Design of a suspension for a bimodal vehicle suitable for road and rail

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Summary

The public transport is a constant innovation process. The demand and desire of passengers are constantly varying. Also the implemented infrastructure varies. To use the implemented infrastructure and make it profitable a new vehicle suspension is designed, which allows the use of road and rail with the same vehicle. To make the vehicle profitable it needs to be efficient: low fuel consumption, low gas emission, comfortable, reliable and on time. To reduce costs an existing vehicle is adapted.

The first and second chapter discuss the existing busses, trains and their characteristics. The characteristics are divided in: legal requirements and technical requirements. The legal requirements are described by the law. The technical aspects look to the construction and kinematics of the vehicles, for example: kind of axles, power train, stability of vehicle, articulation joint and acting forces on track, curving, accelerating and braking situations.

The third chapter discusses an independent suspension for a bimodal vehicle. The choice for an independent suspension is due to better ride characteristics and required building space. The mechanism responsible for the mode switch is integrated in the stub axle between rail and road tire. Both tires are always connected by gear coupling. This coupling results in one brake, steer and power train system, which results in a simpler and lighter system. This vehicle reduces transfer between vehicles and use the available road and rail infrastructure.

The fourth chapter discusses a vehicle where minimal modifications are implemented to allow riding on rails. The existing bus is taken and only suspension and rail wheel are adapted. The suspension needs to be adapted to reach desired ride/comfort characteristics and avoid instability of the vehicle on straight and curving tracks. The rail wheel needs to be adapted to obtain the track width of standard rail. This vehicle will ride only on rails and the passengers will be brought with (small) busses to the nearby change place. This concept solves the problem of public transport connection with a simple vehicle and using the existing rail infrastructure efficiently. The adapted bus is continuously available for rail use meaning more can be done with less money.
<table>
<thead>
<tr>
<th>Variable</th>
<th>Unit</th>
<th>Description</th>
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</thead>
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<tr>
<td>y</td>
<td>[m]</td>
<td>Lateral displacement</td>
</tr>
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<td>l</td>
<td>[m]</td>
<td>Track width</td>
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<tr>
<td>D</td>
<td>[m]</td>
<td>Diameter rail wheel (tape circle)</td>
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<td>r0</td>
<td>[m]</td>
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<td>[m]</td>
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<td>[m]</td>
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<td>[m]</td>
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<tr>
<td>γ</td>
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<tr>
<td>Λ</td>
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<tr>
<td>f</td>
<td>[Hz]</td>
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<td>[m]</td>
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<tr>
<td>δ</td>
<td>[m]</td>
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<td>A,B</td>
<td>[m^{-1}]</td>
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<td>Rw</td>
<td>[m]</td>
<td>Wheel radius (x - longitudinal direction, y – transversal direction)</td>
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<tr>
<td>Rr</td>
<td>[m]</td>
<td>Rail radius (x - longitudinal direction, y – transversal direction)</td>
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<tr>
<td>E</td>
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<tr>
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<td>[-]</td>
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<tr>
<td>G</td>
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<tr>
<td>v_{x,l,r}</td>
<td>[m/s]</td>
<td>Longitudinal creepage</td>
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<tr>
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<td>Lateral creepage</td>
</tr>
<tr>
<td>φ</td>
<td>[1/m]</td>
<td>Spin creepage</td>
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<tr>
<td>c_{11},c_{22},c_{23}</td>
<td>[-]</td>
<td>Kalker coefficients</td>
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<tr>
<td>F_x</td>
<td>[N]</td>
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<tr>
<td>F_y</td>
<td>[N]</td>
<td>Lateral creep force in the contact plane(rail)</td>
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<td>[m/s]</td>
<td>Longitudinal reduced creepage</td>
</tr>
<tr>
<td>τ_y</td>
<td>[m/s]</td>
<td>Lateral reduced creepage</td>
</tr>
<tr>
<td>τ</td>
<td>[m/s]</td>
<td>Reduced creepage</td>
</tr>
<tr>
<td>Y/P</td>
<td>[-]</td>
<td>Nadal limit criterion</td>
</tr>
<tr>
<td>Y</td>
<td>[N]</td>
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</tr>
<tr>
<td>P</td>
<td>[N]</td>
<td>Vertical force acting on the wheel</td>
</tr>
<tr>
<td>μ</td>
<td>[-]</td>
<td>Friction coefficient</td>
</tr>
<tr>
<td>W_e</td>
<td>[rad/s]</td>
<td>Undamped natural frequency</td>
</tr>
<tr>
<td>δ</td>
<td>[degrees]</td>
<td>Steer angle (f - front, c – center, b – back)</td>
</tr>
<tr>
<td>β</td>
<td>[degrees]</td>
<td>Articulation joint angle</td>
</tr>
<tr>
<td>a,b,c,d,e,f,I,j</td>
<td>[m]</td>
<td>Vehicle dimensions</td>
</tr>
<tr>
<td>F</td>
<td>[N]</td>
<td>Gravity forces(1 - front body, 2 - rear body, v - vertical, h - horizontal)</td>
</tr>
<tr>
<td>F_b</td>
<td>[N]</td>
<td>Brake force</td>
</tr>
<tr>
<td>F_y</td>
<td>[N]</td>
<td>Lateral tire force (road tire)</td>
</tr>
<tr>
<td>F_c</td>
<td>[N]</td>
<td>Centrifugal force</td>
</tr>
<tr>
<td>C_g</td>
<td>[-]</td>
<td>Center of gravity</td>
</tr>
<tr>
<td>n</td>
<td>[-]</td>
<td>Number of axles</td>
</tr>
<tr>
<td>ht</td>
<td>[m]</td>
<td>Andentum (Tooth height gear)</td>
</tr>
<tr>
<td>F_z</td>
<td>[N]</td>
<td>Reaction force on axles (f - front, c – center, b – back)</td>
</tr>
<tr>
<td>F_3</td>
<td>[N]</td>
<td>Reaction force on articulation joint (v - vertical, h - horizontal)</td>
</tr>
<tr>
<td>P</td>
<td>[-]</td>
<td>Turning point</td>
</tr>
<tr>
<td>ψ</td>
<td>[degrees]</td>
<td>Centrifugal force angle (cornering)</td>
</tr>
<tr>
<td>R1</td>
<td>[m]</td>
<td>Turning radius body 1</td>
</tr>
<tr>
<td>R2</td>
<td>[m]</td>
<td>Turning radius body 2</td>
</tr>
<tr>
<td>R_a</td>
<td>[m]</td>
<td>Turning radius articulation joint</td>
</tr>
<tr>
<td>R_x</td>
<td>[m]</td>
<td>Turning radius back point of bus</td>
</tr>
<tr>
<td>m</td>
<td>[kg]</td>
<td>Vehicle mass</td>
</tr>
<tr>
<td>a</td>
<td>[m/s^2]</td>
<td>Acceleration</td>
</tr>
<tr>
<td>b_c</td>
<td>[-]</td>
<td>Brake action (0 – none active, 1 – active)</td>
</tr>
</tbody>
</table>
The public transport is a constant innovation process. The demand and desire of passengers are constantly varying. Also the implemented infrastructure varies. But the amount of users increases more for one infrastructure than the other, e.g. the number of road users increases dramatically. A few regions in the Netherlands nowadays have a lower demographic density which makes some of this rail infrastructure obsolete. To make use of the existing, and therefore “cheap” infrastructure, and make it profitable a new public transport is needed.

The company Movares sketched a concept where a rail wheel and a road wheel are fixed on an excenter. By the rotation of this excenter the rail wheel and road wheel can be moved. This concept uses a single drive and brake system for rail and road. To investigate the potential of this concept they started cooperation with Veolia and Technische Universiteit Eindhoven. (Dr. ir. I.J.M. Besselink - Vehicle Dynamics from the Dynamics & Control Section) Started to investigate the technical possibilities of this concept.

To reduce costs an existing bus will be modified. These vehicles are much cheaper than trains or light trains. To be a bimodal vehicle the bus needs to have road wheels and rail wheels. These two types of wheels will be mounted on one axle, which allows the use of one power train, brake system and steer mechanism for each axle. To make a mode switch a mechanism is used. The constructions and mechanisms group was asked to work on such concept in a master thesis.

An independent suspension is designed. This system makes use of one axle, where both wheels are fixed. These axles can substitute all the axles of the bus; and can be driven and steered as necessary. The use of standard components is done as much as possible making a more reliable and cheaper design. The steering is done on the mid plane of the road wheel, thus no extra moments are generated on the suspension when steering. One driven shaft is necessary, for rail and road wheel, which enables the use of one brake system inside the wheel rim. The suspension is done with the existing air springs and shock absorbers. These components are mounted on the position where they work most efficiently. The mode switch mechanism is built between road wheel and rail wheel, and is based on basic principles of a knee mechanism. The use of rods and hydraulic cylinders make it reliable and robust.

The second manner to solve this problem is to make use of the existing system and rail infrastructure. Passengers in small villages will be taken by (small) busses to transfer on the nearby now obsolete rails, where a rail vehicle can take them and bring to a train station or city. The vehicles on rails are busses, whose rail wheel and suspension are adapted to make it safe and comfortable on rails. The suspension can be adapted as necessary. The rail wheel will have a modified “rim”, which makes the bus dimensions suitable for rail infrastructure. This bus can ride efficiently and fastly on rails. The change from busses to rail busses can be done as fast as tram stops in nowadays stations. The technology and implementation are easier and faster to design than to create a bimodal vehicle. After prototyping, acquired experience can be used to design a bimodal vehicle.
1. **Busses**

1.1. **Legal requirements**

The existing busses need to fulfill legal requirements. In The Netherlands these legal requirements are formulated by Dienst Wegverkeer (RDW). The most important aspects imposed by law for the design of an articulated vehicle are listed in Table 1.1.

An important aspect is the turning radius of the vehicle.

![Figure 1.1 – Turning radius of a articulated bus](image)

<table>
<thead>
<tr>
<th>aspect</th>
<th>dimension</th>
<th>observation</th>
</tr>
</thead>
<tbody>
<tr>
<td>Length</td>
<td>18 m</td>
<td></td>
</tr>
<tr>
<td>Width</td>
<td>2.55 m</td>
<td></td>
</tr>
<tr>
<td>Height</td>
<td>4 m</td>
<td></td>
</tr>
<tr>
<td>Vehicle load</td>
<td>28000 kg</td>
<td>With passengers</td>
</tr>
<tr>
<td>Foot brake</td>
<td>4.5 m/s²</td>
<td>deceleration</td>
</tr>
<tr>
<td>Hand brake</td>
<td>1.2 m/s²</td>
<td></td>
</tr>
<tr>
<td>Turning radius outside</td>
<td>12.5 m</td>
<td>Rmax</td>
</tr>
<tr>
<td>Turning radius inside</td>
<td>6.5 m</td>
<td>Rmin</td>
</tr>
</tbody>
</table>

Table 1.1 – Design aspects impose by the law

On Table 1.1 the dimensions are listed and Figure 1.1 shows how it is measured. The vehicle must stay inside the lines given by $R_{max}$ en $R_{min}$. Figure 1.1 shows an articulated bus with only front axle steering. Busses with more steered axles will behave differently when cornering, but the dimension it needs to fit stays the same. In chapter 3 vehicles with more steered axles and their turning radius are shown.

Also a minimal acceleration is desired, which ensures that the bus is suitable for traffic and preserves passengers comfort. The usual acceleration on existing buses is $1.1 \, \text{m/s}^2$ from standstill and tails off gradually to $0.5 \, \text{m/s}^2$ at 90 km/h.

1.2. **Axles**

The most common axles used on the existing busses are shown on Figure 1.2, a portal axle, and Figure 1.3, an independent double wishbone suspension.

The most important components of the rear axle are shown on Figure 1.2. The pull rods (1) have the function to carry the forces in x-direction (ride direction), for example brake forces and acceleration forces. The pull rods (2) have the function to carry the generated forces in y-direction, for example the lateral forces generated by curving or lateral acting wind on the vehicle. The connection between rods and chassis are done...
with silent blocks. These blocks isolate the vehicle body from rod induced vibrations created by the wheel motion. The tires are fixed on the wheel hub (3). This contains the bearings and the drive axle that come from the differential (4). The differential is put lower and on the right side to allow a low-floor bus. The air springs (5) favorably are located in front of and behind of the wheel. The load to be carried by the springs is the same load carried by the wheel. No moment or other forces are generated due to this favorable correct position, in line with the wheels, of the air springs. The shock absorbers (6) are also put at the outside of the wheels, where the velocity is highest. The brake disks (7) are mounted on the wheel hub and are located inside of the tire rim. This axle is equipped with double tire at each side, allowing a load of 12000 kilograms, the total weight of this axle is around 1000 kilogram.

The most important components of the front axle are shown on Figure 1.3. The air springs (1) are located towards the inside of the wheel. This is done to allow large steering angles, up to 45°. The double wishbone (2, 5) carries the forces in x- and y-direction. The wishbones are connected to the chassis with silent blocks. The shock absorbers (3) are placed beside the wheels on the connection arm.

The steering rods (4) connect both tires, which steer according to the ackerman steering principle. This means that the inner and outer wheel on a curve have small angle difference resulting in a better steering property and less tire wear. The king pin (6) allows the rotation of the wheel with respect to the fixed components, thus makes steering possible. The wheels are fixed on the wheel hub (7). The disk brake (8) is mounted on the wheel hub inside the rim. The allowed axle load is 7500 kilograms; the total weight of this axle is around 530 kilogram.

The use of air springs on all axles enables height adjustment when the passengers enter or exit the bus. The use of air springs in the new design enables an adjustable stiffness for road and rail, the stiffness will be dependent of bus load and track characteristics.

### 1.3. Tires

The existing tires used on busses are 275/70R22.5. The characteristics of this tire are listed in Table 1.2. On the two rear axles there are double tires, this means that an articulated bus with 3 axles has 10 tires and could carry a maximum total load of 31500 Kg.

New busses also use the super single tire (SS), for example the 385/55R22.5. In this case an articulated vehicle with 3 axles has 6 tires and can carry a maximum load of 27000 kilograms. Most busses used at this moment used the 275/70 tires due to price and a higher load capacity.

<table>
<thead>
<tr>
<th>aspects</th>
<th>275/70R22.5</th>
<th>385/55R22.5</th>
<th>dimension</th>
</tr>
</thead>
<tbody>
<tr>
<td>load</td>
<td>3150</td>
<td>4500</td>
<td>Kg</td>
</tr>
<tr>
<td>Width</td>
<td>0.287</td>
<td>0.471</td>
<td>m</td>
</tr>
<tr>
<td>Diameter</td>
<td>0.974</td>
<td>1.012</td>
<td>m</td>
</tr>
<tr>
<td>Velocity</td>
<td>110</td>
<td>100</td>
<td>Km/h</td>
</tr>
</tbody>
</table>

Table 1.2 – Tire aspects

Figure 1.3 – Independent front axle (ZF RL 75)
1.4. Power train

The power train of busses consists normally of the components shown on Figure 1.4. The power is delivered by a diesel engine; this is transmitted from the engine to the gear box through a clutch. The gear box, which allows different reduction ratios, is connected by a driveshaft and final ratio to the differential. Between differential and wheel, wheel hub reduction is used to reach weight saving in the driveline.

![Power train diagram](image)

Figure 1.4 – Power train of existing busses

The ratios of the gear box are chosen in such a way that: high torque is available to accelerate the vehicle and, low engine rotation speed and fuel consumption, at higher velocity. Figure 1.5 shows the traction force that can be achieved on the tires of a bus, with a ZF S6 1550 gear box and a 220 KW engine. The line with bullets (fric road) gives the road load up to 110 km/h. The full line (Pconst) gives maximum power and the other six lines represent each possible gear ratio for this specific gear box.

![Traction force graph](image)

Figure 1.5 – Traction force for a 220 KW engine in combination with a ZF S6 1550 gear box

The road load shown on Figure 1.5, is only rolling resistance and air resistance. The next section will show how the road load can be calculated.

1.5. Kinematics and stationary forces on busses

The kinematics of a bus and the analysis of all the stationary forces acting on a bus on static and dynamic situation; on straight and curving tracks are described here. This section contains a short description of the situations analyzed for this project; in appendix A.3 details about equations can be found.
Chapter 1 - Busses

The statics forces are calculated with the use of force and moment balance, determined with free body diagrams. In the static situation only the weight acts resulting in reaction forces on the wheels (Fz) and articulation joint (F3). In appendix A.3 details about equations and illustration of forces can be found.

In the stationary dynamic situation there are more forces on the bus. It is possible to analyze the acting forces by braking and accelerating. Both situations are analyzed and details are shown in appendix A.3.

Figure 1.6 shows the reaction forces for stationary static and dynamics forces acting on a bus. This figure shows the situation for a full bus, 26000 Kg. In the braking situation it is considered that each tire is braked properly in accordance with its normal load to obtain 1,1 m/s² for the vehicle. In the accelerating situation only the back axle is driven to obtain an acceleration of 1,1 m/s². This figure shows that the reaction force is dependent of the situation analyzed.

The dynamics forces will change if the vehicle is cornering. In a curve the reaction forces will be redistributed to the outer side of the curve. Figure 1.7 shows the back side of a bus curving to the left (side) and the most important forces acting on a bus. ∆Fz is the amount of reaction force that is redistributed.

When curving the tires will make a slip angle (α) to generate lateral tire forces (Fy). Figure 1.8 shows a top view of the bus during a steady state curving. The lateral tire forces are perpendicular to the slip angle and are pointing at the actual turning point P1. Force Fc represents the centrifugal force acting on the center of gravity CG. So when driving the described path at the constant speed there will be force equilibrium with forces Fyi and Fc pointing at/from point P1. When speed is increased P2 will become the actual turning point, centrifugal forces increases, slip angles and therefore lateral tire forces increase, and a new equilibrium point arises. When the speed is near zero, point P0 will be the actual turning point, slip angles are zero, and there are no lateral tire forces and no centrifugal forces.

The lateral forces that can be generated at each tire are depending of the slip angle and the cornering stiffness (Cs) of the tire. The cornering stiffness is a specific for a tire design. For low slip angles (< 5°) the cornering stiffness is constant for a constant normal force. For the calculations we use the approximation \( C_j = 0.16 * F_{c,nire} \). With the forces shown in Figure 1.8 equilibrium can be found and the actual turning point can be calculated. Appendix A.4 shows how this is done.
Chapter 1 - Busses

Figure 1.8 – Forces equilibrium during steady state cornering

There are also forces that counter the movement of the vehicle, for example air resistance, rolling resistance and road elevation.

The air resistance \( F_{air} \) depends mainly on the frontal area of the vehicle and speed, and is defined as:

\[
F_{air} = c_w * A * \frac{1}{2} * \rho * V^2
\]  

(1.1)

The rolling resistance \( F_{roll} \) depends on the kind of road. The rolling coefficient \( C_{rr} \), for example for road tire on asphalt varies between 0.006 to 0.01. Rolling resistance is defined as:

\[
F_{roll} = C_{rr} * F_{z, axle} * \cos(\theta)
\]  

(1.2)

Elevation on the road also works counter the vehicle movement; the amount of resistance depends from the road gradient \( \theta \), see Figure 1.9.

\[
F_e = m_{1+2} * g * \sin(\theta)
\]  

(1.3)

In Figure 1.10 and Figure 1.11 the graphs show the influence of each resistance for a vehicle with a mass of 26000 Kg. In Figure 1.10 an elevation of 6% causes the largest resistance, 71%. Air resistance, at higher speeds, takes 19.5 %. And the rolling resistance 9.5 %. If elevation force is not considered, Figure 1.11, rolling resistance corresponds to 42% and air resistance to 58%, at high speeds.

A good analysis of the situations, presented in this chapter, can determine safe speeds for cornering, the minimal torque needed to overcome resistance forces and the forces acting on the suspension. These parameters will be used to design the vehicle for road mode.
Chapter 1 - Busses

1.6. **Articulation Joint**

To make possible that long busses (18 m) can travel into city streets, they have an extra rotation point. This rotation is possible due to an articulation. An example of this articulation is shown on Figure 1.12 and the data are shown on Table 1.3.

<table>
<thead>
<tr>
<th>Parameters</th>
<th>Dimension</th>
</tr>
</thead>
<tbody>
<tr>
<td>Length</td>
<td>1320 mm</td>
</tr>
<tr>
<td>Width</td>
<td>780 mm</td>
</tr>
<tr>
<td>Horizontal angle</td>
<td>+/- 52°</td>
</tr>
<tr>
<td>Vertical angle</td>
<td>+/- 10°</td>
</tr>
<tr>
<td>Weight</td>
<td>675 Kg</td>
</tr>
</tbody>
</table>

![Figure 1.12 – Articulation joint, LYON model from ATG autotechnik](image)

![Table 1.3 – Data from the LYON articulation joint](image)
2. Trains

2.1. Legal requirements

The legal requirements for trains are determined by Inspectie Verkeer en Waterstaat (IVW), in The Netherlands. The main aspects that hold for this project are listed in Table 2.1 and shown in Figure 2.1.

Table 2.1 shows the main design aspects determined legally. The minimal turning radius is for primary rails, but on secondary rails also 125 meter can be found. It is expected that the RegioRailer (bimodal vehicle) will also ride on this rail.

Figure 2.1 shows a detailed front view of the kinematics border profile of the train and bus contour in it. Legal requirements imposed that movable parts attached to the rail vehicle need to remain inside the kinematics border profile in straight and curving tracks. Due to the width of the bus, the outside corner of the road tire needs to be 100 mm above the rails when the vehicle is running in rail mode, as shown in Figure 2.1.

The protection system that needs to be implemented is simpler than the present rail system used on trains and trams, because the bimodal vehicle is not allowed to ride between other traffic on rail. The suggested system is a VECOM, which only gives position of the vehicle on the rails to a central location, but does not communicate with other rail vehicles.

<table>
<thead>
<tr>
<th>aspect</th>
<th>dimension</th>
<th>observation</th>
</tr>
</thead>
<tbody>
<tr>
<td>Length</td>
<td>404 m</td>
<td>Normal rail vehicles</td>
</tr>
<tr>
<td>Width</td>
<td>3 m</td>
<td></td>
</tr>
<tr>
<td>Height</td>
<td>4,680 m</td>
<td></td>
</tr>
<tr>
<td>Axle load</td>
<td>22500 kg</td>
<td>With engine</td>
</tr>
<tr>
<td>Brake</td>
<td>2.8 m/s²</td>
<td>deceleration</td>
</tr>
<tr>
<td>Emergency brake</td>
<td>3.5 m/s²</td>
<td>deceleration</td>
</tr>
<tr>
<td>Turning radius minimum</td>
<td>150 m</td>
<td>Radius for regional rail</td>
</tr>
</tbody>
</table>

Table 2.1 – Design aspects imposed by the law
2.2. **Bogie** [10, 11]

There are a few types of bogies. In this chapter the most important components will be explained based on a *shinkansen* bogie. Each bogie has at least one wheel and axle set as shown in Figure 2.2. On most of the bogies the wheels are coupled with an axle, which enables that trains to stay on the rail in straight and curving tracks. The gauge in Europe is 1435 mm, see Figure 2.2. Most of the bogies have two sets of wheels and axle; the advantage of two axles is decreasing of impact due to track irregularities.

The bogie can have a bolster or not, the difference is the suspension configuration. Figure 2.3 shows where this component is located on a rail vehicle. On the bolster vehicle, the bogie rotates relative to the body on curve, while on straight tracks it has a high rotational resistance which prevents hunting. This is achieved with the centre pivot that serves as the centre of rotation and side bearers that resist the rotation.

To reduce the number of parts and the bogie weight the bolsterless bogie is designed. By this kind of bogies the rotation on curves is provided by the horizontal deformations of bolster springs (secondary suspension). The anti-yawing shock absorber (damper) at the outer side of the bogie frame prevents wheel set from hunting, which reduces the comfort.

The basic components of a bogie are shown on Figure 2.3 and Figure 2.4. The bogie frame is constructed in an “H” shape. To this frame all the other components are attached, for example, axle, springs and car body.

The suspension components are: bolster spring, anti-yaw shock absorber, axle shock absorber and centre pin (centre pivot); which respectively: supports the body, allows the bogie to rotate relative to the car body on curves, isolates the body from vibration generated by the bogie, and transmits traction force from the bogie to the body.

The axle box suspension, Figure 2.5, uses coil springs and cylindrical laminated rubber/wing type spring system. In this system, the vertical load on the axle springs is supported mainly by coil springs. The longitudinal and lateral loads are supported mainly by the cylindrical rubber parts, which also supports and guides the axle box.

The body is supported directly by the air springs, Figure 2.4, permitting considerable horizontal displacement in curves. When the bogie rotates relative to the car body on curves, this relative angular displacement is absorbed through horizontal distortion of the air springs. Longitudinal forces between the car bodies and the bogie are transmitted via the centre pin, which is mounted at the rotational centre of the bogie.
Chapter 2 - Trains

The transmission consists of gears and flexible coupling to transmit the power generated by the motor. The motor is mounted parallel to the axle. The motor is coupled to the gear box and this to the axle.

The most common brake systems used on rail vehicles are: wheel tread brakes and disk brakes. Wheel tread brakes use block-shaped brake shoes that are pushed against the wheel tread. Although they have a simple construction, they generate large amounts of frictional heat at high running speeds. This rises the temperature of the wheel to critical levels, presenting a risk of cracking. Disk brakes are mounted on the axle, and clamped by brake pads and calipers. This makes disk brake more suitable for high speed trains.

![Figure 2.4 – Bolsterless bogie shinkansen](image1)

![Figure 2.5 – Axle box suspension of a bolsterless shinkansen](image2)

The wheels on this bogie usually have a diameter of 860 mm. The axle run in tapered roller bearings. The wheel base of this bogie is 2500 mm with a weight of 6600Kg (unsprung mass 3500 Kg). [1]

2.3. Wheels

The wheels are fixed on a bogie and connected by a rigid common axle (wheelset), as shown on Figure 2.2. The wheel has a conical profile as shown in Figure 2.6. In curving tracks, the conical profile negotiates the difference in angular velocity between the inner and outer wheel, as explained in the next chapter. The position of the contact point when the wheelset is at a central position on the rails determines the so called “tape circle”, where the diameter (D) of the wheel is measured. On the inner side of the wheel, the conical profile has a flange which prevents derailment and guides the vehicle once the available creep forces have been saturated. Typical width of the profile is 125 - 135 mm.

![Figure 2.6 – Main elements of a wheel profile](image3)
and flange height is typically 28 - 30 mm. The flange inclination is normally between $65^\circ$ and $70^\circ$. The conicity (or tread gradient) ($\gamma$) at the tape circle is 1:20 for common rolling stock and 1:40 for high speed rail vehicles to prevent hunting.

Rail wheels are designed for high axle load, where 225kN is usual. For this project tram wheels will be used where maximum axle load of 110 kN is usual. This is lighter and smaller, reducing unsprung mass and the required space on the wheel housing. The wheel tread will be maintained as shown on Figure 2.7, but the “wheel rim” will be optimized so that it can be put on the wheel hub but strong enough for the applied load.

2.4. Kinematics and stationary forces on Trains

The railway vehicle moving on a track is a complex dynamical system. The bodies that form a vehicle can be connected in various ways and a moving interface connects the vehicle with the track. This interface involves the complex geometry of the wheel tread and the rail head and no conservative frictional forces generated by relative motion in the contact area. This section is limited to the linear force acting between geometries.

2.4.1. Concepts of curving

The behavior of coned wheel sets in a curve was understood early in the development of railways. E.g. Redtenbacher (1855) provided an analytical analysis, see Figure 2.8.

Taking into account the geometry in Figure 2.8, it is possible to derive an equation for the outward movement ($y$) of the wheelset as follows:

$$y = \frac{l \times r_0}{2 \times \gamma \times R}.$$  \hspace{1cm} (2.1)

Figure 2.7 – Wheel profile of a tram

Figure 2.8 – Redtenbacher’s formula for the rolling of a coned wheelset on a curve
Redtenbacher formula shows that a wheelset will only be able to move outwards to achieve pure rolling if both the radius of curvature and the flangeway clearance are sufficiently large. The usual flangeway clearance is 5 to 7 mm.

Figure 2.9 shows that it is not possible to achieve pure rolling, for pure conical wheels, because the outward movement of the wheelset is much larger than the flangeway clearance.

A way to improve the performance is to reduce the wheel radius (tape radius $r_0$) or increase the conicity. The wheel radius is constrain by the load, smaller wheel radius allow lower wheel loads. The increase of conicity is attractive, but larger conicity could lead to instability of the rail vehicle as explained in the next section. The existing trains solve this issue with the transition between wheel tread and wheel flange, where the conicity increases and holds the train on the track as explained in section 1.4.4.

2.4.2. Kinematics of a wheelset on straight tracks

If two wheels mounted on a common axle are rolling along the track and slightly to one side, the wheel on the outside side will run on a larger radius and the outsider side will run on a smaller radius. If pure rolling is maintained, the wheelset will move back into the centre of the track and a steering action would be realized. This motion is referred as kinematic oscillation, as shown in Figure 2.10.

Kinematic oscillation was analyzed mathematically for the case of purely coned wheels by Klingel in 1883. Klingel derived the following relationships:

\[ \Lambda = 2 \pi \sqrt{\frac{r_0 \cdot l}{2 \gamma}} \]  
\[ f = \frac{V}{\Lambda} \]  
\[ a_{y,\text{max}} = 4 \pi^2 \gamma_0 f^2. \]

Thus, with Klingel\'s formulation, the wavelength ($\Lambda$) depends on: wheel radius ($r_0$), track width ($l$) and conicity ($\gamma$). The frequency of the movement depends on forward velocity ($V$) and wavelength. Thus a large conicity will lead to a short wavelength and oscillation with higher frequency and higher lateral acceleration ($a_y$) as shown on Figure 2.11, where the vehicle run at a speed of 22 m/s. Passengers comfort decrease significantly for lateral accelerations > 1.1 m/s$^2$. 

---

Figure 2.9 – Relation between outward movement and radius of curvature; $l=1,5m$, $r_0=0,450m$, $\gamma=0,05$ rad

Figure 2.10 – The kinematic oscillation of a wheelset
Therefore, a lower comfort for passengers and the whole system could be unstable; this limit cycling is known as hunting. The stability can be provided by the proper choice of suspension stiffness and wheel profile.

2.4.3. Wheel rail contact

The normal contact

The study of contact between bodies is possible with finite element methods. However, the use of analytical methods can give a good notion about the contact. Hertzian contact is used to describe situation with normal force on the the contact point.

Hertzian contact between two elastic bodies is valid when:
- Elastic behavior;
- Semi-finite spaces;
- Large curvature radius compared to the contact size;
- Constant curvature inside the contact path.

Then:
- the contact surface is an ellipse;
- the contact surface is considered flat;
- the contact pressure is a semi-ellipsoid.

On appendix A.5 is shown how the dimension of the contact surface is calculated.

Forces on wheel

The behavior of adhesion on railways is determined by the forces arising in the contact interface. Due to the surface compression ($\delta$) the contact interface is slightly flatted, creating a region of contact between wheel and rail. Due to traction and braking, tangential forces, the contact will be disturbed. This distortion in the contact interface leads to a region of slip and adhesion, this mix of elastic and local slipping is know as micro creep.

The creep forces are a function of the relative speeds between elastic bodies. The general expression of creep forces take into account stiffness coefficients $c_{ij}$ determined by the linear theory of Kalker, shown on Figure 2.12, and derived as follow:

![Figure 2.11 – Lateral acceleration variation due to conicity, with a speed of 22 m/s](image)

![Figure 2.12 – Kalker rolling contact model with saturation](image)
\[ F_x = -G^* a^* b^* c11^* v_x \]
\[ F_{y,\text{yaw}} = -G^* a^* b^* c22^* v_y \]
\[ F_{y,\text{span}} = -G^* a^* b^* c23^* c^* \varphi \]

(2.5)

On appendix A.6 is described how the coefficients can be calculated and how the saturation will influence the forces.

2.4.4. Contact forces in the railway contact

Equivalent conicity

To understand the wheel-rail forces, a description of the cross sectional profiles of the wheel and rail are required. The conicity to be taken into account on a moving wheelset has a variable value. On the wheel, the flange and wheel tread are connected by a concave part which can frequently be in contact with the rail. This concave contact part, where the cone angle (conicity) is variable, is very important for steering and for stability considerations.

Gravitational stiffness

A description closer to the real shape of the wheels and of the rails is necessary to approach the principle of the gravitational centering mechanism.

First instance the vertical left and right loads \( (P) \) are considered identical, and the profiles are considered the same on each side from rail and wheel, as shown on Figure 2.13. When a wheelset is perfectly conical, the reaction force \( Y \) in relation to the normal load is compensated as far as there is no flange contact.

When a wheelset has concave profiles, the normal loads do not stay symmetrical with the lateral displacement, as shown on Figure 2.14. The reaction force \( Y \) is different for each side due to different conicity at the contact point.

When there is a large difference between the angle values, the profile combination is strongly centering. This is always the case in flange contact, as shown on Figure 2.15. However, the gravitational effect must be efficient even around the central position.

1 Handbook of railway vehicle dynamics, pg 87-112
When the friction is not negligible, Figure 2.13 to Figure 2.15 becomes the force representation incorrect, and Figure 2.16 gives the correct representation. The large value of the contact angle generates a large spin creepage value. A large friction value generates spin torque in the neighborhood of the contact, generating a lateral force which is always diverging. Despite the fact that this torsion value is small, it generates the equivalent of a yaw angle offset.

A second effect of this force will be in the wheelset equilibrium: this spin force, due to friction, is directed mainly upward. Considering the equilibrium between the vertical force, this new force $F_{y,\text{spin}}$ and the reaction force ($F_{y,r}$), it is found that the normal force (N) on a flanging wheel at the equilibrium is reduced by the friction; then the gravitational effect is reduced too. The associations with independent wheels increase the friction force.

**Safety Criteria, Nadal’s Formula (Y/P)**

The Y/P ratio is used as a safety criterion when flanging. In the real case of an attacking wheel, the spin force when flanging is added to the yaw lateral force which together counteract the Y guiding force. For a right wheel:

$\begin{align*}
Y &= N \sin \gamma - F_y \cos \gamma \\
Y &= N \cos \gamma + F_y \sin \gamma
\end{align*}$

When $F_y = \mu N$ (maximum, saturated) a safe level of Y/P has been set by Nadal:

$$\left( \frac{Y}{P} \right)_{\text{max}} = \frac{\tan(\gamma) - \mu}{1 + \mu^* \tan(\mu)}.$$  (2.7)

**Figure 2.17 – Maximum speed versus curve radius**

The first way to ensure safety is larger flange conicity. The second is to reduce the Y force by a good design of the bogie. Another option is to limit the track twist, limiting the reduction of P when flanging. The last way is to reduce the friction coefficient, by lubrication. However, the friction
forces are also limited in the lateral direction by the presence of a longitudinal force. The friction coefficient is shared between the two directions. In the Nadal formula, this sharing effect is not considered, making the formula adequate for independent wheels. This also means that a rigid wheelset, where the longitudinal forces are important on the attack wheel, will be safer than an independent wheel in the same conditions. IVW use as maximum Y/P=1.

For example, assume the follow parameters: P = 45 kN, Y/P=1, mass 26000 kg and 3 axles (n) then a maximum Y can be determined roughly, which means that the results are only based on the derailment criterion. Lateral force that can be carried in each axle must be in equilibrium with the centripetal force; this will result in a maximum speed, shown in Figure 2.17. That can be determined as follow:

\[
F_{cent} = m \frac{V^2}{R}
\]

\[
F_{cent} = Y_{max} \times n
\]

\[
V_{max} = \sqrt{\frac{Y_{max} \times R \times n}{m}}
\]

2.4.5. Dynamics of the rail vehicle on straight and curving tracks

Analysis of the rail vehicle dynamics on a straight track is determined by the vehicle stability. The limit cycling of the vehicle, hunting, happens when the vehicle motion is perturbed by a lateral displacement or yaw angle of the vehicle. These excitations could be so high that the maximum amplitude increases and is finally only restricted by wheel flange contact, which can result in discomfort, premature wear or derailment. Hunting predominantly occurs in empty or lightweight vehicles. The critical hunting speed is highly dependent on the vehicle/track characteristics. Considering the wheel/rail geometry and the creep force saturation, the vehicle/track system under hunting should be treated as nonlinear. Vehicle simulation computer models, which include the processes to solve motion equations, are often used to predict the hunting speed. The conicity of wheel-rail has considerable influence on the vehicle hunting speed. As wheelset conicity increases, the critical speed of hunting decreases.\[3, 8\]

Analysis of rail vehicle on a curving track is influenced by a few characteristics: wheelbase, gauge clearance and bogie rotational resistance. The stiffness of the vehicle suspension has high influence in curve and straight characteristics. Wheelset stability increases with increasing stiffness of the connection to the bogie frame. However, the relationship between suspension stiffness and the mass and conicity of the wheels influences the critical speed. Increasing the longitudinal stiffness of the primary suspension impairs the guidance properties of the wheelset in curves, while increasing the lateral stiffness reduces the ability of the wheelset to safely negotiate large lateral irregularities. The equivalent conicity of the wheel/rail contact should be increased, to make the radii difference available resulting in better curving performance.\[3, 8\]

As a result, requirements for high speed stability on straight track and good curving with safe negotiation of track irregularities are contradictory. The wheel/rail contact combination and suspension stiffness must be selected to give the best compromise for the conditions under which the vehicle will operate.
2.4.6. Friction Forces

Acting friction forces are the same for busses on road. Only the coefficients for rolling resistance will change if the same vehicle rides on rail. The rolling coefficient (Crr) for steel wheels on steel rails varies between 0,001 to 0,0025. Due to this small coefficient the total friction forces are lower than for vehicles on road. The elevation gradients found on rail tracks are also smaller, in The Netherlands the maximum elevation on rail is 6%.

At Figure 2.18 and Figure 2.19 the graphs show the influence of each resistance force for a vehicle mass of 26000 Kg. In Figure 2.18 an elevation of 4 % causes the biggest resistance, 76%. Air resistance, at higher speeds, takes 20,4 %. And the roll resistance 3,6 %. If elevation force is not considered, Figure 2.19, roll resistance corresponds for 16% and air resistance for 84%, at higher speeds.

But compared to a vehicle on road the friction forces, without elevation, are 33% lower, resulting in lower fuel consumption and emissions.

An analysis of the situations, presented on this chapter, can determine safe speeds for curving, the minimal torque needed to overcome resistance forces and the parameters for the design of a suspension.
3. **Independent Suspension**

The first concept is an independent suspension, where road wheel and rail wheel are fixed on one up-right. Both wheels are driven and steered; however steering will only be possible in road mode. This concept can be used on all three axles of the articulated bus, thus one system for the whole bus. Figure 3.1 shows the independent suspension concept. In the next sections the components, functionality and the more important aspects of it will be explained.

![Figure 3.1 – Assembly of the independent suspension concept](image)

There are several types of independent suspension possible. A few options will be shown and discussed.

<table>
<thead>
<tr>
<th>(a) macpherson</th>
<th>(b) multi-link</th>
<th>(c) trailing arm</th>
<th>(d) double wishbone</th>
</tr>
</thead>
</table>

![Figure 3.2 – Independent suspensions configurations](image)
Chapter 3 – Independent suspension

Figure 3.2 shows a few configurations and how a suspension design can be done. These systems are used in passenger cars but nowadays bus and trucks are also improving to such more complex suspensions.

The Macpherson suspension (a) is widely used on the front axle of cars; the wide use is due to the simplicity of the assembly. The under arm looks like a wishbone and the “upper” arm is the spring with a coil-over shock absorber. The disadvantage of this system is the place of spring and shock absorber. If the spring and damper are mounted above the up-right it will require more height in the system and between rail wheel and wheel there is not enough space to place such spring. If the whole system is mounted towards the inside of the rail wheel the steering characteristics are worse and extra forces will be generated in the suspension when steering. This will lead to a heavier and more complex system, which make this system less attractive. The aspiration is to place the steering on the mid plane of the road wheel, which leads to better steering performance and no extra forces will be generated.

The multi-link suspension has separated arms for each degree of freedom, which allow fine tuning of the system. But these arms need to be fixed to the up-right. If this system will be implemented, it requires a complex up-right, which also will require more space. The location of spring and shock absorber need to be positioned to the outside of the system.

Trailing arm (c) is a simple system which can be used with coil springs or torsion springs. This system could be constructed between the wheels, but then steering is not possible anymore.

The double wishbone (d) is the implemented system. A lot of modifications are made. The spring and shock absorber are transferred to a suitable place and the under wishbone is substituted by the steering mechanism, which has a double function now in the concept. The steering line is on the centre line of the road tire, due to a special up-right. The double wishbone seems to be the best option.

3.1. Road Wheel

3.1.1. Tire Choice

Due to rail width and maximum allowed bus width, double tires can not be used. Thus super single tires need to be used. Due to axle load the largest super single tires are desired. However, due to constrained dimensions, wheel housing, the maximum nominal width that can be used is 385 mm tire (400 mm is the maximum width the tire can have according to Goodyear). Figure 3.3 shows a top section view of the bus where the first design dimensions are shown. These are the dimensions fixed by legal requirements and infrastructure requirements. The characteristics of the chosen tire are shown in Table 3.1.

<table>
<thead>
<tr>
<th>Property</th>
<th>385/55R22.5</th>
<th>Dimensions</th>
</tr>
</thead>
<tbody>
<tr>
<td>Load Capacity</td>
<td>4500 Kg</td>
<td></td>
</tr>
<tr>
<td>Width</td>
<td>385 (400 max) m</td>
<td></td>
</tr>
<tr>
<td>Diameter</td>
<td>996 (1012 max) m</td>
<td></td>
</tr>
<tr>
<td>Speed</td>
<td>110 Km/h</td>
<td></td>
</tr>
</tbody>
</table>

Table 3.1 – Tire characteristics(Goodyear)
The tire rim will be a standard rim that allows a fixing face at the most outside of the wheel, leaving space inside of the wheel, e.g. a brake system, as shown in Figure 3.4. The inside rim diameter and rim width, are respectively around 500 and 380 mm.

The rolling radius shown on Figure 3.4 is smaller than the nominal diameter, shown on Table 3.1, due to axle load. The rolling radius will be approximately 45 mm smaller than the nominal diameter, on a fully loaded bus.

3.1.2. Wheel hub
Wheel hub connects the wheel to the vehicle and allows rotation of the wheel by use of bearings. The wheel hub is connected with the driving axle, which transmits the torque to the wheel.

3.1.3. Bearings
The bearings enable the rotation of the wheel with respect to the up-right. The bearings withstand vertical and lateral forces applied on the wheel. The bearings need to be chosen in such way that they can support dynamic forces; it is supposed that the dynamic forces will not be higher than three times the static forces. The maximum static vertical force is dependent of the maximum axle load, which is 9000Kg for the chosen super single tires. The maximum lateral force is dependent on the friction coefficient and vertical load. The friction coefficient for rubber bus tires are between 0.6 and 0.8, this result in a maximum lateral tire force of 36 kN.

The vertical and lateral forces on the tires mean axial en radial forces for the bearings. Due to forces in both directions tapered roller bearings are chosen, and are arranged as O in the wheel hub. Figure 3.6 shows a section view of the wheel hub with bearings in O arrangement. This arrangement of bearings allows ideal preload, which results in higher stiffness of the bearing construction and less play. Due to the load distribution on the wheel hub, the outside bearing (right) can be smaller than the inside bearing (left).
3.1.4. Up-right
The up-right connects the wheels to the suspension, steering and switch mechanism. The road
wheel runs in bearings on the wheel axle and the rail wheel is fixed on the movable stub axle,
which moves with respect to the up-right, as will be explained in section 3.2.

There are 3 connection points on the up-right, connecting it to the steering rods and wishbone.
The upper connection point will be fixed to the wishbone through a pivot ball joint. The two points
on either side of the wheel will be connected to the steering rods. The connection points are in
line with the mid plane of the wheel, thereby less moment will be generated when steering.

3.1.5. Wishbone
The wishbone connects the up-right to the frame, which is fixed to the bus chassis. The motion of
the wishbone is constraint by the use of a spring and shock absorber. To determine the optimum
location of the components, three concepts are evaluated, as shown in

Figure 3.8 – Evaluated wishbone configurations
Chapter 3 – Independent suspension

Figure 3.8 shows the most important components of the evaluated designs. The up-right is symbolized with a triangle; the 3 corners of the triangle give the rotation points, on the mid plane of the road wheel. The showed forces Fz and Fy are respectively, the vertical load and lateral force that acts on the wheel. The ends of the rods are rotating points allow the up-right to move in vertical direction.

The arch system (a) consists of: three rods and the arch that goes around the wheel. The springs are located on the mid plane on either side of the wheel, and the shock absorber can be placed next to the air spring. The location of the spring and shock absorbers are in line with the wheels, which allow both to work efficiently. The disadvantage of this system is the relatively heavy arch that is needed and the dimensions to enclose the air spring. The air spring is enclosed by the arch to ensure that the forces are correctly applied in the arch.

Figure 3.9 shows a concept how it could be done; the shock absorber is not drawn. The structure used could be made with plates, which allows a hollow structure resulting in lower weight but high stiffness. The height over the wheel is limited.

![Arch and Air Spring](image)

(a)Top view  (b)Trimeetric view

The torsion bar concept (b) consists of two rods, a beam and a torsion bar. On this concept the spring is substituted by a torsion bar. The upper beam is worked out as a triangle wishbone, as shown on Figure 3.10. The torsion bar goes through the wishbone tube. At the middle of this tube a spline connection is made to connect tube and torsion bar. The torsion bar is fixed at the end on the frame also with spline connection. The wishbone rotates in silent blocks. Silent blocks are widely used in automotive construction due to high radial stiffness and vibration isolation. The disadvantage of this construction is the extra weight due to the torsion bar dimensions and the extra feature needed to connect the wishbone to the shock absorber.

![Torsion Bar Concept](image)

The L-arm concept (c) is the most attractive concept where the standard air spring and shock absorbers are used and placed on the optimum location. This construction effectively uses the available space, resulting in a compact structure. The arm has the same length (l) in horizontal
and vertical direction, which means a ratio of 1:1, thus the force exercised on the wheel is the same felt by the springs.

Figure 3.11 shows a schematic view of the L-arm. The point where the force (F) acts is the upper fixation point of the up-right, which can move 200 mm up and down. The wishbone (L-arm) transmits this force to the spring and shock absorber. The shock absorber is connected through a rod and rocker. It is mounted vertically on the place where the highest velocity happens. Due to the use of correct ratios the air spring and shock absorber can work properly and efficiently.

Figure 3.12 shows the chosen assembly of the suspension and a section view of the wishbone, air spring and shock absorber. The wishbone is fixed on the frame with silent blocks and with a ball joint to the up-right, which allows a rotation (z-axis) when steering, and a rotation (x-axis) for spring action. The air spring is fixed between wishbone arm and a holder. The shock absorber upper side is fixed in the holder, and the under side is fixed on the rocker. The rocker consists of two metal plates, a sandwich construction and runs in bearings. The rod has rod ends on either side, which allow a rotation (x-axis) between rod and wishbone/rocker. The wishbone is made of plates, which means that it is a hollow welded structure, resulting in a low weight stiff construction.

Figure 3.12 shows the suspension statically loaded, 1/3 of the wheel travel is used; the other 2/3 will be used for dynamics forces acting on riding and support the passenger load.
3.1.6. Air spring and shock absorber
Components used in the design are standard components, as used in existing busses. The location and the functionality as explained in the previous chapter is shown in Figure 3.12.

The use of an air spring has some advantages with respect to other kind of springs. This system is already used in busses which enable to control the height of the bus if passengers enter and exit the bus. The air springs are light and appropriately for either rail or road mode.

The shock absorber used is the same as in existing busses. The position of the shock absorber was discussed in the previous section. The other option is to position it horizontally. Then the shock absorber would be placed in the position of the rod, shown on Figure 3.12. This could make the construction simpler. The prices of shock absorbers that can work horizontally are much higher than the presently used vertical bus shock absorbers.

3.1.7. Frame
The frame connects the subsystems of suspension and steer mechanism to the bus chassis. The frame needs to be strong and stiff enough but also light enough to reduce mass. The frame will be constructed from plates, resulting in a low weight stiff fabrication. The frame will be fixed to an existing chassis where the air spring and shock absorber were fixed on existing busses. The frame width is the same as the ZF middle axle used in existing busses. The dimension on y-direction is larger to enable steering. The corridor in the bus is 800 mm.

3.1.8. Brakes
The brakes will be standard disk brakes used on existing busses. They will be mounted inside the wheel rim as existing busses, see Figure 1.2. The brake caliper will be hydraulic instead of pneumatic. The main reason to use it is the small available space in a wheel rim for a pneumatic actuator. The steering motion on the axle prevents often used remote actuators.

The other possibility was to install a brake on the shaft beside the differential. This design is not chosen, due to safety and the long path from brakes to the wheel, which makes the system sensitive for broken drive shafts. Wheel tread brakes, around the rail wheel, were possible also. Because the rail wheel goes up and down it, is necessary that also the brake system goes up and down. The forces generated by braking need to be supported by the movable stub axle. Resulting in a heavier and more complex movable stub axle, decreasing the robustness and reliability of the system.

3.2. Rail Wheel
The rail wheel is the component that goes up or down during a mode switch. The wheel is fixed on a movable stub axle through the rail wheel hub. The movable stub axle is guided in the up-right, as shown on section 3.2.3.

Figure 3.14 shows an assembly and section view of the rail wheel components; these are the components that will move by mode switch. These components are normally not visible because there are inside the up-right. In the next section each part and its functionality will be discussed separately.
3.2.1. Wheel choice
The rail wheel tread has tram wheel profile; this profile is smaller than normal rail wheels. Resulting in lighter wheels, less required space and lower unsprung mass. The “rim” will be chosen so that it fits on the rail wheel hub and rail. The wheel diameter (tape circle) is 650 mm, the diameter is determined in combination with the power train and axle load. The wheel and hub are separate components, which eases production and installation of the wheels by bolting.

3.2.2. Wheel Hub
On the wheel hub are fixed: rail wheel, cardan joint, bearings and gear. The wheel hub will run on tapered roller bearings as shown on Figure 3.15, which will carry lateral and longitudinal forces applied on the rail wheel. The torque produced by the engine will come from the differential via the cardan joint to the rail wheel hub and to the gear. The drive path will be explained in section 3.4.
3.2.3. Bearings and movable stub axle guides

The tapered roller bearings are preloaded in an O arrangement. The preload is adjusted with the lock nut, providing low play and high stiffness.

The movable stub axle runs in cross roller guide inside the upright. The cross roller guide allows one degree of freedom, a vertical translation in this situation. The rails are fixed as shown in Figure 3.16, with standard bolts on the internal holders. The holders will be welded first in the upright and after that, they will be reworked. The rail will not be adjustable. This is done to ensure robustness and reliability of the system avoiding play. Due a not adjustable rail, the tolerances of the holders need to be chosen carefully, to ensure that rails can be mounted correctly and precisely.

The cross roller guide is compact and can carry forces in x- and y-direction; these properties make it suitable for this application. The forces that will act on these guides were treated in chapter 2, where the lateral forces (Fy) will be as large as the normal force acting on the axle, which is maximal 45 kN. To calculate the Fy force that the rail supports only the half of the rollers can be used. This is because the roller is positioned intercalated in the roller cage, as shown in Figure 3.17.a. This means that the Fy force that the guide can carry is half of the Fx that can be carried. There are other options to guide the movable stub axle. Making use of bearings running in rails, sleeve bearings or cross ball. At all this options dimensions are too large or have too small load capacity.
3.2.4. Movable stub axle
The movable stub axle can make a vertical translation of 200 mm during a mode switch. On the movable stub axle will be fixed: rail wheel, planetary gear and first gear step, rails and lower switching rods; as shown on Figure 3.14 and Figure 3.18.
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The lower switching rod responsible for the translation will be fixed on the movable stub axle, lower rod fixation. The working of the switching mechanism will be explained in section 3.5. On the stub axle, the wheel hub and corresponding components will be fixed, as was shown on Figure 3.15. The seal plate (1), in Figure 3.19, will be used to seal the oil reservoir from the sliding gear system, and the seal plate (2), in Figure 3.19, will be used to close the up-right from particles and moisture coming from outside.

Figure 3.19 shows the rail use position of the movable stub axle in the up-right and the seal plates. The bearings of the planet arms and the axle, that couples the sliding gear system with the gear ratio, will be supported by the bearing housing (section 3.4.2), as was shown in Figure 3.14.

3.3. Steering Mechanism

3.3.1. Steering rods

The steering is done with the use of rods and beams as shown in Figure 3.20. The steering mechanism is also a part of the suspension, which means that it needs to be strong and stiff enough to endure acting dynamic forces shown in Figure 3.21.
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The lateral forces ($F_y$) will be carried by rod 1 and the beam. Rod 1 and beam will work in pull/push. The steer beam need to support bending forces applied in the y-direction. This is the reason for the diamond shape. These lateral forces will have roughly the same magnitude in road and rail mode, as explained in chapter 1 and 2. In road mode this force is dependent on the friction coefficient and normal force acting in the road wheel. In rail mode this force will be maximal when flanging (Nadal criterion, on chapter 2). Which means that the steer beam bearing needs to support $R_{F_y}$ which will be 45 kN maximal.

The longitudinal force will be carried by the beam and rod 2 as shown in Figure 3.21. The longitudinal force will be also 45 kN maximal. Rod 2 will work in pull/push, dependent of the direction of force $F_x$, braking or acceleration. The shape of the beam, triangle back to back, has this form due to applied bending force. Knowing the forces direction and magnitude it is possible to design the rods, beams and to select the suitable bearings that can carry the applied forces.

3.3.2. Bearings and rod ends
The bearings are chosen separately for each joint of the steering mechanism, which dependents of direction and amplitude of applied forces. Figure 3.22 shows detail views of the bearing and position of it in the design.

Detail 1 show the supporting of the chassis fixation block. Fixation block connects the steering mechanism to the frame. The bearings used are: needle bearing and needle/ball bearing. Where radial force ($F_x$) will be carried by the needle rollers and the axial forces ($F_y$) will be carried by the angular contact ball bearing.

Detail 2 shows how the beam is supported. The joints between rods, beam, steer beam and the base plates are also supported by needle bearing and needle/ball bearing.
At the end of the rod 1 and the beam, rod ends are used (detail 3) to connect the steer mechanism to the up-right. The rod ends can carry high radial forces and enable rotation on the z axis and a smaller rotations on the x and y axis, around 25 degrees. Figure 3.23 shows the vertical displacement of the steer mechanism due to the wheel motion. The steer mechanism will rotate on the fixation block, the vertical displacement (\(\Delta z\)) is 200 mm, and the arm length (l) is 782 mm that will lead to a maximum angle (\(\alpha\)) of 14 degrees which is smaller than the allowed angle of 25° for the rod ends.

### 3.3.3. Hydraulic cylinder

The steering is actuated by a hydraulic cylinder. The hydraulic cylinder is fixed on one end side to the steer beam and the other side between the base plates. The cylinder stroke is dependent of the steer angle. The design has a maximum steer angle of 22°, which enables the bimodal vehicle to follow legally determined turning radius. To ensure safety the hydraulic cylinder is equipped with non-return valves; this prevents leakage from broken pressure hose.
Given the available space and applied forces three cylinder positions are investigated as shown in Figure 3.24

![Figure 3.24 – Investigated cylinder positions](image)

(a)  (b)  (c)

The first position (a) is the cylinder behind the steer beam. The cylinder is positioned at a distance $l_1$ from the rotation point of the steer beam, to minimize required cylinder force. The cylinder stroke is as long as the displacements of the steer beam. Leading to a too long cylinder, requiring too much space.

The second position (b) is the cylinder on an angle ($\alpha$) with respect to the steer beam, reducing necessary space compared with (a). Due to the movement of other components is this option less attractive.

The third position (c) is the cylinder parallel to steer beam. To achieve that the steer beam design is adapted, with an extra “triangle”. In the design the triangle is shorter than the $\frac{1}{2}$ steer beam, it has a ratio 1:1.4. This position allows a better compromise between forces and available space.
3.3.4. Turning radius

The minimal turning radius of a vehicle is determined by law. The bus must fulfill the requirements presented on section 1.1. This means, \( R_{\text{max}} = 12.5 \text{ m} \) and \( R_{\text{min}} = 6 \text{ m} \). The turning radius is dependent of vehicle dimensions, steering angles and number of steered axles. The bus dimensions are known by the bus producer, the steer angle is dependent of the design, the chosen design allows a maximum steer angle of 22 degrees.

First a vehicle with two axles will be analyzed. The turning radius can be determined with trigonometry, as shown in Figure 3.25. In the first situation (a) only the front axle steers. This means that the turning point is in line with the rear axle. The turning radius (\( R \)) can be calculated as follows:

\[
R = \frac{w_b}{\tan(\delta_f)} \quad (3.1)
\]

On the second situation (b) also the rear axle can steer. The steer angle can be chosen, in such way that it fulfills legal requirements or to improve ride properties. The minimal turning radius is reached if the rear steer angle (\( \delta_r \)) is chosen equal to the front steer angle. Then the turning point will goes upwards and to the left side, as shown in Figure 3.25.b. The turning radius (\( R \)) is defined as follows:

\[
R = \frac{w_b}{2 \tan(\delta)} \quad (3.2)
\]

Thus, with equal steer angle the turning radius reduces 50 %. The articulated bus has two bodies, where the three axles will be steered. The first body behaves as the vehicle shown in Figure 3.25.b. The second body will turn on the turning point of the first body, to ensure that the articulation angle (\( \beta \)) and the turning radius are not exceeded, this body is steered. Figure 3.26 shows the bus dimensions in meters. With the dimensions of the vehicle and the legal requirements, the steer angles: front (\( \delta_f \)), center (\( \delta_c \)) and back (\( \delta_b \)), can be calculated.
Figure 3.26 – Bus dimensions [m]

Figure 3.27 – Turning radius of a bus equipped with independent suspension design, with 3 steered axles

Figure 3.27 shows a top view of a bus (bicycle model) in a steady state cornering. Table 3.2 shows the results achieved with 3 steered axles. Notice that the front and center steering angles are chosen equal and the back steer angle is smaller. Different steering angles are free to be chosen, as long as legal requirements are respected. The necessary equations to calculate the steer angles are shown on Appendix A.2.

<table>
<thead>
<tr>
<th></th>
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<th>dimensions</th>
</tr>
</thead>
<tbody>
<tr>
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<td>[°]</td>
</tr>
<tr>
<td>$\delta_c$</td>
<td>21</td>
<td>[°]</td>
</tr>
<tr>
<td>$\delta_b$</td>
<td>10</td>
<td>[°]</td>
</tr>
<tr>
<td>$\beta$</td>
<td>42</td>
<td>[°]</td>
</tr>
<tr>
<td>$R_{\text{max}}$</td>
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<td>[m]</td>
</tr>
<tr>
<td>$R_{\text{min}}$</td>
<td>7.9</td>
<td>[m]</td>
</tr>
</tbody>
</table>

Table 3.2 – Achieved results for a 3 steered axle with independent suspension design
3.4.  Power Train

The power train carries for the transmitted power from the cardan joint to the rail wheel and road wheel. Figure 3.28 shows a flow diagram of the power train.

![Power train flow diagram](image)

**Figure 3.28 – Power train flow diagram**

The power produced by the engine goes into the gearbox and than transmitted to the differential. The differential divides the torque between the wheels. Drive shaft and cardan joint connect the differential to the wheels. The cardan joint is connected to the rail wheel hub. In rail mode the drive is directly connected to driven wheel, see Figure 3.29.a. In road mode the torque goes through two gear ratios to the road wheel, see Figure 3.29.b.

![Power train components and power line](image)

**Figure 3.29 – Power train components and power line**

To achieve equal circumferential velocities of the wheels, gear ratios are used. The circumferential velocity is different between wheels, due to wheel diameters and the sliding gear system. The sliding gear system enables the movement of the rail wheel during a switch mode without losing the power. However, this system introduces a ratio in the system. To correct it, extra gear ratio and a suitable rail wheel diameter are chosen. The circumferential velocities need to be almost equal to ensure a smooth mode switch.
3.4.1. Gear ratio

The gear ratio consists of two spur gears. Figure 3.30 shows the position of the gears in the movable stub axle.

The gear, Figure 3.31 – detail 1, is mounted on a splined axle and fixed with a ring. The ring fixes the gear with respect to the axle, through a bolt. The axle connects the gear ratio to the sliding gear system, as shown in Figure 3.14 and Figure 3.30. The axle runs in a needle/ball bearing, which carry the radial loads and the small axial load, which could be generated for example due to gear misalignment.

The gear, Figure 3.31 – detail 2, is connected, fixated and mounted on the rail wheel hub, with bolts.

The gear ratio (1:2.2) corrects the ratio created by the use of the sliding gear system (3.4:1) and the difference in wheel diameters (0.65:1).
3.4.2. Sliding gear system

The sliding gear system ensures that the transmitted power is not interrupted during a mode switch. The rail wheel and the road wheel are constantly connected, which also allows the use of one brake system inside the road wheel rim.

This system is explained in lecture note of “Constructieprincipes” [Hoek, W. van der, 1984, page 13.208]. The system is based on the planetary gear system, with two planet gears. Spur gears are used in the switching gear system. The ring gear \( r_r \) is not movable (only rotation), and the sun gear \( r_s \) can translate inside the ring. The maximum stroke \( z \) is dependent of the diameters of these gears. The planet gears \( r_p \) will move together with the sun, but will make \( \frac{1}{2} \) stroke \( (z/2) \) of the sun gear. This principle is shown in Figure 3.32. On the left side (a) the middle position is shown. On the right side (b) the upper position is shown, this position corresponds with the road mode.

![Switching gear system](image)

The dimensions of the gears are dependent of the needed stroke and chosen gear radius. The gear dimension shown refers to the pitch circle diameter. The dimensions can be determined as follow:

\[
2r_p = r_r - r_s \\
z = 2r_s - r_r - 4h_t
\]

(3.3)

For this project a stroke of 200 mm is used. This value is determined due to the minimal height (100 mm) of the road wheel with respect to the rail and the desire of symmetry in the mode switch system. For the design of the switching system, a sun gear diameter of 100 mm is chosen. The term \( 4h_t \) is a safety factor, \( h_t \) refers to the addendum; the sun gear must never engage in the ring gear, due to different angular velocity of the gears. The addendum is equal to the gear module (m). The chosen module is 4 mm, resulting in a ring gear diameter of 340 mm and planet gear diameter of 120 mm.

The sliding gear system is implemented inside the up-right and towards the movable stub axle, as shown in Figure 3.14, Figure 3.30 and Figure 3.35.
The planets gears are fixed between a sandwich construction, which runs plain bearing as shown in Figure 3.36. To create a “non-rotating” axle a needle/ball bearing is mounted inside the axle, as shown in Figure 3.36. Because the displacement of the planet gear is small and not frequent plain bearings are used for the sandwich construction, see Figure 3.36. The end stops of the planet gears are in the movable stub axle, as was shown in Figure 3.18.

The ring gear is divided in: tooth ring, tooth holder and stiffness plate, as shown in Figure 3.34.b. This division eases production and the assembly of the gear. The stiffness plate will increase the stiffness of the tooth ring and help to maintain it cylindrical. The tooth holder is connected to the tooth ring with bolts.

(a) section view of switching mechanism  
(b) section view of ring gear

Figure 3.34 – Section view of switching mechanism and ring gear
Figure 3.35 – Top section view of switching mechanism assembly

Figure 3.36 – Top section view of switching mechanism, zoomed on sun bearings

Figure 3.37 – Top section view of switching mechanism, zoomed in the planet bearing
Figure 3.37 shows how the planet gear bearing is fixed to the sandwich construction and the needle/ball bearing position. The bearing is mounted between the sandwich constructions and fixed with a bolt. The bearing axle is divided in two parts, the face of this two parts have a lead for centering.

### 3.4.3. Driven axles

The number of driven axles is determined by the minimal acceleration required in each mode. The required acceleration $(1.1 \text{ m/s}^2)$ is equal for rail and road. Taking into account the acceleration, the normal load acting on the wheel and the friction coefficient in each mode, can be determined the number of axles and the most suitable to be driven.

The maximum wheel load is 4500 kg, which leads to a maximal normal load of 45 kN. The friction coefficient is dependent of the mode, for dry road the typical value is 0.8 and for rail it is usual 0.45.

If road load is not counted and one axle is driven the maximal acceleration on road and rail will be respectively 2.7 and 0.8 m/s$^2$. This means that in rail mode more axles need to be driven. Normally the back axle of the bus, push bus, is driven. It is possible to drive the middle axle with a cardan joint from the back axle (differential). However, due to low floor and the articulation joint this option is technically complicated. A better option could be a hydraulic motor or an electrical motor. Electric motors are more used and the technology is available on the market. This motor can be driven with batteries or a generator that is coupled with the main engine. This electric motor could also be positioned on the front axle, due to load distribution the middle axle will be more loaded most of the time, which makes it more suitable to be driven.

### 3.5. Mode Switch

The mode switch is the process where the vehicle goes from road to rail or rail to road. The rail wheel goes up or down on a switching place. The switching place consists of a special profile of road and rail, as shown on Figure 3.49 and Figure 3.50. The switch is a synchronized combination of road profile and actuation. The mechanism carries only the weight of the switching mechanism and does not carry the weight of the bus during a mode switch. This allows a lighter and simpler construction that can be positioned inside the up-right.

The mode switch can be done in various forms. First, it can be chosen to move both wheels on a mode switch. Second, the wheel can be moved e.g. in a translation or circular movement. Finally, the construction can carry the bus weight or only the switch components during a switch mode.

It is chosen to move one wheel, the rail wheel. The movement of one wheel will require a simpler mechanism and could lead to a lighter mechanism. The rail wheel has smaller dimensions than the road wheel, requiring less space for the movement.

The switch can be done in a vertical translation or a circular movement. A circular movement is possible with the use of an excenter. However, this excenter move in vertical and horizontal direction during a mode switch. The horizontal displacement is equal to the vertical displacement, which requires

![Figure 3.38 – Schematic view of the switch mechanism](image-url)
more space. Vertical translation is chosen for less required space.

The translation can be done in different ways. One possibility is the use of hydraulic cylinders, positioned vertically. Due to the stroke of 200 mm and the required cylinder dimensions, this did not fit in the up-right. Therefore, the use of rods is more attractive. The actuation of the rods can be done with hydraulic cylinders. The cylinder is used to move the mechanism to the upper position and to lock the mechanism. Figure 3.38 shows a schematic view of the switching mechanism and the components.

3.5.1. Operating principle mode switch

The switching mechanism use an actuator and two rods on either side, a so called “knee mechanism”, as shown in Figure 3.38. The operating principle is a combination of the rod mechanism and road profile. These two systems are coupled during a mode switch.

Starting with the whole mechanism locked on the upper position, as shown on Figure 3.39. In this position, the vehicle is in road mode. The system is locked, the rotation point between rods (B) is positioned at the inside (10 degrees) of the rotating point of the system (A), thus the weight of the movable stub axle in combination with the hydraulic cylinder force will ensure that the system is locked in the upper position. When switching is desired, the vehicle goes to the switching place, as described in section 3.5.5. After arriving at that location, the system will unlock the rods. The cylinder will contract, point B will move to the outside of the system; due to the gravitational forces the system will lower, if the oil goes controlled out the cylinder, the speed of the process can be tuned. This process will take place in phase 2 and 3 of Figure 3.49. This happens until the rods are in line, Figure 3.38, after that the cylinder is actuated and locks the system in the lower position, as shown in Figure 3.40. Thus, point B is rotated 10 degrees to inside with respect to the point A.

To switch from the lower position to the upper position the whole process will be repeated. The vehicle is running in the rail mode and arrives to the switching place, as shown on Figure 3.50. The cylinders will unlock the rods in phase 1. The system goes upwards due to the road profile, in phase 2, and the cylinders are free (oil can move free in/out). The system is pushed upwards until the system passed the dead zone, cylinder in line with point A and B, as shown in Figure 3.41. After that, the cylinder is actuated and this moves the mechanism to the upper locked position, as shown in Figure 3.39.

The cylinders are coupled to ensure that the system goes synchronized. The cylinders are
equipped with non-return valve, the valve ensures that the system does not move if an pressure hoses breaks.

3.5.2. Rods
The rods are an essential part of the mechanism. In the upper position, the applied force is smaller compared to the applied force in the lower position. The rods are fixed on points A, B and C, see Figure 3.38. These points are explained separately in the next paragraphs.

The point A is the connection of the rod with the up-right as shown in Figure 3.42. The upper fixation is done with the fixed axle which has a plain bearing at the outside. Plain bearing is suitable due to low frequency, slow rotation and mainly radial acting forces.

Figure 3.43 shows a section, side and front view of the link between hydraulic cylinder and rods at position B. The hydraulic cylinder is fixed to the connection pivot, as shown in Figure 3.43 and Figure 3.45.b (right fixation). The connection pivot has a cylinder at either side, with a thread for the transverse bolt. The transverse bolt provides a pivot in plain bearings.

The lower rod shape is shown in Figure 3.44.a. This shape enables to fit the rod in the available space and make it suitable for the applied forces.

Figure 3.44 shows a front, side and a section view of the lower rod. The fixation point C is where the lower rod is fixed to the movable stub axle. The upper part of the lower rod, has a “gap”, this is due to the moment that the system will be moved to the upper position, then the upper rod needs to fit between this gap. The middle part of the lower rod is a tubular square rod, which can carry the load and is small enough to be mounted between the other components in the up-right. The fixation on the movable stub axle is done with the fixation axle, which has a plain bearing at the outer side to enable rotation of the rod.
3.5.3. Actuators

The switch mechanism has an actuator which locks the mechanism at the outer positions and lifts the mechanism during a mode switch from rail to road. It is important that the in and out moving can be controlled; this allows the system to work at a safe speed. The actuator chosen is a hydraulic cylinder. An other possible actuator could be a ball screw with an electric/hydraulic motor. Due to available space and applied forces the hydraulic cylinder is chosen.

This actuator is at one end, fixed to the switch mechanism, as explained in the previous section, and at the other end to the up-right. The fixation of the cylinder to the up-right (left fixation) is done with an axle, as show in Figure 3.45, which has a plain bearing at the outside to enable rotation during a mode switch.

The hydraulic cylinder is equipped with non-return valve due to safety. This cylinder is also coupled with the switch system and switch place. That ensures that the cylinder works at the right moment as explained in the section 3.5.1.
3.5.4. Oil reservoir

To enable a mode switch, a switching gear system is used, as explained in the previous section. To lubricate the gears an oil reservoir is needed. The difficulty of this lubrication is that the available space is small, and the movable stub axle makes a vertical translation. The translation of the movable stub axle creates an opening in the up-right at the rail wheel side, as was shown in Figure 3.19. On the up-right rubber is used, the shape of this rubber is shown in Figure 3.46. This rubber is clamped on the plate edge. The rubber and the seal plate (2) close the up-right from external factors, sand and water, and damp plate vibration which can lead to irritating noise for driver and passengers.

The oil reservoir seal designed for the switching gear system protects the gears from the outside particles and prevents oil leak. The oil reservoir is fixed to the up-right and seals to the movable stub axle, as shown in Figure 3.47.a. This translation does not happen frequently and it is almost a static condition. The oil that could escape will leak in the up-right, and does not contaminate the environment.

Figure 3.47 shows a section view (a) of the oil reservoir and a front view (b) which shows the shape of the oil reservoir. Oil scrapers are used to prevent that the oil moves with the ring gear,
otherwise too much oil is needed for lubrication. The oil reservoir is not totally cylindrical at the bottom, allowing more oil, to prevent strong temperature increase.

![Diagram of Oil Reservoir](image)

**Figure 3.47 – Oil reservoir front and section view**

The sealing is done in two steps. First, a scraper is positioned at the outside, which wipes dirt, moisture and foreign particles from the movable stub axle. This sealing acts in both directions and can operate in severe conditions and heavy attack of dirt, the scraper used is shown in Figure 3.48.a. The outward lip works as scraper, and the inward lip helps also by sealing the oil.

The second step is done with a seal that prevents the oil leak. The lip seal works in one direction, the lip seal used is shown in Figure 3.48.b. The U shape is mounted to the side of the oil. The rubber seals the oil with the help of the internal helical spring. This seal works also on surfaces that are not ideal and has higher contact pressure due to the internal spring.

![Diagram of Seals](image)

**Figure 3.48 – Oil seals used (Trelleborg)**
The mounting manner of the seals is shown in Figure 3.48.c. The two seals are designed to work in dynamic and static situations and work in a dirty environment. Because the seals are designed for extreme situations than expected by the use of bimodal bus, oil leak and frequent maintenance are not expected.

3.5.5. Necessary platform
To make de switch mode possible a special platform is necessary. It can be used for road/rail or rail/road switch, as shown in Figure 3.49 and Figure 3.50. On Figure 3.49 the recommended elevations for road are indicated. The length of this switch place should be at least 28 m for a switch velocity of 10 km/h and a switch time of 10 seconds. Dashed wheel represents not in contact with road or rail, and full line represents a wheel that is in contact.

Figure 3.49 shows the road to rail switch mode. In this case the rail wheel is on the upper position, locked. When the vehicle is at the switch place, the system will unlock the rail wheel from the upper position, this happens in phase 1. In phase 2 the road goes in upward direction. In this phase and phase 3 the rail wheel can go downwards without obstruction. If the rail wheel is totally down and locked, the road elevation is ended and the vehicle can travel on the rail mode.

Figure 3.49 – Switch mode from road to rail

Figure 3.50 shows the rail to road switch mode. In this case the rail wheel is on the lowest position, locked. When the vehicle arrives at the switch place and the road wheel is in contact with the road (first full line tire) the rail wheel will be unlocked. In phase 2 the bus goes downwards due to the profile of the road and the unlocked rail wheel will press in upwards direction. On phase 3 the hydraulic cylinder will bring the rail wheel (first dashed rail wheel) to the upper position and lock it. After that the vehicle can travel in the road mode.

Figure 3.50 - Switch mode from rail to road
4. Rail wheel directly on current bus axles

There are two ways to solve the problem addressed in this part of the study: create a new public transport or solve the problem with the existing technologies. The concept presented in this chapter also tries to make use of the existing rail infrastructure with minimal modifications.

Nowadays small villages are served with (small) busses due to the low number of passengers in these places; this is done with success in the Netherlands. In the neighborhood of these villages, most of the times also older rails are available, but not always stations. If a simple light, low cost small capacity vehicle could be modified to ride on these rails, a more efficient use of the existing infrastructure could be made. The (small) busses could bring the passengers from the villages to change places; there the passengers could step over to the economic light rail-vehicle. Because this (small) busses schedule will be coupled to a rigid time schedule for the new rail vehicle, connection time for passengers can be as fast as normal metro transfer, allowing villagers to reach larger cities.

The rail vehicle could be a bus as shown in the previous chapters. But in this chapter the existing bus axles will be used. To fulfill the requirements, the rail wheel and/or suspension can be adapted. The modifications needed guarantee safety, provide ride/comfort performance and fit the bus on the rail dimensions. The bus needs to be equipped with safety equipment, which can be the same that is used for the bimodal vehicle.

Because it was not the scope of this project to create such rail vehicle, the concept is not worked out in details. However this chapter will analyze the basic principles and minimal modifications on a middle axle. In the following sections the modifications that are needed to fulfill the requirements imposed by law and vehicle performance will be discussed. Figure 4.1 shows a concept for a middle axle of an existing bus with an adapted rail wheel.

![Figure 4.1 – Assembly of the suggested design](image)

4.1. Operating principle

The standard bus will be taken with standard axles, where the rail wheel will be adapted in such way that the bus can ride on rails and the suspension will be adapted to achieve ride/comfort performance and safety. The axles discussed in chapter 2 will be used and the rail wheel profile discussed in section 2.3 will be used.
The axles are optimized for road performance. On the rail other requirements and track characteristics are found than on the road. It is expected that rail irregularities are less then road irregularities, thus a smaller suspension displacement is needed. On the road the lateral stiffness is determined mostly by tires characteristics. Thus the tires are the “lateral spring” of road vehicles, and the suspension has a relatively high stiffness in lateral direction. On the rail vehicles the primary suspension fulfills this task, thus the existing bus suspension needs to be adapted in the lateral direction. The rail vehicle has a totally different behavior on curving than road vehicles. The description of steady state cornering in chapter 3.3.4 can not be applied to the rails. The curving of rail vehicles can be analyzed with the theory presented in chapter 2. Where Y/P (Nadal criterion, chapter 2) relation will restrict the vehicle speed in curving tracks. On straight sections the hunting effect will be a dominant factor and will be an important design factor.

4.2. Axle

The existing axles are optimized for road use. Thus some components need to be changed, in such way that this vehicle is safe and comfortable on rails. Two situations are analyzed: straight and curving track. On the straight track, it is important that the vehicle is stable, thus no hunting effect. On curving track, the vehicle must be able to follow the curvature of the rail. The way to achieve the desired results is explained in section 2.4.5.

The amount of driven axles for such rail vehicle are two, as mentioned on chapter 3, when the same bus is riding on rail mode. One axle could be driven with the existing engine and the second axle with an electric motor. The energy for the electric motor can come from batteries of a generator coupled to the main engine. The second option, generator, is attractive, because less power is needed on rail than installed on busses nowadays. Due to less friction it is possible for this motor to drive an axle and a generator for the electric motor.

The driven axles would have a lockable differential, which enables the bus to make kinematic oscillation, to stay on rail on straight tracks and follow the curvature of the rail on curving tracks. The front axle of a bus, has independent suspension, is steered and does not have a differential. For this axle two concepts are possible. One of them is to change to a fixed axle, which will have consequences for the low floor of the bus; the axle will go through the bus corridor. Another possibility is to fix the steering and synchronize both wheels on one axle with a high speed shaft, which allows a stiff coupling with a thinner shaft. With the use of a gear ratio, thus e.g. on rail wheel or wheel hub, axles and cardan coupling is possible to connect right and left wheels. The steering will be fixed.

Figure 4.2 shows a sketch of how the system could be implemented. The rail wheel is fixed on the wheel hub, which rotates free on the stub axle. An internal gear is mounted inside the rail wheel which is coupled with a second gear. The second gear is fixed on an axle which bearing housing is fixed to the stub axle. To allow suspension a cardan coupling is needed, and it needs to be extendible. On the other side of the axle also a cardan coupling is used which is fixed to a axle whose bearing housing is fixed to the chassis of the vehicle. The whole system is symmetric, which means that on both side the same components are placed.
4.3. Rail wheel

If the normal rail wheel profiles were to be put directly on the bus axle, the rail wheel width would be larger than rail width. The solution is to adapt the rim. The wheel thread profile is the same used for existing trams.

Figure 4.3 shows a section view of the designed rail wheel. The inside diameter is big to allow the use of the brakes already installed on the bus axle. Figure 4.4 shows a section view of the complete assembly of a middle axle, where it is shown there is enough space for the standard brake components. On Figure 4.1 and Figure 4.4 it is possible to see the arms of the axle. These arms make it possible that the air spring and shock absorber can be placed in place where the efficiency of these components is the highest. In this design those components and rod fixations can stay at the same place.

This concept will cause less modification on the bus. The most important tuning is in the rail wheel profile and possibly the rods, to get the appropriate dynamics of the vehicle. The design of this vehicle can be made faster; the exploitation knowledge acquired with this vehicle can be used for the development of a bimodal vehicle. Initially a single prototype can be produced with low cost and tested without passengers. In case of a later project failure the converted vehicle could be returned to their original bus state and sold off.
Conclusion and Recommendations

The necessity of a new form of public transport could be realized in several ways. The technical aspects of the two possible solutions are discussed in this report. Riding characteristics and safety aspects need to be investigated in detail before this vehicle can be effectively used; however the results of this study show that it is technically feasible. A prototype for testing without passengers will offer much information about the performance of this new vehicle and is required to ensure users safety. The two concepts presented in this report are developed with two different scopes: a new bimodal vehicle and a rail vehicle with use of existing slightly modified busses on rail.

The first concept is a bimodal vehicle. This concept is a compromise between road and rail characteristics, resulting in a flexible vehicle. The axle designed will substitute all the axles resulting in one universal steered axle that can be driven. To achieve the legal requirements all the axles will be steered, but with different steer angles. The total mass allowed will be less due to the use of super single tires instead of double tires. The mode switch will be done without stopping the vehicle. After arriving at a change place the whole system will be activated and the mode switch will be done, which will ask less capability from the driver and is safer for the users. The mode switch mechanism is light it does not carry the weight of the bus when switching, which results in a simpler and more robust system. The bus needs to be adapted to fit this new wheel suspension, the corridor will be a little smaller and, at the front, axle location needs to be changed due to a totally different axle. A special platform is needed for mode switch.

The second concept is a rail vehicle which makes use of the existing bus technology. This concept makes use of minimal modifications and is an efficient use of the present rail infrastructure. Busses (small) collect the passengers and bring them to the nearby new rail transfer platform. There the passengers transfer to the new rail vehicle which will bring the people to a train station or city. The rail bus vehicle has minimal modifications on the suspension rods and rail wheel. These modifications ensure that the vehicle is suitable for rail dimensions and has satisfactory ride and comfort characteristics. The rail characteristics need to be a compromise between performance on straight and curving tracks. Due to contradictory desires on straight and curving tracks, wheel conicity and suspension stiffness needs to be chosen carefully. The development phase and the experience acquired with this concept can be used to develop a bimodal vehicle.

For both concepts it needs to be proven that also the dynamic characteristics fulfill all the requirements. To investigate that, computer models are required. Rail/wheel contact needs to be investigated and modeled to be included in existing articulated road vehicle dynamics modeling software, as well as the stability of the vehicle on road and rail and also curving/straight track performance. Finite element methods can further reduce weight of components and evaluate ideas to optimize the shape of components for the various loads. The construction of a prototype and measurements is required to prove that the concepts outlined in this report could be implemented and are safe for passengers.
# References


[15] [www.carbible.com](http://www.carbible.com), visited on 06/08/2009

Appendix

A.1. List of bearings

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Table A.1 – List of bearings used
Appendix

A.2. Turning radius of an articulated bus

Figure A.1 – Bus dimensions

Figure A.2 – Turning radius of a articulated bus
With the use of trigonometry, the turning radius of all points on the bus can be calculated. Some parameters must be respected, for example outside turning radius, inside radius and maximum angle at the articulation joint.

If holds:
→ \( wb = b \)
→ \( \delta_y = \delta_z = \delta \)

than:
→ \( R_1 = \frac{wb}{2 \cdot \sin(\delta)} \)
→ \( R_a = \sqrt{R_1^2 + (c + \frac{wb}{2})^2} \)

**Without** steering third axle
→ \( R_2 = \sqrt{R_a^2 - d^2} \)
→ \( R_x = \sqrt{(R_2 + \frac{l}{2})^2 + e^2} \)
→ \( \beta = a \cos(\frac{R_1}{R_a}) + a \cos(\frac{R_2}{R_a}) \)

**With** steering third axle
→ \( R_2' = R_2 + d \cdot \sin(\delta_b) \)
→ \( R_x' = R_x + d \cdot \sin(\delta_b) \)

<table>
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<tr>
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</tr>
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<td>f</td>
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<td>[m]</td>
</tr>
<tr>
<td>l</td>
<td>2.55</td>
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<td>R1</td>
</tr>
<tr>
<td>Ra</td>
</tr>
<tr>
<td>( \delta )</td>
</tr>
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</table>

**Without steering**

| R2                  | 10.36  | [m] |
| Rx                  | 12.02  | [m] |
| \( \beta \)        | 50 [°] |     |

**With steering**

| R2'                 | 10.44  | [m] |
| Rx'                 | 12.13  | [m] |
| \( \beta' \)      | 40 [°] |     |
| \( \delta_b \)    | 10 [°] |     |

Table A.2 – Turning characteristics
A.3. Kinematics and stationary forces on buses

With:
- $F_1$ – weight load on body 1
- $F_2$ – weight load on body 2
- $F_{zf}$ – Reaction force on front axle
- $F_{zc}$ – Reaction force on center axle
- $F_{zb}$ – Reaction force on back axle
- $F_3$ – Vertical force on the articulation joint
- $a, b, c, i, j, w_1$ – vehicle dimensions

**Static equations**

\[ F_1 = m_1 \cdot g \]
\[ F_2 = m_2 \cdot g \]

\[ F_{zf} = \frac{b}{w_1} F_1 - \frac{c}{w_1} F_3 \]

\[ F_{zc} = \frac{a}{w_1} F_1 + \left(\frac{w_1 + c}{w_1}\right) F_3 \]

\[ F_{zb} = \frac{i}{(i + j)} F_2 \]

\[ F_3 = \frac{j}{(i + j)} F_2 \]
Figure A.4 – Free body diagram during braking

With:
- CG1 – Center of gravity body 1
- CG2 – Center of gravity body 2
- F1 – weight load on body 1
- F1h – horizontal load on body 1
- F2 – weight load on body 2
- F2h – horizontal load on body 2
- Fzf – Reaction force on front axle
- Fzc – Reaction force on center axle
- Fzb – Reaction force on back axle
- F3v – Vertical force on the articulation joint
- F3h – Horizontal force on the articulation joint
- Fbf – Brake force on front axle
- Fbc – Brake force on center axle
- Fbb – Brake force on back axle
- a, b, c, i, j, w1, h1, t1, h2, t2 – vehicle dimensions
Dynamics equations

Braking

→ \( F_{1h} = m_1 \cdot a_x \)
→ \( F_{2h} = m_2 \cdot a_x \)
→ \( F_{3f} = bc \cdot \mu \cdot F_{cf} \)
→ \( F_{bc} = bc \cdot \mu \cdot F_{zc} \)
→ \( F_{bb} = bc \cdot \mu \cdot F_{zb} \)

Moment around the forces \( F_{3v} \) and \( F_{3h} \):

\[ \sum M = 0 \]
\[ i \cdot F_2 - (h_2 - t_2) F_{2h} - (i + j) \cdot F_{zb} - t_2 \cdot F_{bb} = 0 \]

Use equation \( F_{bb} \)

\[ \rightarrow F_{zb} = \frac{1}{(i + j + bc \cdot \mu \cdot t2)} [i \cdot F_2 - (h_2 - t_2) \cdot F_{2h}] \]

For \( F_{3h} \) force balance can be used

\[ \rightarrow F_{3h} = F_{2h} - F_{bb} \]

Moment around the forces \( F_{zv} \) and \( F_{bb} \):

\[ \sum M = 0 \]
\[ t_2 \cdot F_{3h} + (i + j) F_{3v} - j \cdot F_2 - h_2 \cdot F_{2h} = 0 \]

\[ \rightarrow F_{3v} = \frac{1}{i + j} [j \cdot F_2 + h_2 \cdot F_{2h} - t_2 \cdot F_{3h}] \]

Moment around the forces \( F_{zf} \) and \( F_{bf} \):

\[ \sum M = 0 \]
\[ -(h_1 - t_1) \cdot F_{1h} + a \cdot F_1 - w_1 \cdot F_{zc} + (w_1 + c) \cdot F_{3v} - (h_1 - t_1) \cdot F_{3h} = 0 \]

\[ \rightarrow F_{zc} = \frac{1}{w_1} [a \cdot F_1 - (h_1 - t_1) \cdot F_{1h} + (w_1 + c) \cdot F_{3v} - (h_1 - t_1) \cdot F_{3h}] \]

Moment around the forces \( F_{zc} \) and \( F_{bc} \):

\[ \sum M = 0 \]
\[ w_1 \cdot F_{3f} - b \cdot F_1 - h_1 \cdot F_{1h} + c \cdot F_{3v} - (h_1 - t_1) \cdot F_{3h} = 0 \]

\[ \rightarrow F_{3f} = \frac{1}{w_1} [b \cdot F_1 + h_1 \cdot F_{1h} - c \cdot F_{3v} + t_1 \cdot F_{3h}] \]
Appendix

Figure A.5 – Free body diagram of the load distribution under acceleration

With:
- CG1 – Center of gravity body 1
- CG2 – Center of gravity body 2
- F1 – weight load on body 1
- F1h – horizontal load on body 1
- F2 – weight load on body 2
- F2h – horizontal load on body 2
- Fzf – Reaction force on front axle
- Fzc – Reaction force on center axle
- Fzb – Reaction force on back axle
- F3v – Vertical force on the articulation joint
- F3h – Horizontal force on the articulation joint
- Fab – Drive force on back axle
- a, b, c, i, j, w1, h1, t1, h2, t2 – vehicle dimensions
Appendix

**Accelerating**
Dynamic forces due to acceleration
→ \( F_{ab} = \mu \cdot F_{cb} \)

Vertical force balance on body 2
\[
\sum F = 0
\]
\[
F_{3v} + F_{zb} - F_z = 0
\]
→ \( F_{3v} = F_z - F_{zb} \)

Horizontal force balance on body 2
\[
\sum F = 0
\]
\[
F_{3h} + F_{ab} - F_{2h} = 0
\]
→ \( F_{3h} = F_{2h} - F_{ab} \)

Moment balance around CG2 on body 2
\[
\sum M = 0
\]
\[
i \cdot F_{3v} - j \cdot F_{zb} + (h_z - t_z) \cdot F_{3h} + h_z \cdot F_{ab} = 0
\]
Fill in vertical and horizontal force balance on body 2
\[
i \cdot (F_2 - F_{3h}) - j \cdot F_{2h} + (h_z - t_z) \cdot (F_{2h} - F_{ab}) + h_z \cdot F_{ab} = 0
\]
Fill in Fab
\[
i \cdot (F_2 - F_{3h}) - j \cdot F_{2h} + (h_z - t_z) \cdot (F_{2h} - \mu \cdot F_{2h}) + h_z \cdot \mu \cdot F_{2h} = 0
\]
→ \( F_{zb} = \frac{1}{[c + d - (h_z \cdot \mu + (h_z - t_z) \cdot \mu)]} [i \cdot F_2 + (h_z - t_z) \cdot F_{2h}] \)

Vertical force balance on body 1
\[
\sum F = 0
\]
\[
F_{3e} + F_{2e} - F_1 - F_{3v} = 0
\]
→ \( F_{3e} = F_3 + F_1 - F_{2e} \)

Moment balance around CG1 on body 1
\[
\sum M = 0
\]
\[
\alpha \cdot F_{3v} - b \cdot F_{2e} + (c + b) \cdot F_{3v} - (h_1 - t_1) \cdot F_{3h} = 0
\]
Fill in vertical force balance on body 1
\[
\alpha \cdot (F_3 + F_1 - F_{2e}) - b \cdot F_{2e} + (c + b) \cdot F_{3v} - (h_1 - t_1) \cdot F_{3h} = 0
\]
→ \( F_{3e} = \frac{1}{[w_1]} [(e + w_1) \cdot F_3 + \alpha \cdot F_1 - (h_1 - t_1) \cdot F_{3h}] \)
Lateral load transfer takes place when cornering

Moment around roll axis

\[ \sum M = 0 \]

\[
m_1 \cdot a_y \cdot h_1' + m_2 \cdot a_y \cdot h_2' + m_1 \cdot g \cdot h_1' \cdot \sin(\varphi) + m_2 \cdot g \cdot h_2' \cdot \sin(\varphi) - c_{pf} \cdot \varphi - c_{pc} \cdot \varphi - c_{pb} \cdot \varphi = 0
\]

with

Small \( \varphi \rightarrow \sin(\varphi) = \varphi \)

\[
m_1 \cdot a_y \cdot h_1' + m_2 \cdot a_y \cdot h_2' + (m_1 \cdot g \cdot h_1' + m_2 \cdot g \cdot h_2' - c_{pf} - c_{pc} - c_{pb}) \cdot \varphi = 0
\]

\[
\varphi = -\frac{m_1 \cdot a_y \cdot h_1' + m_2 \cdot a_y \cdot h_2'}{(m_1 \cdot g \cdot h_1' + m_2 \cdot g \cdot h_2' - c_{pf} - c_{pc} - c_{pb})}
\]

Figure A.6 – Load transfer on steady state cornering

Moment around CG

\[ \sum M = 0 \]

\[
s \cdot \left( \frac{1}{2} \cdot F_z - \Delta F_z \right) - s \cdot \left( \frac{1}{2} \cdot F_z - \Delta F_z \right) + \frac{1}{2} \cdot h_i \cdot (F_{yr} + F_{yl}) - c_{p,axle} \cdot \varphi = 0
\]

\[
\Delta F_z = \frac{1}{w_b} [h_i \cdot F_{y,axle} - c_{p,axle} \cdot \varphi]
\]
Figure A.7 – Back view of a bus on steady state cornering

With:
- Fzf – Reaction force on axle
- $\Delta Fz$ – load transfer due to cornering
- g – gravity constant
- mi – body mass
- $ay$ – lateral acceleration
- $c_{\phi,\text{axle}}$ – roll stiffness of the axle
- Fyr – lateral force on right tire
- Fyl – lateral force on left tire
- s, wb, hi, ti, wb – bus dimension
A.4. Lateral forces acting on steady state cornering

With:
- \( F_{c1} \) – centrifugal force acting on body 1 when cornering
- \( F_{c2} \) – centrifugal force acting on body 2 when cornering
- \( F_{yf} \) – lateral force on front axle (i- inner, o –outer)
- \( F_{yc} \) – lateral force on center axle (i- inner, o –outer)
- \( F_{yb} \) – lateral force on back axle (i- inner, o –outer)
- \( \alpha_f \) – slip angle on front axle (i- inner, o –outer)
- \( \alpha_c \) – slip angle on center axle (i- inner, o –outer)
- \( \alpha_b \) – slip angle on back axle (i- inner, o –outer)
- \( a, b, c, i, j, w1 \) – bus dimensions
- \( \beta \) – turning angle body 2
- \( P_0, P_1 \) – center of rotation

**Lateral forces for each tire**
To calculate the lateral forces the follow steps will be done:
1. First calculate the \( F_z \) on each tire
2. Use the relation between normal force and corner stiffness \( C_s=0.16 \times F_{z,tire} \)
3. Calculate the slip angle for each tire, based on trigonometry
4. Calculate \( F_y \)
5. Tune first \( x \) and then \( y \) to reach force balance
Figure A.9 – Forces acting on each axle/tire
Appendix

1. Thus right and left $F_z$ will be (when turning to left)
   \[ \rightarrow F_{zf} = \frac{1}{2} * F_{z, axle} + \Delta F_{z, axle} \]
   \[ \rightarrow F_{zf o} = \frac{1}{2} * F_{z, axle} - \Delta F_{z, axle} \]

2. Cornering stiffness
   \[ \rightarrow C_s = 0.16 * F_{z, tire} \]

3. Slip angles
   Front axle
   \[ \rightarrow \alpha_{fi} = \tan^{-1} \left( \frac{w_i - x}{R - s - y} \right) \]
   \[ \rightarrow \alpha_{fo} = \tan^{-1} \left( \frac{w_i - x}{R + s - y} \right) \]

   Center axle
   \[ \rightarrow \alpha_{ci} = \tan^{-1} \left( \frac{x}{R - s - y} \right) \]
   \[ \rightarrow \alpha_{co} = \tan^{-1} \left( \frac{x}{R + s - y} \right) \]

   Back axle
   \[ \rightarrow \beta = \frac{w_i}{R} \]
   \[ k = (i + j) * \sin(\beta) \]
   \[ P = \frac{R - k - y}{\cos(\beta)} \]
   \[ \rightarrow \alpha_{bi} = \tan^{-1} \left( \frac{x}{P - s} \right) \]
   \[ \rightarrow \alpha_{bo} = \tan^{-1} \left( \frac{x}{P + s} \right) \]

4. Lateral force
   \[ F_{y, axle, tire} = C_s, tire * \alpha_{axle, tire} \]

   It must holds:
   \[ \sum_{i=2}^{i=2} F_{ei} = \sum_{axle=f}^{axle=sh} F_{y, axle, tire} \]

   with:
   - axle - front (f), center (c), back (b)
   - tire - inner (i), outer (o)
Also the centrifugal forces \((F_c)\) will change direction if the position of the center of rotation changes. The following equation holds for the centrifugal force:

\[
\begin{align*}
\theta_1 &= \tan^{-1}\left(\frac{b - x}{R - y}\right) \\
\theta_2 &= \tan^{-1}\left(\frac{x}{P}\right)
\end{align*}
\]

Body 1:
\[
\begin{align*}
F_{cx1} &= F_{c1} \sin(\theta_1) \\
F_{cy1} &= F_{c1} \cos(\theta_1)
\end{align*}
\]

Body 2:
\[
\begin{align*}
F_{cx2} &= F_{c2} \cos(\theta_2) \\
F_{cy2} &= F_{c2} \sin(\theta_2)
\end{align*}
\]
A.5. Hertzian ellipse contact

To determine the dimensions of the contact point the relative curvatures are introduced and defined as follows:

\[
A = \frac{1}{2} \left( \frac{1}{R_{xx}} + \frac{1}{R_{yy}} \right)
\]

\[
B = \frac{1}{2} \left( \frac{1}{R_{wx}} + \frac{1}{R_{wy}} \right)
\]

Where the curvature radius on the contact interfaces from the rail and wheel are \( R_{xx}, R_{yy}, R_{wx}, \) respectively, see Figure A.10. The semi axis \( a \) and \( b \) and \( \delta \) the reduction of the distance between the bodies are defined as follow:

\[
a = m \times \left( \frac{3}{2} N \times \frac{1-v^2}{E} \times \frac{1}{A+B} \right)^{\frac{1}{3}}
\]

\[
b = n \times \left( \frac{3}{2} N \times \frac{1-v^2}{E} \times \frac{1}{A+B} \right)^{\frac{1}{3}}
\]

\[
\delta = r \times \left( \left( \frac{3}{2} N \times \frac{1-v^2}{E} \right)^2 (A+B) \right)^{\frac{1}{3}}
\]

with \( E \) being Young’s modulus and \( v \) the Poisson’s ration, assuming the same material for rail and wheel. Factors \( m, n \) and \( r \) are tabulated and a function of \( \theta \) that is defined as follow:

\[
\cos(\theta) = \frac{B-A}{B+A}
\]

\[
\theta^\circ \quad 0 \quad 5 \quad 10 \quad 30 \quad 60 \quad 90 \quad 120 \quad 150 \quad 170 \quad 175 \quad 180 \\
\lambda = A/B \quad 0.0019 \quad 0.0077 \quad 0.0717 \quad 0.3333 \quad 3.0 \quad 13.93 \quad 130.6 \quad 524.6 \quad \infty \\
n/m \quad 0.0212 \quad 0.0470 \quad 0.1806 \quad 0.4826 \quad 1 \quad 2.0720 \quad 5.5380 \quad 21.26 \quad 47.20 \quad \infty \\
m \quad \infty \quad 11.238 \quad 6.612 \quad 2.731 \quad 1.486 \quad 0.7171 \quad 0.4931 \quad 0.311 \quad 0.2381 \quad 0 \\
r \quad 0.2969 \quad 0.4280 \quad 0.7263 \quad 0.9376 \quad 1 \quad 0.9376 \quad 0.7263 \quad 0.4280 \quad 0.2969 \quad 0
\]

Table A.3 – Hertz coefficient from 0 to 180°
A.6. Kalker coefficients\[^{[3]}\]

**Longitudinal creepage**

The static longitudinal creepage in the general case (quasi-static conditions with small creepages) holds:

\[ v_{xl} = - \frac{\Delta r}{r_0}. \]

**Lateral creepage**

The lateral creepage in quasi static conditions, with small creepages, is simply the yaw angle common to the two wheels:

\[ v_y = -\alpha. \]

**Spin creepage**

In quasi static conditions, the rail speed is equal to zero and the general expression is simplified. The spin creepage is:

\[ \varphi = \frac{\sin(\gamma)}{r_0}. \]

The general expression of the creep forces takes into account stiffness coefficients c\(_{ij}\) expressed in the linear theory of kalker by:

\[ F_x = -G * a * b * c_{11} * v_x, \]
\[ F_y_{yaw} = -G * a * b * c_{22} * v_y, \]
\[ F_y_{spin} = -G * a * b * c_{23} * c \varphi. \]

With:

\[ c = \sqrt{a * b}. \]

The stiffness coefficients are defined as follow:

\[ c_{11} = 3.2893 + \frac{0.975}{b/a} - \frac{0.012}{(b/a)^2}, \]
\[ c_{22} = 2.4014 + \frac{1.3179}{b/a} - \frac{0.02}{(b/a)^2}, \]
\[ c_{23} = 0.4147 + \frac{1.0184}{b/a} - \frac{0.0565}{(b/a)^2} - \frac{0.0013}{(b/a)^3}. \]

Kalker’s equations above do not take account saturation. There are many ways to define a saturation, most described in \[1\](*Handbook of railway Dynamics*(pg. 102-103)), all the saturation law defined in the book use the concept of reduced creepage defined as follows:

\[ \tau_x = \frac{G * a * b * c_{11} * v_x}{\mu * N}, \]
\[ \tau_y = \frac{G * a * b * c_{22} * v_y}{\mu * N}, \]
\[ \tau = \sqrt{\tau_x^2 + \tau_y^2}. \]
The exponential saturation law is been chosen to be used in this report and is defined as follow:

\[
\frac{F}{\mu N} = 1 - \exp(\tau)
\]

** Theory described in book *Handbook of railway Dynamics* [3] (pg97 – 103)