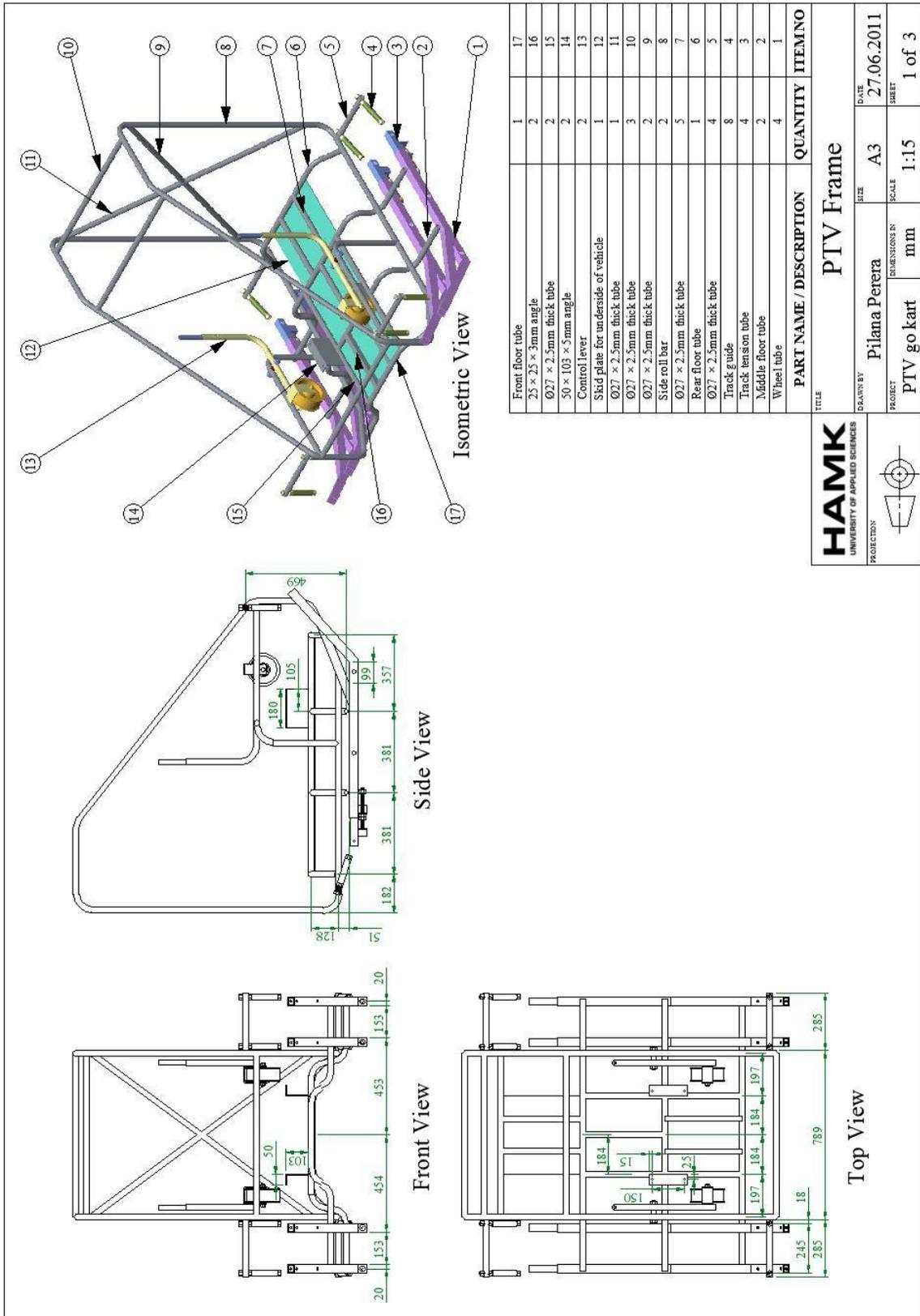
$$\sigma_c = \frac{M}{S} + \frac{F}{A} \quad (\text{Mott 2006, 117})$$

Where: F is the axial force.
A is the cross-section area.

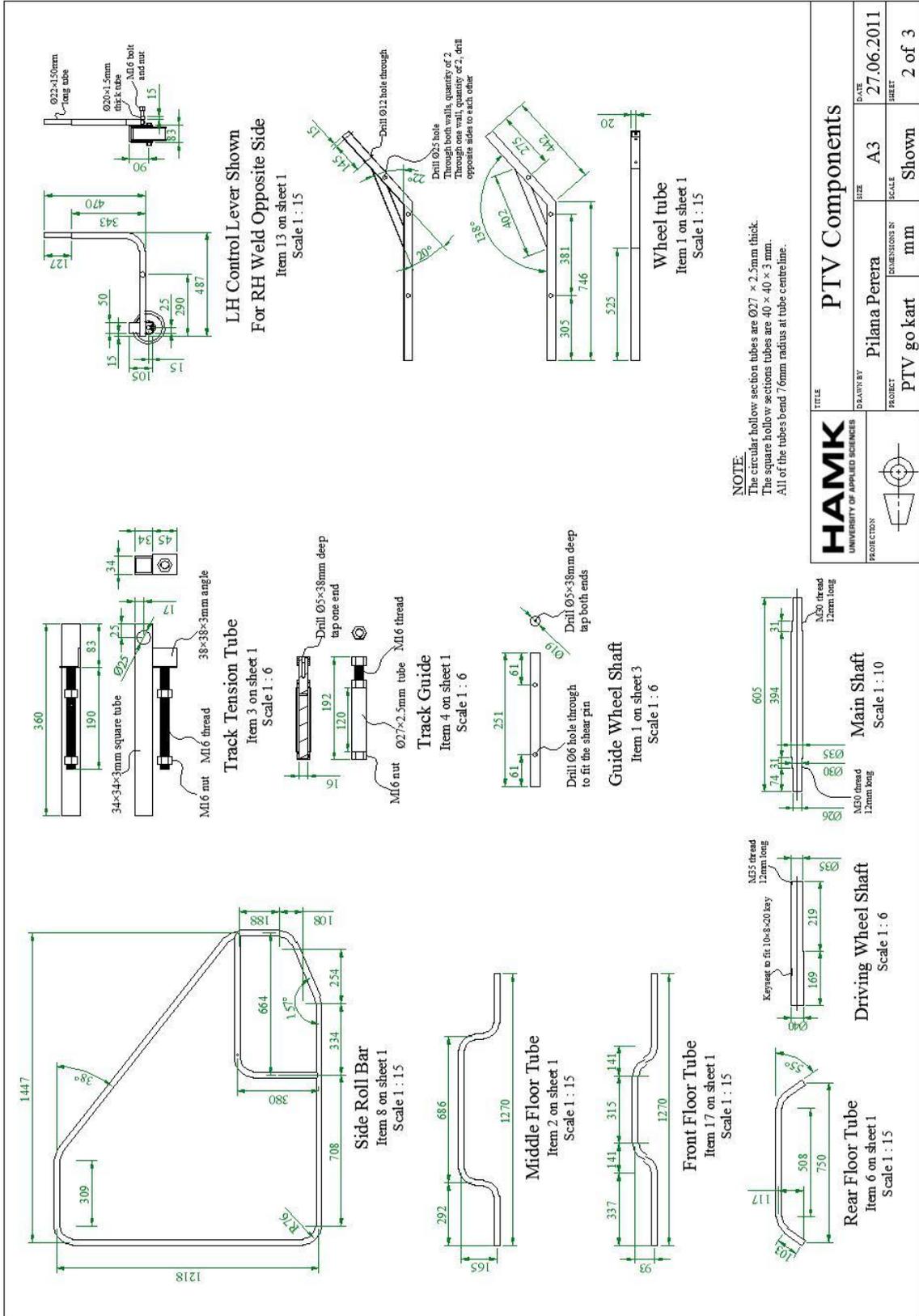
The tubes have both bending moment and axial force. The maximum combined normal stresses at the nodes were calculated. Section modulus and cross-section areas are taken from the catalogues for the selected tube. Refer to Table 3.9, Table 3.10 and Table 3.11.

CAD DRAWINGS

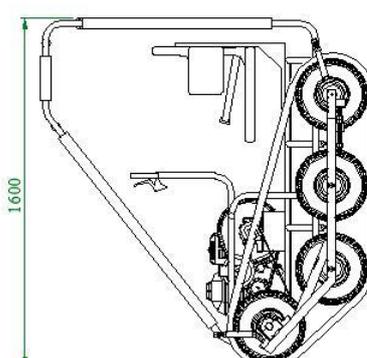
PTV Frame:



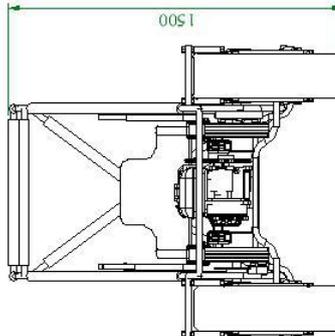
PTV Components:



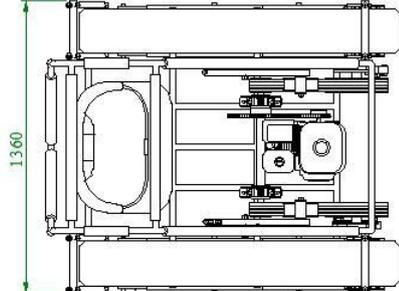
PTV Go Kart:



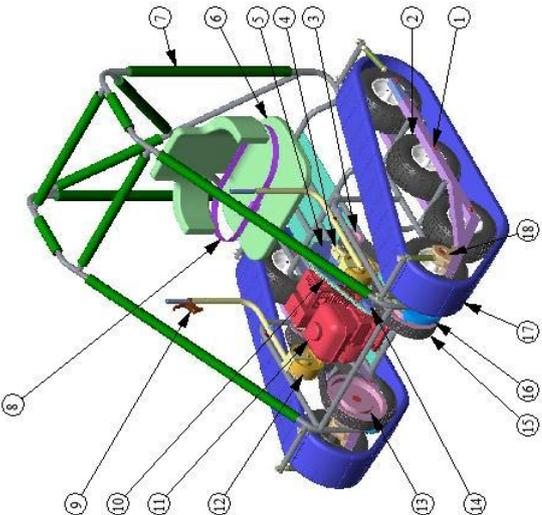
Side View



Front View



Top View



Isometric View

SOURCES:
 The bearings and bearing housings are downloaded from SKF website.
 The idler is downloaded from Fenner Drive website.
 The engine and tires are downloaded from 3D Content Centre website.

PART NAME / DESCRIPTION	QUANTITY	ITEMNO
Ø35mm bearing and housing	4	18
Track made out of 42 inch diameter tire	2	17
200mm diameter disc brake	2	16
A-section, 1530mm long V-belt	8	15
23 teeth, 08B-1 sprocket	1	14
A-section, 4 groove, Ø280mm pulley	2	13
74mm width, 152mm diameter idler	2	12
6.5HP horizontal shaft engine	1	11
91 pitches, 08B-1 Chain	1	10
Bicycle hand brake	1	9
Seat belt	1	8
Foam padding	10	7
Seat	1	6
70 teeth, 08B-1 sprocket	1	5
Ø30mm bearing and housing	2	4
A-section, 4 groove, Ø90mm pulley	2	3
13 x 5.00 - 6 wheel	8	2
Guide wheel shaft	6	1

HAMK
UNIVERSITY OF APPLIED SCIENCES

PROJECTION



TITLE		PTV Go Kart	
DESIGNED BY	SIZE	DATE	SHEET
Pilana Perera	A3	27.06.2011	3 of 3
PROJECT	DIMENSIONS IN	SCALE	SHEET
PTV go kart	mm	1:18	3 of 3