CHAPTER 3

DESIGN OF THE STEERING DEVICE

One of the principal aspects to be concerned about is the way the Ruck ´n Roll is going to be steered. Making a recall to the premise of space and weight minimization, there is a need to evaluate between several options which kind of device is the most suitable to be adapted to the Ruck ´n Roll.

3.1 Requirements

The steering device must be design under the following concerns:

1. Manipulation must be located in the Module 1 of the Ruck ´n Roll, it means in the rear axle.
2. Feet must not be involved in the operation of the steering device.
3. Hands must be in a secure position to provide stability and a high security perception.
4. Steering must be achieved by a weight-shifting motion of the driver with a specified minimum torque range from 10 N·m to 20 N·m and a maximum steering angle of 5°

3.2 Products already in the market: principles of steering

To find a proper way to achieve steering in this project, it is necessary to research in the market what products with similar functionality already exist, what their steering principles are and compare advantages and disadvantages between each one of them.

3.2.1 Skateboard

A skateboard (figure 3.1) is an entertainment transporting device where the user provides the power through a leg´s motion against the floor impulsing the skateboard forward.

Figure 3.1 Skateboard

The skateboard has the following components (Wikipedia, 2005[14]):

1. Board: usually made by placing 7 layers of wood one over another. It is where the feet of the user are located.
2. Axles: two of this parts coupled by the sides, without reaching the perimeter line of the board. Generally made of aluminum, they support the wheels and perform the steering. They have some subcomponents like the Kingpin (principal screw), rubs and covers.
3. Rollers: 8 of them for each wheel. They are classified under ABEC norms.

4. Binders: Sticked in the top surface of the board, they provide the roughness to avoid feet slipping while driving.
5. Screw set: 8 screws are needed to keep the axis and the board together and 4 nuts to prevent the wheels of getting out of place.

3.2.2 Skateboard principle of steering

In this device, the two sets of wheels are attached to the board via tilted steering axes (Figure 3.2). While the main purpose of this construction is to give the rider the ability to steer, it also provides stability as it is shown below (Wisse , 2005[15])

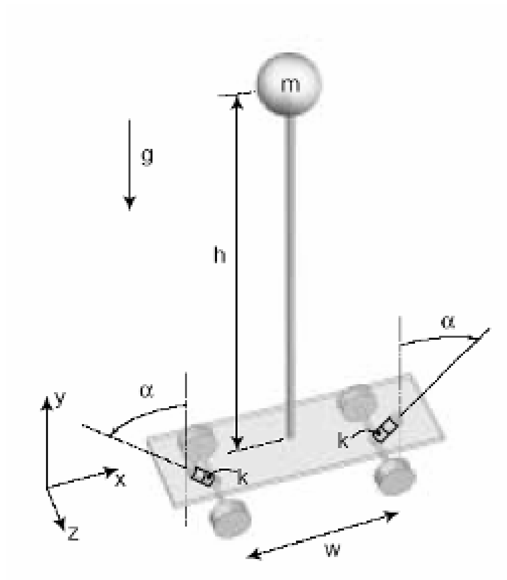


Figure 3. 2 Settings for the analysis of
the skateboard, from M.Wisse, A. L. Schwab, Skateboards,
Bicycles, and Three-dimensional BipedWalking Machines:
Velocity dependent Stability by Means of Lean-to-yaw Coupling

The following assumptions are to be stated (Wisse , 2005[16]):

1. The board and the wheels are assumed to be massless.
2. Height between the board and the floor is neglected.
3. There is always contact between all four wheels and the floor, it means that the model cannot tip over.

4. Rider is modeled as a single point mass at height h above the floor, rigidly attached to the skateboard.
5. The distance between the front and rear wheels is *w*.
6. The steering axes are mounted at an angle *a* with respect to vertical, so steering in that direction is caused by sideways leaning of the rider.
7. The steering axes are equipped with rotational springs with stiffness *k*.
8. The model presents fore-aft and sideways symmetry.

The skateboard is a non-holonomic system, it means it can not slip sideways but it can move to a sideways position by a sequence of steering actions. Due to this it has a smaller velocity space (lean and ride) than coordinate space (lean, x- and y- position and orientation in plane). The linearized equations of motion are considered, so the forward velocity can be considered as a parameter (Wisse, 2005[17]). The linearized model has a single degree of freedom: the sideways lean angle of the rider θ (figure 3.3).

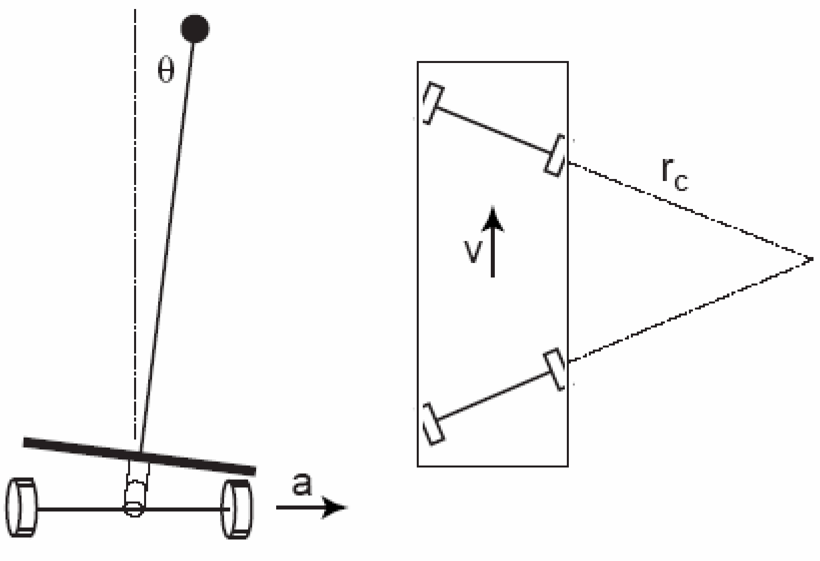


Figure 3. 3 Lean angle and its yaw effect, from M.Wisse, A. L. Schwab, Skateboards, Bicycles, and Three-dimensional BipedWalking Machines: Velocity dependent Stability by Means of Lean-to-yaw Coupling

With zero velocity in the forward direction, it presents the same behavior than of an inverted pendulum under the action of a destabilizing gravity torque and a stabilizing spring torque. Note that the tilt angle a cancels out in their projected torques, although the two springs act on the tilted joints.

While being ridden with a velocity v, the skateboard performs a turn only with a non-zero lean θ. Both the velocity and the radius of curvature r_c (Figure 3.3) determine the sideways acceleration of the board according to

$$a = \frac{v^2}{r_c} = \frac{2v^2}{w \tan \alpha}\theta \qquad \text{(Wisse , 2005[18])} \qquad (3.1)$$

The linearized model´s system equation is given by

$$mh^2\ddot{\theta} + \left(2k - mgh + \frac{2mh}{w \tan \alpha}v^2\right)\theta = 0 \quad \text{(Wisse , 2005[19])} \qquad (3.2)$$

This means that if the spring k is high enough to counteract the instability presented in the inverted pendulum, the system is never unstable. Otherwise, it can always be made stable by velocity means. The required velocity is directly proportional to the angle α. The expression for the critical velocity is

$$v_{\min} = \sqrt{\frac{(mgh - 2kw)\tan \alpha}{2mh}} \qquad \text{(Wisse , 2005[20])}$$

(3.3)

3.3.1 Bicycle

The bicycle (Figure 3.4) is a vehicle with two wheels, usually of the same size. It is used for transporting one or more people, due to the force exerted on the pedals, transmitted to the pinion in the rear wheel through a circular chain. It was introduced in Europe in the nineteenth century (Wikipedia, 2005[21]).

Figure 3. 4 Bycicle (wordsports.com)

The main parts of the bicycle are:

1. Chasis: the structure that supports the rest of the components and the rider.
2. Wheels
3. Machinery: Group of elements (chain, pinions, plates, connecting rods, axles and pedals) that take advantage of the mechanical force, converting it into displacement.
4. Brakes
5. Steering
6. Handle-bar

3.3.2 Bycicle principle of steering

Starting from the static equilibrium case, it is easy to notice the high unstability of a bicycle at rest, and that a moderate speed can easily stabilize it. From some studies done by Hubbard (Wisse , 2005[22]) it is shown that an uncontrolled bike can be asymptotically stable in a certain speed range.

Considering one of the simplest bicycle models, an uncontrolled bike with a rigid rider attached. In the study *Skateboards, Bicycles, and Three-dimensional Biped Walking Machines: Velocity dependent Stability by Means of Lean-to-yaw Coupling,* done by M.Wisse and A. L. Schwab, it is defined as an example of a dynamically coupled lean-to yaw motion due to the hands free operation of the bike and it is based in a recent bicycle benchmark publication by Schwab, Meijaard and Papadopoulos (Wisse , 2005[23]).

As it is shown in figure 3.5, the mechanical model of the bicycle consists of four rigid bodies:

1. The rear frame (with rider rigidly attached to it)
2. The front frame (formed by the front fork and the handlebar assembly)
3. Two knife-edge wheels

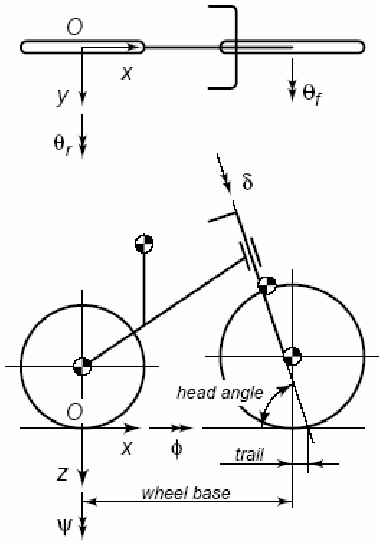


Figure 3. 5 Settings for mechanical model of the bycicle, from M.Wisse, A. L. Schwab, *Skateboards, Bicycles, and Three-dimensional BipedWalking Machines: Velocity dependent Stability by Means of Lean-to-yaw Coupling*

The bodies referred above are interconnected by revolute hinges at the steering head between the rear frame and the front frame and at the two wheel hubs. The modeling of the contact between the stiff non-slipping wheels and the flat surface is done by non-holonomic constraints in the normal direction and by non-holonomic constraints in the longitudinal and lateral directions.

Some assumptions to be set in this analysis are:

1. There's no friction, apart from the idealized friction between non-slipping wheels and the surface
2. There´s no propulsion and
3. There´s no rider control (hands-free coasting operation)

Three degrees of freedom are present in this mechanical model of the bicyle:

1. The lean angle **f** of the rear frame
2. The steering angle **d** and
3. The rotation θ of the rear wheel with respect to the rear frame

Velocity in the forward direction is given by

$$v = -\dot{\theta}_r R_{rw} \qquad \text{(Wisse , 2005[24])} \qquad (3.4)$$

Where R_{rw} is the radius of the rear wheel. The configuration of the system is described by four extra kinematic coordinates together with the degrees of freedom, due to the non-holonomic constraints (Schwab, 2003[25]). The four kinematic coordinates are taken here as the cartesian coordinates X and Y of the rear wheel contact point, the yaw angle y and the rotation θ of the front wheel taking the front frame as reference.

For effects of the quoted study, the dimensions and mechanical properties of the benchmark model are those of a regular 18 kg bicycle with an average 76 kg rider. The complete set of parameters can be referred to Schwab, Meijaard and Papadopoulos (Schwab, 2005[26]).

The linearized model of the bicycle is considered at constant forward speed in this study. In this model only two degrees of freedom remain, $\mathbf{q} = (\varphi, \delta)^T$. The equations of motion are

$$M\ddot{q} + [C1 \cdot v]\dot{q} + [K0 + K2 \cdot v^2]q = f$$

(3.5)

Where

M - constant mass matrix

C1 - “damping” matrix, proportional to the forward speed v

K0 – constant part of stiffness matrix

K2 – part of the stiffness matrix which is proportional to the square of the forward speed.

From the consulted study (Schwab, 2005[27]) the following typical values for the entries in these matrices are given

$$M = \begin{bmatrix} 80.812 & 2.3234 \\ 2.3234 & 0.30127 \end{bmatrix}$$
$$C1 = \begin{bmatrix} 0 & 33.774 \\ -0.84823 & 1.7070 \end{bmatrix}$$
$$K0 = \begin{bmatrix} -794.12 & -25.739 \\ -25.739 & -8.1394 \end{bmatrix} \tag{3.6}$$
$$K2 = \begin{bmatrix} 0 & 76.406 \\ 0 & 2.6756 \end{bmatrix}$$

Figure 3. 6 Typical values for entries in matrices in equation of motion of the bycicle, from M.Wisse, L. Schwab, *Skateboards, Bicycles, and Three-dimensional Biped Walking Machines: Velocity dependent Stability by Means of Lean-to-yaw Coupling*

where the standard units are kg, m, and s. The action–reaction lean moment between the fixed space and the rear frame forces, together with the action–reaction steering moment between the rear frame and the front frame is represented by the forces **f** on the right-hand side. The torque that would be applied by a rider's hands or a controller is the latter. For the effects of the quoted study, both of these moments were taken to be zero.

The stability of the upright steady motion is started from the homogeneous linearized equations of motion. For small variations in the degrees of freedom an exponential motion with respect to time is assumed, taking the form $\mathbf{q} = \mathbf{q}0 \exp(\lambda t)$. This involves an eigenvalue problem which characteristic equation is a polynomial in the eigenvalues λ of order four.

The solutions of the characteristic polynomial for a range of forward speeds in this study are the root loci of the eigenvalues λ, shown in the next figure taken from the study *Skateboards, Bicycles, and Three-dimensional Biped Walking Machines: Velocity dependent Stability by Means of Lean-to-yaw Coupling,* done by M. Wisse and A. L. Schwab.

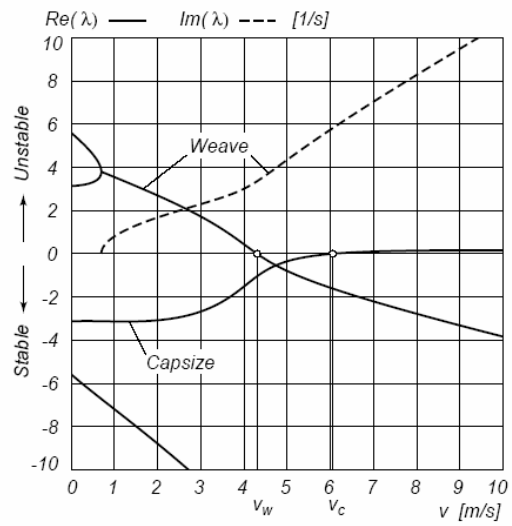


Figure 3. 7 Solutions for the characteristical polinomial of mechanical model of the bicyle, from M.Wisse, A. L. Schwab, *Skateboards, Bicycles, and Three-dimensional BipedWalking Machines: Velocity dependent Stability by Means of Lean-to-yaw Coupling*

According to the root loci analysis done in this study, eigenvalues with a positive real part correspond to unstable motions, whereas eigenvalues with a negative real part result in asymptotically stable motions. The third option, complex conjugated eigenvalues, give rise to oscillatory motions.

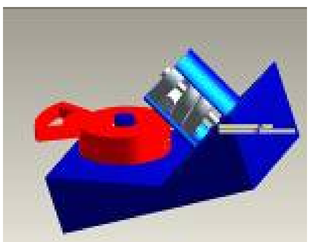
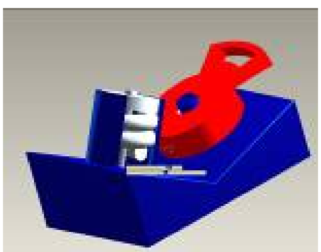
3.3.3 Conceptual design and selection of steering device

Starting from a non-holonomic premise, the following conceptual designs are shown: As required in chapter two, neither hands nor feet are involved in steering action. Safe position of hands will be set in handlers in module 2.

a) Torsion spring device

This device is formed by one torsion spring fixed by a metal tip and allocated through a screw in a cylindrical space. Figure 3.8 shows some screenshots of it.

The functioning is based in the effect of a moment caused by a weight shifting due to leaning by the rider. In figure 3.9 one can appreciate how the torsion spring free ends are then directed to the spaces in the cylindrical structure that contains the spring. These ends are the responsible of the back-to position motion in the rear axle after steering, and the full spring-tip set is responsible of the non-holonomic response of the device, it means that any lean done by the driver will be reflected as a yaw steering. Each free end will be connected to one side of the rear axle. The tip will be fixing the spring in the axial direction and also will keep the rear axle in its place, allowing the spring to undergo torsion and the rear axle to be steered.

(a) (b)

(c)

Figure 3. 8 a, b, c First possible solution to the steering issue in the Ruck ´n Roll problem

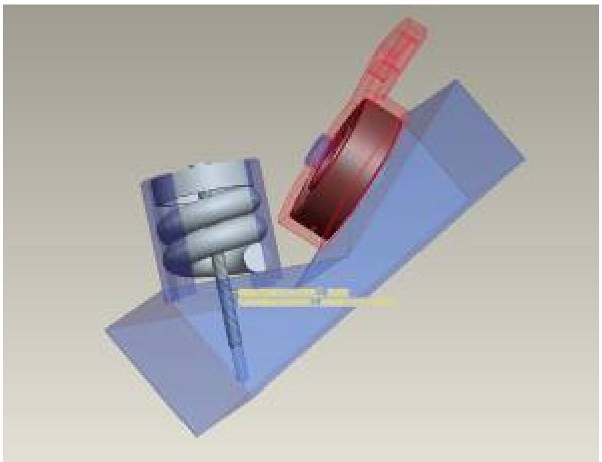


Figure 3.9 Detailed view of torsion spring-tip device

In figure 3.10 it is shown how a ring top is located over and concentrically to a bearing, both mounted in a pin fixed in the main blue structure. Basically the ring top is a yaw amplification tool. The cavity through the circular sector located in the farest part of the ring in reference to the spring will provide a center of gyration for the rear axle while steering. As referred in the study of the skateboard, a holonomic system fuctions under the premise that an external event can cause a motion in the desired direction, even when there is no degree of freedom in that direction. So the tip will keep fixed both the rear axle and the spring in the axial direction, but by effects of leaning, torsion of the spring will result in twist of the rear axle, due to a finger-alike pin that will be fixed in the center of the rear axle, and will always be in contact with the cavity or the ring top, allowing the rear axle to twist around the point of contact.

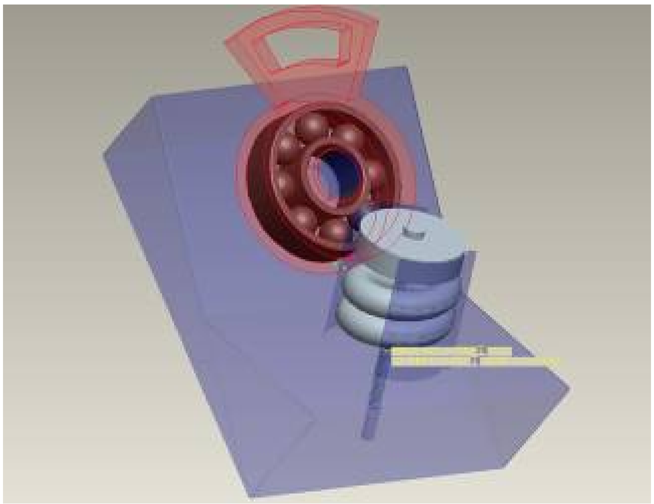


Figure 3.10 Detail of the bearing-ring top mounting

The following pictures show a preliminary set up for mounting the device in the end of the main beam that will be supporting the weight of the rider. Fig 3.11 shows a tentative configuration for mounting the device.

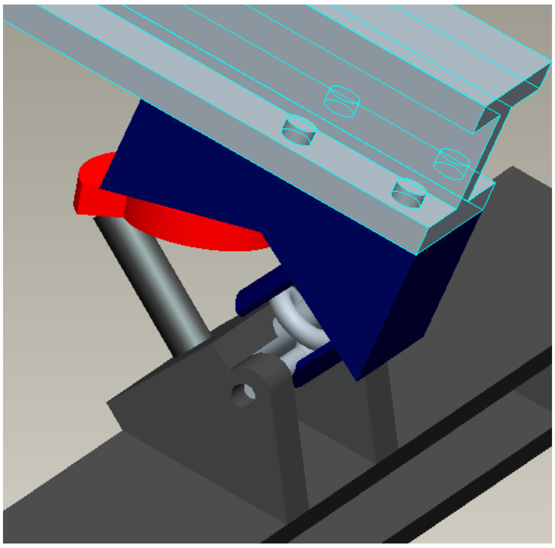


Fig. 3.11 Purposed configuration for steering device-to-beam-and rear-axle assembly

The free ends of the torsion spring will be located and fixed to the normal-to-rear-axle profile with a hole to ensure fixation. Steering device will be fixed to main beam by four screws.

b)Two-parallel-compression-spring device

Main configuration for steering device consists in a two-spring set oriented at 45º respect to x axis and at 90º respect to rotation axis pin. A general overview is shown in figure 3.8

Steering device operates under spring forces and friction forces that exert a certain torque during weight shifting action. When weight is shifted from a vertical axis, this causes a clockwise torque in rear axle. At the same time, friction forces in the wheels will cause a counterclockwise torque in rear axle. Steering will be achieved whenever the friction torque overcomes the spring torque.

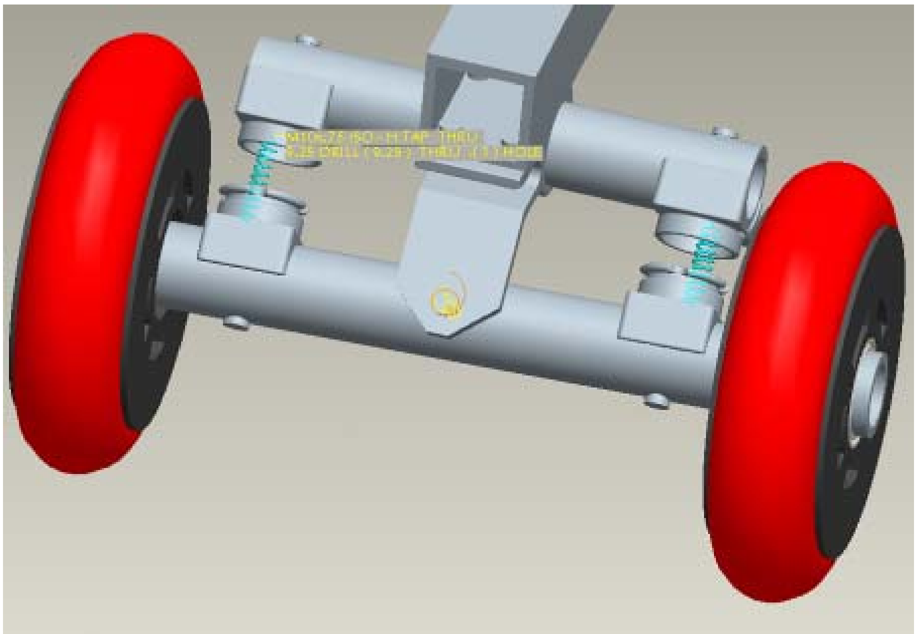
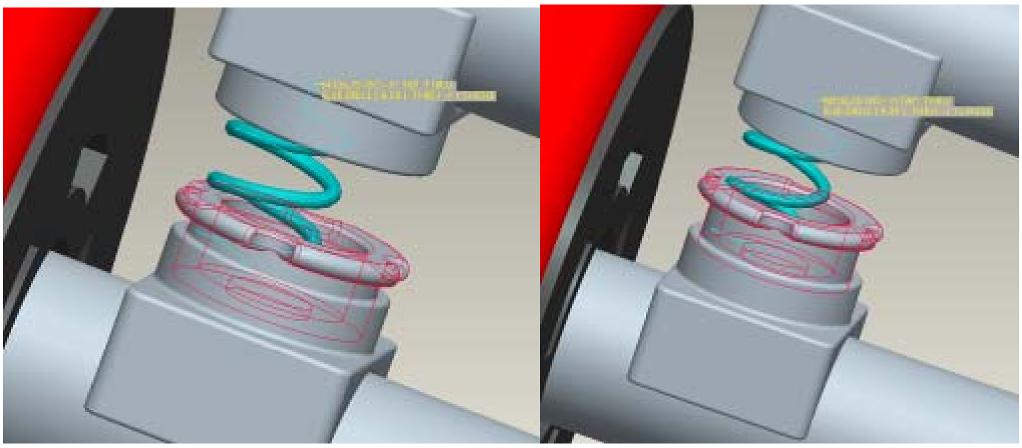


Fig. 3.9 General overview of steering device

Also steering easiness could be adjusted by compressing or releasing each spring, with a manually- activated springing adjuster, that has a thread connection with the spring support. By rotating the adjuster springs will be compressed or releaser according to driver’s preferences. This adjusted is shown in figure 3.6.



(a) (b)

Figure 3.6 Springing adjuster (a) Regular position (b) Compressing spring after rotating

3.3.3.2 Selection of design

The set of data in chart 3.1 gives some relevant points to be observed when choosing the design.

Chart 3.1 Features of each design**.**

Design	No. of components	No. of supports	Adjustable steering	Available simulation	Does simulation work?
a)	7	1	No	No	-
b)	10	2	Yes	Yes	Yes

A SWOT (Strength-Weakness-Opportunities-Threats) matrix is purposed in chart 3.2 and 3. for both designs.

From these matrices it is observed that the design premises of security and weight lessing are main characteristics to be considered. Design b) offers more features to increase security perception than design a). Threat in design b is not necessarily a fact: due to density of parts weight changes can be managed. As it is shown, most important benefit is security perception in this comparison. Design choice for this project is b).

Chart 3.2 SWOT matrix for design a)

Design a)

Strength	Opportunity
Lowest number of components	Decreasing weight possibility in assembly
Weakness	**Threat**
No Functional Available simulation	Doesn't show it works, decreasing security perception
No Adjustable steering	Security perception can be lost

Chart 3.3 SWOT matrix for design b)

Design b)

Strength	Opportunity
2 supports	More stability, increasing security perception
Functional Available simulation	It shows it works even without physical model, promotes security perception
Adjustable steering	Target market can adequate steering effort, increasing security perception
Weakness	**Threat**
Biggest number of components	Increasing-weight possibility

3.4 Purposed solution for steering device in Ruck´n Roll project

Distance between springs axis is 0.02 m, giving a gyration radius of 0.01 m. As shown in figure, rotation of axle will take place in a plane at 45º respect to horizontal plane. Figure 3.9 illustrates the deflection issue for such springs.

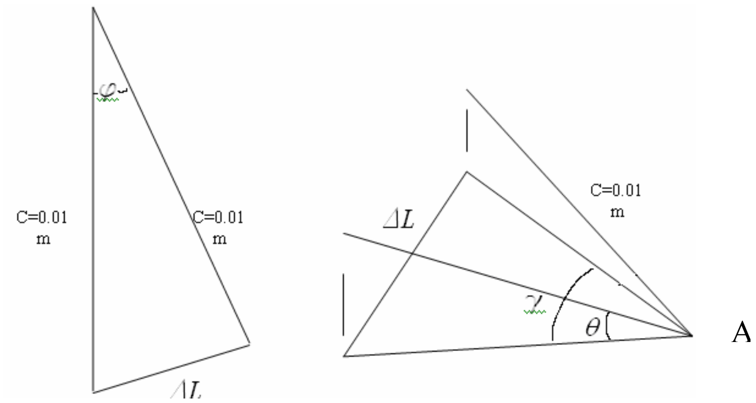


Figure. 3.9 Triangle referring deflection issue.

C represents one half of the axle length. Its joint is in point A. ΔL is length change of the springs during steering action. γ is projected-to ground steering angle. θ is orientation of gyration plane respect to ground and f is steering angle. Given the following relations

$$\frac{C\cos\theta}{\sin 90°} = \frac{0.5\Delta L}{\sin(\gamma/2)} \qquad (3.7)$$

$$\gamma = 2\sin^{-1}\left(\frac{0.5e}{C\cos\theta}\right) \qquad (3.8)$$

Given

$$\theta = 45° \qquad (3.9)$$

$$C = 0.01m \qquad (3.10)$$

From equation 3.8 it is got an expression for actual steering angle f given by a effective steering angle $\theta = 5°$

$$\varphi = \cos^{-1}\left(1 - \frac{2[(\sin(0.5\gamma))(C\cos\theta)]^2}{C^2}\right) = \cos^{-1}\left(1 - \frac{2[(\sin(0.5(5°))(0.01m\cos 45°)]^2}{(0.01m)^2}\right) = 3.54°$$

(3.11)

e is got from the cosine law solution for figure 3.9

$$e = \Delta L = \sqrt{2C^2(1-\cos\varphi)} = \sqrt{2(0.01m)^2(1-\cos 3.54°)} = 0.001m \qquad (3.12)$$

Figure 3.10 gives a view which normal corresponds to the rotation axis, and the view is a top perspective of the inclined plane of gyration.

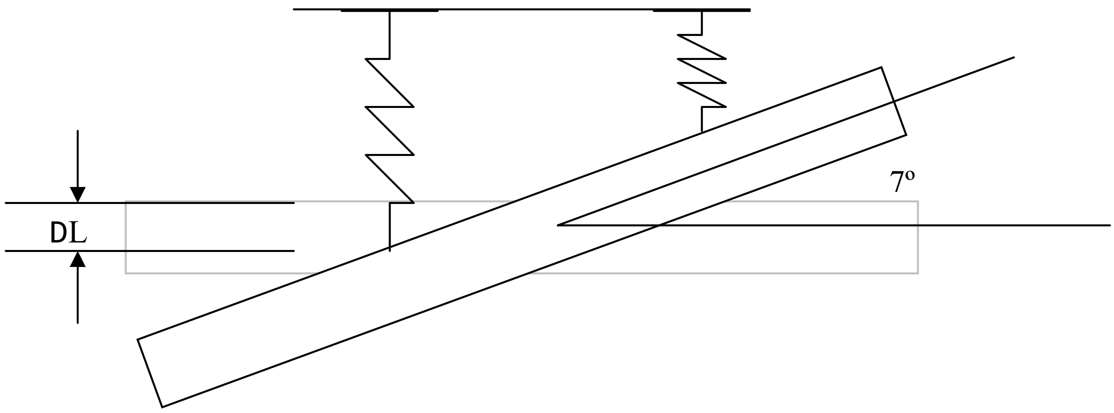


Figure. 3.10 View of plane of gyration

Setting a operative shifting weight force of DF=323 N, it is got a value for stiffness

$$K = \frac{\Delta F}{\Delta L} = \frac{323N}{0.001m} = 284111\frac{N}{m} \tag{3.13}$$

3.4.1 Diameters

Estimated medium diameter Dm is given a value of 0.030m, according to the space available to locate the spring. Wire diameter D_w is got from the equation derived from shear stress equation (Mott, 2000[28])

$$Dw = \left[\frac{8KF_0D_m}{\pi\tau_d}\right] \tag{3.14}$$

Selecting ASTM A231 steel for springs, chart 3.1 shows that, for hard service, design shear stress comes to a value of 895E6 Pa. Substituting such value in (13)

$$Dw = \left[\frac{8(12)(323N)(0.030m)}{\pi(896E6Pa)}\right]^{1/3} = 0.003298m \tag{3.15}$$

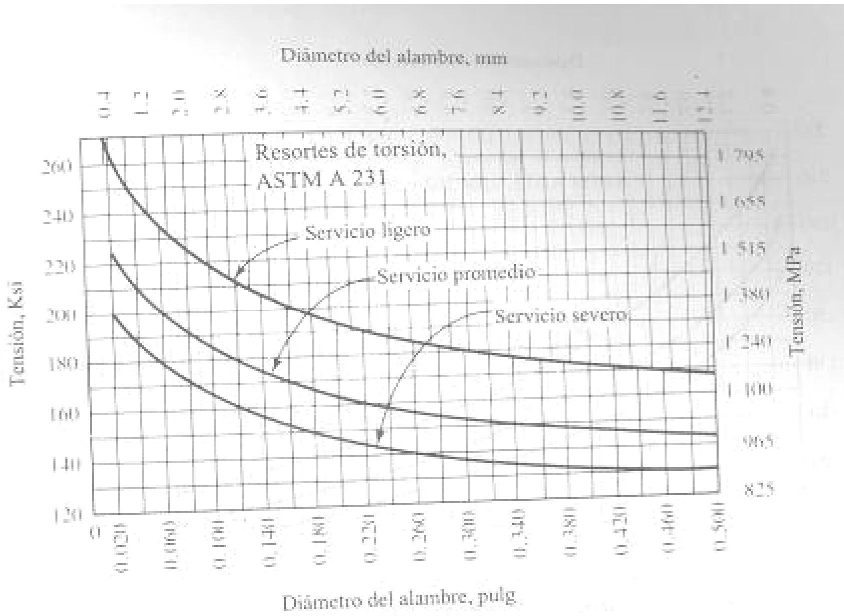


Figure 3.11 Design stress for ASTM A 231 steel under various service conditions.(Mott, 1996)

Such value is standardized to a reported value of 0.0035 m, according to chart 3.2. Outside and inside diameters, necessary for the hole where springs will be located and which geometry is shown in figure 3.11, are calculate with the expressions

$$OD = D_m + D_w = 30mm + 3.5mm = 33.5mm$$
$$ID = D_m - D_w = 30mm - 3.5mm = 25mm \tag{3.16}$$

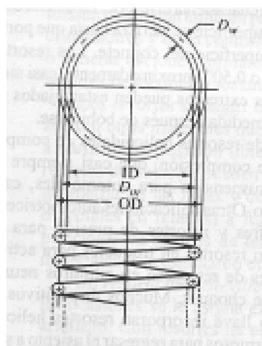


Fig. 3.11 Outside and inside diameters(Mott, 1995)

Chart 3.2 Standard diameters for spring wires (Mott, 1996)

Número de calibre	Número de calibre de alambre U. S. Steel (pulg)[a]	Calibre de alambre instrumentos musicales (pulg)[b]	Calibre Brown & Sharpe (pulg)[c]	Diámetros métricos recomendables (mm)[d]
7/0	0.490 0	—	—	13.0
6/0	0.461 5	0.004	0.580 0	12.0
5/0	0.430 5	0.005	0.516 5	11.0
4/0	0.393 8	0.006	0.460 0	10.0
3/0	0.362 5	0.007	0.409 6	9.0
2/0	0.331 0	0.008	0.364 8	8.5
0	0.306 5	0.009	0.324 9	8.0
1	0.283 0	0.010	0.289 3	7.0
2	0.262 5	0.011	0.257 6	6.5
3	0.243 7	0.012	0.229 4	6.0
4	0.225 3	0.013	0.204 3	5.5
5	0.207 0	0.014	0.181 9	5.0
6	0.192 0	0.016	0.162 0	4.8
7	0.177 0	0.018	0.144 3	4.5
8	0.162 0	0.020	0.128 5	4.0
9	0.148 3	0.022	0.114 4	3.8
10	0.135 0	0.024	0.101 9	3.5
11	0.120 5	0.026	0.090 7	3.0
12	0.105 5	0.029	0.080 8	2.8
13	0.091 5	0.031	0.072 0	2.5
14	0.080 0	0.033	0.064 1	2.0
15	0.072 0	0.035	0.057 1	1.8
16	0.062 5	0.037	0.050 8	1.6
17	0.054 0	0.039	0.045 3	1.4
18	0.047 5	0.041	0.040 3	1.2
19	0.041 0	0.043	0.035 9	1.0
20	0.034 8	0.045	0.032 0	0.90
21	0.031 7	0.047	0.028 5	0.80
22	0.028 6	0.049	0.025 3	0.70

where

OD – outside diameter [mm]

ID – inside diameter [mm]

D_m –medium diameter of spring [mm]

D_w – cross-section diameter of spring [mm]

3.4.2 Spring index

Medium diameter to wire diameter ratio is called spring index.

$$C = \frac{D_m}{D_w} = \frac{0.03}{.0035} = 8.57 > 5 \qquad (3.17)$$

As the spring index is higher than 5, manufacturing of the spring will be outstandable and severe deformation it may undergo will not be likely to fracture it.

3.4.3 Maximum shear stress

It is given by the expression

$$\tau = \frac{8KFC}{\pi D_w^2} \tag{3.18}$$

K, Wahl factor, is a term of curvature, related with C

$$K = \frac{4C-1}{4C-4} + \frac{0.615}{C} = \frac{4(8.57)-1}{4(8.57)-4} + \frac{0.615}{8.57} = 1.17 \tag{3.19}$$

$$\tau = \frac{8KFD_m}{\pi D_w^3} = \frac{8(1.17)(323N)(0.030m)}{\pi(0.0035m)^3} = 673E6Pa \tag{3.20}$$

Value for F was taken from operative force results in chapter 4 for dynamic analysis. Comparing with initial design stress, this value is secure.

3.4.4 Coils spacing

Number of active coils is got from the following expression

$$Na = \frac{GD_w}{8kC^3} = \frac{1240000Pa(0.0035m)}{8(284111N/m)(8)^3} = 0.19 \text{ rounded to 1 active coil} \tag{3.21}$$

However, let us consider 8 coils for the actual spring. Coils spacing during operating compression comes to

cc=(Lo –Ls)/Na =(1.234mm)/6=0.15mm (3.22)

3.4.5 Mounting cylinder diameter

It is recommended

Dmount>=OD+Dw/10= 33.5mm+0.35mm=33.85 mm (3.23)

Installing length is assigned a value of 33.85 mm

Li=33.85mm

So operative length is

$$Lo= Li-DL=33.85 \text{ mm}-1.234\text{mm}=32.62 \text{ mm} \tag{3.24}$$

3.5 Driving Torque

Consider figure 3.13

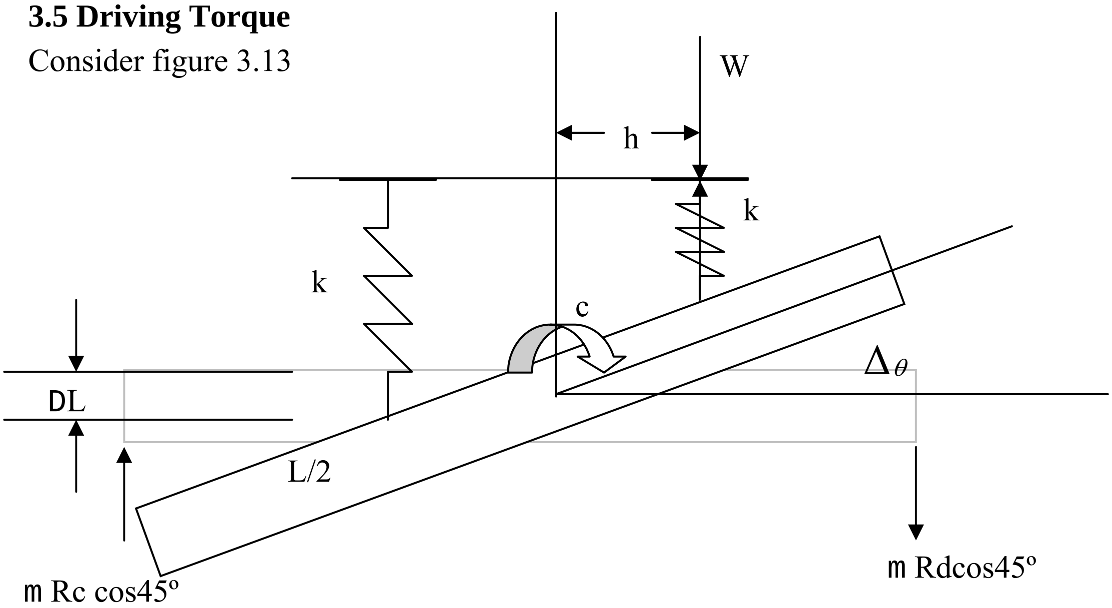


Fig. 3.13 Driving Torque

Equation of angular motion of rear axle is

$$I\ddot{\theta} + +c\dot{\theta} + 2kC\theta = W \cdot h - \mu R_c L \cos 45° \tag{3.25}$$

Mathematical solution for (3.25) is not relevant for this work except from the torque input in the right side of equation. Torque input is expressed in terms of weight and friction forces.

Numerical substitution comes to:

$$I\ddot{\theta} + +c\dot{\theta} + 2kC\theta = W \cdot C - \mu R_c L \cos 45° = 981N \cdot 0.02m - \mu(350.52N)(0.04m)\cos 45° \text{ (2}$$

5)

This fact agrees with the actual behavior and let us appreciate that the steering device response is dependant of friction forces. A plot of the driving torque range is shown in figure 3.14, going form 19.7 to 9.2 N.m.

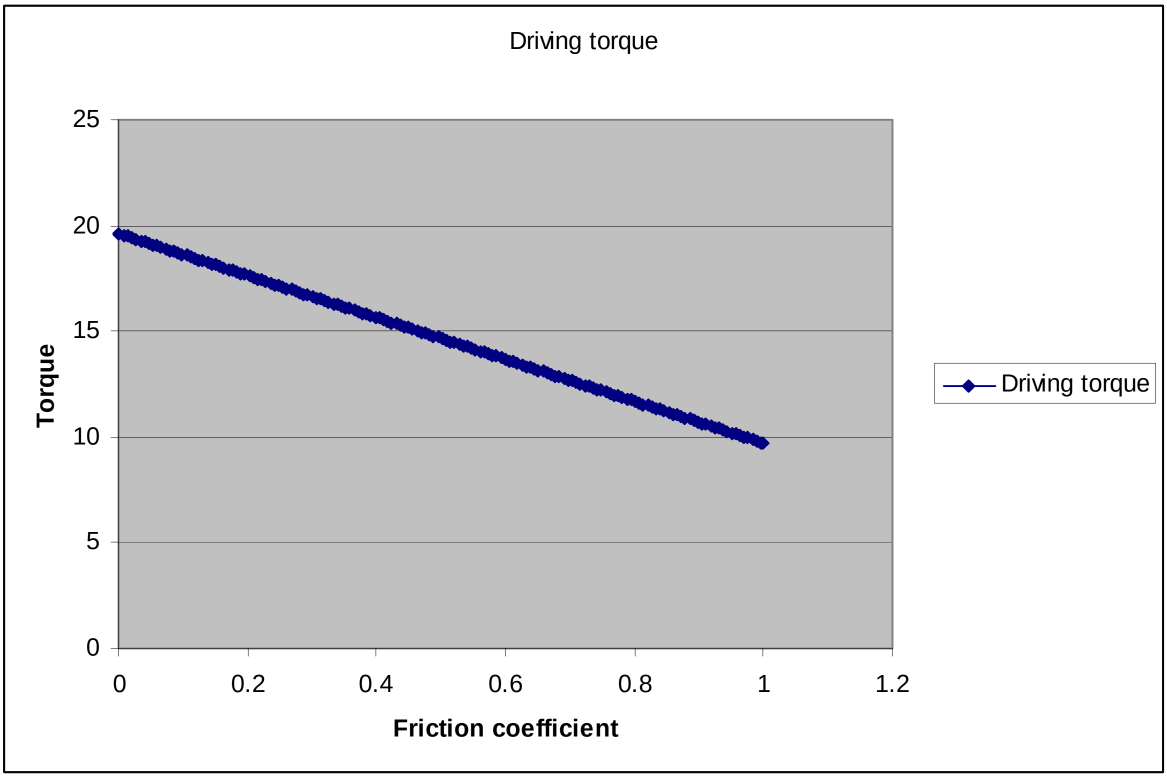
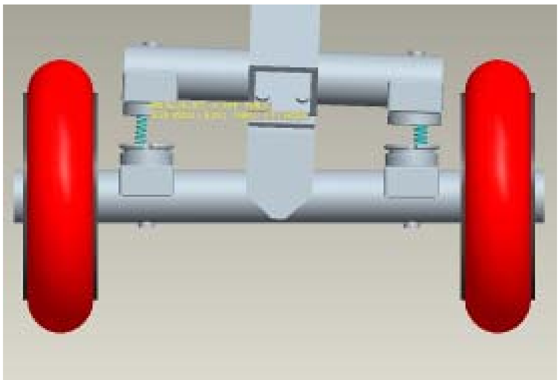


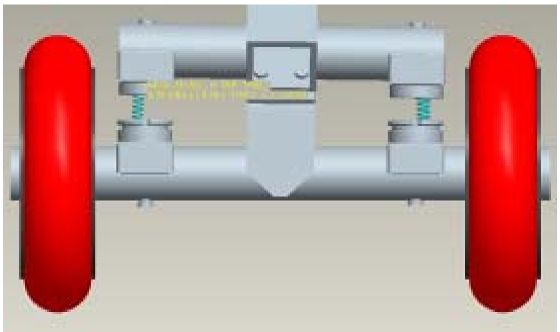
Fig 3.14 Driving torque

3.6 Steering mechanism

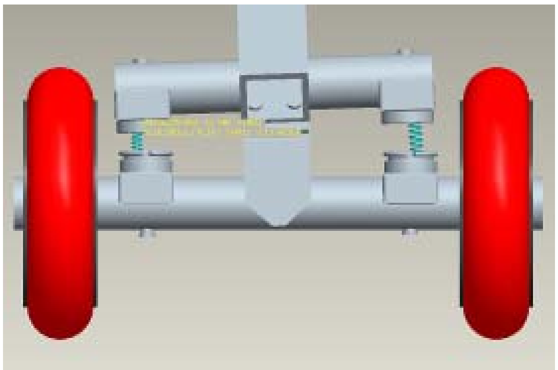
Consider figure 3.14 (a), (b) and (c)



(a)



(b)

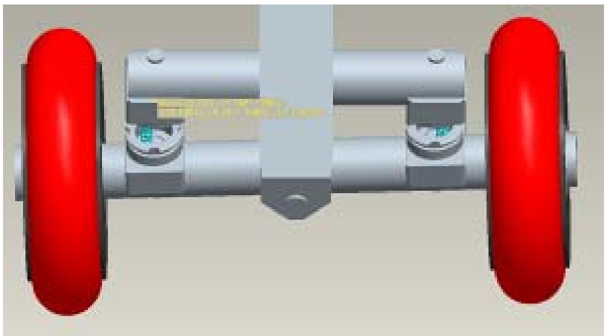


(c)

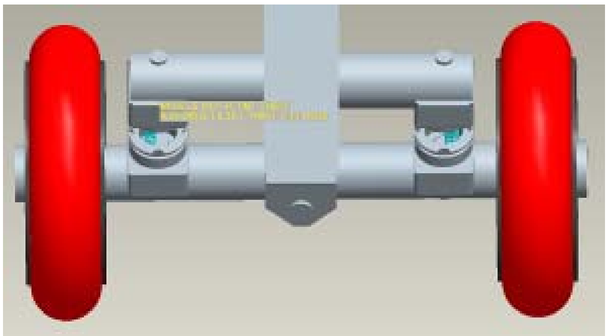
Fig. 3.14 Spring vertical deflection during steering action.

Figure 3.6.1 (a), (b) and (c) show the sequence in a back view of the rear axle. In such sequence it can be seen how the springs are deflected a certain amount due to the weight shifting action of driver. Animation of this action, available in the CD included in this project, was developed in ProEngineer™ Wildfire.

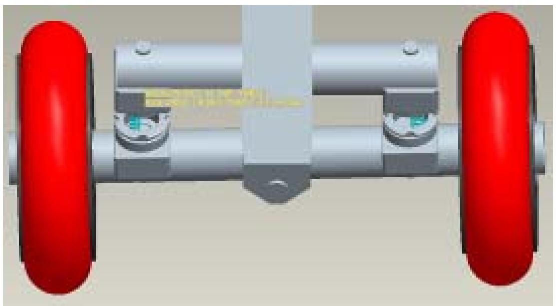
Consider figure 3.15 (a), (b) and (c)



(a)



(b)



(c)

Figure 3.15 Top view of steering action

As it can be seen in the figure, rotation about the 45° allocated axis promotes a forward-back displacement of wheels that will result in steering of the vehicle. Animation of this action, available in the CD included in this project, was developed in ProEngineer™ Wildfire. This animation was done taking as input data maximum steering angle and spring stiffness, using the geometry developed in the referred software, and just for illustrative purposes.