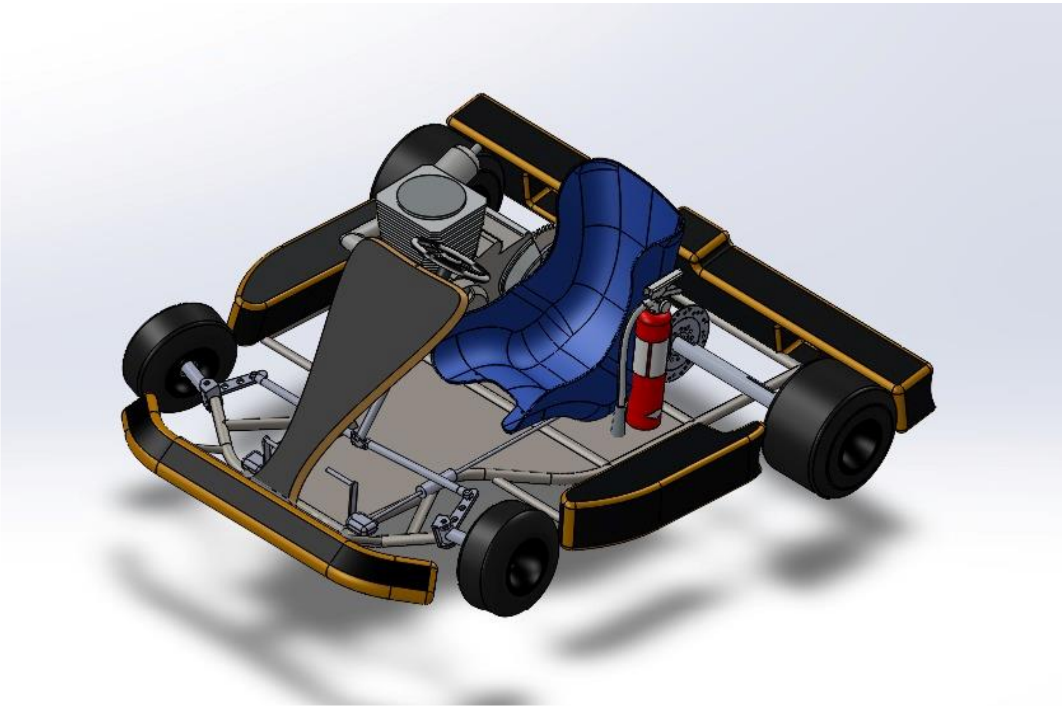
CALCULATION REPORT

ABSTRACT

The team Xtreme Squad focuses on a technically sound vehicle which is backed by a profound design and good manufacturing practices. The team's primary objective is to design a safe and functional vehicle based on rigid and torsion free frame. The design is chosen such that the Kart is easy to fabricate in every possible aspect. The report explains approach, reasons, selecting criteria and expected working of the vehicle parameters. The procedural way of explanation is used for different parts of the vehicle, which starts from approach with the help of known facts, then the design and calculation procedure has been explained. The best way known had been use to go on to the final result of all parameters.



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TRANSMISSION SYSTEM

SPECIFICATIONS OF ENGINE

Parameters	Specifications
Engine name	Hero hunk 150
Displacement	149.2 cc
No. Of cylinders	1
Max power	15.82 PS @8500 rpm
Max torque	13.5 Nm @7000 rpm
Gear box	5 speeds
Bore	57.3 mm

TABLE 1 ENGINE SPECIFICATIONS

Primary Reduction		3.3500 (67/20)
Final reduction		3.0714 (43/14)
Gear ratio	1st	3.0769 (40/13)
	2nd	1.7895 (34/19)
	3rd	1.3043(30/23)
	4th	1.0909(24/22)
	5th	0.9375(30/32)

TABLE 2 GEAR RATIO OF THE ENGINE TRANSMISSION

ENGINE CALCULATIONS

- Driving Torque:

$$Driving\ Torque\ =\ Engine\ Torque\ *\ Reduction\ *\ Efficiency$$

$$=\ 13.5\ *\ 3.5\ *\ 3 * 2.5\ * 0\ .98$$

$$\boxed{Driving\ Torque =\ 347.2875\ Nm}$$

- Friction force:

$$Friction\ force$$
$$=\ co - efficient\ of\ friction * mass\ of\ the\ vehicle$$
$$*\ acceleration\ due\ to\ gravity\ =\ 0.02 * 170 * 9.81$$

$$\boxed{Friction\ force =\ 33.354\ N}$$

- Driving force:

$$Driving\ Force\ = \frac{Driving\ Torque}{Radius\ of\ the\ Wheel} = \frac{347.2875 * 1000}{5.5\ *\ 25.4}$$

$$\boxed{Driving\ Force =\ 2485.95\ N}$$

❖ Tractive force:

$$Tractive\ force\ = driving\ force\ - friction\ force\ = 2485.95 - 33.354$$

$$\boxed{Tractive\ force = 2452.596\ N}$$

❖ Acceleration:

$$Acceleration\ = \frac{Tractive\ Force}{Mass\ of\ the\ vehicle} = \frac{2452.596}{170}$$

$$\boxed{Acceleration = \ 14.42ms^{-2}}$$

❖ Speed:

- **SPROCKET-1 (26 TEETH): GEAR RATIO = 1.857**

$$\text{Speed}\ = \frac{\text{circumference of the tyre} * \text{rpm of the Rear shaft}}{\text{Primary reduction} * \text{Gear ratio} * \text{Fifth gear reduction} * 60000}$$

$$\text{Speed} = \frac{2 * \pi * 5.5 * 25.4 * 8500}{3.35 * 1.857 * 0.9375 * 60000}$$

$$Speed = 21.32\text{ms}^{-1} = 76.752\ \text{kmph}$$

- **SPROCKET-2 (43 TEETH): GEAR RATIO = 3.0714**

$$\text{Speed}\ = \frac{\text{circumference of the tyre} * \text{rpm of the Rear shaft}}{\text{Primary reduction} * \text{Gear ratio} * \text{Fifth gear reduction} * 60000}$$

$$\text{Speed} = \frac{2 * \pi * 5.5 * 25.4 * 8500}{3.35 * 3.0714 * 0.9375 * 60000}$$

$$Speed = 12.891\text{ms}^{-1} = 46.408\ \text{kmph}$$

PARAMETER		VALUE
Drive Torque		347.2875Nm
Drive Force		2485.95N
Acceleration		$14.42ms^{-2}$
Speed	SPROCKET-1 (26 TEETH)	76.752 kmph
	SPROCKET-2 (43 TEETH)	46.408 kmph

TABLE 3 ENGINE AND TRANSMISSION CALCULATION RESULTS

SHAFT CALCULATIONS

❖ Disc location

The aim was to determine the location of the disc on the shaft. This was carried out based on the forces acting on the shaft which was analysed and by the known position of the sprockets by means of engine position.

$$Weight\ of\ disc\ =\ weight\ of\ brake\ disc + \ weight\ of\ disc\ hub = 0.744 + 0.626$$

$$weight\ of\ disc = \ 1.37\ kg$$

$$\begin{aligned} &Weight\ of\ sprocket \\ &\quad = \ weight\ of\ big\ sprocket + \ weight\ of\ small\ sprocket \\ &\quad + \ weight\ of\ sprocket\ hub = \ 0.658 + 0.844 + 0.766 \end{aligned}$$

$$weight\ of\ sprocket = \ 2.268\ kg$$

Component	Mass (kg)	Radius (mm)	Distance from R_A (mm)
Left bearing	3.476	55	0
Disc	1.37	105	x
Sprocket	2.268	82.5	197.5
Right bearing	3.476	55	L=795

TABLE 4 DETAILS OF COMPONENT DIMENSIONS IN SHAFT CALCULATION

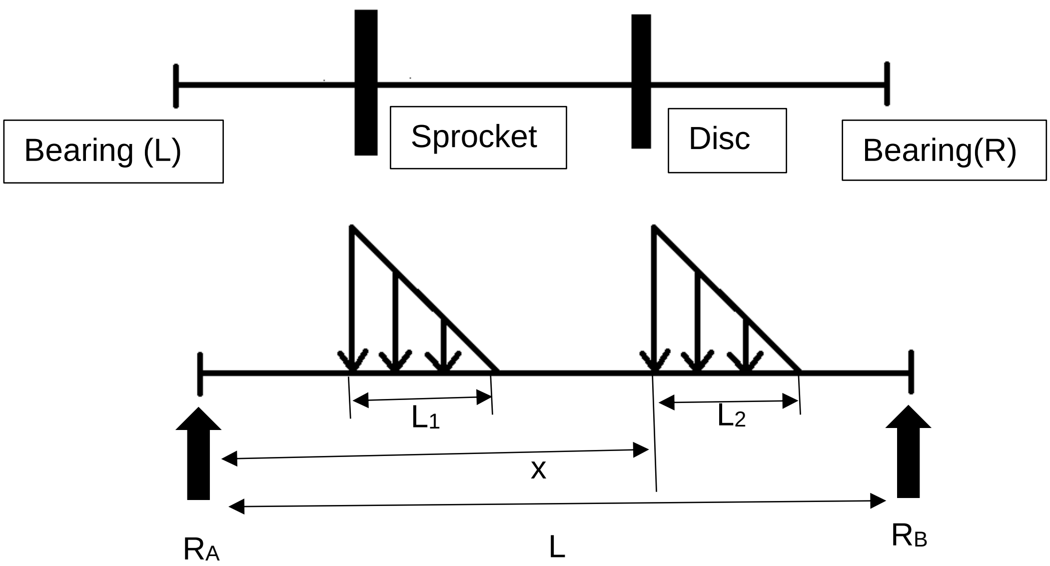

DIAGRAM 1 REPRESENTATION OF SHAFT (FRONT VIEW) AND LOADS ACTING ON IT

Taking moment about R_A (by convention clockwise moment is positive)

$$738.43 * x = R_B * 795 - (22.2264 * 90) * 197.5$$

$$x = \frac{795 * R_A - 395074.26}{738.43}$$

$$x = \frac{395074.26}{738.43}$$

$$x = 535.019\, mm$$

i.e.., brake disc is 535.019 mm from left wheel – sprocket side (or) 259.981mm (approx. 260 mm) from right wheel – disc side

❖ Keys calculation

According to design data handbook (1) for 30mm diameter shaft from table 17-1 (1) respective specifications for keys were taken.

Keyway depth, In shaft = 5 mm

In hub = 3.3 mm

Key dimensions are

Height = 8.3 mm

Width = 5 mm

Length of the keys were based on the disc, wheel hubs and sprockets cross-sectional area on the shaft.

BRAKING SYSTEM

The braking system is the critical sub-system of an automobile. Braking system that meet the requirements of rulebook were selected and calculations were done to achieve an efficient braking system.

BRAKING SYSTEM SPECIFICATIONS:

Brake type	Single disc at rear axle
Brake fluid	DOT 3
Brake disc diameter	190 mm
Master cylinder diameter	15.19 mm
Diameter of piston inside caliper	27.44 mm
Area of master cylinder	0.0001999 mm^2
Area of caliper piston	0.000638 mm^2
Co-efficient of friction between tire and road (µ)	0.7 (dry roads)
Co-efficient of friction between brake pads and brake disc	0.4
F/R Distribution	40/60
Diameter of rear tire	11 inches
Wheelbase (L)	1020 mm
Front track width	960 mm
Rear track width	1040 mm
Cg Height (h)	260 mm

TABLE 5 BRAKING SYSTEM SPECIFICATIONS

Static Analysis:

Overall weight of the kart including the driver was 170kg, taken based upon the appropriate approximation.

❖ Mean Effective Radius

$$R = \frac{(\mathrm{r1} + \mathrm{r2})}{2} = \frac{0.095 + 0.075}{2}$$

$$\boxed{R = 0.085\ m}$$

❖ Static Load at front axle:

Reaction force at front

$$R_f \ = Total\ weight\ of\ kart * \frac{F}{R} Ratio * g = \ 170 * 0.4 * 9.81$$

$$\boxed{R_f \ = 667.08\ N}$$

❖ Static load at rear axle:

Reaction force at rear

$$Rr \ = (Total\ weight\ of\ kart * g) - Static\ load\ at\ front\ axle = \ (170 * 9.81) - \ 667.08$$

$$\boxed{R_r \ = \ 1000.62\ N}$$

Dynamic analysis:

❖ Acceleration:

The acceleration was calculated upon the force required to overcome the frictional force and reaction force of the kart due to its own weight.

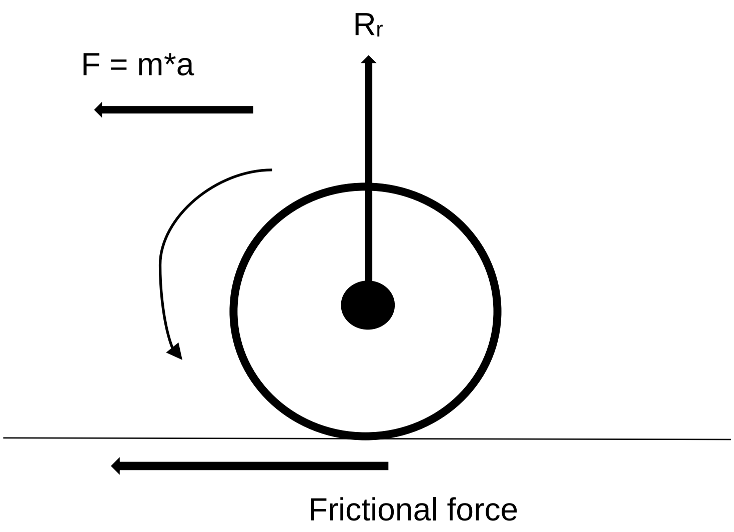

DIAGRAM 2 FREE BODY REPRESENTATION OF FORCES ON A WHEEL

$$Maximum\ frictional\ force \ = \ \mu * Rr \ = 1000.62 * 0.7$$

$$= \ 7000.43\ N$$

$$Acceleration \ = \frac{Maximum\ frictional\ froce}{\text{Mass of kart}} = \frac{700.434}{170}$$

$$\boxed{a = 4.12 \ m/s^2}$$

❖ Load transfer (LT):

During straight line cruise there will be transfer of load from the front axle to the rear axle due to inertia effect.

$$L_T = \frac{W * a * h}{1.02} = \frac{170 \times 4.12 \times 0.26}{1.02}$$

$$\boxed{Load\ transfer = 178.53\ N}$$

❖ Dynamic load at rear axle:

$$Dynamic\ load\ at\ rear\ axle = Static\ load\ at\ rear\ axle + Load\ transfer$$

$$= 1000.62 + 178.53$$

$$\boxed{Dynamic\ load\ at\ rear\ axle = 1179.15\ N}$$

❖ Dynamic load at front axle:

$$Dynamic\ load\ at\ front\ axle = Static\ load\ at\ front\ axle - Load\ transfer$$

$$= 667.08 - 178.53$$

$$\boxed{Dynamic\ load\ at\ front\ axle = 489.27\ N}$$

❖ Frictional force at rear:

$$Frictional\ force\ at\ rear = Dynamic\ load\ at\ rear * \mu = 1179.15 * 0.7$$

$$\boxed{Frictional\ force\ at\ rear = 825.4\ N}$$

❖ Torque on tire:

$$Torque\ on\ tire = Frictional\ force * radius\ of\ tire = 825.4 * 0.1397$$

$$\boxed{Torque\ on\ tire = 115.3\ Nm}$$

❖ Braking force:

$$[\,Torque\ on\ tire = Torque\ on\ disc\,]$$

$$Braking\ force = \frac{Torque\ on\ disc}{Effective\ radius} = \frac{115.3}{0.085}$$

$$\boxed{Braking\ force = 1356.47\ N}$$

This is the required braking force of the kart that is needed.

Deceleration and Stopping distance:

❖ Force on master cylinder (F_{mc}):

Drivers' effort = 220 N

Pedal ratio = 5:1

$$Force\ on\ master\ cylinder\ =\ Drivers\ effort * Pedal\ ratio = \ 220 * 5$$

$$\boxed{F_{mc} = 1100\ N}$$

❖ Pressure in master cylinder (P_{mc}):

$$Pressure\ in\ master\ cylinder\ = \frac{Force\ at\ master\ cylinder}{Area\ of\ master\ cylinder} = \frac{1100}{0.0001999}$$

$$\boxed{P_{mc} = 5.5027 * 10^{6}\ \ N/mm2}$$

❖ Force on the caliper (F_c)

$$Pressure\ in\ master\ cylinder\ (Pmc)\ =\ Pressure\ in\ the\ caliper\ (Pc)$$

[Neglecting frictional losses in the brake lines]

$$F_c =\ Pc * Area\ of\ caliper\ piston =\ 5.00025 * 10^{6} * 0.000638$$

$$\boxed{Fc = 3510.75\ N}$$

❖ Clamping force:

Multiplies by 2 because the caliper consists of 2 pistons thereby both the piston with equal force will be acting on the disc.

$$Clamping\ force\ =\ Fc *\ 2 =\ 3510.75 * 2$$

$$\boxed{clamping\ force = 7021.51\ N}$$

❖ Actual braking force:

$$Actual\ braking\ force\ =\ Clamping\ force * Friction\ between\ disc\ and\ pads$$

$$=\ 7021.51 * 0.4$$

$$\boxed{= Actual\ braking\ force =\ 2808.6\ N}$$

Actual braking force > Required braking force

2808.6 N > 1356.47 N

As we see the calculated actual braking force is more than that of required braking force that is calculated, the kart will stop easily without any disturbance

❖ Torque of the disc:

$$Torque\ of\ the\ disc\ =\ Braking\ force * Effective\ radius\ of\ the\ disc$$

$$= 2808.6 * 0.085$$

$$\boxed{= \ Torque\ of\ the\ disc = \ 238.73\ Nm}$$

❖ Deceleration:

$$Deceleration\ = \frac{Braking\ force}{Mass\ of\ the\ kart} = \frac{2808.6}{170}$$

$$\boxed{= Deceleration = -16.52\ m/s^2}$$

❖ Stopping distance(s):

$$Assume\ the\ kart\ is\ moving\ with\ a\ velocity\ of\ 45\ km/hr\ (u\ =\ 12.5\ m/s)$$

$$v^2 - u^2 = 2 * \ a * s$$

$$0 - \ (12.5)^2 = 2 * (-16.52) * s$$

$$\boxed{s\ = 4.72\ m}$$

❖ Stopping time (t):

$$v\ =\ u + (a * t)$$

$$0\ =\ 12.5 + (-16.52 * t)$$

$$\boxed{t\ =\ 0.75\ s}$$

This result is for a driver effort of 220 N and fixed pedal ratio of 5:1. For optimum result by increasing the driver's effort stopping distance can be minimized.

Drivers' effort (N)	Stopping distance (m)	Stopping time (sec)
150 N	6.9	1.1
200 N	5.2	0.83
220 N	4.7	0.75
250 N	4.61	0.66

TABLE 6 REPRESENTATION OF STOPPING DISTANCE AND STOPPING TIME OVER DIFFERENT DRIVER EFFORT CONDITION

Thermal analysis:

Due to the frictional force that is created between the disc and the caliper brake shoes, the temperature increases.

❖ Kinetic energy (KE):

$$Kinetic\ energy\ of\ the\ vehicle\ = \frac{1}{2} * m * v^2 = \frac{1}{2} * 170 * (12.5)^2$$

$$\boxed{Kinetic\ energy\ of\ the\ vehicle = \ 13281.25\ J}$$

❖ Brake power (BP):

$$Brake\ power\ = \frac{KE}{Brake\ time} = \frac{13281.25}{0.832}$$

$$\boxed{= \ Brake\ power = \ 15963.04\ W}$$

❖ Rubbing area:

$$Rubbing\ area\ on\ one\ side\ of\ the\ disc\ = \frac{\pi}{4} * (D^2 - d^2) = \frac{\pi}{4} * (0.19^2 - \ 0.17^2)$$

$$\boxed{Rubbing\ area\ on\ one\ side\ of\ the\ disc\ = \ 5.654 * 10^{-3}\ m^2}$$

❖ Total rubbing area:

$$Total\ rubbing\ area = 5.654 * 10 - 3 * 2$$

$$\boxed{= Total\ rubbing\ area = 0.0113\ m^2}$$

❖ Heat flux (q):

$$Heat\ flux = \frac{Brake\ power}{total\ rubbing\ area} = \frac{15963.04}{0.0113}$$

$$\boxed{q \ = \ 1411442.6\ W/m^2}$$

Parameter	Value
Brake Pedal lever Ratio	5:1
Stopping distance	4.72 m
Stopping time	0.75 s
Brake power	15963.04 W

TABLE 7 BRAKING CALCULATION RESULTS

STEERING SYSTEM CALCULATIONS:

The calculations were done to achieve an Ackerman geometry close to 100% by maintaining minimum turning radius.

Wheel base (B)	1020 mm
Track width (W)	960 mm
Distance between c-clamp to c-clamp (c)	760 mm
Tie-rod length (a)	716 mm
Steering arm length (h)	120 mm
Castor angle	8^0
King pin inclination (KPI)	10^0
Camber angle	0^0

TABLE 8 STEERING SYSTEM SPECIFICATION

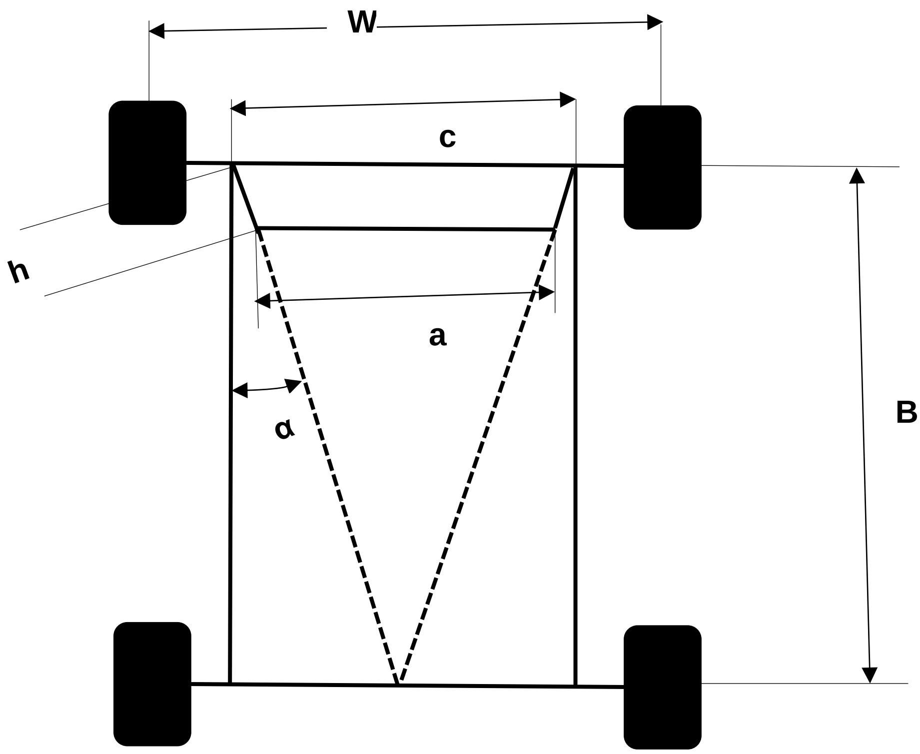

DIAGRAM 3 ACKERMANN AND VARIOUS OTHER PARAMETERS REPRESENTATION

Ackerman angle (α):

From geometry,

Ackerman angle is given by,

$$\alpha \ = \tan^{-1}(\frac{c-a}{h}) = \tan^{-1}(\frac{760-716}{120})$$

$$\boxed{\alpha \ = \ 20.136°}$$

Castor Trial:

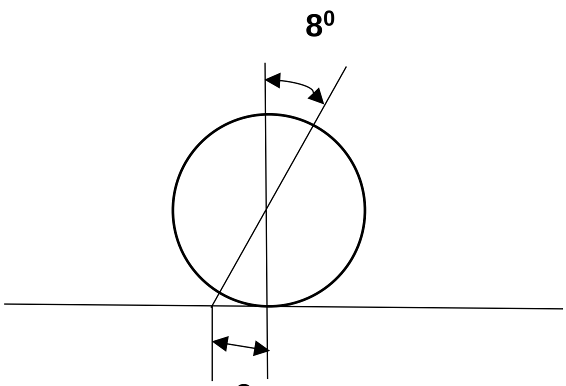

DIAGRAM 4 REPRESENTATION OF WHEEL FOR CASTOR TRAIL CALCULATION

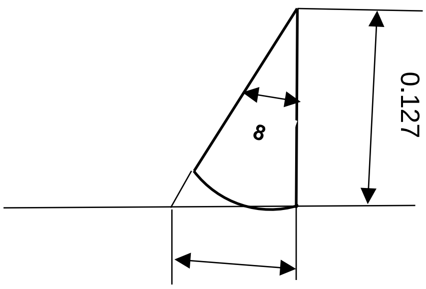

DIAGRAM 5 REPRESENTATION OF WHEEL FOR CASTOR TRAIL CALCULATION

Radius of wheel = r_f = 5 inch = 0.127 m

From figure the castor trial is given by,

tan (castor angle) = $\frac{c}{r^f}$

$$Castor\ trial\ (c) = \ rf * \tan(castor\ angle) = \ 0.127 * tan\ (80)$$

$$\boxed{Castor\ trial\ (c) \ = \ 0.0178\ m}$$

Scrub Radius:

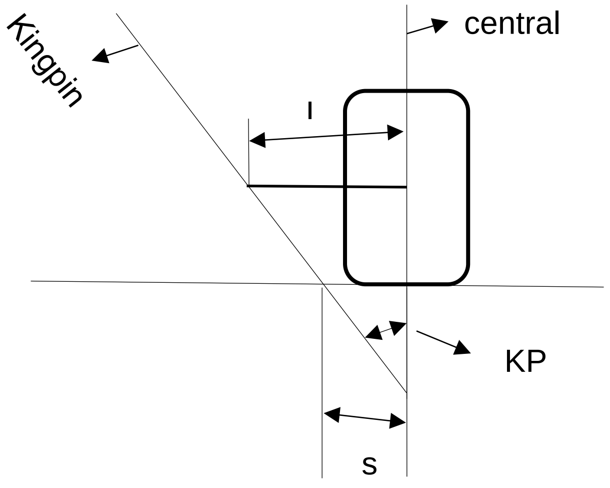

DIAGRAM 6 SCRUB RADIUS CALCULATION

$$l = spindle\ length = 0.1\ m$$

$$s = scrub\ radius$$

$$rf = radius\ of\ tire = 0.127\ m$$

$$KPI = 10$$

From figure the scrub radius is given by,

$$s = l - (rf \times tan(KPI)) = 0.1 - (0.127 \times tan(100))$$

$$\boxed{s = 0.077\ m}$$

Turning radius and steer angles:

The aim was to minimize the turning radius while maintaining an Ackerman geometry close to 100 %.

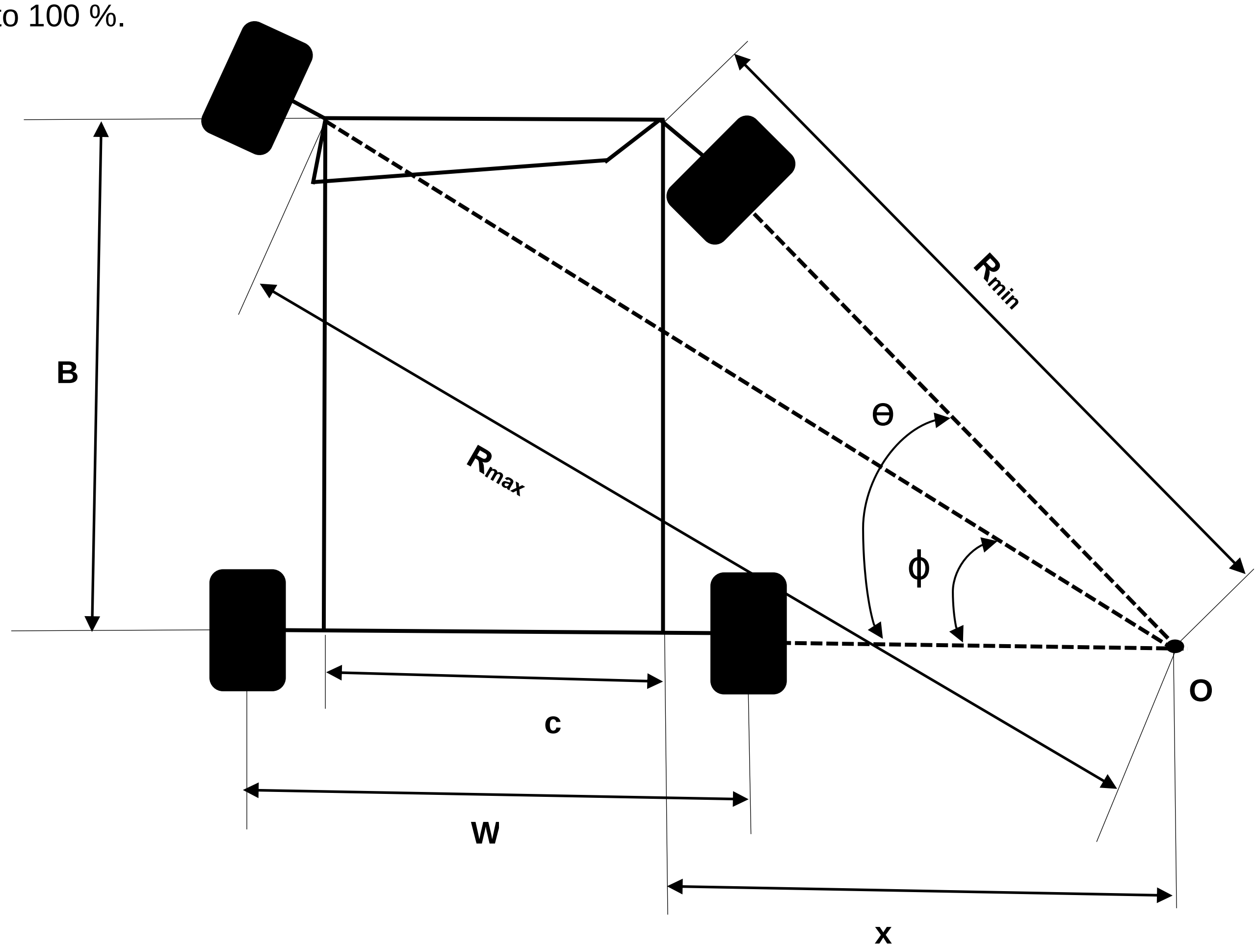

DIAGRAM 7 REPRESENTATION OF WHEELBASE

❖ The outer lock angle(ϕ):

By fixing the inner lock angle(ɵ) to 35^{0},

The outer lock angle from the figure is given by,

$$cot(\phi) - cot(\theta) = \frac{c}{B}$$

$$cot(\phi) - cot(350) = \frac{760}{1020}$$

$$\boxed{\therefore \phi = 24.70}$$

From figure by substituting the steer lock angles, the axis of the tires will intersect at a particular point called instantaneous center ‘o’.

$x = 1457.64\,mm$ (The distance between the instantaneous center ‘o’ to right side of the c-clamp axis at the rear)

❖ Inner turning radius (R_{min}):

From geometry,

$$R^2 min \; = B^2 + x^2 = 1020^2 + 1457.64^2$$

$$\boxed{Rmin \; = \; 1779.08 \, mm}$$

❖ Outer turning radius (R_{max}):

From geometry,

$$R^2 max \; = B^2 + (\, x + c \,)^2 = 1020^2 + (\, 1452.64 + 760 \,)^2$$

$$\boxed{Rmax \; = \; 2440.97 \, mm}$$

❖ Turning radius (R):

$$R^2 = B^2 + \left(x + \frac{c}{2}\right)^2 = 1020^2 + \left(1457.64 + \frac{760}{2}\right)^2$$

$$\boxed{R \; = \; 2101.74 \, mm}$$

❖ Turning radius w.r.t cg:

$$R^2 cg \; = (B - cg \; distance \; from \; the \; front \; axle)^2 + \left(x + \frac{c}{2}\right)^2$$

$$= (1020 - 548.47)^2 + \left(1457.64 + \frac{760}{2}\right)^2$$

$$\boxed{Rcg \; = \; 1897.2 \, mm}$$

Ackerman percentage (%):

Ackerman = $\tan^{-1}\left(\frac{B}{\frac{B}{\tan(\phi)} - c}\right) = \tan - 1\left(\frac{1020}{\frac{1020}{\tan(24.75)} - 760}\right)$ = 35.077°

Therefore, Ackerman % = $\frac{\text{Inner lock angle}}{\text{Ackerman}} * 100 = \frac{35}{35.077} * 100$

= 0.99 * 100

$\boxed{\text{Ackerman \% = 99.7 \%}}$

Thus 99.7 % ≅ 100 %, which is very much efficient during the cornering and gives high performance in overall.

Steering effort:

$$Considering \; turning \; velocity, v \; = \; 20 \; kmph$$

$$Turning\ velocity\ (v)\ =\ 5.55\ m/s$$

$$At\ turning\ radius, R\ =\ 1897.2\ mm$$

$$Lateral\ acceleration, a_y\ =\ \frac{v^2}{R}\ =\ \frac{5.55^2}{1.897}\ =\ 16.23\ m/s^2$$

- ✓ Weight transfer during cornering(ΔW),

$$\Delta W\ =\ \frac{Weight\ at\ front\ axle *\ a * cg\ height}{front\ track\ width}\ =\ \frac{68 * 16.23 * 0.26}{0.96}$$

$$\Delta W\ =\ 298.9\ N$$

Weight on each front tire= $W_f\ =\ 0.4 * 170 * 9.81 * \frac{1}{2}\ =\ 333.54\ N$

Weight on inner front tire = $Wi\ =\ Wf - \Delta W\ =\ 333.54 - 298.9\ =\ 35\ N$

Weight on outer front tire = $Wo = Wf\ +\ \Delta W\ =\ 333.54\ +\ 298.9\ =\ 632.44\ N$

- ✓ $Lateral\ force\ on\ inner\ front\ tire\ =\ \frac{weight\ on\ inner\ front\ tire}{g}\ \times\ \frac{v^2}{R}$

$$=\ \frac{35}{9.81}\ \times\ 16.23$$

$$Lateral\ force\ on\ inner\ front\ tire\ =\ 57.9\ N$$

- ✓ $Lateral\ force\ on\ outer\ front\ tire\ =\ \frac{weight\ on\ outer\ front\ tire}{g}\ \times\ \frac{v^2}{R}$

$$=\ \frac{632.44}{9.81}\ \times\ 16.23$$

$$Lateral\ force\ on\ outer\ front\ tire =\ 933.9\ N$$

- ✓ $Moment\ due\ to\ lateral\ force\ =\ Lateral\ Force\ x\ Wheel\ Radius\ x\ tan\ (caster\ angle)$

For Inner Wheel, $MLateral\ I\ =\ 57.9\ x\ 0.127\ x\ tan\ (8°)\ =\ 1.0334\ Nm$

For outer Wheel, $MLateral\ o\ =\ 933.9\ x\ 0.127\ x\ tan\ (8°)\ =\ 16.67\ Nm$

- ✓ $Moment\ at\ Kingpin\ due\ to\ lateral\ force\ =\ MLateral\ i\ +\ MLateral\ o\ =\ 17.7\ Nm$

Now, $Moment\ Due\ to\ Self - Aligning\ torque\ =\ Lateral\ Force\ x\ \frac{Contact\ Patch}{KPI}$

For Inner Wheel, $MSAT\ I\ =\ 57.9\ x\ \frac{0.009}{10}\ =\ 0.052\ Nm$

For Outer Wheel, $MSAT\ O\ =\ 1046.3\ x\ \frac{0.009}{10}\ =\ 0.941\ Nm$

- ✓ $Moment\ at\ Kingpin\ due\ to\ Self-Aligning\ torque\ =\ MSAT\ i\ +\ MSAT\ o$

$$=\ 0.052\ +\ 0.941 =\ 0.993\ Nm$$

- ✓ $Net\ Moment\ at\ Kingpin\ =\ Moment\ due\ to\ Lateral\ Forces\ +\ Moment\ due\ to\ Self-Aligning\ torque$

$$Net\ Moment\ at\ Kingpin\ =\ 18.693\ Nm$$

- ✓ $Force\ in\ Steering\ Arm\ =\ \frac{Net\ Moment\ at\ Kingpin}{Length\ of\ steering\ Arm} = \frac{18.6}{0.12}$

$$Force\ in\ Steering\ Arm\ =\ 155\ N\ (Perpendicular\ direction)$$

- ✓ $Horizontal\ Force\ along\ tie-rods\ =\ Force\ in\ steering\ column\ \times \sin(Ackerman\ angle)$

$$= 155\ \times \sin(20.13)$$

= 54.1 N

$$Horizontal\ Force\ along\ tie-rods =\ 54.1\ N$$

- ✓ $Moment\ at\ Steering\ Column\ =\ Force\ Along\ tie-rod\ x\ Tripod\ radius$

$$=\ 54.1 \times\ 0.1$$

$$Moment\ at\ Steering\ Column =\ 5.41\ Nm$$

- ✓ $Steering\ Effort\ =\ \frac{Moment\ at\ steering\ column}{Steering\ Wheel\ Radius} = \frac{5.41}{0.14}$

$$\boxed{Steering\ Effort\ =\ 38.64\ N}$$

This is the maximum steering effort that is required in order to turn the wheels.

Parameter	Value
Steering	Mechanical linkage
Ackermann angle	20.136°
Ackerman Percentage	99.7 %
Inner lock angle	35°
Outer lock angle	24.7°
Turning radius	2101.74 mm
Steering ratio	1:1
Scrub radius	0.077 m
Steering effort	38.64 N

TABLE 9 STEERING CALCULATION RESULTS

Reference

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[2] Rulebook, IKC – 5

[3] DETAILED ESIGN CALCULATIONS & ANALYSIS OF GO-KART VEHICLE-vinash Barve[1], Vivek Gurve[2], Gaurao Tapre[3], Arvind Totey[4] Scholar Mechanical Engineering, S. B. Jain Institute of Technology Management and Research, Nagpur, India[1, 2, 3] Assistant Professor, Mechanical Engineering, S. B. Jain Institute of Technology Management and Research, Nagpur, India[4]

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[5] Design and Fabrication of Race Spec Go-Kart Simranjeet Singh, Aniket Badgujar, Pushparaj Patil, Gaurang Kadam 1,2(Automobile, Rajendra Mane College of Engineering and Technology, India)

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