ME412 Automotive Engineering
Project deliverable

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Contents

Chapter 1

Requirements

1.1 Target Segment

1.1.1 Main competitors

Our target segment is economy cars. The main competitors in the economy car market consist of Mitsubishi Mirage, Chevrolet Spark, Honda Fit, and Suzuki Celerio. These cars averaged to approximately $40,000$ sold units in the year of 2019.

Figure 1.1: Main competitors: Mitsubishi Mirage(top left), Chevrolet Spark(top right), Honda Fit(bottom left), Suzuki Celerio(bottom right)

1.2 Vehicle Purpose

Economy cars are widely used for commuting short to medium distances. They are excellent means of enjoying another one's company and running daily tasks such as commuting to work or picking up groceries. Also, since they can load several small sized luggage, they are also suitable for traveling nearby places.

1.3 Car Attributes

Our design maintains the affordability of economy cars, but will capture a modern, cutting-edge aesthetic. The design is made for urban settings and for functionality; it is compact and efficient in its build, as well as has immense functionality due to its small and lightweight structure. This allows the automobile to be easily parked and maneuvered in ordinarily difficult parking situations. The car's design is a great balance of refinement and affordability.

1.4 Main specifications

The design we are creating must satisfy several criteria. It must be able to carry $2-4$ passengers. Its cargo carrying capacity is limited to a small load amount (such as groceries or small sized suitcases), and the car's range is limited to approximately 160km. Its climb gradient is maintained at around $30-40\%$. It can accelerate from 0km/h to 100km/h in a time span of $7-8$ seconds. The car can travel approximately 15km/L, and its maximum dimensions are limited to $3-3.5\text{m} \times 1.5\text{m} \times 1.5\text{m}[L \times W \times H]$.

Chapter 2

Vehicle packaging

2.1 Ideation Skecthes

Figure 2.1: Ideation sketches

2.2 Key targets specifications

Overall

- Length: 3700mm
- Width: 1600mm
- Height: 1500mm

Drivers height

- H30: 250mm
- H5: 450mm
- L113: 440mm
- Forward vision angles: $7 - 14$ up, 7 down
- Front shoulder room: 1400mm
- Lateral location: 350mm

Rear occupants

- Rear shoulder room: 1350mm
- Rear lateral location: 350mm

Powertrain

- Engine: Front transverse, FWD, 4 cylinders
- Engine size: 670×695mm
- Fuel tank: Situated below rear passenger seats

Cargo space

- Maximum cargo width: 1200mm due to suspension
- Minimum cargo width: 1150mm
- Aperture height: 850mm
- Lift over height: 650mm
- Cargo volume: 0.5m^3

Wheel locations

- Wheel outer diameter: 600mm
- Wheelbase: 2500mm
- Front track: 1490mm
- Rear track: 1470mm
- Ground clearance: 150mm at ML3
  - ML1 Curb mass: 950kg
  - ML2 Design mass: 1100kg
  - ML3 Gross vehicle mass: 1400kg

2.3 Orthographic side and end views

Below is an orthographic side and end views that shows the vehicle outlines and the basic layouts. It includes the dimensions for occupancy packaging, such as H5, H30, forward vision angles, effective headroom, shoulder room, and lateral locations. The figure also shows the cargo storage areas and the location of engine, gearbox, and fuel tank. It also indicates the overall dimension of the vehicle that are tabulated above.

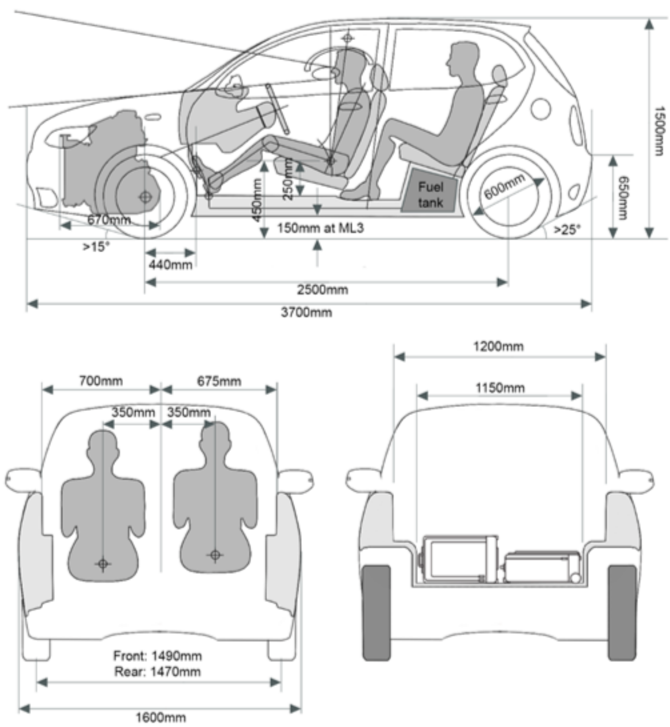

Figure 2.2: Orthographic side and end views

Chapter 3

Power requirements

An analysis on power requirements is pivotal when designing a car. Based on the power requirements analysis, suitable engine is selected.

3.1 Resistance analysis

In order to estimate the power requirements of the vehicle, analysis on the resistance force components at the required state must be made. The main resistance forces that impact the power requirements of the vehicle are rolling resistance F_R, aerodynamic drag F_{aero}, climbing resistance F_{Cl}, and acceleration resistance F_a.

Rolling resistance The rolling resistance of the vehicle can be calculated as

$$F_R = f_R(m_V + m_{PL})g$$

The vehicle will be in most times operated on dry concrete or asphalt roads. This assumption can be taken into account by letting the rolling resistance coefficient $f_R = 0.015$.
The curb mass m_V of our vehicle is 950kg. Since our vehicle should be capable of taking in 4 passengers and some cargo, the maximum payload m_{PL} is decided to be 400kg. Considering that Honda Fit has maximum payload of 385kg and Mitsubishi Mirage of 408kg, this value is reasonable.

Aerodynamic drag From the equation

$$F_{aero} = \frac{1}{2}\rho v^2 c_D A_x$$

the air density is $\rho = 1.225\text{kg/m}^3$, and the drag coefficient c_D of the vehicle is assumed to be 0.29 at the worst case scenario. Also, based on the dimensions of the vehicle mentioned above, the frontal area A_x is approximated to be 2.04m^2.

Acceleration resistance Reasonable assumptions on the rotational inertia coefficients k_{m_i} from the following equation must be made:

$$F_a = (k_{m_i} \cdot m_V + m_{PL}) \cdot a_x$$

Our vehicle will be powered by an internal combustion engine, operating with 5 gears. Assumed values of rotational inertia coefficients k_{m_i} associated with each gear level are tabulated below.

Gear i	1	2	3	4	5
k_{m_i}	1.32	1.15	1.10	1.07	1.06

3.2 Target specifications on power requirements

Different operating conditions will be considered to determine the maximum power requirements. First, the power required to maintain top velocity of $v_r = 200\text{km/h}(= 55.56\text{m/s})$ at no acceleration and no grade will be considered. Second, the power required to accelerate from 0 to 100km/h in required time $t_r = 7.2s$ will be considered. Lastly, the power required to maintain the speed of $v = 180\text{km/h}(= 50\text{m/s})$ at the climb gradient of 2%, which is equivalent to $\theta = 1.146$, will be considered.

Top speed operation condition Net resistance force acting on the vehicle under the top speed operation condition is

$$\begin{aligned} F &= F_R + F_{aero} \\ &= f_R(m_V + m_{PL})g + \frac{1}{2}\rho v^2 c_D A_x \\ &= 0.015(950 + 400)\text{kg} \cdot 9.81\text{m/s}^2 + \frac{1}{2} \cdot 1.225\text{kg/m}^3 \cdot (55.56\text{m/s})^2 \cdot 0.29 \cdot 2.04\text{m}^2 \\ &= 1317.21\text{N} \end{aligned}$$

Corresponding power requirement is

$$P_{max} = Fv = 1317.21\text{N} \cdot 55.56\text{m/s} = 73.18\text{kW}$$

Acceleration from 0 to 100 For a vehicle moving at speed v and accelerating at a, the maximum power requirement is

$$\begin{aligned} P_{max} &= (F_R + F_{aero} + F_a)v \\ &= \left[f_R(m_V + m_{PL})g + \frac{1}{2}\rho v^2 c_D A_x + (k_{m_i} m_v + m_{PL})a_x \right] v \end{aligned}$$

Rearranging in terms of acceleration,

$$a_x(P_{max}, v) = \frac{\frac{P_{max}}{v} - \frac{1}{2}\rho v^2 c_D A_x - f_R(m_V + m_{PL})g}{k_{m_i} m_v + m_{PL}} = \frac{\Delta v}{\Delta t}$$

Therefore we can find the relationship between the time required to accelerate from 0 to 100km/h and the maximum power requirement as following:

$$\begin{aligned} t_{0-100\text{km/h}} &= \sum_{t=0}^{t(v=100\text{km/h})} \Delta t = \sum_{v=0\text{km/h}}^{100\text{km/h}} \frac{1}{a_x(P_{max}, v)} \Delta v \\ &= \sum_{v=0\text{km/h}}^{100\text{km/h}} \frac{k_{m_i} m_v + m_{PL}}{\frac{P_{max}}{v} - \frac{1}{2}\rho v^2 c_D A_x - f_R(m_V + m_{PL})g} \Delta v \\ &= f(P_{max}) \end{aligned}$$

For example, the time taken until the vehicle with $P_{max} = 100\text{kW}$ attains 100km/h can be calculated by integrating $\Delta t/\Delta v = 1/a_x(100\text{kW}, v)$ with respect to v. It is assumed that the gear is changed to higher lever for every 10m/s of speed increase.

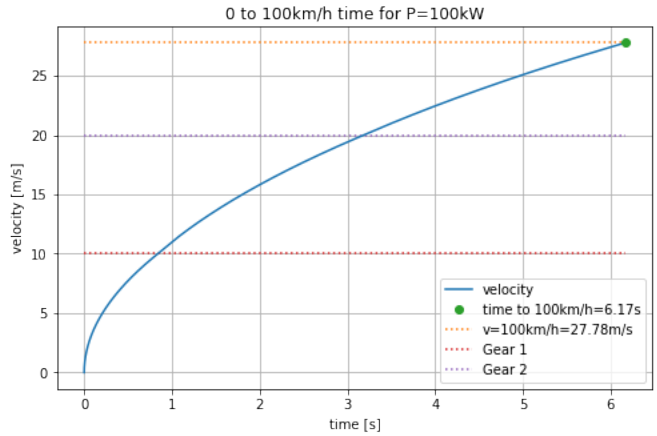

Figure 3.1: 0 to 100km/h time for P=100kW

This calculation is repeated with different values of P_max, and the result is plotted. From the calculated result, the power requirement for $t_r = 7.2s$ is interpolated and found to be

$$P_{max} = 86.69\text{kW}$$

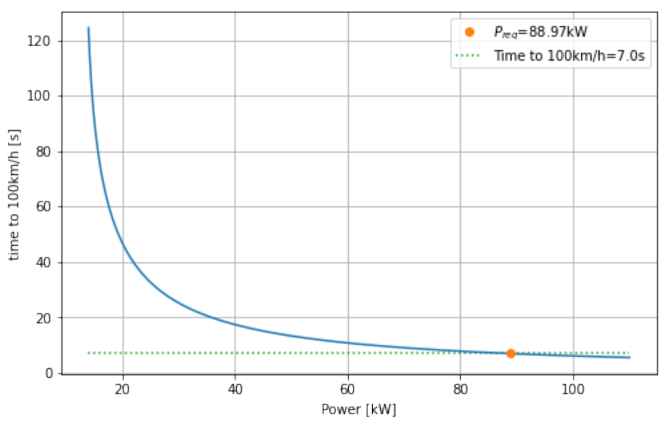

Figure 3.2: Power-time graph

Climb gradient Net resistance force acting on the vehicle under the climb gradient condition of 2% is

$$\begin{aligned}
F &= F_R + F_{aero} + F_{Cl} \\
&= f_R(m_V + m_{PL})g + \frac{1}{2}\rho v^2 c_D A_x + (m_V + m_{PL})g\sin\theta \\
&= (f_R + \sin\theta)(m_V + m_{PL})g + \frac{1}{2}\rho v^2 c_D A_x \\
&= (0.015 + \sin(1.146°))(950 + 400)\text{kg} \cdot 9.81\text{m/s}^2 + \frac{1}{2} \cdot 1.225\text{kg/m}^3 \cdot (50\text{m/s})^2 \cdot 0.29 \cdot 2.04\text{m}^2 \\
&= 1582.08\text{N}
\end{aligned}$$

Corresponding power requirement is

$$P_{max} = Fv = 1582.08\text{N} \cdot 50\text{m/s} = 79.10\text{kW}$$

3.3 Net power requirements

From the analysis above, it can be found that the maximum net power requirements for the vehicle occurs under the acceleration operating condition.

$$P_{max} = 86.69\text{kW}$$

Considering that 2018 Honda Fit Sport CVT has an engine power of 95.4kW, 2020 Mitsubishi Mirage has 58.2kW, and 2019 Chevrolet Spark has 73.1kW, it can be concluded that the calculated power is reasonable for a sporty economic hatchback.

Chapter 4

Tractive force diagram

4.1 Choosing an engine

Given our power requirements, engine 2 was chosen from the ICE Generic Engines Catalog as the engine implemented into our car design. The specification of engine 2 is as follows.

Work cycles per rotation	$i = 0.5$
Displacement	$V_D = 1.4\text{L}$
Power	$P_{max} = 112\text{kW@5000rpm}$
Torque	$M_{max} = 250\text{Nm@}1500 - 3500\text{rpm}$
4 cylinder in-line	Gasoline | Turbocharged

4.2 Top speed calculation

$$
\begin{aligned}
P_{max} \cdot \eta_G &= F \cdot v_{max} \\
&= (F_R + F_{aero}) v_{max} \\
&= (f_R m_V g + \frac{1}{2} \rho v_{max}^2 c_D A_x) v_{max} \\
&\frac{1}{2} \rho c_D A_x v_{max}^3 + f_R m_V g v_{max} - P_{max} \cdot \eta_G = 0 \\
&\frac{1}{2} \cdot 1.225\text{kg/m}^3 \cdot 0.29 \cdot 2.04\text{m}^2 \cdot v_{max}^3 + 0.015 \cdot 950\text{kg} \cdot 9.81\text{m/s}^2 \cdot v_{max} - 112000\text{W} \cdot 0.95 = 0 \\
v_{max} &= 64.53\text{m/s}
\end{aligned}
$$

4.3 Choosing gear ratios

Largest ratio calculation The tire used in the following calculation is model 185/65 R 14, whose diameter is calculated to be

$$14\text{in} \cdot 25.4\frac{\text{mm}}{\text{in}} + 2 \times (185\text{mm} \cdot 0.65\%) = 596.1\text{mm}$$

which is close to the 600mm of tire diameter from the packaging plan.
The maximum gear ratio was selected so that the vehicle would be able to drive off the 50% of grade.

$$\begin{aligned} i_{A_{max}} &= \frac{r_{dyn} \cdot (m_V + m_{PL}) \cdot g \cdot (f_R \cdot cos\theta + sin\theta)}{T_{M_{max}} \cdot \eta_G} \\ &= \frac{0.289\text{m} \cdot (950 + 400)\text{kg} \cdot 9.81\text{m/s}^2 \cdot (0.015 \cdot \cos 27° + \sin 27°)}{380\text{Nm} \cdot 0.95} \\ &= 4.95 \end{aligned}$$

Smallest ratio calculation Contrary to standard economic cars, we designed our vehicle to be sporty and support a higher acceleration rate. Thus, we chose to use v_{max} optimal as the method of choice for the smallest gear ratio calculation.

$$\begin{aligned} i_{A_{min}} &= \frac{n_N \cdot r_{dyn} \cdot 2\pi}{v_{max}} \\ &= \frac{4000rpm \cdot \frac{1\text{min}}{60\text{s}} \cdot 0.289\text{m} \cdot 2\pi\text{rad/rev}}{68.92\text{m/s}} \\ &= 1.76 \end{aligned}$$

Intermediate gears with progressive gear steps calculation Progressive gear steps are utilized for passenger car transmissions. The following values are obtained through the relationship between the ratio of two neighboring gears.

ϕ_{G1}	ϕ_{G2}	$i_{G_{tot}}$	i_1	i_2	i_3	i_4	i_5
1.204	1.05	2.8125	4.950	3.553	2.677	2.118	1.760

Using the evaluated quantities above, a saw profile diagram for progressive gear steps is plotted in Figure 5.

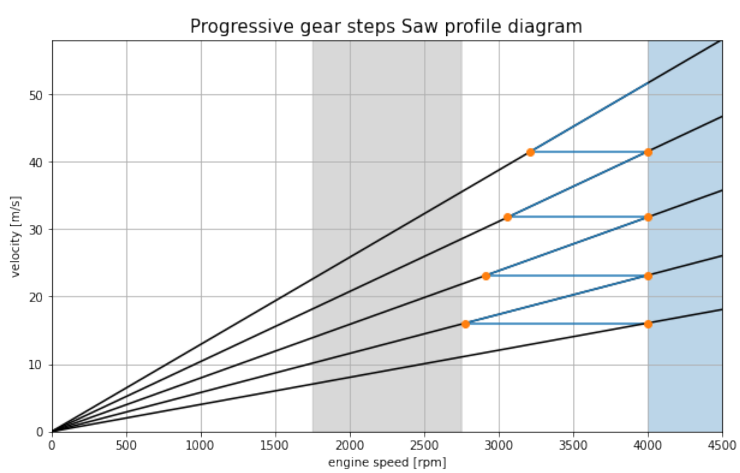

Figure 4.1: Progressive gear steps Saw profile diagram for $\phi_{G2} = 1.05$

4.4 Tractive force diagram

Based on the maximum power and the power demand for level road and no payload, following tractive force diagram is obtained. Note that the available force includes 95% of transmission efficiency.

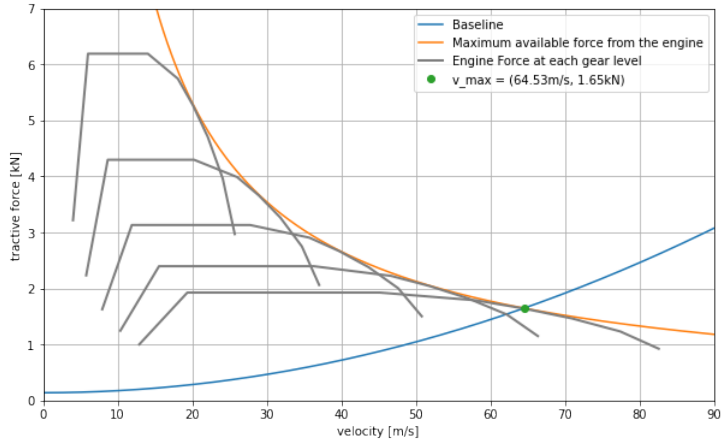

Figure 4.2: Tractive force diagram

Chapter 5

Longitudinal driving dynamics

Driving performance

5.1 Drive limits

The driving concept our vehicle operates at is front-wheel drive. In order to achieve sufficient acceleration, a larger load distribution to the rear axle is required. We've designated values:

$$l_f/l(curbweight) = 0.35$$
$$l_f/l(fullyloaded) = 0.45$$
$$h_g/l = 0.20$$
$$\mu = 1.0$$

to satisfy our functional requirements.
The resulting traction limits are as follows:

$$a_{limit}(fullyloaded) = 6.3765\mathrm{m/s^2}$$
$$grade_{limit}(curbweight) = 55\%$$

5.2 Excess force

The excess force was calculated from the tractive force diagram of our fully loaded and curb weight vehicle. The plot displays the excess tractive force for each respective gear of our curb weight and fully loaded vehicle.

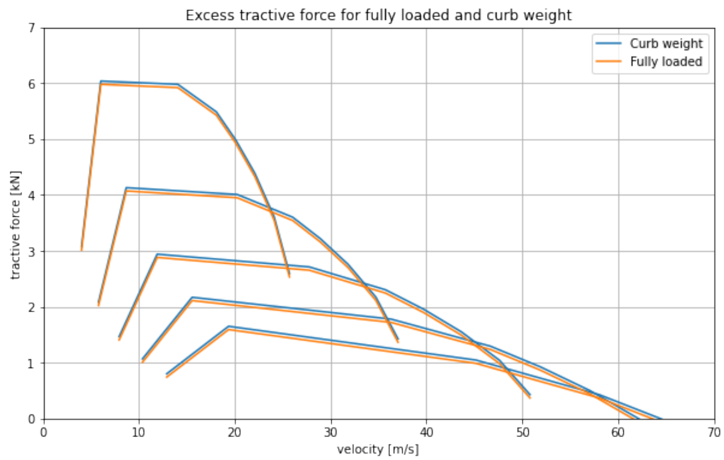

Figure 5.1: Excess tractive force

5.3 Maximum grade

The maximum grade was determined over driving speed of our fully loaded vehicle. This was iterated over five different gears. From the plot, it is observed that there is a maximum grade of 50% for our vehicle, which is below the traction limit for gradeability of 55%.

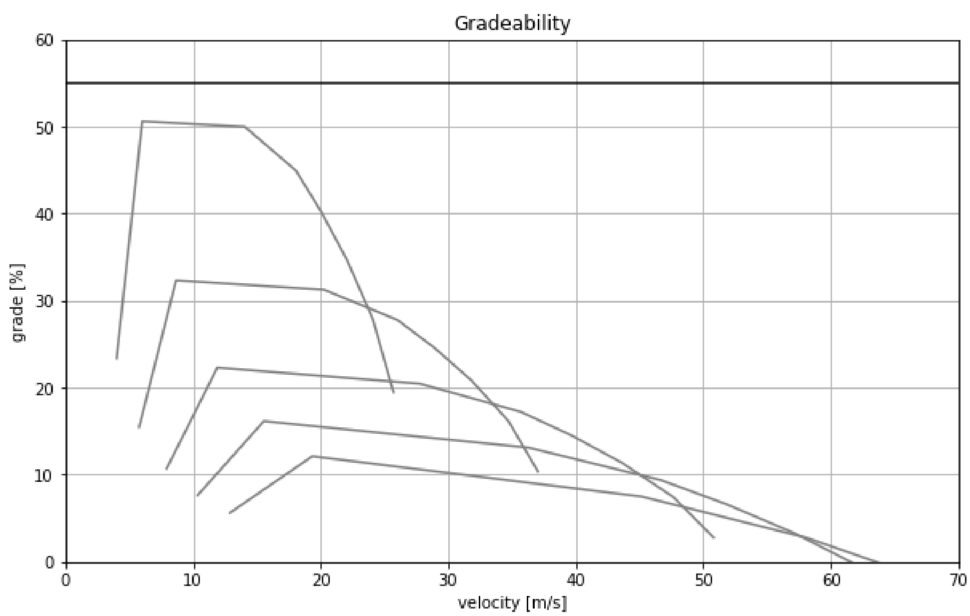

Figure 5.2: Gradeability

5.4 Maximum acceleration

The maximum acceleration of our vehicle is calculated with no payload in effect. We also assume a transmission efficiency of 95%. The estimated traction limit for acceleration is $6.3765 m/s^2$. It requires 6.2076 s to accelerate from 0-100 km/h and 3.1884 s to accelerate from 80-120 km/h.

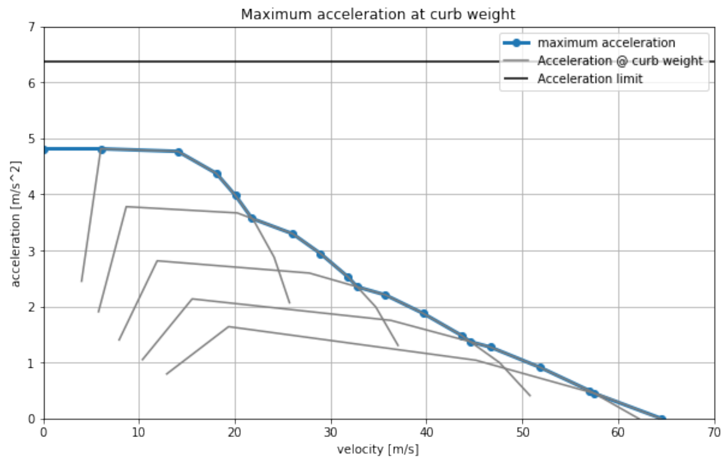

Figure 5.3: Maximum acceleration and acceleration limit

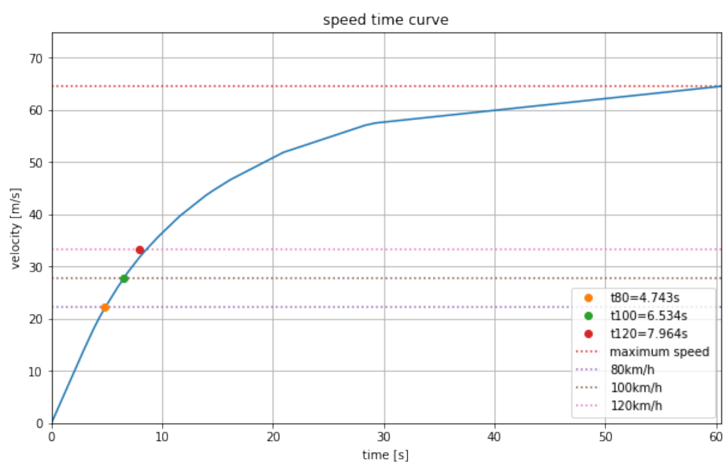

Figure 5.4: Time required to accelerate

5.5 Fuel consumption

For the fuel consumption calculation, we similarly assume curb weight for our vehicle. The best fuel consumption was calculated for our vehicle at 50 km/h and 100 km/h.

$$FC = \frac{B_{time}}{v} = \frac{BSFC \cdot BMEP \cdot V_D \cdot n_M \cdot i}{\rho_{fuel} \cdot v}$$

Best fuel consumption at 50 km/h: 2.9430 L/100 km
Best fuel consumption at 100 km/h: 4.9051 L/100 km

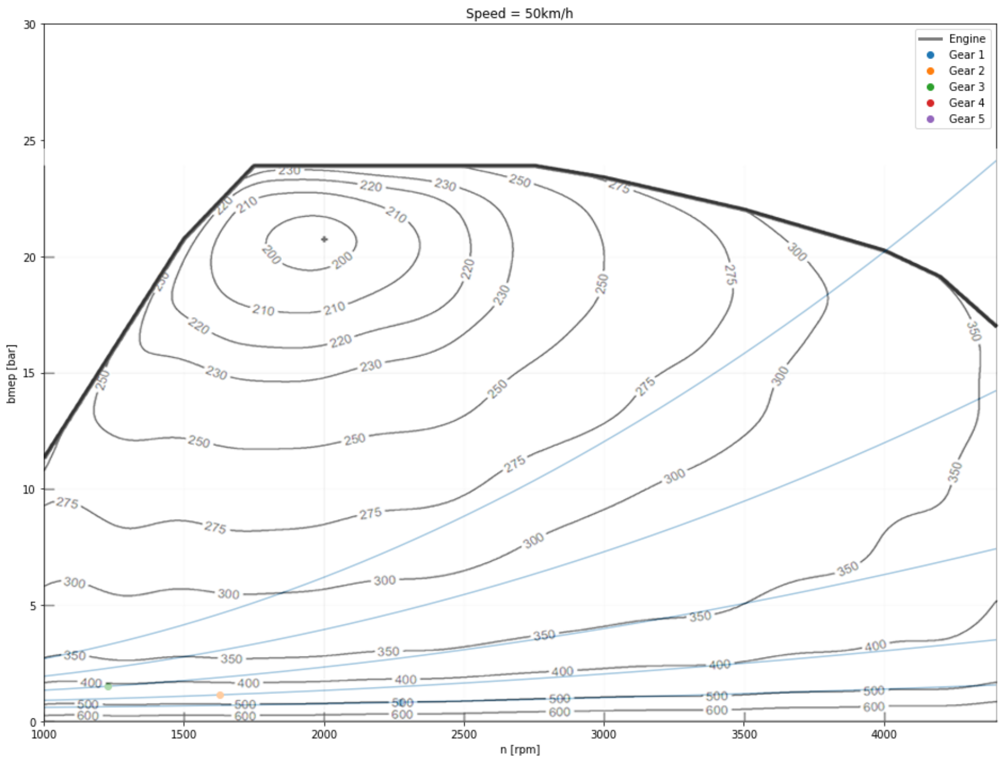

Figure 5.5: BMEP for v=50km/h

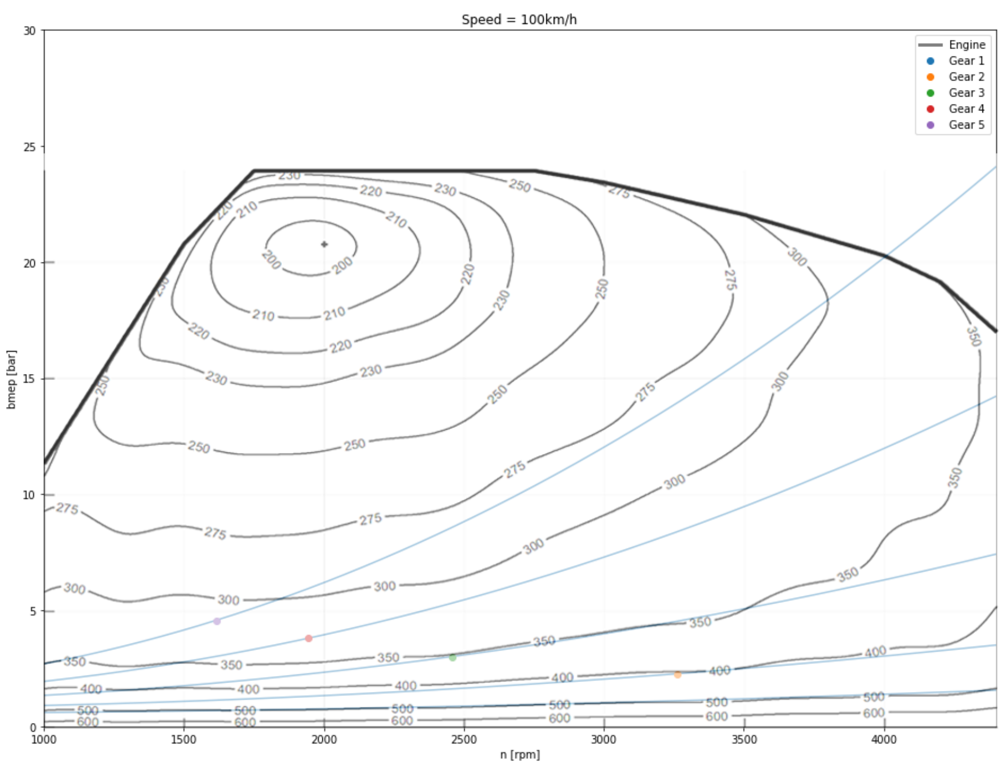

Figure 5.6: BMEP for v=100km/h

Chapter 6

Brakes

6.1 Brake force distribution

6.1.1 Brake force distribution diagram for empty and full vehicle

Following assumptions are made for the calculation of the brake force distribution.

$$\psi_{CW} = \frac{l_{f,CW}}{l} = \frac{G_{R,CW}}{G} = 0.35$$
$$\psi_{FL} = \frac{l_{f,FL}}{l} = \frac{G_{R,FL}}{G} = 0.45$$
$$\chi = \frac{h_G}{l} = 0.2$$

Below is the brake force distribution diagram for the curb weight and fully loaded vehicle.

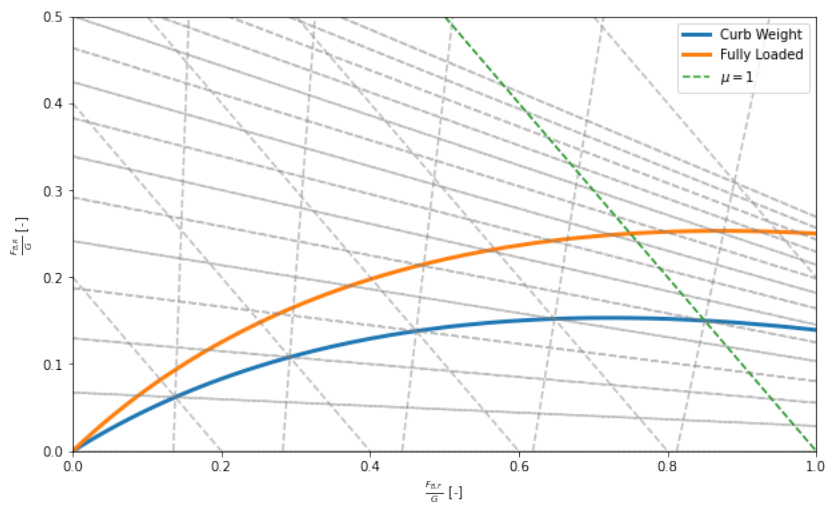

Figure 6.1: Brake force distribution diagram

6.1.2 Brake force control design

The brake force control strategy is plotted below. A limited distribution for the fully loaded vehicle is adopted to reduce the loss in brake power. Also, load sensing is used to limit the rear brake force so that the distribution curve can stay in the stable region, i.e. the controlled distribution can remain under the ideal distribution curve.

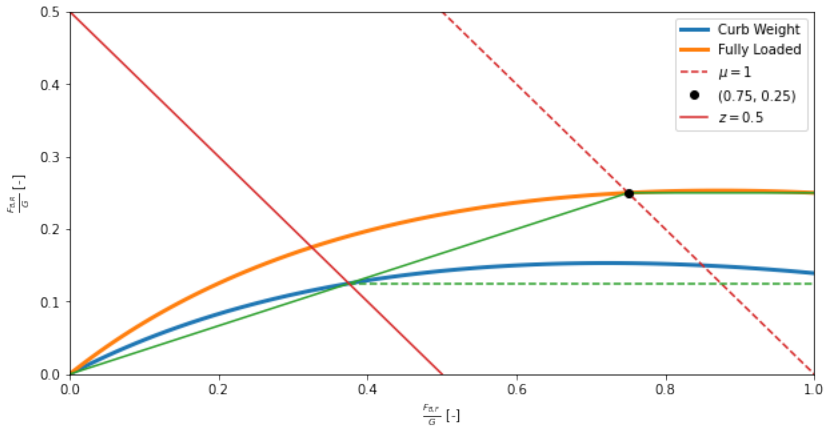

Figure 6.2: Brake force control strategy

6.2 Brake system design

The following brake system design is based on the fully loaded vehicle.

$$\text{Main cylinder diameter} : 25.5 \text{ mm}$$
$$\text{Disk diameter front} : 290 \text{ mm}$$
$$\text{Piston diameter front} : 38 \text{ mm}$$
$$\text{Disk diameter rear} : 175 \text{ mm}$$
$$\text{Piston diameter rear} : 28 \text{ mm}$$
$$\text{Max brake force front} : 9.973 \text{ kN}$$

$$\begin{aligned}
\ddot{x}_{max} &= \mu \cdot g + f_R \cdot g \\
&= 9.96\text{m/s}^2 \\
F_{max,F} &= \mu \cdot mg \cdot \frac{l_r}{l} + m \cdot \ddot{x}_{max} \cdot \frac{h_g}{l} \\
&= 1.0 \cdot [1350\text{kg} \cdot 9.81\text{m/s}^2 \cdot (1 - 0.45) + 1350\text{kg} \cdot 9.96\text{m/s}^2 \cdot 0.2] \\
&= 9.973kN
\end{aligned}$$

$$\text{Max brake force rear} : 3.311 \text{ kN (derived from brake force distribution diagram)}$$

$$\begin{aligned}
p_{max,F} &= \frac{F_{max,F}}{A_{pist,F}} = \frac{F_{max,F}}{\pi \cdot (\frac{d_{pist,F}}{2})^2} \\
&= \frac{9973\text{N}}{\pi \cdot (\frac{.038\text{m}}{2})^2} \\
&= 8794\text{kPa} \\
p_{max,R} &= \frac{F_{max,R}}{A_{pist,R}} = \frac{F_{max,R}}{\pi \cdot (\frac{d_{pist,R}}{2})^2} \\
&= \frac{3311\text{N}}{\pi \cdot (\frac{.028\text{m}}{2})^2} \\
&= 5377\text{kPa}
\end{aligned}$$

Max brake pressure front: 8794 kPa
Max brake pressure rear: 5377 kPa

Chapter 7

Vertical driving dynamics

7.1 Initial tuning of body springs

The stiffness of the body spring is calculated with the following assumptions. Based on the packaging plan, wheel travel on compression is assume to be limited to 0.08m for the front axle and 0.10m for the rear axle. Also, the mass of a single tire is assumed to be 30kg. Total change in load will be calculated and be divided by the wheel travel to find the stiffness.

7.1.1 Body spring rates front

Consider the front axle. First, load variation due to the loading state must be known. Mass on the front axle of curb weight vehicle is:

$$m_{F,CW} = m_V \cdot (1 - \psi_{CW}) = 950\text{kg} \cdot (1 - 0.35) = 617.5\text{kg}$$

Mass on the front axle of a fully load vehicle is:

$$m_{F,FL} = (m_V + m_{PL}) \cdot (1 - \psi_{FL}) = (950\text{kg} + 400\text{kg}) \cdot (1 - 0.45) = 742.5\text{kg}$$

Therefore the load variation is

$$\Delta m_{max,F} = 62.5\text{kg}$$

This can be used for the calculation of the spring rate for the dynamic load.

$$\begin{aligned}
\Delta G_{total} &= \Delta G_{dyn} + \Delta m_{max,F} \cdot g \\
&= \mu \cdot \chi \cdot m \cdot g + \Delta m_{max,F} \cdot g \\
&= \mu \cdot \chi \cdot \frac{m_V + m_{PL}}{2} \cdot g + \Delta m_{max,F} \cdot g \\
&= 1 \cdot 0.2 \cdot \frac{950\text{kg} + 400\text{kg}}{2} \cdot 9.81\text{m/s}^2 + 62.5\text{kg} \cdot 9.81\text{m/s}^2 \\
&= 1937.475\text{N}
\end{aligned}$$

Finally,

$$\begin{aligned}
c_{B,F} &= \frac{\Delta G_{total}}{\Delta U_{max}} = \frac{1937.475\text{N}}{0.08\text{m}} \\
&= 24218.4375\text{N/m}
\end{aligned}$$

7.1.2 Body spring rates rear

The calculation of the body spring rate for the rear axle can be carried out in a similar manner. The load variation can be calculated as follows:

$$
\begin{aligned}
m_{R,CW} &= m_V \cdot \psi_{CW} = 950\text{kg} \cdot 0.35 = 332.5\text{kg} \\
m_{R,FL} &= (m_V + m_{PL}) \cdot \psi_{FL} = (950\text{kg} + 400\text{kg}) \cdot 0.45 = 607.5\text{kg} \\
\Delta m_{max,R} &= 137.5\text{kg}
\end{aligned}
$$

Now, the spring rate for the dynamic load can be calculated.

$$
\begin{aligned}
\Delta G_{total} &= \mu \cdot \chi \cdot \frac{m_V + m_{PL}}{2} \cdot g + \Delta m_{max,R} \cdot g \\
&= 1 \cdot 0.2 \cdot \frac{950\text{kg} + 400\text{kg}}{2} \cdot 9.81\text{m/s}^2 + 137.5\text{kg} \cdot 9.81\text{m/s}^2 \\
&= 2673.225\text{N} \\
c_{B,R} &= \frac{\Delta G_{total}}{\Delta U_{max}} = \frac{2673.225\text{N}}{0.10\text{m}} \\
&= 26732.25\text{N/m}
\end{aligned}
$$

7.2 Initial tuning of shock absorbers

Damping parameter for the safety is calculated by minimizing the dynamic wheel load variations, and the damping parameter for the optimal comfort is calculated by minimizing the hub motion of the body and its acceleration. Based on these values, the optimal damping parameter was calculated.
The spring stiffness was assumed to be $220,000\text{N/m}$ for the front axle and $200,000\text{N/m}$ for the rear axle.

7.2.1 Damping parameter front

Optimal safety The calculation of the damping parameter for the optimal safety is as follows.

$$
\begin{aligned}
k_{B_s,F} &= \sqrt{(c_{B,F} + c_{T,F}) \cdot m_T} \\
&= \sqrt{(24218.4375\text{N/m} + 220000\text{N/m}) \cdot 30\text{kg}} \\
&= 2706.76\text{N} \cdot \text{s/m}
\end{aligned}
$$

Optimal comfort The calculation of the damping parameter for the optimal comfort is as follows.

$$
\begin{aligned}
k_{B_c,F} &= \sqrt{2 \cdot c_{B,F} \cdot m_{B,F}} \\
&= \sqrt{2 \cdot c_{B,F} \cdot \left(\frac{m_{F,CW}}{2} - m_T\right)} \\
&= \sqrt{2 \cdot 24218.4375\text{N/m} \cdot \left(\frac{617.5\text{kg}}{2} - 30\text{kg}\right)} \\
&= 3674.48\text{N} \cdot \text{s/m}
\end{aligned}
$$

7.2.2 Damping parameter rear

Optimal safety The calculation of the damping parameter for the optimal safety is as follows.

$$\begin{aligned}
k_{B_s,R} &= \sqrt{(c_{B,R} + c_{T,R}) \cdot m_T} \\
&= \sqrt{(26732.25\text{N/m} + 200000\text{N/m}) \cdot 30\text{kg}} \\
&= 2608.06\text{N} \cdot \text{s/m}
\end{aligned}$$

Optimal comfort The calculation of the damping parameter for the optimal comfort is as follows.

$$\begin{aligned}
k_{B_c,R} &= \sqrt{2 \cdot c_{B,R} \cdot m_{B,R}} \\
&= \sqrt{2 \cdot c_{B,R} \cdot \left(\frac{m_{R,CW}}{2} - m_T\right)} \\
&= \sqrt{2 \cdot 26732.25\text{N/m} \cdot \left(\frac{332.5\text{kg}}{2} - 30\text{kg}\right)} \\
&= 2698.99\text{N} \cdot \text{s/m}
\end{aligned}$$

7.3 Initial tuning of anti-roll bar

The roll angle at the operation condition of driving velocity of 16m/s and a cornering radius of 50m is calculated. If the roll angel exceeds 3°, an anti-roll bar will be implemented, whose spring rate will be calculated.

The spring ratio between the body spring and the equivalent tire spring is assumed to be 0.6. Also, based on the track width from the specification, the spring track width for both front and rear axle s_s is assumed to be 1.05m. Finally, the roll center is assumed to be at the ground, which will simplify the calculation of the distance between the CoG of the body and the roll center.

The distance between the CoG of the body and the roll center is as follows.

$$\begin{aligned}
\Delta h &= \frac{h_G \cdot m - 4 \cdot m_T \cdot r_{dyn}}{m_B} \\
&= \frac{\chi \cdot l \cdot (m_V + m_{PL}) - 4 \cdot m_T \cdot r_{dyn}}{(m_V + m_{PL} - 4 \cdot m_T} \\
&= \frac{0.2 \cdot 2.5\text{m} \cdot (950\text{kg} + 400\text{kg}) - 4 \cdot 30\text{kg} \cdot 0.289\text{m}}{(950\text{kg} + 400\text{kg}) - 4 \cdot 30\text{kg}} \\
&= 0.521\text{m}
\end{aligned}$$

Next, the actual body spring stiffness is calculated from the equivalent body spring stiffness.

$$\begin{aligned}
c_{B,E} &= c_{B,F} + c_{B,R} = 50950.6875\text{N/m} \\
c_B &= \frac{c_{B,E}}{i_c^2} = \frac{50950.6875\text{N/m}}{0.6^2} = 141529.6875\text{N/m}
\end{aligned}$$

This can be used for the calculation of the roll angle at the given operation condition.

$$\begin{aligned}
\phi &= \frac{2 \cdot \Delta h}{c_B \cdot s_s^2} \cdot F_{C,B} = \frac{2 \cdot \Delta h}{c_B \cdot s_s^2} \cdot \frac{(m_V + m_P L) \cdot v^2}{r} \\
&= \frac{2 \cdot 0.521\text{m}}{141529.6875\text{N/m} \cdot (1.05\text{m})^2} \cdot \frac{(950\text{kg} + 400\text{kg}) \cdot (16\text{m/s})^2}{50\text{m}} \\
&= 2.41^\circ
\end{aligned}$$

Since the roll angle at the given operating condition is already smaller than the maximum roll angle of $3°$, anti-roll bar is not required in the vehicle.

Chapter 8

Lateral driving dynamics

The bicycle model is used to analyze the lateral driving characteristic of the vehicle. Yaw moment of inertia of $J_z = 2000kg \cdot m^2$ is used for the calculation.

8.1 Self-steering behavior

8.1.1 Stationary self-steering behavior

The loads on the front and rear axles are calculated from the quantities:

$$F_{z,F} = m \cdot g \cdot (1 - \psi)$$
$$F_{z,R} = m \cdot g \cdot \psi$$

For the curb weight, $m = 950kg$, $\psi = 0.35$, and for the fully loaded vehicle, $m = 1350kg$, $\psi = 0.45$. Using the graph below, the cornering stiffness for both axles were estimated.

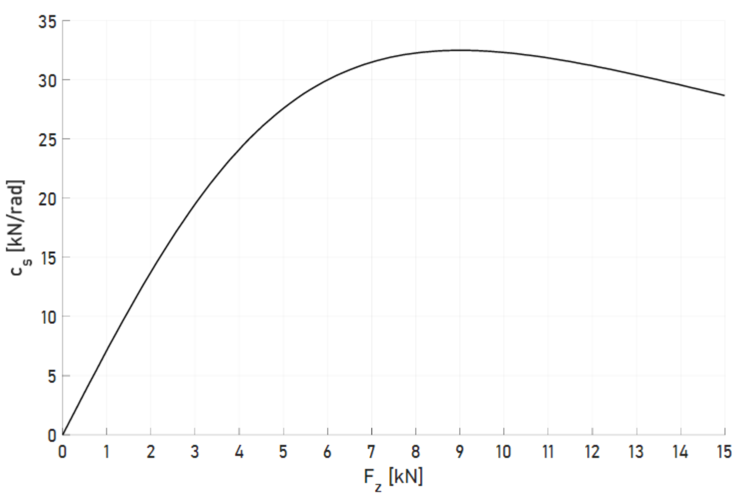

Figure 8.1: Axle loads vs cornering stiffness

Load condition	$F_{z,F}$	$c_{s,F}$	$F_{z,R}$	$c_{s,R}$
Curb weight	6057.675N	30 kN/rad	3261.825N	20 kN/rad
Fully loaded	7283.925N	32 kN/rad	5959.575N	30 kN/rad

Yaw eigenfrequency, Damping ratio, Static yaw gain

From the calculated cornering stiffness, eigenfrequency, damping ratio, and static yaw gain are obtained as functions of speed.

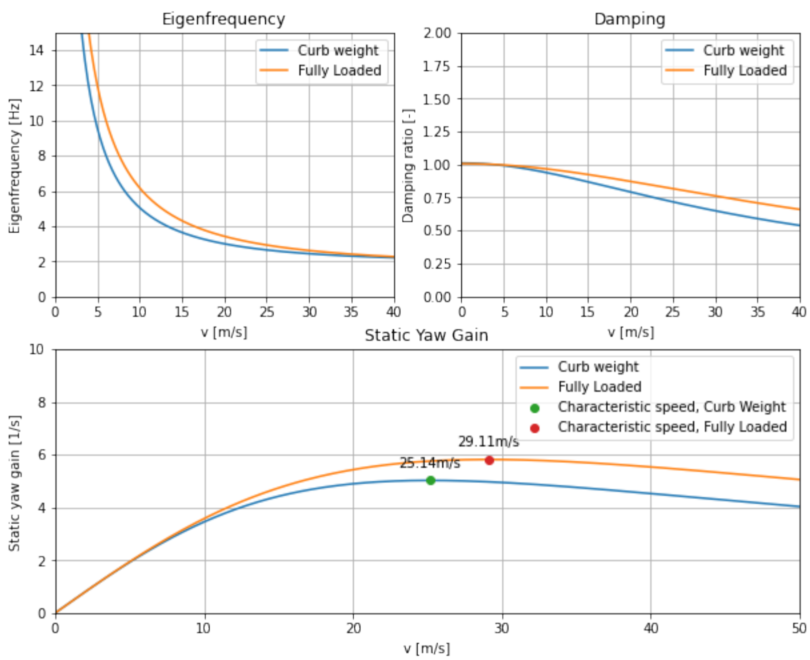

Figure 8.2: Eigenfrequency, damping ratio, static yaw gain vs. speed

From the plots, it can be seen that the vehicle has reasonable eigenfrequency and damping ratio. For the speed range between $20m/s$ and $30m/s$, the eigenfrequencies of both the curb weight and fully loaded vehicle is within the range between 2 and 4, which implies that the vehicle is not sluggish in terms of yaw response. Also, the damping ratio for the same speed range is around 0.7 for the curb weight and 0.8 for the fully loaded. Since the damping ratio for the curb weight vehicle is slightly below the target damping of 0.8, the car can exhibit a bit nervous behavior when empty.

Characteristic speed

From the figure above, it can be seen that the vehicle shows an understeering behavior. Thus, the critical speed is not present in the damping ratio curve, but the characteristic speed can be found from the static yaw gain plot. The characteristic speeds for the curb weight and fully loaded vehicle are determined by finding the maximum value of the static yaw gain, and their values are as follows:

$$v_{ch,CW} = 25.14m/s$$
$$v_{ch,FL} = 29.11m/s$$

The characteristic speed of a fully loaded vehicle is slightly larger than the desired range of 18 to 28m/s. This can be enhanced by reducing the value of ψ_{FL}. If ψ_{FL} is reduced from 0.45 to 0.43, the axle loads becomes $F_{z,F} = 7549N$ and $F_{z,R} = 5695N$. The corresponding cornering stiffness are $c_{s,F} = 32kN/rad$ and $c_{s,R} = 28kN/rad$. Using this cornering stiffness, the characteristic speed of the fully loaded vehicles can be reduced to 27.48m/s, as it can be observed from the plot below.

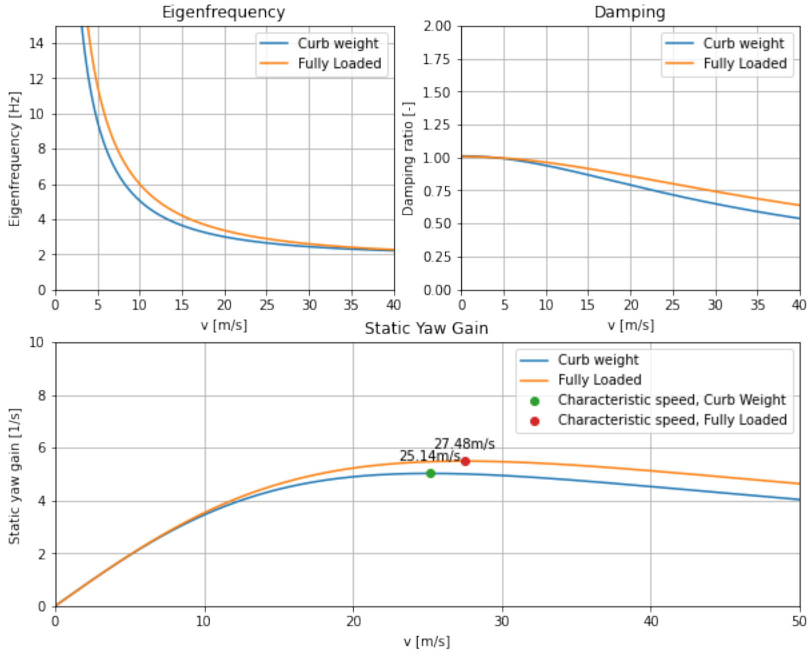

Figure 8.3: Eigenfrequency, damping ratio, static yaw gain curves with $\psi_{FL} = 0.43$

8.2 Steering design

The steering design we've chosen for our vehicle is rack-and-pinion steering with dual-pinion EPS. Rack-and-pinion steering is very reliable because it is used extensively for passenger vehicles. Additionally, it has fewer parts than other types of steering, which contributes to greater steering precision and a lighter vehicle. A more responsive steering system will improve control and overall feedback.

8.2.1 Front and rear axle design

The front axle for our vehicle is F2, McPherson suspension, and the rear axle is R8, rear McPherson suspension. According to the design matrix, these axles are extremely economic and maintain sufficient performing ability. Competitors such as the Mitsubishi Mirage and the Honda Fit opted for R5, torsion-type twist beam axle, as their rear axle. Their rear axle lacks in performing ability, but is more economic. Our vehicle aims to have superior kinematic and handling ability compared to our competitors, in exchange for a slight amount of economic consideration.

Appendix A

Links

A.1 Link to Google Colab Jupyter Notebook

`https://colab.research.google.com/drive/1qDDLOP1KD_rThgtA6hmgC3WqzFRZwH0_#scrollTo=gW2y8Ja5-elK`