Alternative lower rail design to lower injury level of occupant caused by frontal crash

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Summary

With the increase in traffic density the chance of getting involved in a traffic accident grows. Numbers have shown that car crash causes the most deaths compared with other road users. Motozawa, Kamei and Witteman have introduced a deceleration pulse to minimize the injury level of occupants. Where Motozowa and Kamei use a static design of a lower rail, uses Witteman an adaptive system to minimize deceleration and thereby the appearing injuries.

This report presents the design of a lower rail with 3 phases to decelerate a car with the deceleration pulse proposed by Witteman used as basis. The deceleration pulse contains three stages: high, low and high. To achieve this behavior four basic elements have been used; three tubes and a hydraulic cylinder. With help of the hydraulic cylinder the normal deformation sequence of tube 1, 2 and 3 can be altered in 2, 1 and 3.

Simulations have shown that the sequence of high, low and again high deceleration is very suitable for high speed collisions; the altered sequence is more applicable for lower speed collisions. With the limitations and simplified model of Motozawa and Kamei’s is it cleared that this design will lower the HIC value and the thereby correlated AIS level.
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1 General introduction

With the increasing of the vehicle fleet in the world, the demands of vehicle safety are growing. Nowadays vehicle fleet contains a lot of variety in car models, from small grocery cars till large SUV vehicles. These differences in model do not stop at their appearance; also the structure properties can make a huge dissimilarity. Most cars are equipped with a crash zone to absorb most of the energy. This crash zone has to increase the chance of survival or lower the risk of injury.

1.1 Problem description
The deformation of the crash zone creates an almost constant deceleration during a crash. This can lead to intrusion of the occupant’s survival space or too high forces are created. Both situations are unwanted with the aim to lower the amount of casualties.

1.2 Goal of research
The aim of this research is to provide a design of a frontal axial beam which properties are helping to reduce the maximum deceleration of the occupant of a car.

1.3 Outline
The report contains the following chapters: Chapter 2 contains the literature survey which describes the necessity of replacing the traditional lower rail. Chapter 3 describes the new lower rail design, followed by chapter 4. Herein the design will be simulated and compared with simulations of a traditional lower rail. Finally conclusions of the design are provided in chapter 5.
2 Literature survey

2.1 Social consequences of a crash

Car crashes are a well-known accepted fact in modern society. Not only can a crash cause several injuries, but also social costs. In figure 2.1 is the comparison made between fatal injuries of car accidents and other road users.

![Registered road kills in The Netherlands](image)

Fig 2.1 Registered road kills in The Netherlands [1]

It is seen that fatal injuries of road accidents are the majority of all deaths. Social costs of traffic accidents can be found in figure 2.2.

![Traffic accidents’ costs of 1997 in The Netherlands](image)

Fig 2.2 Traffic accidents’ costs of 1997 in The Netherlands [2]

To lower social costs and fatal injuries car crashes have to be prevented. When a crash is unavoidable intrusion of the occupant’s department and excessive high decelerations have to be averted.

2.2 Research of other investigators

One way to prevent high decelerations and intrusion is to install an adaptive front structure in cars. Several investigators have discussed various options to achieve this. Motozawa and Kamei presented a two mass model to calculate the relationship between an occupant, vehicle and seat belts.
The equations of motions for this system can be expressed as:

\[ m \ddot{X}_m = -k(X_m - X_f) \]  
\[ m \ddot{X}_f = k(X_m - X_f) + F \]

(1)  
(2)

The deceleration pulse of a conventional vehicle is depicted in figure 2.4.

The maximum occupant’s deceleration is twice the acceleration of the car. To minimize this deceleration a adaption has to be made to the car’s deceleration. Motozawa and Kamei used a cosine type mathematical solution to produce the desired behavior of the occupant. Their solution can be found in figure 2.5.

The car’s deceleration curve contains three stages: high-low-high. First the crush zone has to produce a high deceleration followed by a low value and ended by another high deceleration level. To achieve this solution a fundamental model is constructed and tested in a finite element program. The deformation process is depicted in figure 2.6.
This model is tested at a velocity of 15.56 m/s. An example of their solution is implanted in a production vehicle body.

Witteman [4] has noticed this system has its limitations. The load control beam satisfies only one specific situation. In this case the control beam is tested with a earlier mentioned speed of 15.56 m/s. What will happen when the kinetic energy level is higher or lower then stated in Motozawa and Kamei’s investigation? Intrusion of the occupant’s department or to high deceleration will occur. To avoid this limitation Witteman [4] proposed a system with variable stiffness. He also used a car deceleration with three stages [5] to minimize injury risk. Figure 2.8 shows the optimal pulse for different speeds.
Fig 2.8 Optimal pulse for different speeds [5]

First phase corresponds to the crash initiation phase, second the airbag deployment and third the occupant contact phase [6]. Witteman [7] proposed an adaptable system which energy absorption can be varied. Two stiff beams move backwards sliding along friction plates. The amount of friction force can be altered.

Fig 2.9 Two stiff beams with variable friction force [7]

Wågström, Thomson and Pipkorn [8] also concluded the call for an adaptive front structure. Their project aims at finding guidelines about how to choose deformation behavior that lead to less harmful acceleration pulses during low speed frontal collisions and to maintain intrusion resistance at high speed frontal collisions. To calculate the maximum deceleration a mass spring model has been used.

Fig 2.10 Mass spring model Wågström, Thomson and Pipkorn [8]

After evaluation is concluded that the model predicts the peak acceleration at the chosen crash velocities. To calculate the amount of required energy absorption formula 3 is used.
The total spring energy is given by:

\[ E_k = \frac{m_1v_1^2}{2} + \frac{m_2v_2^2}{2} \]  

(3)

When car 1 and 2 have the same deformation length of 700 mm and same stiffness the adjusted stiffness \( k \) can be calculated by:

\[ k = \frac{E_k}{x_{\text{max}}^2} = \frac{E_k}{0.49} \]  

(5)

With the help of adaptive frontal stiffness value \( k \) simulations can be made. Crashes are simulated between three types of vehicles: light, medium and heavy cars.

Figure 2.11 shows the results of different simulations with various speeds. L stands for Light, M for medium and H for Heavy weight vehicles. Generally spoken has an adaptive frontal stiffness a positive influence on decreasing the maximum deceleration. The frontal deformation properties have to be altered in prior to impact according to figure 2.12.

All investigations have shown that an adaptation in properties of the crush zone have a very good influence in controlling the deceleration of the occupant during crash.
3 Designing a new front structure

As seen in the previous chapter several investigations have concluded a structural adjustment in the design of the frontal crash zone will have its benefit in lowering the maximum deceleration of the occupant. This chapter will propose a construction which properties are significantly different compared with a traditional lower rail. A traditional lower rail will be seen as a structure which causes a constant deceleration behavior of a car during crashing.

3.1 Goal of the front structure design

When a car crash occurs several events can happen [9]:

- unacceptable high deceleration
- crushing of the occupant compartment
- impact with part of the vehicle interior
- ejection of the occupant

High deceleration and compartment crushing are the two main events that are dealt with in this research. According to Tomas and Sparke [12] a deceleration curve must accommodate three conflicting requirements.

1. In low speed collisions minimum vehicle damage has to occur.
2. For most accidents minimum deceleration and therefore occupant loading.
3. High load capacity for high speed collisions.

The construction that will be proposed will deal with the two last named requirements. The first requirement is an unpleasant event, but will not lead to any fatal injuries.

3.2 Structure demands

A car crash can happen in various ways. The following facts will have to be taken into account:

- Frontal crash
- Vehicle speed is 64 km/h
- Offset of the collision
- Angle of impact
- The total car mass is 1020 kg and occupant 80 kg
- Maximum installation width of 100 mm

To provide maximum protection against fatal injuries the amount of energy absorption is of great importance. When too little energy is absorbed, the chance of intrusion is very high. This situation occurs when only one lower rail is loaded; the offset is more than 50%. Thereby the new lower rail design must be sufficient enough to absorb energy and prevent intrusion. Also the angle of impact plays an important role. With the deceleration curve of Witteman table 3.1 can be reproduced.
<table>
<thead>
<tr>
<th>Crash velocity</th>
<th>32 km/h</th>
<th>56 km/h</th>
<th>64 km/h</th>
</tr>
</thead>
<tbody>
<tr>
<td>Phase 1</td>
<td>Deceleration</td>
<td>9 g</td>
<td>32 g</td>
</tr>
<tr>
<td></td>
<td>Deformation length</td>
<td>170 mm</td>
<td>188 mm</td>
</tr>
<tr>
<td></td>
<td>Time duration</td>
<td>12.5 ms</td>
<td>12.5 ms</td>
</tr>
<tr>
<td>Phase 2</td>
<td>Deceleration</td>
<td>9 g</td>
<td>9 g</td>
</tr>
<tr>
<td></td>
<td>Deformation length</td>
<td>Total 448 mm</td>
<td>416 mm, total 386 mm</td>
</tr>
<tr>
<td></td>
<td>Time duration</td>
<td>100.7 ms</td>
<td>42.5 ms, total 55 ms</td>
</tr>
<tr>
<td>Phase 3</td>
<td>Deceleration</td>
<td>9 g</td>
<td>23 g</td>
</tr>
<tr>
<td></td>
<td>Deformation length</td>
<td>138 mm, total 724 mm</td>
<td>176 mm, total 762 mm</td>
</tr>
<tr>
<td></td>
<td>Time duration</td>
<td>35 ms, total 90 ms</td>
<td>39.4 ms, total 89.4 ms</td>
</tr>
</tbody>
</table>

Table 3.1 Parameters of three accidents velocities [5]

The amount of energy that has to be dissipated can be calculated by:

\[ E = m \times a \times s \]  

<table>
<thead>
<tr>
<th>Phase</th>
<th>Energy (Nm)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>91291.86</td>
</tr>
<tr>
<td>2</td>
<td>38653.36</td>
</tr>
<tr>
<td>3</td>
<td>43681.97</td>
</tr>
<tr>
<td>Total</td>
<td>173627.19</td>
</tr>
</tbody>
</table>

Table 3.2 Dissipated energy

Table 3.2 exists of the amount of dissipated energy during each phase. A car contains various systems to absorb this amount of energy. Figure 3.1 shows a schematic drawing of an automotive body.

![Schematic representation of an automotive body](image)

Fig 3.1 Schematic representation of an automotive body [11]

In the above figure two lower rails can be seen. In this report will be stated that the requested energy dissipation of table 3.2 will be handled by only one new lower rail to avoid intrusion of the compartment.

### 3.3 Proposal of lower rail design

With all requirements taken into account the following design has been made. It contains three tubes and a hydraulic cylinder.
3.3.1 Deforming sequence of lower rail explained

Figure 3.2 depicts the lower rail with the three phases indicated.

During a crash the lower rail will deform. First the tube of phase one has to crumble to get the desired deceleration. Tube 1 has a smaller cross area compared with tube 2 and also contains a trigger element, therefore it has a higher stress concentration and will deform first. When the buckling deformation has proceeded through the length of the zone, the valve of the hydraulic cylinder will open. Tube 2 is able to deform, the deceleration level of phase two will be achieved. When the cylinder’s stroke has reached its maximum length, the load increases and tube 3 finally deforms.

3.3.2 Lower rail dimensions explained

As seen in figure 3.2 the lower rail contains 3 tubes and a hydraulic cylinder as basic components. To choose the right dimensions, the amount of energy that a tube can dissipate has to been known. Therefore figure 3.4 has been used. It contains the test results of the energy absorption of different profiles (FeP03) and is taken from Witteman [14].
By using formula 7 the tube’s dimensions can be calculated to absorb the requested amount of energy:

\[ E \approx t^\frac{5}{3} \times H^\frac{1}{3} \quad (7) \]

\[ t = \text{thickness} \]
\[ H = \text{column width} \]

This formula is based on the theory of Wierzbicki. A tube with the dimensions 75 x 75 x 2 will be taken as a reference tube.

**Tube 1 and 3**

Witteeman stated that a tube must have an original length of 27.5% longer than the necessary deformation size stated in table 3.1. Tube 1 also contains an initiator to establish a stable force level during crushing. Triggering with a 10 % bead leads to an increase of 8.6 % in energy absorption [14]. With the help of formula 7 and the increase percentage, tube 3 has been chosen to be 100 x 100 x 4 x 225. With these dimensions the cross area of tube 3 becomes 1536 mm². The cross section of tube 1 is set so that the deformation load will be slightly less than the maximum load of tube 3. In figure 3.4 can be seen that a tube of 75 x 75 x 2 x 188 absorbs 15770 Nm.

According to formula 7 the E value corresponds to 13.38. To absorb 91291.86 Nm of phase 1, the E value must raise to 71.37. Tube 1 must therefore succeed the following enumeration:

\[
\begin{align*}
H1: & \quad t^\frac{5}{3} \times H^\frac{1}{3} = 71.37 \\
H2: & \quad H^2 - (H - 2t)^2 < 1536 \\
H3: & \quad H \geq 2t
\end{align*}
\]

The enumeration is visualized in figure 3.5.
Fig 3.5 Enumeration of tube 1

De solution is lying on line H1 and is limited by H2 and H3. The dimensions 70 x 70 x 5.5 x 240 satisfy these conditions.

**Tube 2**

With formula 4 is it possible to calculate the desired stiffness of each tube. Table 3.3 shows a dispute. Tube 2 has to deform second, but it’s the “weakest” compared with the other two tubes.

<table>
<thead>
<tr>
<th>Tube</th>
<th>Stiffness [kN/m]</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>0.96 x 10^3</td>
</tr>
<tr>
<td>2</td>
<td>0.08 x 10^3</td>
</tr>
<tr>
<td>3</td>
<td>2.81 x 10^3</td>
</tr>
</tbody>
</table>

Table 3.3 The required stiffness of all tubes

To get the requested behavior of the lower rail, a hydraulic cylinder has been used to act as a controllable stiffness. The stroke of the piston must have same length as the deformation length of phase 2. Therefore the dimension of tube 2 is minimal 796 mm; it also must contain space to house the attachments of the cylinder. By using formula 7, an energy absorption of 38653 [Nm] and figure 3.4, the dimensions eventually becomes 100 x 100 x 2 x 850.

**Hydraulic cylinder**

During deformation of the lower rail the hydraulic cylinder undergoes several forces. During phase one the cylinder will be loaded with the highest force. Bending resistance must be obtained in this situation. The critical bending force [15] can be calculated by:

\[
F_{cr} = \frac{\pi^2 EI}{(KL)^2} \tag{8}
\]

E = young modulus
I = inertia
K = effective length factor
L = length of the column

To compute the smallest critical force, some assumptions will be made. The highest deceleration occurs during deformation of tube 1, namely 45g. This leads to a force of
486 kN. It will be assumed that this force will load the cylinder. The smallest critical force happens when the hydraulic cylinder is at its maximum length. When the inertia is taken from the cylinder, the critical force becomes 944 kN. Bending is avoided.

To ensure a safe wall thickness of the cylinder, formula 9 [16] can be used.

\[ \sigma = \frac{P}{d} \times \frac{R}{d} \quad (9) \]

\( \sigma \) = stress  
\( P \) = pressure in cylinder  
\( R \) = inner radius of cylinder  
\( d \) = wall thickness

With a thickness taken of 5mm, the stress doesn’t become too high.

While deforming during phase 2, the pivot is moving back in the cylinder. Oil will flow through the exit valve. This oil stream causes a damping force of the cylinder. Using formula 10 [17] this amount can be calculated:

\[ F_d = 0.241 \times \rho \times l_r \times \left( A_{cil} - A_{piv} \right) \times D^{0.25} \times \left( y_{gem}^* \right)^{1.75} \quad (10) \]

\( \rho \) = density  
\( l_r \) = length of the restriction  
\( A_{cil} \) = Area of the cylinder  
\( A_{piv} \) = Area of the connecting rod  
\( d_r \) = diameter of the restriction  
\( \nu \) = viscosity of the oil  
\( y_{gem}^* \) = speed of piston

With chosen parameters, the damping force becomes 40 N, a small part compared with the 38653 N of phase 2. For this reason the damping force will be neglected.

The inertia of the profiles is of great importance to guarantee good energy absorption when a crash occurs under an angle. This can be seen in picture 3.6 [14]. Figure 3.6 contains the test results of energy absorption of profiles with a load direction of 30 degrees. Profiles with a lower inertia level are more sensitive to load direction than higher ones. Tube 3 of the lower rail design has the highest inertia of all three tubes. It will be placed as far as possible to the point of impact to secure a good resistance against bending.

With the specified dimensions of all used elements is trying to satisfy all demands of a crush zone. Chapter 4 will continue with the simulation of the lower rail in Matlab to show its potential in lowering the deceleration of the occupant.
Fig 3.6   Energy absorption of different profiles with a load direction of 30 degrees
4 Lower rail simulated in Matlab

The lower rail is designed to produce the deceleration curve of 64 km/h seen in figure 2.8. The earlier mentioned model of Motozawa and Kamei will be used to simulate the behavior of car and occupant during a crash. Appendix B will explain the model of Motozawa and Kamei. The simulation conditions are:

- frontal collision
- offset of 40 % (one lower rail is deformed)
- all crash energy absorbed by lower rail

In this chapter different crash speeds will be simulated and compared with a constant deceleration of the vehicle. The constant deceleration is obtained by using a speed of 64 km/h and a crash zone of 762 mm.

The new design of the lower rail contains a hydraulic cylinder with valve that can be activated. The deformation sequence can be altered by activating the valve before the crash occurs, phase one and two will be shifted. The phase shifting will also be investigated to see its effects on the occupant’s deceleration.

4.1 Simulation of lower rail with 32, 56 and 64 km/h

The occupant is exposed to a deceleration produced by the lower rail. Its impact on the occupant’s deceleration can be found below.

![Simulation results](image)

Fig 4.1 Results of simulations with 32, 56 and 64 km/h
Figure 4.1 shows a variety of deceleration curves. By each collision speed are the three crash beams investigated. The beam with a constant deceleration level will be named “normal”. Results with the new construction are indicated as “with rail”. Followed by the results of the new construction containing the adjusted deformation sequence and indicated as “with phase shifted”. The following paragraph will link a number to the simulations results to judge which curve has the most positive influence on the occupant.

4.2 Simulation outcomes judged by HIC value

The simulation model of Motozawa and Kamei represents the occupant as a single mass. Therefore undergoes all body parts the same deceleration. An important formula to judge the movement of the head is the HIC:

$$HIC = \max \left( t_2 - t_1 \right) \int_{t_1}^{t_2} \frac{1}{t_2 - t_1} \, a(t) \, dt \right]^{2.5}$$

In the above expression, $a$ is the resultant deceleration expressed in g’s and $t_1$ and $t_2$ are two points in time. In typical automotive applications, the maximum window of $t_2 - t_1$ is at most 36 ms. The calculated HIC values can be found in table 4.1.

<table>
<thead>
<tr>
<th>Lower rail situation/speed</th>
<th>32 km/h</th>
<th>56 km/h</th>
<th>64 km/h</th>
</tr>
</thead>
<tbody>
<tr>
<td>traditional lower rail</td>
<td>355</td>
<td>353</td>
<td>354</td>
</tr>
<tr>
<td>new lower rail</td>
<td>653</td>
<td>113</td>
<td>200</td>
</tr>
<tr>
<td>new lower rail with phase shifting</td>
<td>83</td>
<td>797</td>
<td>553</td>
</tr>
</tbody>
</table>

Table 4.1

Once a HIC value is obtained, it can be correlated to an AIS value used in table 4.2 [17]. In general, a HIC of 1,000 indicates a high probability of moderate brain injury and has to be avoided. Values above 1,500 correspond to a high probability of severe brain injury.

<table>
<thead>
<tr>
<th>AIS</th>
<th>Severity</th>
<th>Type of injury</th>
<th>HIC</th>
</tr>
</thead>
<tbody>
<tr>
<td>0</td>
<td>none</td>
<td>None</td>
<td></td>
</tr>
<tr>
<td>1</td>
<td>minor</td>
<td>Superficial injury</td>
<td>&lt; 250</td>
</tr>
<tr>
<td>2</td>
<td>moderate</td>
<td>recoverable</td>
<td>&lt; 750</td>
</tr>
<tr>
<td>3</td>
<td>serious</td>
<td>Possible recoverable</td>
<td>&lt; 1250</td>
</tr>
<tr>
<td>4</td>
<td>severe</td>
<td>Not fully recoverable without care</td>
<td>&lt; 1750</td>
</tr>
<tr>
<td>5</td>
<td>critical</td>
<td>Not fully recoverable with care</td>
<td>&lt; 2500</td>
</tr>
<tr>
<td>6</td>
<td>maximum injury</td>
<td>Fatal</td>
<td>&gt; 2500</td>
</tr>
</tbody>
</table>

Table 4.2

The “type of injury” description in the table is associated with each AIS value and the same description would be applicable to AIS levels obtained from other biomechanical responses. Because of this the AIS scale is independent of the location of the injury. The highlighted values in table 4.1 shows the lowest HIC value and thereby the lowest AIS level.
5 Conclusion

In this report a proposal is given for a design of a new lower rail in order to minimize the peak occupant deceleration with a collision of 64 km/h. This is done by using profiles with different stiffness placed in series. According to Witteman three different stages must be go trough. First stage contains the highest deceleration and therefore must include the stiffest profile. This is in contrast with the second stage; the lowest deceleration must be achieved. In order to prevent unwanted deformation of profile 2, a hydraulic cylinder has been used to act as a controllable stiffness.

As stated before the design of the lower rail has been optimized to minimize the occupant’s deceleration of a collision with 64 km/h. To investigate how this rail will react on collisions with lower speed, different simulations have been made. Chapter 4 shows various results. From these plots and calculated HIC values it can be concluded that at certain speeds the new lower rail design has a positive contribution in minimizing the peak deceleration. Also the influence of shifting the deformation of profile 1 and 2 is examined. At lower speeds it is recommended first to deform tube 2 before 1 by opening the exit valve of the hydraulic before malformations takes places.

While evaluating this investigation a few notes have to be made about the lower rail design and simulating it in Matlab. First the design will be treated. The lower rail is designed to produce a prescribed deceleration curve. All the energy produced by an accident will be dissipated by the lower rail. This is not the case in real live. Witteman [14] mentioned that a front structure of a car absorbs energy as stated in figure 5.1. This absorption will influence the deceleration curve. The installation of the new lower rail in figure 5.1 will result in a too stiff crush zone.

Also the amount of overlap during a crash is of great importance in absorbing energy. Most of the accidents will take place with an overlap. The higher the overlap value, the more components will be involved in absorbing energy.

![Fig 5.1 absorption of energy according to Witteman [14]](image)

The draft of the lower rail contains a hydraulic cylinder to control the stiffness of the weakest tube. The cylinder contains an exit valve to make it possible to guide the fluid
out of the cylinder. The opening of this valve will take time. This duration is not included in the simulation of the lower rail. The opening of the valve will influence the transition of the deformation of phase 1 to 2.

The total length of the lower rail has become 1325 mm. It is thinkable that this length is too big to fit in a motor compartment of a car. To reduce the length it is possible to use a 3 stage hydraulic cylinder, like figure 5.2.

![3 Stage hydraulic cylinder](image)

The length of the cylinder will decrease even as the length of the tube of phase 2. This also has its benefit on when a crash occurs under an angle. The bending moment will decrease.

The simulation model of Motozawa and Kamei is a simplification to investigate the behavior of an occupant. In this simplification several cases are not considered. The occupant and car are represented as a simple point mass which can move in one direction. In real live this is not the case; they can translate and rotate in several directions. This will have its influences on the deceleration of car and occupant. Also the model does not limit the movement of the occupant and no airbag is presented. There is only limited space available in real car. When this limit has been reached, the occupant must hit a hard surface or the deployed airbag. Figure 4.1 shows frequently the maximum deceleration at the end of the simulation. In this time the maximum movement in the available space is already reached. When hitting a hard surface, higher decelerations will occur.

As earlier mentioned the model only simulates the movement of car and occupant, no intrusion of parts is calculated. This intrusion can lead to serious injury consequences.

This report presented a lower rail design which contains several profiles with different stiffness. The simulation demonstrated an improvement to control the maximum deceleration of an occupant compared with a constant deceleration of now a days cars. Also the possibility to control the deformation sequence has a positive effect to minimize the occupant’s AIS level.
6 References

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Appendix A

Lower rail concept

Front View

Side view

3-D view
Appendix B

Simulink model of Motozawa and Kamei

The model (figure B.1) used to simulate the behavior of the occupant during a crash with constant deceleration is shown in figure B.2.

![Fig. B.1 Two mass one dimensional model [3]](image)

The model is based on two equations of motion:

\[
\begin{align*}
    m\ddot{X}_m &= -k(X_m - X_f) & (1) \\
    m\ddot{X}_f &= k(X_m - X_f) + F & (2)
\end{align*}
\]

At \(t = 0\) the initial conditions are:

\[X_m = X_f = 0 \quad \text{and} \quad \dot{X}_m = \dot{X}_f = -v_0\]

To ensure the best vehicle crash energy absorption efficiency, it is currently thought to be ideal for crash load \(F\) to be constant [3]. If the mass of the car is much higher than the mass of the occupant, formula 2 can be simplified as:

\[
\ddot{X}_f = \frac{F}{M + m}
\]

The parameters taken in the simulink model are:

\[
\begin{align*}
    m_1 &= \text{"mass of occupant"} = 80 \text{ [kg]} \\
    m_2 &= \text{"mass of car"} = 1020 \text{ [kg]} \\
    k_{\text{belt}} &= \text{"stiffness of occupant belt"} = 10000\text{[N/m]}
\end{align*}
\]
To fit the behavior of the new rail design in, several adaptations will be made. The constant deceleration will be replaced by a three-stage deceleration. Figure B.2 will be substituted by figure B.3.

The simulink model contains two switches. They will switch when the deformation length of phase 1 and phase 2 are passed through every stage. In other words “switch” will activate phase 2 when the deformation length of 188 mm has been passed and
“switch1” after 586 mm. This stands for the addition of phase 1 and 2, see table 3.1. To investigate the deceleration of the occupant when phase 1 and 2 are shifted, the activation length of “switch” will be altered into 398 mm. Force F1 replaces F2 and vice versa.